Contributions on the use of dynamic absorbers in a vehicle engine

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Abstract: - In the study presented in the paper, a theoretical calculation will be made regarding engine vibration dampers. These will be validated with the experimental model. These validations bring results in static and dynamic balance through the reduction of torsional vibrations in the engine. Torsional vibrations are caused by various factors as well as engine operating conditions. The paper focuses on the reduction of torsional vibrations from a theoretical and experimental point of view. Therefore, in this study, a dynamic absorber was considered which was integrated into the system considered by the paper and was analyzed both from the point of view of its static and dynamic behavior.

Key-Words: - Dynamic absorbers, experimental engine stand, R.M.S. vibration measurements, torsional vibrations, engine measurements points.

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1 Introduction

In most cases where engine vibrations occur, vibration reduction or elimination methods such as vibration isolation and/or the use of a dynamic vibration damper are used. Other studies regarding these balancers propose improving the shape and construction of cars through the structures on which these shock absorbers are mounted. This fact will lead to their redesign, and this is an expensive option. In this regard, there are methods to reduce vibrations by using insulating materials, but this is ineffective in this case because only low and medium amplitude vibrations (eg 1-7mm) are eliminated. For balancing the components of a structure to vibrations with large amplitudes above 20 mm, only the use of a dynamic absorber can effectively solve the problem of their reduction. For this reason, the study of the paper focuses on these aspects. The analyzed structure consists of a crankshaft with moving crew components supported by it and contains a primary system that is unloaded with an auxiliary component known as a dynamic absorber.

By adding the dynamic absorber, the number of degrees of freedom of the primary system increases by one, and, as a result, the shape of the resonance curve changes.

If the parameters of the dynamic absorber are chosen correctly, the resonance curve of the primary system-dynamic absorber assembly has a pronounced minimum in the frequency range where only the primary system has a maximum, [1]. This results in a significant damping of the vibrations of the primary system in the region near its resonance, [1].

Usually, the dynamic absorber is superior to any isolation system because it can be easily tuned to operate in the area of the frequencies that need elimination. The use of the dynamic absorber is recommended when the natural frequency of the primary system is close to the frequency of the disturbing forces. At its point of application, the dynamic absorber acts on the primary mass with a force of the same frequency as the disturbing force, but in the opposite direction. That is why it can be used to reduce the unbalanced forces that appear in the operation of various machine parts and they balance it.

The analyzed mechanical system includes a flywheel, and the mode of absorption of engine vibrations is checked by incorporating a dynamic absorber of small mass at the end of the shaft, as shown in Figure 2.

The dynamic absorber can be designed together with the structure whose vibrations need to be eliminated, or it can be added later. In addition, it may act on the entire structure or it can be attached to a certain element, whose operation produces the overall vibrations of the structure (like in the paper).

The effectiveness of the dynamic absorber depends on its parameters: mass, elastic constant, and damping constant. Optimal results are achieved when using large masses attached to the system. However, the use of masses exceeding 25% of the primary mass presents serious inconveniences from the point of view of assembly and dimensions. This aspect leads to special attention regarding the second option, namely the damping of the system. The research undertaken to develop dynamic absorbers with a small mass, but high efficiency has led to the improvement of the damping optimization methods, [1], [2].

In this study, our focus is on the torsional vibrations generated in the motor shaft. Therefore, it has been studied that the risk of increasing torsional vibrations, [1], [3], above a certain value is minimal. This risk appears in the functionality of the motors, namely, in the shaft system, when the torques acting on it show important periodic variations in a relatively short time, [4], [5].

In the specialized literature, [2], [6], [7], [8], [9] and [10], the presence of a pendulum element attached to the shaft is discussed. This element acts as a dynamic absorber, aiming to balance the movement of the shaft and reduce engine vibrations. However, this element, although in theory it behaves well, in practice takes up a significant amount of space on the motor shaft or flywheel, not having enough room for movement, leading to an unwanted increase in the dimensions of the motor frame.

Unlike the specialized literature presented regarding dynamic absorbents, the dynamic absorbent presented in the paper is of very small size (m5 = 150g). The simulation of its movement under the effect of vibrations will be simulated in Matlab.

Therefore, the paper presents an experimental study to validate the theoretical simulation. In this study, a dynamic absorber is constructed as a 150 g mass washer, mounted with clearance on a bolt directly on the engine flywheel. This design ensures that the space occupied by the dynamic absorber is almost negligible, effectively reducing vibrations.

Engines with fewer cylinders [6], [8], [9], [11] and [12] are known to transmit higher vibrations to engine attached components and these are also influenced by road infrastructure. on which the vehicle drives and the working conditions in which it operates, etc.

An internal combustion engine was studied [13], in which the crankshaft is subjected to the action of torque moments that vary according to the variation of inertial forces.

