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A Dynamic Model for Gears

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Abstract: Nearly all models studying the dynamic of gearing with parallel axes are based on classical mechanical models and known which are studying the spinning vibration of shafts gears and determine their own beats and strains of shafts spinning. These classical dynamic models are very useful, but they didn't take in consideration the dynamic events formed between the two teeth in contact (or more pairs of teeth in contact). It's not seen the physiology of the mechanism itself with toothed gears. In this study we do not only account for the impact of teeth (collisions between teeth) but we also consider all the principal dynamic events that are occurring in top gear plane. This article will present an original model that explores the dynamic events originated and taking place in the plane geared couple of the parallel axes geared transmissions. The first measure to be taken for the protection of the environment of pollution of the over a billion motor vehicles with heat engines already in circulation, is to improve the motors and their drives.

Keywords: Gears, Gears Dynamic, Toothed Wheel, Dynamic Synthesis, Forces, Velocities, Powers, Efficiency, Dynamic Coefficient

Introduction

Gears are the most used mechanical systems of transmissions. They have been used since 2000 years ago to today to reduce operating force or rotation speed from the motor to the working machine. A more than 2000 years old Antikythera mechanism, from ancient Egypt, was built with gears, cams and planetary gears. Here, trasmission motion with spur wheels to irrigate crops and worm gears to process cotton were used for the first time. As early as 230 years BC, in the town of Alexandria in Egypt, they had been used the wheels with several levers and equipment rack. Such gears were made and used, since the earliest times, to lift significant heavy masses vessels anchors and to load the catapults used on the battlefields (Mirsayar *et al*., 2017).

Afterward, gears have been utilized for wind and water driven machines to reduce or multiply rotation speed at the pumps from windmills or watermills. In the fifteenth century with the spur wheel of the Engineer Leonardo da Vinci began a new gears generation. He created new kinematics and dynamics gears, introducing new fundamental findings about superposition of independent movements.

Patented after 1882, Benz was the first motor with a sprocket gearing transmission and chain gear (), however, the first patent about gears and gearing wheels with chain was made and introduced in the 1870 by two British men, Starley and Hillman.

After 1912, in Cleveland USA, specialized wheels and gears (worm, cylindrical, conical, with straight teeth, curved or inclined) have began to be industrially produced.

Today, the gears are used everywhere in the industrial's world (in car industries, in energy industries, in electro-technique equipments and electronics etc).

Considering the industrial importance of gears, it is imperative to search for a new gears dynamic model able to account for all important and real physical phenomena occurring in gears.

Blankenship and Singh (1995) have presented a model that describes mesh force transmissibility in a helical gear pair. They have developed a spectral stiffness and a transmissibility matrix, based on a linear theory, which completely represent the forced response of a helical gear pair.

One method for determining the dynamic load between two rotating elastic helical gears was presented by Andersson and Vedmar (2003). The stiffness of the

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gear teeth was calculated using the finite element method that analyzed an elliptic tooth load. To make sure that the new incoming contacts (who are the principal source of excitation properly simulated), the necessary deformation of the gears was determined by utilizing the gears geometry and positions at any time step of the dynamic calculation. This allows the contact to be positioned outside the plane of action.

Elastic couplings have been introduced by shaft finite elements for gear body distortions with time and position-varying properties to tooth deflections (Ajmi and Velex, 2005).

The variability of the dynamic behavior induced by transmission error of a mass production gear pair, which is generated by manufacturing errors, it has been demonstrated in a research paper (Driot and Liaudet, 2006).

Three different methods for solving load distribution of gear teeth for static transmission error are presented (Wink and Serpa, 2008).

A study showing the interaction between the external forcing and the eccentricity deriving from the experimentally fluctuation in gear rattle response with a rig consists of a 1:1 ratio steel spur gear pair and the input gear is being controlled in displacement with the output gear being under no load, is presented by Ottewill *et al*. (2009).

A nonlinear model with dynamic time-varying is proposed for predict the sidebands modulation to the planetary gear sets (Inalpolat and Kahraman, 2010).

The effects of the friction and dynamic backlash on the multi-degree of freedom nonlinear dynamic gear system, which included time varying stiffness, were investigated by Siyu *et al*. (2011).

