CAM GEARS DYNAMICS IN THE CLASSIC DISTRIBUTION

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ABSTRACT

The paper presents an original method to determine the general dynamics of mechanisms with rotation cams and followers, particularized to the plate translated follower. First, one presents the dynamics kinematics. Then one solves the Lagrange equation and with an original dynamic model with one degree of freedom, with variable internal amortization, it makes the dynamic analysis of two models.

Keywords: cam mechanisms; rotary cam; plate translated follower; dynamics; classic distribution

1.   INTRODUCTION

In conditions which started to magnetic motors, oil fuel is decreasing, energy which was obtained by burning oil is replaced with nuclear energy, hydropower, solar energy, wind, and other types of unconventional energy, in the conditions in which electric motors have been instead of internal combustion in public transport, but more recently they have entered in the cars world (Honda has produced a vehicle that uses a compact electric motor and electricity consumed by the battery is restored by a system that uses an electric generator with hydrogen combustion in cells, so we have a car that burns hydrogen, but has an electric motor), which is the role and prospects which have internal combustion engines type Otto or Diesel?

Internal combustion engines in four-stroke (Otto, Diesel) are robust, dynamic, compact, powerful, reliable, economic, autonomous, independent and will be increasingly clean.

The Otto engines or those with internal combustion in general, will have to adapt to hydrogen fuel. It is composed of the basic hydrogen which can be extracted industrially, practically from any item (or combination) through nuclear, chemical, photonic by radiation, by burning process, etc... (Most easily hydrogen can be extracted from water by breaking up into constituent elements, hydrogen and oxygen; by burning hydrogen one obtains water again that restores a circuit in nature, with no losses and no pollution). Hydrogen must be stored in reservoirs cell (a honeycomb) to ensure that there is no danger of explosion; the best would be if we could breaking up water directly on the vehicle, in which case the reservoir would feed water (and there were announced some successful).

As a backup, there are trees that can donate a fuel oil, which could be planted on the extended zone, or directly in the consumer court. With many years ago, Professor Melvin Calvin, (Berkeley University), discovered that "Euphora" tree, a rare species, contained in its trunk a liquid that has the same characteristics as raw oil. The same professor discovered on the territory of Brazil, a tree which contains in its trunk a fuel with properties similar to diesel.

During a journey in Brazil, the natives driven him (Professor Calvin) to a tree called by them "Copa-Iba".

At the time of boring the tree trunk, from it to begin flow a gold liquid, which was used as indigenous raw material base for the preparation of perfumes or, in

concentrated form, as a balm. Nobody see that it is a pure fuel that can be used directly by diesel engines.

Calvin said that after he poured the liquid extracted from the tree trunk directly into the tank of his car (equipped with a diesel), the engine functioned irreproachable. In Brazil the tree is fairly widespread. It could be adapted in other areas of the world, planted in the forests, and the courts of people.

From a jagged tree is filled about half of the tank; one covers the slash and it is not open until after six months; it means that having 12 trees in a courtyard, a man can fill monthly a tank with the new natural diesel fuel.

Some countries (USA, Brazil, Germany) producing alcohol or vegetable oils, for their use as fuel.

In the future, aircraft will use ion engines, magnetic, laser or various micro particles accelerated. Now, and the life of the jet engine begin to end. In the future the jet will be accomplished with atomic particle or nuclear.

Even in these conditions internal combustion engines will be maintained in land vehicles (at least), for power, reliability and especially their dynamics. Thermal engine efficiency is still low and, about one third of the engine power is lost just by the distribution mechanism. We must improve the distribution mechanism.

The paper proposes an original dynamic model illustrated for the rotating cam with plate translated follower. One presents the dynamic kinematics (the original kinematics); the variable velocity of the camshaft obtained by an approximate method is used with an original dynamic system having one degree of freedom and a variable internal amortization; it tests two movement laws, one classic and an other original.