The paper wants to bring forward the importance of using a dynamic shock absorber mounted on the flywheel of the four-stroke engine, both from the point of view of its shape and dimensions, as well as from the point of view of the purpose it must fulfil, namely the reduction of the given torsional vibrations engine [6], [7], [8], [9] and [14].

These vibrations correspond to the masses in translational movement and parallel plane movement that the mobile crews of the shaft behave. All are carried out in a cycle of the thermal process taking place in the engine cylinders (three cylinders).

The lack of a dynamic absorber in an engine will make it an unbalanced engine, and the addition of the absorber in the engine structure will lead to its static and dynamic balancing, which is desirable and is the object of this paper.

Since no study can take into account all the factors that influence the whole, some simplifying hypotheses will be taken into account, such as:

- The equivalent system that will be reached is, in fact, the initial mechanical system, but in which it is considered that the shaft is elastic, rectilinear, of negligible mass and loaded with reduced circular masses (m_1, m_2, m_3, m_4, and m_5), [7], [8], and [9].
- The elastic constant of the shaft is influenced by two reduced and consecutive masses, as well as by their axial mechanical moments relative to the axis of symmetry of the shaft (Fig.2 and Fig.3). The mechanical energy of the real system originating from the vibrations of the crankshaft and the attached mobile crews must be equal to that of the low-mass system.
- The theoretical analysis considered that the angular speed of the axis is constant, in reality, it has variations [1].
- A piston and internal combustion engine cannot achieve a uniform movement of the shaft [1], [6] for the reasons:

• One of the engine piston strokes is an active stroke, in which the engine accumulates kinetic energy;

• The second reason is given by the fact that the kinematics of the connecting rod-crank mechanism of the piston and its strokes have an alternative movement and variable speed.

To be able to experience this study and to put into value the simplifying hypotheses, an engine will be used as a test stand, a spark ignition engine with three cylinders and in line (Fig.1).

Fig. 1: Engine stand of experimental studies [2].

Research in the field [1], [2], [3], [6], [7], [8], [9], [10], [11], [14], [15] and [16] presents solutions regarding combating torsional vibrations of internal combustion engines such as:

- Changing the shape of the crankshaft;

-The amplitudes of the harmonics due to the torsional vibrations corresponding to the different engine cylinders. These can be reduced in the in-line engine, only for a limited range of critical revolutions and only by changing the sequence of the shaft levers, as well as the ignition order in the engine cylinders.

In comparison to these, the paper uses to a small extent only the first option given by the researchers, not by modifying the crankshaft, but by adding an auxiliary mass called a dynamic absorber. Previous theoretical studies [14], showed that this additional mass does not need to weigh more than 150 g and was added to the end of the flywheel shaft at a small distance/radius, to gain space in the engine housing.

Next, the paper will present a theoretical case, and validated simulation study in Matlab.

2 Mathematical modeling

It will consider the system of four masses denoted reduced masses: m_1, m_2, m_3, and m_4, and represented in Figure 2. These are connected by the reduced section of the shaft and have the equivalent elastic constants denoted by u_{12}, u_{23}, u_{34}, and u_{45}. The reduced masses are corresponding to the system (crankshaft) and the mobile crews including the flywheel.

The dynamic absorber has the characteristic dimensions l_1 and l_2 and the mass M is attached to the last reduced mass m_4. Harmonic of order (x) acting on the mechanical system will be applied to the reduced masses m_1, m_2, and m_3 in the form of an excitation force noted $P_x\cos(\Omega_x t-\varphi)$, where r – is the radius of the reduction circle from the motor shaft to the reduced mass (respectively, half of the stroke of the piston) (Fig.2).

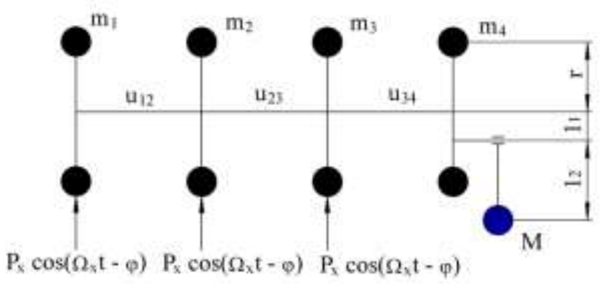

Fig. 2: Mechanical system.

It seeks to simplify the studied system by its modeling with an equivalent system (Fig. 3) whose solutions can be found more easily. Thus, the dynamic absorber is replaced by a mechanical system consisting of the reduced mass m_4 and the reduced section of the shaft having the elastic constant u_{34} (the constant u_{34} imposes on the system the condition of being dynamically equivalent to the dynamic absorber), respectively the application of the reduced mass m_4 with the same moment as the dynamic absorber.