An original model of planetary gears investigates planet position errors and simulates their contribution to the quasi-static and dynamic load sharing amongst the planets (Gu and Velex, 2012).

A more recent research paper describes a model for the analysis of the forces of contact and of the deformations transmissions with spur gear. Each gear deformation contact point is defined as a combination of a general and a local mode (Fernandez Del Rincon *et al*., 2013).

An original dynamic model describing the conditions to avoid the interferences was presented by Petrescu and Petrescu (2014). All gears: Velocities, forces, powers and mechanical yield are considered and discussed in this study (Mirsayar *et al*., 2017).

A unique relationship to obtain an optimal profile relief with respect the transmission error was presented by Bruyere and Velex (2014).

Dynamic Model

The new presented gears dynamic model starting from the gears position presented on the Fig. 1.

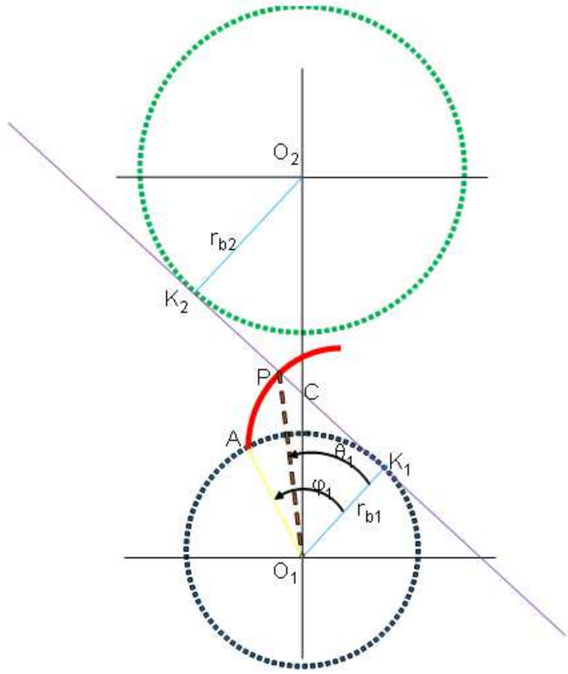


Fig. 1. Angles characteristic at a position of one tooth from the driving wheel, in gearing

A tooth of the lead-gearing wheel 1 in a defined position on the segment of gearing K_1K_2 it is shown in Fig. 1. This is characterized by the two angles θ_1 and φ_1, the first showing the vector O_1P position (the contact vector) in relation to fixed vector O_1K_1 and the second presenting how much is turned the tooth (leading wheel 1) in relation to O_1K_1.

Between the two angles holds the correlation of Equation 1:

$$\phi_1 = tg\theta_1 \quad \theta_1 = arctg\phi_1 \tag{1}$$

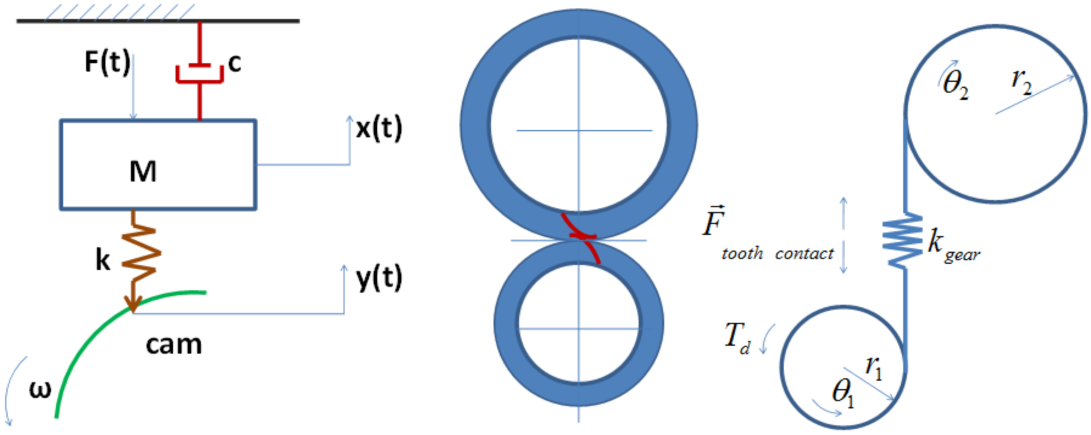
Since φ_1 is the sum of angles θ_1 and γ_1, where the angle γ_1 represents the known function invθ_1 we have:

$$\phi_1 = \theta_1 + \gamma_1 = \theta_1 + inv\theta_1 = \theta_1 + (tg\theta_1 - \theta_1) = tg\theta_1 \tag{2}$$