2.   DYNAMIC OF THE CLASSIC DISTRIBUTION MECHANISM

2.1.   Precision Kinematics in the Classic Distribution Mechanism

In the fig. number one, it presents the kinematic schema of the classic distribution mechanism, in two consecutive positions (HAIN K., 1971; GIORDANA F., 1979); with an interrupted line is represented the particular position when the follower is situated in the lowest possible plane, (s=0), and the cam which has a clockwise rotation, with constant angular velocity, ω, is situated in the point A^0, (the fillet point between the base profile and the rise profile), a particular point that marks the beginning of the rise movement of the follower (ANGELAS J. and Lopez-Cajun C., 1988), imposed by the cam-profile (DAN K., 2009); with a continue line is

represented the higher joint in a certain position of the rise phase (PETRESCU F. I., 1987).
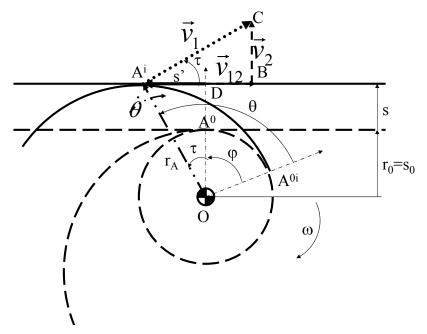

Figure 1. The kinematics of the classic distribution mechanism

The point A^0, which marks the initial higher pair, represents in the same time the contact point between the cam and the follower in the first position. The cam is rotating with the angular velocity, ω (the camshaft angular velocity), describing the angle φ, which shows how the base circle has rotated clockwise (together with the camshaft); this rotation can be seen on the base circle between the two particular points, A^0 and A^{0i}.

In this time the vector r_A=OA (which represents the distance between the centre of cam O, and the contact point A), has rotated anticlockwise with the angle τ. If one measures the angle θ, which positions the general vector, r_A, in function of the particular vector, r_{A0}, one obtains the relation (0):

$$\theta = \varphi + \tau \tag{0}$$

where r_A is the module of the vector $\vec{r}_A$, and θ_A represents the phase angle of the vector $\vec{r}_A$.

The angular velocity of the vector $\vec{r}_A$ is $\dot{\theta}_A$ which is a function of the angular velocity of the camshaft, ω, and of the angle φ (by the movement laws s(φ), s'(φ), s''(φ)).

The follower isn't acted directly by the angle φ and the angular velocity ω; it's acted by the vector $\vec{r}_A$, which has the module r_A, the position angle θ_A and the angular velocity $\dot{\theta}_A$. From here we deduce a particular (dynamic) kinematics, the classical kinematics being just static and approximate kinematics (ANTONESCU P., 1985).

Kinematics we define the next velocities (Fig. 1): $\vec{v}_1$ =the cam's velocity; which is the velocity of the vector $\vec{r}_A$, in the point A; now the classical relation (1) becomes an approximate relation, and the real relation takes the form (2).

$$v_1 = r_A \cdot \omega \qquad (1)$$

$$v_1 = r_A \cdot \dot{\theta}_A \qquad (2)$$

The velocity $\vec{v}_1 = AC$ is separating into the velocity $\vec{v}_2$ =BC (the follower's velocity which acts on its axe, vertically) and $\vec{v}_{12}$ =AB (the slide velocity between the two profiles, the sliding velocity between the cam and the follower, which works along the direction of the commune tangent line of the two profiles in the contact point).

Because usually the cam profile is synthesis for the classical module C with the AD=s' known, it can write the relations:

$$r_A^2 = (r_0 + s)^2 + s'^2 \qquad (3)$$

$$r_A = \sqrt{(r_0 + s)^2 + s'^2} \qquad (4)$$

$$\cos \tau = \frac{r_0 + s}{r_A} = \frac{r_0 + s}{\sqrt{(r_0 + s)^2 + s'^2}} \qquad (5)$$

$$\sin \tau = \frac{AD}{r_A} = \frac{s'}{r_A} = \frac{s'}{\sqrt{(r_0 + s)^2 + s'^2}} \qquad (6)$$

$$v_2 = v_1 \cdot \sin \tau = r_A \cdot \dot{\theta}_A \cdot \frac{s'}{r_A} = s' \cdot \dot{\theta}_A \qquad (7)$$

Now, the follower's velocity isn't $\dot{s}$ ($v_2 \neq \dot{s} \equiv s' \cdot \omega$), but it's given by the relation (9). In the case of the classical distribution mechanism the transmitting function D is given by the relations (8):

$$\begin{cases} \dot{\theta}_A = D \cdot \omega \\ D = \dfrac{\dot{\theta}_A}{\omega} \end{cases} \tag{8}$$

$$v_2 = s' \cdot \dot{\theta}_A = s' \cdot D \cdot \omega \tag{9}$$

The determining of the sliding velocity between the profiles is made with the relation (10):

$$v_{12} = v_1 \cdot \cos\tau = r_A \cdot \dot{\theta}_A \cdot \frac{r_0 + s}{r_A} = (r_0 + s) \cdot \dot{\theta}_A \tag{10}$$

The angles τ and θ_A will be determined, and also their first and second derivatives.