It is mentioned that the reduced masses m_1-m_5 from the theoretical modeling simulate the values of the experimental model. In addition, the elasticity constants (u_{ij}) i = 1-4, j = 2-5 are close to those of the engine and are in accordance with the values given by specialized books [16].

So that, in the work, the value of the calculated reduced masses and the other initial dimensions are:

$$
\begin{array}{ll}
m_1 = 2661.19 \text{ g} & m_2 = 2655.46 \text{ g} \\
m_3 = 2658.40 \text{ g} & m_4 = 5600 \text{ g} \\
M = 150 \text{ g} & \\
P_x = 10705 \text{ N} & \\
l_1 = 40 \text{ mm} & l_2 = 10 \text{ mm} \\
r = 40.7 \text{ mm} & \\
x = 0,5 \div 6 & \omega_0 = 177.93 \text{ rad/s} \\
u_{12} = u_{23} = u_{34} = 495204.67 \text{ N/mm} &
\end{array} \tag{1}
$$

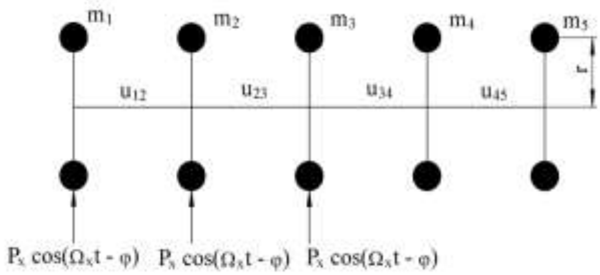

Fig. 3: Equivalent mechanical system.

In the case of an internal combustion engine, the crankshaft is subjected to the action of torques. These torques, within a cycle of the thermal process that takes place in the engine cylinders, vary greatly. The change in their values is due to the variation of the inertial forces corresponding to the masses of the mobile crews of the crankshaft in translational and parallel plane movements. To be able to evaluate the torsional vibrations of such a complex elastic system, it is necessary that, beforehand, it is transformed into an equivalent dynamic system, namely formed in a rectilinear elastic shaft of negligible mass, loaded with reduced circular masses [1], [2].

The elastic constants of the crankshaft elements depend on the four reduced masses. Also, the axial mechanical moments of the reduced masses must be chosen so that the kinetic energy and the potential energy of the shaft of the real, vibrating system (consisting of the crankshaft with the parts fixed on it and the related mobile crews) are equal with a kinetic energy and a potential energy equivalent to the reduced system.

System (1) follows the mechanical model in figure 3 and is written in matrix form as follows:

$$\begin{bmatrix} m_1 & 0 & 0 & 0 & 0 \\ 0 & m_2 & 0 & 0 & 0 \\ 0 & 0 & m_3 & 0 & 0 \\ 0 & 0 & 0 & m_4 & 0 \\ 0 & 0 & 0 & 0 & m_5 \end{bmatrix} \cdot \begin{Bmatrix} \ddot{a}_1 \\ \ddot{a}_2 \\ \ddot{a}_3 \\ \ddot{a}_4 \\ \ddot{a}_5 \end{Bmatrix} + \\ + \begin{bmatrix} u_{12} & -u_{12} & 0 & 0 & 0 \\ -u_{12} & u_{12}+u_{23} & -u_{23} & 0 & 0 \\ 0 & -u_{23} & u_{23}+u_{34} & -u_{34} & 0 \\ 0 & 0 & -u_{34} & u_{34}+u_{45} & -u_{45} \\ 0 & 0 & 0 & -u_{45} & u_{45} \end{bmatrix} \cdot \begin{Bmatrix} a_1 \\ a_2 \\ a_3 \\ a_4 \\ a_5 \end{Bmatrix} = \\ \begin{Bmatrix} P_x cos(\Omega_x t-\varphi) \\ P_x cos(\Omega_x t-\varphi) \\ P_x cos(\Omega_x t-\varphi) \\ 0 \\ 0 \end{Bmatrix} \tag{2}$$

The elongations of the torsional vibrations produced by the five reduced masses will be noted according to the amplitudes A_i of the same torsional vibrations:

$$a_i = A_i \cos(\Omega_x t-\varphi) \quad \text{where} \quad i = 1 \div 5 \tag{3}$$

The expressions (3) and their second-order derivatives for the torsional vibration amplitudes are introduced in the differential equation system, and by simplification, with the trigonometric function $cos(\Omega_x t-\varphi)$, it obtains the algebraic equations system (3).