From the correlation (1) one obtains the Equation 3:

$$\begin{aligned} &\dot{\phi}_1 = (1+tg^2\theta_1)\cdot\dot{\theta}_1 = (1+\phi_1^2)\cdot\dot{\theta}_1 \\ &\dot{\theta}_1 = \frac{\dot{\phi}_1}{1+\phi_1^2} = D_1\cdot\omega_1; D_1 = \frac{1}{1+\phi_1^2} = \frac{1}{1+tg^2\theta_1} \\ &\ddot{\theta}_1 = \dot{D}_1\cdot\omega_1 = D_1^{'}\cdot\omega_1^2; D_1^{'} = \frac{-2\cdot\phi_1}{(1+\phi_1^2)^2} = \frac{-2\cdot tg\theta_1}{(1+tg^2\theta_1)^2} \end{aligned} \tag{3}$$

The dynamic model shown in Fig. 2a and 2b is similar to that of the gears with cam, as geared wheels are derived from those with camshaft; practically the toothed wheel is a cam multiplied, where each tooth is a cam with only the phase up of lifting.



(a) (b)

Fig. 2. (a) The dynamic model. Forces, displacements and elasticity of the system (b) The dynamic model: Forces, displacements and elasticity of the system

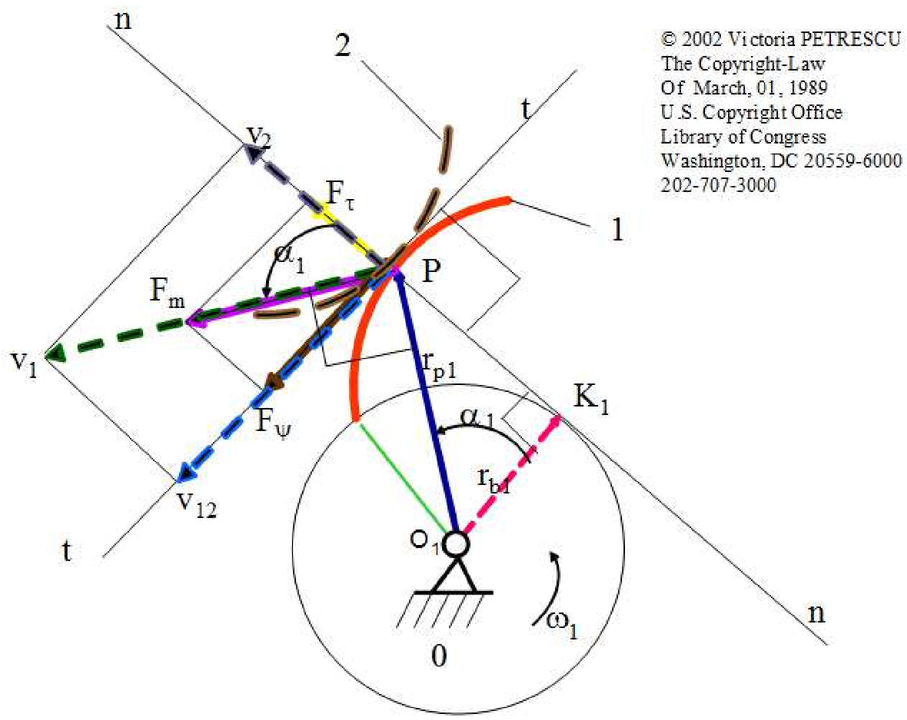


Fig. 3. Forces and velocities characteristic at a position of one tooth from the driving wheel, in gearing