The τ angle has been determined from the triangle ODA^i (Fig.1) with the relations (11-13):

$$\sin\tau = \frac{s'}{\sqrt{(r_0 + s)^2 + s'^2}} \tag{11}$$

$$\cos\tau = \frac{r_0 + s}{\sqrt{(r_0 + s)^2 + s'^2}} \tag{12}$$

$$tg\,\tau = \frac{s'}{r_0 + s} \tag{13}$$

One derives (11) in function of φ angle and obtains (14):

$$\tau' \cdot \cos\tau = \frac{s'' \cdot r_A - s' \cdot \dfrac{(r_0 + s) \cdot s' + s' \cdot s''}{r_A}}{(r_0 + s)^2 + s'^2} \tag{14}$$

The relation (14) will be written in the form (15):

$$\tau' \cdot \cos\tau = \frac{s'' \cdot (r_0 + s)^2 + s'' \cdot s'^2 - s'^2 \cdot (r_0 + s) - s'^2 \cdot s''}{[(r_0 + s)^2 + s'^2] \cdot \sqrt{(r_0 + s)^2 + s'^2}} \tag{15}$$

From the relation (12) one extracts the value of $\cos\tau$, which will be introduced in the left term of the expression (15); then one reduces $s'' . s'^2$ from the right term of the expression (15) and obtains the relation (16):

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$$\tau' \cdot \frac{r_0 + s}{\sqrt{(r_0 + s)^2 + s'^2}} = \frac{(r_0 + s) \cdot [s'' \cdot (r_0 + s) - s'^2]}{[(r_0 + s)^2 + s'^2] \cdot \sqrt{(r_0 + s)^2 + s'^2}} \tag{16}$$

After some simplifications the relation (17), which represents the expression of τ', is finally obtained:

$$\tau' = \frac{s'' \cdot (r_0 + s) - s'^2}{(r_0 + s)^2 + s'^2} \tag{17}$$

Now, when τ' has been explicitly **(**deduced), the next derivatives can be determined. The expression (17) will be derived directly and obtains for the beginning the relation (18):

$$\tau'' = \frac{[s''' \cdot (r_0 + s) + s'' \cdot s' - 2s' \cdot s''] \cdot [(r_0 + s)^2 + s'^2] - 2[s'' \cdot (r_0 + s) - s'^2] \cdot [(r_0 + s) \cdot s' + s' \cdot s'']}{[(r_0 + s)^2 + s'^2]^2} \tag{18}$$

The terms from the first bracket of the numerator (s'.s'') are reduced, and then one draws out s' from the fourth bracket of the numerator and obtains the expression (19):

$$\tau'' = \frac{[s''' \cdot (r_0 + s) - s' \cdot s''] \cdot [(r_0 + s)^2 + s'^2] - 2 \cdot s' \cdot [s'' \cdot (r_0 + s) - s'^2] \cdot [r_0 + s + s'']}{[(r_0 + s)^2 + s'^2]^2} \tag{19}$$

Now it can calculate θ_A, with its first two derivatives, $\dot{\theta}_A$ and $\ddot{\theta}_A$. We will write θ instead of θ_A, to simplify the notation. Now one can determine (20) which is the same of (0):

$$\theta = \tau + \varphi \tag{20}$$

One derives (20) and obtains the relation (21):

$$\dot{\theta} = \dot{\tau} + \dot{\varphi} = \tau' . \omega + \omega = \omega . (1 + \tau') = D . \omega \tag{21}$$

One derives twice (20), (or derives 21) and obtains (22), with (relation 24) and with ω=ct.:

$$\ddot{\theta} = \ddot{\tau} + \ddot{\varphi} = \frac{d^2\tau}{dt^2} + \frac{d^2\varphi}{dt^2} = \frac{d}{dt}\left(\frac{d\tau}{dt}\right) + \frac{d}{dt}\left(\frac{d\varphi}{dt}\right) = \frac{d}{dt}\left(\frac{d\tau}{d\varphi} \cdot \frac{d\varphi}{dt}\right) + \frac{d}{dt}(\omega) =$$

$$= \frac{d}{dt}(\tau' \cdot \omega) + \frac{d\omega}{dt} = \frac{d\tau'}{dt} \cdot \omega + 0 = \frac{d\tau'}{d\varphi} \cdot \frac{d\varphi}{dt} \cdot \omega = \tau'' \cdot \omega \cdot \omega = \tau'' \cdot \omega^2 = D' \cdot \omega^2 \qquad (22)$$