$$\begin{aligned}
&\left(\Omega_x^2-\frac{u_{12}}{m_1}\right)A_1+\frac{u_{12}}{m_1}A_2=-\frac{P_x}{m_1}\\
&\frac{u_{12}}{m_2}A_1+\left(\Omega_x^2-\frac{u_{12}}{m_2}-\frac{u_{23}}{m_2}\right)A_2+\frac{u_{23}}{m_2}A_3=-\frac{P_x}{m_2}\\
&\frac{u_{23}}{m_3}A_2+\left(\Omega_x^2-\frac{u_{23}}{m_3}-\frac{u_{34}}{m_3}\right)A_3+\frac{u_{34}}{m_3}A_4=-\frac{P_x}{m_3}\\
&\frac{u_{34}}{m_4}A_3+\left(\Omega_x^2-\frac{u_{34}}{m_4}-\frac{u_{45}}{m_4}\right)A_4+\frac{u_{45}}{m_4}A_5=0\\
&\frac{u_{45}}{m_5}A_4+\left(\Omega_x^2-\frac{u_{45}}{m_5}\right)A_5=0
\end{aligned} \tag{4}$$

The system (3) can be written in the matrix form:

$$\begin{bmatrix} \Omega_x^2-\frac{u_{12}}{m_1} & \frac{u_{12}}{m_1} & 0 & 0 & 0 \\ \frac{u_{12}}{m_2} & \Omega_x^2-\frac{u_{12}}{m_2}-\frac{u_{23}}{m_2} & \frac{u_{23}}{m_2} & 0 & 0 \\ 0 & \frac{u_{23}}{m_3} & \Omega_x^2-\frac{u_{23}}{m_3} & \frac{u_{34}}{m_3} & 0 \\ 0 & 0 & \frac{u_{34}}{m_4} & \Omega_x^2-\frac{u_{34}}{m_4}-\frac{u_{45}}{m_4} & \frac{u_{45}}{m_4} \\ 0 & 0 & 0 & \frac{u_{45}}{m_5} & \Omega_x^2-\frac{u_{45}}{m_5} \end{bmatrix} \cdot \begin{Bmatrix} A_1 \\ A_2 \\ A_3 \\ A_4 \\ A_5 \end{Bmatrix} = \begin{Bmatrix} -\frac{P_x}{m_1} \\ -\frac{P_x}{m_2} \\ -\frac{P_x}{m_3} \\ 0 \\ 0 \end{Bmatrix} \tag{5}$$

For the mechanical system (Fig. 2) formed of reduced mass m_5 (M – absorber mass) and the reduced crankshaft with the elastic constant u_{45} to be dynamically equivalent with the initial dynamic absorber [15], [16], it is necessary and sufficient that the next relations to be fulfilled:

$$m_5 = M \cdot \frac{(l_1+l_2)^2}{r^2} \tag{6}$$

The parameters l_1, l_2, and r have the characteristic values according to the model in Figure 2.

$$u_{45} = M \cdot \left[\frac{(l_1+l_2)^2}{r^2} \cdot \frac{l_1}{l_2} \cdot \omega_0^2\right] \tag{7}$$

where - ω_0 [rad/s] represents the constant angular velocity with which rotates the reduced shaft.

u_{45} – represents the elasticity constant corresponding to the dynamic absorber (fig.3), its value depends on the constructive parameters of the chosen system. The value of this constant influences the A_4 amplitude of the system (respectively, the last mass to which the dynamic absorber is attached).

Results that:

$$\frac{u_{45}}{m_4} = \frac{1}{m_4} \cdot M \cdot \frac{(l_1+l_2)^2}{r^2} \cdot \frac{l_1}{l_2} \cdot \omega_0^2 \tag{8}$$

$$\frac{u_{45}}{m_5} = \frac{l_1}{l_2} \cdot \omega_0^2 \tag{9}$$

The elasticity constant u_{45} depends both on the mass of the flywheel m_4 and on the mass m_5 (mass of the equivalent system), but also on the constructive parameters of the system shown in Figures 2 and 3. Ratios (8) and (9) are needed to calculate the solutions of the matrix system (5).

It is mentioned that the harmonic pulsation (Ω) of order x has the expression:

$$\Omega_x = x \cdot \omega_0 \tag{10}$$

where, (x) represents the order of the harmonic, i.e. the number of oscillations that the harmonic executes in the time interval T_0, during which the shaft executes a complete rotation of 2π [rad].