The contact between two teeth represents an interaction between a rotation cam and a follower. Consequently, similarly to models with cams, this interaction will determine precision kinematic (dynamics kinematic) as well as for gears with parallel axes. The leading vector at the driving wheel 1 (in the dynamics kinematic) is the vector O_1P (vector of contact), with the position angle θ_1 and angular velocity ($\dot{\theta}_1$). To the driven wheel 2, one forwards the speed v_2 (see schedule kinematics in Fig. 3 and Equation 4):

$$\begin{cases} v_2 = -v_1 \cdot \cos\theta_1 = -r_{p1} \cdot \dot{\theta}_1 \cdot \cos\theta_1 = \\ = -r_{b1} \cdot \dot{\theta}_1 = -r_{b1} \cdot D_1 \cdot \omega_1 \\ but: \quad v_2 = r_{b2} \cdot \omega_2 => \omega_2 = -\frac{r_{b1}}{r_{b2}} \cdot D_1 \cdot \omega_1 \end{cases} \quad (4)$$

By derivation can be determined the angular acceleration, at the wheel 2 (Equation 5) and integrating the movement of the wheel 2 (expression 6):

$$\varepsilon_2 = -\frac{r_{b1}}{r_{b2}} \cdot D_1^{'} \cdot \omega_1^2 \tag{5}$$

$$\phi_2 = -\frac{r_{b1}}{r_{b2}} \cdot arctg(\phi_1) = -\frac{r_{b1}}{r_{b2}} \cdot \theta_1 \tag{6}$$

Reduced Force (engines and resistance) at the lead wheel 1, is equal to the elastic force (of the couple, where at the lead wheel 2 does not works and a strong technological force; expression 7):

$$\begin{cases} F^* = K \cdot \left(r_{b1} \cdot \phi_1 - r_{b2} \cdot \phi_2\right) \\ = K \cdot \left(r_{b1} \cdot \phi_1 - r_{b2} \cdot \frac{r_{b1}}{r_{b2}} \cdot \theta_1\right) \\ = K \cdot r_{b1} \cdot \left(tg\theta_1 - \theta_1\right) \end{cases} \tag{7}$$

Minus sign was already taken, so φ_2 is replaced just in module (in the expression of 7) and K means the constant elastic of teeth (being in contact; is measured in [N/m]). In dynamics the equation of motion is written with Equation 8:

$$M^* \cdot \ddot{x} + \frac{1}{2} \cdot \frac{dM^*}{dt} \cdot \dot{x} = F^* \tag{8}$$

Reduced mass M^* is determined by the relationship (9):

$$\begin{cases} M^* = \left(J_1 + \frac{1}{i^2} \cdot J_2\right) \cdot \frac{1}{r_{p1}^2} = \\ = \left(J_1 + \frac{1}{i^2} \cdot J_2\right) \cdot \frac{\cos^2 \theta_1}{r_{b1}^2} = \\ = \frac{J_1 + \frac{1}{i^2} \cdot J_2}{r_{b1}^2} \cdot \cos^2 \theta_1 = \\ = C_M \cdot \cos^2 \theta_1; C_M = \frac{J_1 + \frac{1}{i^2} \cdot J_2}{r_{b1}^2} \end{cases} \tag{9}$$

where, J_1 and J_2 represent the moments of inertia (mass, mechanical), reduced to wheel 1, where i is the transmission ratio module (from the wheel 1 to the wheel 2; see the relation 10):

$$i = \frac{r_{b2}}{r_{b1}} = -\frac{\omega_1}{\omega_2} \tag{10}$$

The moving x of the wheel 2 on the gearing segment, writes as Equation 11:

$$\begin{cases} x = r_{b2} \cdot \phi_2 = r_{b2} \cdot -\frac{r_{b1}}{r_{b2}} \cdot arctg\phi_1 \\ = -r_{b1} \cdot arctg\phi_1 = -r_{b1} \cdot \theta_1 \end{cases} \tag{11}$$

The speed (Equation 12) and corresponding accelerations (Equation 13) can be written:

$$\dot{x} = -r_{b1} \cdot \dot{\theta}_1 = -r_{b1} \cdot \frac{1}{1 + tg^2\theta_1} \cdot \omega_1 \tag{12}$$

$$\begin{cases} \ddot{x} = -r_{b1} \cdot \ddot{\theta}_1 = -r_{b1} \cdot \frac{-2 \cdot tg\theta_1}{\left(1 + tg^2\theta_1\right)^2} \cdot \omega_1^2 \\ = 2 \cdot r_{b1} \cdot \frac{tg\theta_1}{\left(1 + tg^2\theta_1\right)^2} \cdot \omega_1^2 \end{cases} \tag{13}$$