One can write now the transmission functions, D and D' (for the classical module, C), in the forms (23-24):

$$D = \tau' + 1 \qquad (23)$$

$$D' \equiv D^I = \tau'' \qquad (24)$$

To calculate the follower's velocity (25) we need the expression of the transmission function, D (where ω is the general rotation speed of the cam, the average value, or the input value, which is a constant, and w is the momentary rotation speed of the cam which is a variable value).

$$v_2 = s' \cdot w = s' \cdot \dot{\theta}_A = s' \cdot \dot{\theta} = s' \cdot D \cdot \omega = \dot{s} \cdot D \qquad (25)$$
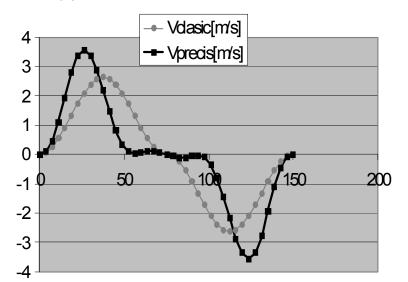
where

$$w = D.\omega \qquad (26)$$

For the classical distribution mechanism (Module C), the variable w is the same as $\dot{\theta}_A$ (see the relation 25). But in the case of B and F modules (at the cam gears where the follower has a roll), the transmitted function D and w take complex forms.

Now, one can determine the acceleration of the follower (27).

$$\ddot{y} \equiv a_2 = (s'' \cdot D + s' \cdot D') \cdot \omega^2 \qquad (27)$$

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Figure 2 represents the classical and dynamic kinematics; the velocities (a), and the accelerations (b).

a

b

Figure 2. The classical and dynamic kinematics; a-velocities and b-accelerations of the follower

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To determine the acceleration of the follower, are necessary to be known s' and s'', D and D', τ' and τ''.

The dynamic kinematics diagrams of v_2 (obtained with relation 25, see Fig. 2a), and a_2 (obtained with relation 27, see Fig. 2b), have a more dynamic aspect than one kinematic. One has used the movement law SIN, a rotational speed of the crankshaft n=5500 rpm, a rise angle $\varphi_u=75^0$, a fall angle $\varphi_d=75^0$ (identically with the ascendant angle), a ray of the basic circle of the cam, $r_0=17$ mm and a maxim stroke of the follower, $h_T=6$ mm.

Anyway, the dynamics is more complex (NORTON R. L., 1999), having in view the masses and the inertia moments, the resistant and motor forces, the elasticity constants and the amortization coefficient of the kinematic chain, the inertia forces of the system, the angular velocity of the camshaft and the variation of the camshaft's angular velocity, ω, with the cam's position, φ, and with the rotational speed of the crankshaft, n.

2.2. Solving Approximately the Lagrange Movement Equation

In the kinematics and the static forces study of the mechanisms one considers the shaft's angular velocity constant, $\dot{\varphi} = \omega$=constant, and the angular acceleration null, $\ddot{\varphi} = \dot{\omega} = \varepsilon = 0$. In reality, this angular velocity ω isn't constant, but it is variable with the camshaft position, φ.

In mechanisms with cam and follower the camshaft's angular velocity is variable as well. One shall see further the Lagrange equation, written in the differentiate mode and its general solution.

The differentiate Lagrange equation has the form (28):

$$J^* \cdot \ddot{\varphi} + \frac{1}{2} \cdot J^{*I} \cdot \dot{\varphi}^2 = M^* \qquad (28)$$

Where J* is the inertia moment (mass moment, or mechanic moment) of the mechanism, reduced at the crank, and M* represents the difference between the motor moment reduced at the crank and the resistant moment reduced at the crank; the angle φ represents the rotation angle of the crank (crankshaft). J^{*I} represents the

derivative of the mechanic moment in function of the rotation angle φ of the crank (29).

$$\frac{1}{2} \cdot J^{*I} = \frac{1}{2} \cdot \frac{dJ^*}{d\varphi} = L \tag{29}$$

Using the notation (29), the equation (28) will be written in the form (30):

$$J^* \cdot \ddot{\varphi} + L \cdot \dot{\varphi}^2 = M^* \tag{30}$$

We divide the terms by J* and (30) takes the form (31):

$$\ddot{\varphi} + \frac{L}{J^*} \cdot \dot{\varphi}^2 = \frac{M^*}{J^*} \tag{31}$$