If the dynamic absorber is dimensioned in such a way that:

$$\frac{l_1}{l_2} = x^2 \tag{11}$$

in addition, the conditions (8) and (9) that must be fulfilled by the reduced mechanical system (continuing the reduced mass m_5 and the reduced crankshaft with the elastic constant u_{45}) are taken into account. To be dynamically equivalent (balanced) with the initial dynamic absorber [1], [2] the expressions of the five determinants Δ, Δ_1, Δ_2, Δ_3, and Δ_4 of it become:

$$\Delta = \begin{vmatrix} \Omega_x^2 - \frac{u_{12}}{m_1} & \frac{u_{12}}{m_1} & 0 & 0 & 0 \\ \frac{u_{12}}{m_2} & \Omega_x^2 - \frac{u_{12}}{m_2} - \frac{u_{23}}{m_2} & \frac{u_{23}}{m_2} & 0 & 0 \\ 0 & \frac{u_{23}}{m_3} & \Omega_x^2 - \frac{u_{23}}{m_3} - \frac{u_{34}}{m_3} & \frac{u_{34}}{m_3} & 0 \\ 0 & 0 & \frac{u_{34}}{m_4} & \Omega_x^2 - \frac{u_{34}}{m_4} - \frac{u_{45}}{m_4} & \frac{u_{45}}{m_4} \\ 0 & 0 & 0 & \frac{u_{45}}{m_5} & \Omega_x^2 - \frac{u_{45}}{m_5} \end{vmatrix} \tag{12}$$

$$\Delta = -\frac{u_{34}}{m_3}\frac{u_{34}}{m_4}\left[\left(x^2\omega_0^2 - \frac{u_{12}}{m_1}\right)\left(x^2\omega_0^2 - \frac{u_{12}}{m_2} - \frac{u_{23}}{m_2}\right) - \frac{u_{12}}{m_1}\frac{u_{12}}{m_2}\right] - \frac{1}{m_4} m \frac{(l_1+l_2)^2}{r^2}\left(\frac{l_1}{l_2}\omega_0^2\right)^2 \left\{\left(x^2\omega_0^2 - \frac{u_{23}}{m_3} - \frac{u_{34}}{m_3}\right) \left[\left(x^2\omega_0^2 - \frac{u_{12}}{m_1}\right)\left(x^2\omega_0^2 - \frac{u_{12}}{m_2} - \frac{u_{23}}{m_2}\right) - \frac{u_{12}}{m_1}\frac{u_{12}}{m_2}\right] - \frac{u_{23}}{m_2}\frac{u_{23}}{m_3}\left(x^2\omega_0^2 - \frac{u_{12}}{m_1}\right)\right\} \tag{13}$$

$$\Delta_1 = \frac{P_x}{m_1}\frac{1}{m_4} m \frac{(l_1+l_2)^2}{r^2}\left(\frac{l_1}{l_2}\omega_0^2\right)^2 \left[\left(x^2\omega_0^2 - \frac{u_{12}}{m_2} - \frac{u_{23}}{m_2}\right) \left(x^2\omega_0^2 - \frac{u_{23}}{m_3} - \frac{u_{34}}{m_3}\right) - \frac{u_{23}}{m_2}\frac{u_{23}}{m_3}\right] - \frac{P_x}{m_2}\frac{1}{m_4} m \frac{(l_1+l_2)^2}{r^2} \left(\frac{l_1}{l_2}\omega_0^2\right)^2 \frac{u_{12}}{m_1}\left(x^2\omega_0^2 - \frac{u_{23}}{m_3} - \frac{u_{34}}{m_3}\right) + \frac{P_x}{m_3}\frac{u_{12}}{m_1}\frac{u_{23}}{m_2}\frac{1}{m_4} m \frac{(l_1+l_2)^2}{r^2}\left(\frac{l_1}{l_2}\omega_0^2\right)^2 \tag{14}$$

$$\Delta_2 = \frac{1}{m_4} m \frac{(l_1+l_2)^2}{r^2}\left(\frac{l_1}{l_2}\omega_0^2\right)^2 \left[\frac{P_x}{m_1}\frac{u_{12}}{m_2}\left(x^2\omega_0^2 - \frac{u_{23}}{m_3} - \frac{u_{34}}{m_3}\right) - \frac{P_x}{m_2}\left(x^2\omega_0^2 - \frac{u_{12}}{m_1}\right)\left(x^2\omega_0^2 - \frac{u_{23}}{m_3} - \frac{u_{34}}{m_3}\right) + \frac{P_x}{m_3}\frac{u_{23}}{m_2}\left(x^2\omega_0^2 - \frac{u_{12}}{m_1}\right)\right] \tag{15}$$

$$\Delta_3 = \frac{1}{m_4} m \frac{(l_1+l_2)^2}{r^2}\left(\frac{l_1}{l_2}\omega_0^2\right)^2 \left\{-\frac{P_x}{m_1}\frac{u_{12}}{m_2}\frac{u_{23}}{m_3} + \frac{P_x}{m_2}\frac{u_{23}}{m_3}\left(x^2\omega_0^2 - \frac{u_{12}}{m_1}\right) - \frac{P_x}{m_3}\left[\left(x^2\omega_0^2 - \frac{u_{12}}{m_1}\right)\left(x^2\omega_0^2 - \frac{u_{12}}{m_2} - \frac{u_{23}}{m_2}\right) - \frac{u_{12}}{m_1}\frac{u_{12}}{m_2}\right]\right\} \tag{16}$$

$$\Delta_4 = 0 \tag{17}$$

Δ_5 is not important, so it is not represented in the paper.