Then it derives the mass (reduced) and one obtains the relationship (14):

$$\frac{dM^*}{dt} = -2 \cdot C_M \cdot \frac{\cos\theta_1 \cdot \sin\theta_1}{1 + tg^2\theta_1} \cdot \omega_1 \tag{14}$$

The equation of motion (8) takes now the form (15), which can be arranged and in form (16):

$$3 \cdot C_M \cdot r_{b1} \cdot \frac{tg\theta_1}{\left(1 + tg^2\theta_1\right)^3} \cdot \omega_1^2 = K \cdot r_{b1} \cdot \left(tg\theta_1 - \theta_1\right) \tag{15}$$

$$\begin{cases} \theta_1^d \equiv \theta_1 = tg\theta_1 - \frac{3 \cdot C_M \cdot tg\theta_1 \cdot \omega_1^2}{K \cdot \left(1 + tg^2\theta_1\right)^3} \\ = tg\theta_1 \cdot \left[1 - \frac{3 \cdot \left(J_1 + \frac{r_{b1}^2}{r_{b2}^2} \cdot J_2\right) \cdot \omega_1^2}{r_{b1}^2 \cdot K \cdot \left(1 + tg^2\theta_1\right)^3}\right] \end{cases} \tag{16}$$

Relation (16) represents the solution for the motion equation of the superior couple; at an rotation angle of the wheel 1, φ_1, known, which is corresponding a pressure angle θ_1 known, the relation (16) produces an angle of pressure (dynamic), θ_1^d .

In terms of the constant of elasticity (for the teeth in contact), K is large enough only if the radius of the base circle of the wheel 1 don't decrease too much (z_1 needs to be greater than 15-20), for the normal and even higher (but not too large) velocities, the ratio in parenthesis of expression 16 keeps under the value 1 and the expression 16 can be engineering estimated to the natural short form (17):

$$\theta_1^d = tg\theta_1 = \phi_1^c \equiv \phi_1 \tag{17}$$

The Angular (Dynamic) Speed at the Lead Wheel 1

Now it may determine the temporary angular speed of the lead wheel 1 (expression 19); requirements and the intermediate expression (18):

$$\begin{cases} \frac{\Delta\omega_1}{\omega_m} = \frac{\Delta\phi_1}{\phi_1} \Rightarrow \Delta\omega_1 = \frac{\Delta\phi_1}{\phi_1}\cdot\omega_m = \frac{\phi_1^d - \phi_1}{\phi_1}\cdot\omega_m \\ = \frac{tg(\theta_1^d) - \phi_1}{\phi_1}\cdot\omega_m = \frac{tg(\phi_1) - \phi_1}{\phi_1}\cdot\omega_m = \frac{inv\phi_1}{\phi_1}\cdot\omega_m \end{cases} \quad (18)$$

$$\begin{cases} \omega_1 = \omega_m + \Delta\omega_1 = \left(1 + \frac{inv\phi_1}{\phi_1}\right)\cdot\omega_m \\ = \frac{tg(\phi_1)}{\phi_1}\cdot\omega_m = \frac{tg(tg\theta_1)}{tg\theta_1}\cdot\omega_m = R_{d1}\cdot\omega_m \end{cases} \quad (19)$$

It defines the dynamic coefficient, R_{d1}, as the ratio between φ_1 tangent and φ_1 angle, or the ratio: $\frac{tg(tg\theta_1)}{tg\theta_1}$, expression 20:

$$R_{d1} = \frac{tg(tg\theta_1)}{tg\theta_1} \quad (20)$$

Dynamic synthesis of gearing with axes parallel can be realized with the help of expressions (20). The necessity to have a dynamic factor as low (close to the value 1), asks the limiting of the maximum pressure angle, θ_{1M} and the normal angle α_0 and increasing the minimum number of teeth of the leading wheel 1 (z_{1min}).