The term with $\dot{\varphi}^2$ will be moved to the right side of the equation and the form (32) will be obtained:

$$\ddot{\varphi} = \frac{M^*}{J^*} - \frac{L}{J^*} \cdot \dot{\varphi}^2 \tag{32}$$

Replacing the left term of the expression (32) with (33) one obtains the relation (34):

$$\ddot{\varphi} = \frac{d\dot{\varphi}}{dt} = \frac{d\dot{\varphi}}{d\varphi} \cdot \frac{d\varphi}{dt} = \frac{d\dot{\varphi}}{d\varphi} \cdot \dot{\varphi} = \frac{d\omega}{d\varphi} . \omega \tag{33}$$

$$\omega \cdot \frac{d\omega}{d\varphi} = \frac{M^*}{J^*} - \frac{L}{J^*} \cdot \omega^2 = \frac{M^* - L \cdot \omega^2}{J^*} \tag{34}$$

Because, for an angle φ, ω is different from the nominal constant value ω_n, one can write the relation (35), where dω represents the momentary variation for the angle φ; the variable dω and the constant ω_n lead us to the needed variable, ω:

$$\omega = \omega_n + d\omega \tag{35}$$

In the relation (35), ω and dω are functions of the angle φ, and ω_n is a constant parameter, which can take different values in function of the rotational speed of the drive-shaft, n. At a moment, n is a constant and ω_n is a constant as well (because ω_n is a function of n). The angular velocity, ω, becomes a function of n too (see the relation 36):

$$\omega(\varphi, n) = \omega_n(n) + d\omega(\varphi, \omega_n(n)) \tag{36}$$

With (35) in (34), one obtains the equation (37):

$$(\omega_n + d\omega) \cdot d\omega = [\frac{M^*}{J^*} - \frac{L}{J^*} \cdot (\omega_n + d\omega)^2] \cdot d\varphi \qquad (37)$$

The relation (37) takes the form (38):

$$\omega_n \cdot d\omega + (d\omega)^2 = \frac{M^*}{J^*} \cdot d\varphi - \frac{L}{J^*} \cdot d\varphi \cdot [\omega_n^2 + (d\omega)^2 + 2 \cdot \omega_n \cdot d\omega] \qquad (38)$$

The equation (38) will be written in the form (39):

$$\omega_n \cdot d\omega + (d\omega)^2 - \frac{M^*}{J^*} \cdot d\varphi + \frac{L}{J^*} \cdot d\varphi \cdot \omega_n^2 +$$
$$+ \frac{L}{J^*} \cdot d\varphi \cdot (d\omega)^2 + 2 \cdot \frac{L}{J^*} \cdot d\varphi \cdot \omega_n \cdot d\omega = 0 \qquad (39)$$

The relation (39) takes the form (40):

$$(\frac{L}{J^*} \cdot d\varphi + 1) \cdot (d\omega)^2 + 2 \cdot (\frac{L}{J^*} \cdot d\varphi + \frac{1}{2}) \cdot \omega_n \cdot d\omega -$$
$$- (\frac{M^*}{J^*} \cdot d\varphi - \frac{L}{J^*} \cdot d\varphi \cdot \omega_n^2) = 0 \qquad (40)$$

The relation (40) is an equation of the second degree in dω. The discriminate of the equation (40) can be written in the forms (41) and (42):

$$\Delta = \frac{L^2}{J^{*2}} \cdot (d\varphi)^2 \cdot \omega_n^2 + \frac{\omega_n^2}{4} + \frac{L}{J^*} \cdot d\varphi \cdot \omega_n^2 + \frac{L \cdot M^*}{J^{*2}} \cdot (d\varphi)^2$$
$$+ \frac{M^*}{J^*} \cdot d\varphi - \frac{L^2}{J^{*2}} \cdot (d\varphi)^2 \cdot \omega_n^2 - \frac{L}{J^*} \cdot d\varphi \cdot \omega_n^2 \qquad (41)$$

$$\Delta = \frac{\omega_n^2}{4} + \frac{L \cdot M^*}{J^{*2}} \cdot (d\varphi)^2 + \frac{M^*}{J^*} \cdot d\varphi \qquad (42)$$

One keeps for dω just the positive solution, which can generate positives and negatives normal values (43), and in this mode only normal values will be obtained for ω; for $\Delta < 0$ one considers dω=0 (this case must be not seeing if the equation is correct).

$$d\omega = \frac{-\frac{L}{J^*} \cdot d\varphi \cdot \omega_n - \frac{\omega_n}{2} + \sqrt{\Delta}}{\frac{L}{J^*} \cdot d\varphi + 1} \qquad (43)$$

Observations: For mechanisms with rotative cam and follower, using the new relations, with M^* (the reduced moment of the mechanism) obtained by the writing of the known reduced resistant moment and by the determination of the reduced motor moment by the integration of the resistant moment one frequently obtains some bigger values for $d\omega$, or zones with Δ negative, with complex solutions for $d\omega$. This fact gives us the obligation to reconsider the method to determine the reduced moment.