The amplitudes of the torsion vibrations executed by the four reduced masses A_i (in this case i =1~4) are provided by the expressions:

$$A_i = \frac{\Delta_i}{\Delta} \quad \text{(i=1~4)} \tag{18}$$

The vibration amplitudes of the reduced masses m_i (i =1~ 4) have the expressions (18). It follows from these expressions after the replacement of the relations of the corresponding determinants that:

$$A_1 \neq 0 \qquad A_2 \neq 0 \qquad A_3 \neq 0 \qquad A_4 = 0 \tag{19}$$

The amplitudes A_1, A_2, A_3, and A_4 refer to the vibrations of the reduced masses m_1, m_2, m_3, and m_4 (these are a result of applying the method of reduction of masses and lengths on the crankshaft). The amplitude A_5 refers to the torsion vibrations executed by the fictive reduced mass m_5 that was introduced on the reduced crankshaft with elasticity constant u_{45}, to propose a study the effect of the dynamic absorber on the vibrations of the crankshaft represented of the reduced mechanical mass.

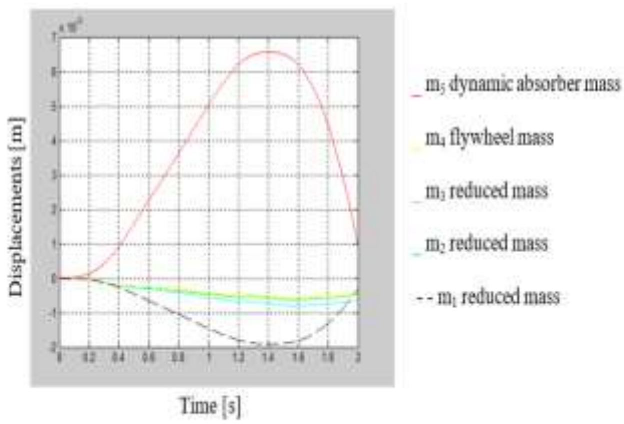

Fig. 4: The mobile crews' displacements for the reduced masses.

It is aimed to obtain amplitudes of the steering wheel (A_4) as close as possible to zero value. The theoretical solutions from relation (19) (of system 5) are validated by the simulated model given in Figure 4 and are in accordance with the purpose of the paper.

The amplitude A_4 simulated in Matlab according to Figure 4, validates the theoretical calculation, its value (yellow color) approaches much to zero.

Theoretical validations (Fig. 4) including simulation will be compared with the measurements performed on the engine and presented in the following.

3 Experimental research studies

3.1 Measuremet conditions

In the study of the paper, a three-cylinder engine was chosen to analyze its dynamics, simultaneously with the analysis of the additional fuel consumption, which occurs in unbalanced engines. It is considered in this case that the ignition in the cylinders is in the order 1-3-2. If we want to explain this fact, the numbering of the cylinders according to the standard in force, [17], starts with the coupling flange between the crankshaft and the drive shaft. What appears in addition, with this type of engine, is an angular gap of 120° between two ignitions.

- We mention that the analysis of a vehicle with reduced power 50-70 CP implies less pollution, but accelerations and decelerations, especially when they involve torsional vibrations of the engine, will lead to an increase in fuel consumption.
- It is mentioned that the material from which the motor shaft is made can influence the reduction or transmission of torsional vibrations.
- The experimental stand consists of an engine with three cylinders in line. The engine will be supported at five points (two on each side and one on the engine housing) with the help of elastic plates that fix it to the vehicle housing (Fig. 5), [3].

The following constructive and functional characteristics of the engine were studied:

Engine 1200 – normal standard; Number of cylinders 3 in line; Piston stroke 77 mm, Cylinder capacity of engine is 1198 cmc (centimetres cube capacity); Maximum power 64 CP; Maximum couple of 4000 RPM.

Fig. 5: Points of engine support [2].

The use of a dynamic absorber in a structure, in the present case on the engine flywheel, contributed to the stability of the system both from a static point of view, but especially from a dynamic point of view and presents as validation of the results from the vibration measurements that focused on these two types of balancing (static and dynamic). We mention that before starting the measurements, the engine was calibrated from a dynamic point of view with the help of a balancing program called IMPAQ and with BENSTONE equipment. In the following, it will be presented what the method of dynamic balancing of the four-stroke engine with three cylinders consisted of:

- Two planes called P1 and P2 were used for measurements, setting the type of sensors, and setting the balancing clearance according to ISO 1940. A corrective mass was chosen for balancing in the P1 plane, and another mass in the P2 plane of the engine. Following these balances, we can state that the imbalance of the rotating bodies is caused by a force that arises for the following reasons:
- The centre of gravity of the body is not on the axis of rotation;
- The main axis of inertia of the body does not coincide with its axis of rotation.