Result and Discussion (Dynamics of the Wheel 2)

The first measure to be taken for the protection of the environment of pollution of the over a billion motor vehicles with heat engines already in circulation, is to improve the motors and their drives.

At this stage we may determine the temporary (dynamic) angular velocity at the led wheel 2 (relationship 28) and the angular values of displacement, velocity, acceleration, in situations: Classical kinematic, precision kinematic and dynamic (with: c = kinematic, cp = precision kinematic, d = dynamic; see the relations: 21-31):

$$\phi_2^c = -\frac{r_{b1}}{r_{b2}}\cdot\phi_1 \quad (21)$$

$$\omega_2^c = -\frac{r_{b1}}{r_{b2}}\cdot\omega_1 \quad (22)$$

$$\varepsilon_2^c = -\frac{r_{b1}}{r_{b2}}\cdot\varepsilon_1 = 0 \quad (23)$$

$$\phi_2^{cp} = -\frac{r_{b1}}{r_{b2}}\cdot arctg\phi_1 = -\frac{r_{b1}}{r_{b2}}\cdot\theta_1 \quad (24)$$

$$\omega_2^{cp} = -\frac{r_{b1}}{r_{b2}}\cdot\frac{1}{1+\phi_1^2}\cdot\omega_1 = -\frac{r_{b1}}{r_{b2}}\cdot\frac{1}{1+tg^2\theta_1}\cdot\omega_1 \quad (25)$$

$$\varepsilon_2^{cp} = -\frac{r_{b1}}{r_{b2}}\cdot\frac{-2\cdot\phi_1}{(1+\phi_1^2)^2}\cdot\omega_1^2 = -\frac{r_{b1}}{r_{b2}}\cdot\frac{-2\cdot tg\theta_1}{(1+tg^2\theta_1)^2}\cdot\omega_1^2 \quad (26)$$

Dynamics of the conducted wheel 2 can be calculated with expressions (27-31):

$$\phi_2^d = -\frac{r_{b1}}{r_{b2}}\cdot\int\frac{tg\phi_1}{\phi_1+\phi_1^3}d\phi_1 \quad (27)$$

$$\begin{cases} \omega_2^d = -\frac{r_{b1}}{r_{b2}}\cdot\frac{1}{1+\phi_1^2}\cdot\frac{tg\phi_1}{\phi_1}\cdot\omega_1 = \\ = -\frac{r_{b1}}{r_{b2}}\cdot\frac{1}{1+tg^2\theta_1}\cdot\frac{tg(tg\theta_1)}{tg\theta_1}\cdot\omega_1 \end{cases} \quad (28)$$

$$\varepsilon_2^d = -\frac{r_{b1}}{r_{b2}}\cdot\frac{(1+tg^2\phi_1)\cdot(\phi_1+\phi_1^3) - tg\phi_1\cdot(1+3\cdot\phi_1^2)}{(\phi_1+\phi_1^3)^2}\cdot\omega_1^2 \quad (29)$$

With:

$$\begin{cases} \phi_{1m} = tg\theta_{1m} = \\ = \frac{(z_1+z_2)\cdot\sin\alpha_0 - \sqrt{z_2^2\cdot\sin^2\alpha_0 + 4\cdot z_2 + 4}}{z_1\cdot\cos\alpha_0} \end{cases} \quad (30)$$

$$\phi_{1M} = tg\theta_{1M} = \frac{\sqrt{z_1^2\cdot\sin^2\alpha_0 + 4\cdot z_1 + 4}}{z_1\cdot\cos\alpha_0} \quad (31)$$

Now may be defined to wheel 2 the coefficient Rd2 (dynamic), (see the expressions 28 and 32):

$$R_{d2} = \frac{1}{1+\phi_1^2}\cdot\frac{tg\phi_1}{\phi_1} = \frac{1}{1+tg^2\theta_1}\cdot\frac{tg(tg\theta_1)}{tg\theta_1} \quad (32)$$