If we take into consideration M^*_r and M^*_m, calculated independently (without integration), one obtains for the mechanisms with cam and follower normal values for $d\omega$, and $\Delta \geq 0$.

In paper (PETRESCU, F.I., Petrescu, R.V. 2005) one presents the relations to determine the resistant force (44) reduced to the valve, and the motor force (45) reduced to the ax of the valve:

$$F_r^* = k \cdot (x_0 + x) \tag{44}$$

$$F_m^* = K \cdot (y - x) \tag{45}$$

The reduced resistant moment (46), or the reduced motor moment (47), can be obtained by the resistant or motor force multiplied by the reduced velocity, *x'*.

$$M_r^* = k \cdot (x_0 + x) \cdot x' \tag{46}$$

$$M_m^* = K \cdot (y - x) \cdot x' \tag{47}$$

2.3.   The Dynamic Relations Used

The dynamics relations used (48-49), have been deduced in the paper [PETRESCU, F.I., Petrescu, R.V. (2005)]:

$$\Delta X = (-1) \cdot \frac{\left(k^2 + 2 \cdot k \cdot K\right) \cdot s^2 + 2 \cdot k \cdot x_0 \cdot (K+k) \cdot s + \left[\dfrac{K^2}{K+k} \cdot m_S^* + (K+k) \cdot m_T^*\right] \cdot \omega^2 \cdot (Ds')^2}{2 \cdot \left(s + \dfrac{k \cdot x_0}{K+k}\right) \cdot (K+k)^2} \tag{48}$$

$$X = s - \frac{\left[\dfrac{K^2}{K+k} \cdot m_S^* + (K+k) \cdot m_T^*\right] \cdot \omega^2 \cdot (Ds')^2}{2 \cdot \left(s + \dfrac{k \cdot x_0}{K+k}\right) \cdot (K+k)^2} - \frac{\left(k^2 + 2 \cdot k \cdot K\right) \cdot s^2 + 2k \cdot x_0 \cdot (K+k) \cdot s}{2 \cdot \left(s + \dfrac{k \cdot x_0}{K+k}\right) \cdot (K+k)^2} \tag{49}$$

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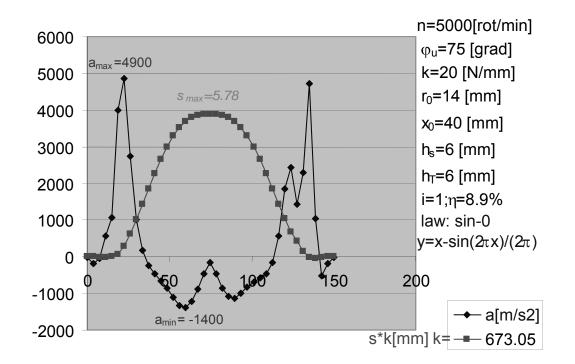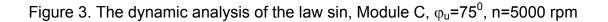## 3.   THE DYNAMIC ANALYSIS

The dynamic analysis or the classical movement law sin (BAGEPALLI B. S., 1991), can be seen in the diagram from figure 3, and in figure 4 one can see the diagram of an original movement law (C4P) (module C) (SATYANARAYANA M., 2009). It was considered a motor shaft speed of 5000 [rpm] (rot. per minute).

The angle of climb had been taken 75 [deg] (degrees sexazecimale). Constant spring valve spring was considered to be 20 [N/mm]. The basic circle radius of the cam is 14 [mm]. Valve spring pretension is 40 [mm] (GE Z., 2011). Valve travel is 6 [mm]. Tappet race all 6 [mm]. Use has been made of a transmission ratio of rocker arm equal to the unit.

The program of calculation used was written in excel that is extremely affordable.