Unbalance is therefore the condition a rotor is in when centrifugal forces are transmitted to the bearings as a force or vibratory motion. Unbalance can vary with rotor velocity; it can also be static, quasi-static or dynamic.

Fig. 6: The areas chosen for the points where engine vibration measurements were made [2].

Root Mean Square accelerations denoted R.M.S or r.m.s (acceleration most suitable for the intended purpose of the work) were measured.

The engine vibration measuring device was digital and measured accelerations in three directions. The most relevant measurement was along the Oz axis, i.e. perpendicular to the motor axis [18].

The engine speed was fixed at 1600 RPM, at which dynamic engine balancing was performed. A bolt is mounted on the flywheel and a large washer (m = 150g) is added to act as a dynamic damper. Thus, the washer rotates freely on the screw shaft without coming out of the screw.

The peaks given by the RMS (Root Mean Square) accelerations were followed by the vibration measurements.

For the experiment, from the practical experience of drivers and car maintenance mechanics, five measurement points on the engine casing will be chosen, the measurement points marked: *Point 1, Point 2, Point 3, Point 4* and *Point 5* represented in Figure 6 .

The RMS acceleration of the values obtained for all five marked points, are obtained with the vibration device, [9], [10], [12], [15] and are measurements at an integration of 10 s. The points were chosen according to the technical problems that arose in practice.

Point 1 is located next to the alternator, on the engine block. This point was chosen because the alternator component consists of a rotor and stator, the rotor rotating at high velocities (5400 RPM), and is supported in a bearing on a bearing.

Point 2 is located on the engine crankcase between the engine alternator and the oil filter, in the immediate location of this point there are no factors that lead to a significant increase in vibrations.

Point 3 is located in the plane containing the camshaft, near a gear between the camshaft and camshaft.

Point 4 is located on the crankcase of the engine block in an area that is not influenced by additional sources of vibration.

Point 5 is located near the engine flywheel and the clutch housing, an area strongly influenced by flywheel vibrations. Also, between points 4 and 5 is the water pump, the water pump having a pulsating mode of operation being driven by a cam on the engine camshaft and adding additional vibrations.

From the measurements made on channel 1 of the SVAN 958 device, Figures 7-9 show the r.m.s accelerations for the five studied points, because the main kinematic characteristic of the vibration measurement is acceleration [16].

If we want to briefly explain what a thermal process of the engine is, we can say that the processes that make up the engine cycle are called thermal processes, and in four-stroke engines, the thermal processes, except for combustion, roughly coincide with the displacements of the piston. For this reason, cylinder strokes are named according to the thermal process: *intake stroke*, *compression stroke*, *expansion stroke* and *exhaust stroke*. The combustion process takes place in the compression stroke, which takes about 1/3 of the total time, partly in the expansion stroke and about 2/3 in the last stroke.

3.2 Analysis of the results obtained from the measurements

The following figures (Fig. 7 – Fig. 9) will show the RMS accelerations measured at all five study points for the statically and dynamically unbalanced or balanced engine.

In the following, the graphs (Fig.7 - Fig.9) will be presented resulting from the simulation of the motion of the unbalanced or statically and dynamically balanced engine casing (Matlab software). the results are compared between statically and dynamically balanced motors against an unbalanced one, and all these RMS acceleration peaks are reproduced in Table 1.

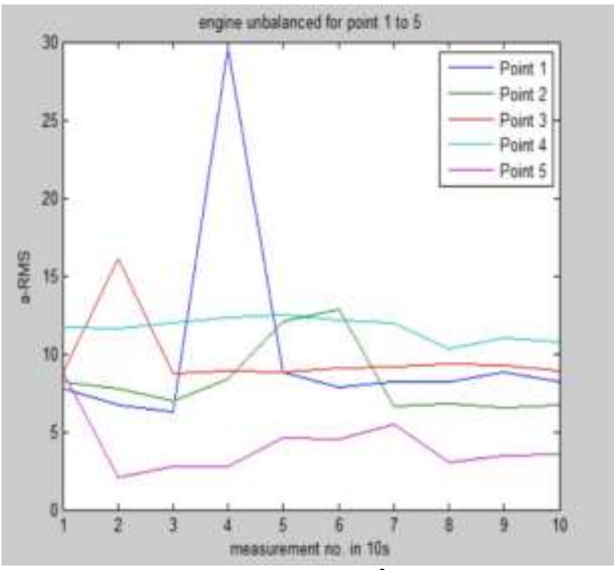

Fig. 7: RMS accelerations [mm/s^2] of an unbalanced, statically, and dynamically balanced engine (Point 1 to Point 5).