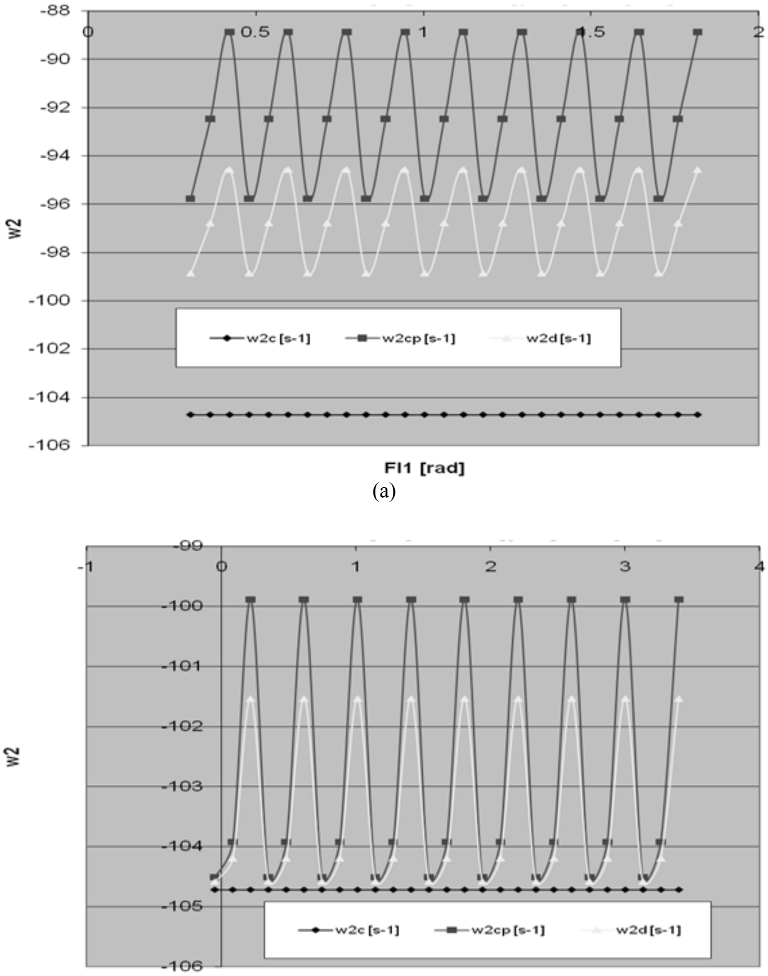
Conclusion

The first measure to be taken for the protection of the environment of pollution of the over a billion motor vehicles with heat engines already in circulation, is to improve the motors and their drives.

Representation of angular velocity, ω_2, depending on the angle φ_1, for r_{b1} and r_{b2} known (z_1, z_2, m and α_0 imposed), for a certain value of the angular speed input constant (imposed by the speed of the shaft which is mounted wheel leading 1), may be seen in the Fig. 4a and 4b.

Observe the appearance of vibration at the dynamic angular velocity, ω_2.

Start with equal rays and 20 degrees all for, α_0 (Fig. 4a) and stay on the chart last rays equal and α_0 reduced to 5 degrees (Fig. 4b). This method may be used in place of the cumbersome calculations, which may determine the torsional vibrations at the gears.



(a)

(b)

Fig. 4. Dynamics of wheel 2; ω_2 cinematic, ω_2 in precision cinematic, ω_2 dynamic

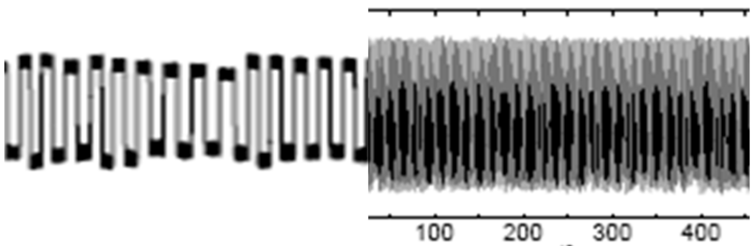


Fig. 5. Dynamics of wheel 2; ω_2 dynamic (vibrations obtained experimental in two different ways)

In Fig. 5 it may see the experimental vibration [1, 2]. Presented method is simply, directly and original.

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Author's Contributions

All the authors contributed equally to prepare, develop and carry out this work.

Ethics

This article is original. Author declares that are not ethical issues that may arise after the publication of this manuscript.

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