Figure 3. The dynamic analysis of the law sin, Module C, $\varphi_u = 75^0$, n=5000 rpm
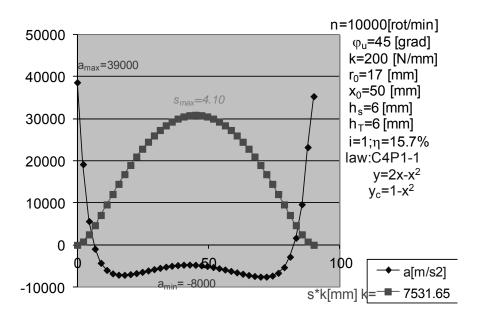
Figure 4. The dynamic analysis of the new law, C4P, Module C, $\varphi_u=45^0$, n=10000 rpm

Using a law of movement of original tappet can increase engine speed to superior (TARAZA D., 2002). Increasing the speed can be increased more than if we use another module cam follower, namely the module B (see the Figure 5-6).
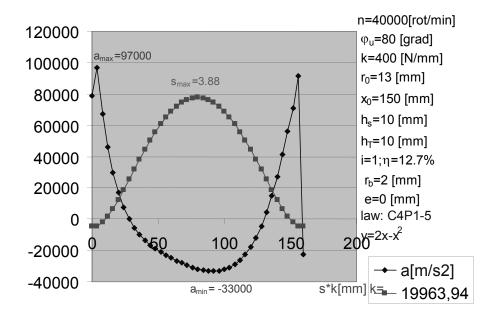

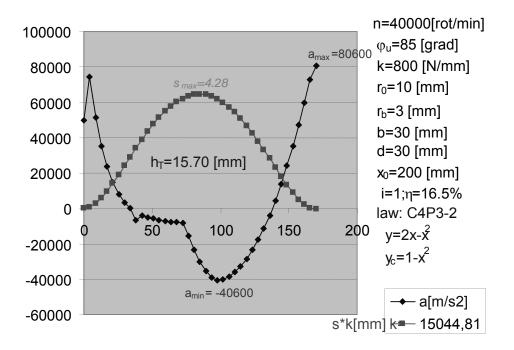Figure 5. Law C4P1-5, Module B, $\varphi_u=80^0$, n=40000 rpm

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Figure 6. Law C4P3-2, Module F, $\varphi_u=85^0$, n=40000 rpm

4. CONCLUSIONS

If we improve the cam dynamics, it can synthesize high-speed cam, or high-performance camshafts (PETRESCU F. I. and Petrescu R. V., 2005, 2008).

In this paper one presents an original method to determine the dynamic parameters at the camshaft (the distribution mechanisms). It makes the dynamics, of the rotary cam and plate tappet with translational motion, with a great precision.

The presented method is the most elegant and direct method to determine the kinematics and dynamic parameters. The dynamic synthesis can generate a cam profile which will work without vibrations. Processes robotization increasingly determine and influence the emergence of new industries, applications in specific environmental conditions, approach new types of technological operations, handling of objects in outer space, leading teleoperator in disciplines such as medicine, robots that covers a whole larger service benefits our society, modern and computerized. In this context, this paper seeks to contribute to the scientific and technical applications in dynamic analysis and synthesis of cam mechanisms.

Using the classical movement laws, the dynamics of the distribution cam-gears depreciate rapidly at the increasing of the rotational speed of the shaft. To

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support a high rotational speed it is necessary the synthesis of the cam-profile by new movement laws, and for the new Modules.

A new and original movement law is presented in the pictures number 4, 5 and 6; it allows the increase of the rotational speed to the values: 10000-20000 rpm, in the classical module C presented (Fig. 4). With others modules (B, F) one can obtain 30000-40000 rpm (see Figs. 5, 6).

Dynamic module B (CHANG Z., 2001, 2011) is similar to that of a conventional module C presented in work. But the geometry module B changes, as well as cinematic joined forces, dynamic cinematic (precision), in such a way that particular dynamic module B is changed. In Figure 7 is presented the module B.   Why   we must have high speeds? For construction of an internal combustion modern engine, must be modified the main mechanism, built an engine compact and high speeds. This goal can be achieved with virtually a new module for distribution, such as the module B. On the other hand all module B is the one with which it may be possible to make the timing mechanisms in high yields in operation (PETRESCU F.I., PETRESCU R.V., 2013a,b,c).
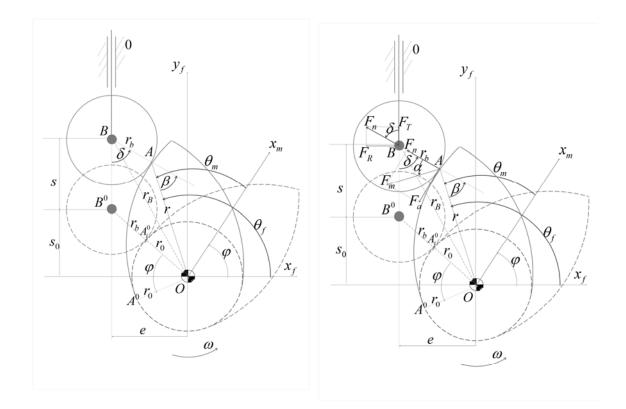

Figure 7. The geometry and the forces at the module B.

High yields for the mechanisms of distribution, as a matter of indifference laws of movement imposed tappet. It is important module type chosen (in our case B) and adjustments valve spring.

The used law is the classical law, cosine law. The synthesis of the cam profile can be made with the relationships (50) when the cam rotates clockwise and with the expressions from the system (51) when the cam rotates counterclockwise (trigonometric).

$$\begin{cases} x_C = (-e - r_b \cdot \sin\delta) \cdot \cos\varphi - [(s_0 + s) - r_b \cdot \cos\delta] \cdot \sin\varphi \\ \\ y_C = [(s_0 + s) - r_b \cdot \cos\delta] \cdot \cos\varphi + (-e - r_b \cdot \sin\delta) \cdot \sin\varphi \end{cases} \tag{50}$$

$$\begin{cases} x_c = (-e + r_b \cdot \sin\delta) \cdot \cos\varphi + (s_0 + s - r_b \cdot \cos\delta) \cdot \sin\varphi \\ \\ y_c = (s_0 + s - r_b \cdot \cos\delta) \cdot \cos\varphi - (-e + r_b \cdot \sin\delta) \cdot \sin\varphi \end{cases} \tag{51}$$

The r_0 (the radius of the base circle of the cam) is 0.013 [m]. The h (the maximum displacement of the tappet) is 0.020 [m]. The angle of lift, φ_u is $\pi/3$ [rad]. The radius of the tappet roll is r_b=0.002 [m]. The misalignment is e=0 [m]. The cosine profile can be seen in the fig. 8.
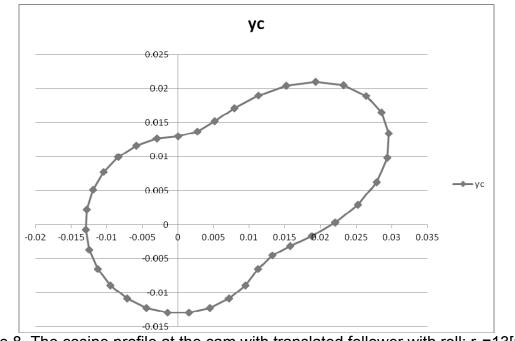

Figure 8. The cosine profile at the cam with translated follower with roll; r_0=13[mm], h=20[mm], φ_u=$\pi/3$[rad], r_b=2[mm], e=0[mm].

The obtained mechanical yield (obtained by integrating the instantaneous efficiency throughout the climb and descent) is 0.39 or η=39%. The dynamic diagram can be seen in the fig. 9 (the dynamic setting are partial normal). Valve spring

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preload 9 cm no longer poses today. Instead, achieve a long arc very hard (k=500000[N/m]), require special technological knowledge.
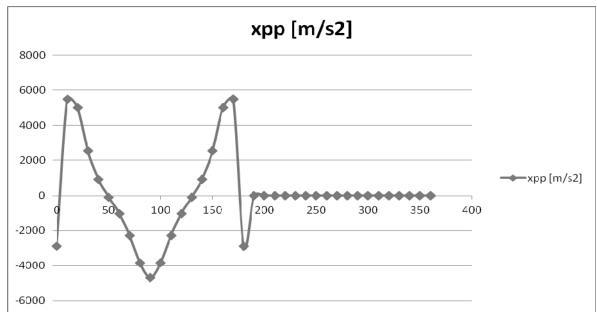

Figure 9. The dynamic diagram at the cosine profile at the cam with translated follower with roll; r_0=13[mm]; h=20[mm]; φ_u=π/3[rad]; r_b=2[mm]; e=0[mm]; n=5500[rpm]; x_0=9[cm]; k=500[kN/m]

The yield distribution mechanisms can be increased even more, reaching even to rival gears (PETRESCU F.I., PETRESCU R.V., 2013a).

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