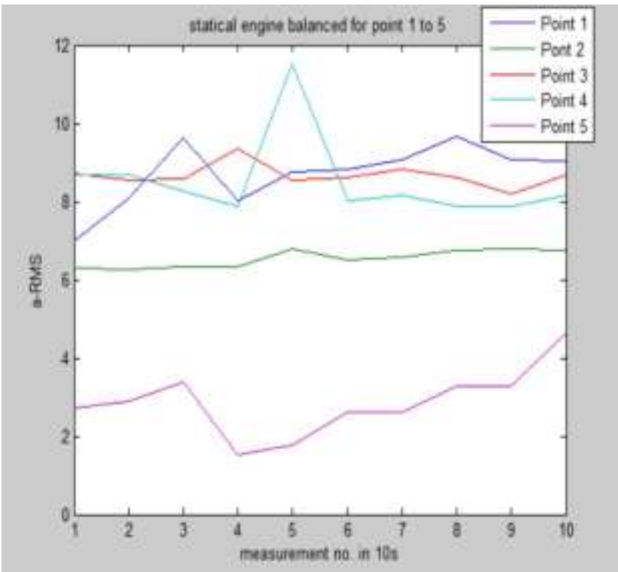

Fig. 8: RMS accelerations [mm/s^2] of a static balanced engine (Point 1 to Point 5).

Figure 8 shows an evolution from the point of view of vibration transmission from an unbalanced engine. The higher peak is in the Point 4 (a_{RMS} = 11.2 mm/s^2). The other peaks versus to positions of points are close, except Point 5 which has very low values.

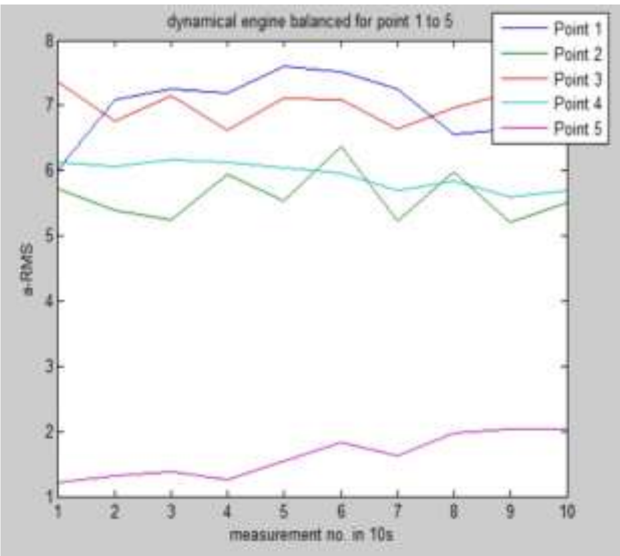

Fig. 9: RMS accelerations [mm/s^2] of a dynamic balanced engine (Point 1 to Point 5).

Figure 9 shows an evolution from the point of view of vibration transmission from a dynamically balanced engine. The higher peaks for all positions of points are close, except Point 5 which has very low values.

By the experimental data obtained, the balancing method with a dynamic absorber of 150g is validated in a percentage of 90% (the difference of 10% is represented by the factors that intervene in the measurements, such as the choice of experimental points, etc.).

The RMS peaks of the accelerations obtained for the engine with unbalanced and statically and dynamically balanced shock absorbers (especially the dynamically balanced one) are given in Table 1 (the peaks appear due to the technical operating conditions influenced by possible external factors, mounting, roadway, temperature, etc.).

The rms acceleration peaks obtained from the measurements according to Table 1 for the 5 studied points do not influence the operation of the dynamic absorber in the system.

Table 1. Acceleration a_{RMS} (peaks) of the engine in the measurement points.

a_{RMS} [mm/s^2]in the points	*Measure point position*	*Unbalanced engine*	*Statically engine balanced*	*Dynamically engine balanced*
Point 1	(on the engine block and located near the alternator)	30	9.5	7.6
Point 2	(on the engine block, more precisely on the engine crankcase)	13	6.8	6.4
Point 3	(located in the plane containing the distributor axis)	16	9.4	7.4
Point 4	(on the crankcase of the engine block)	12.5	11.5	6.2
Point 5	(on the crankcase of the engine block near the engine flywheel as an additional source of vibration production)	9	4.5	2

4 Conclusions

The new theoretical approach to the dynamic absorber in vehicles shows the ideal case where a dynamic absorber could be built so that it does not take up space, or in other words, the square of the harmonic order (x) is equal to the ratio of the distance at which the front absorber is mounted of the axis of rotation of the crankshaft and the length of the dynamic absorber. In this theoretical case, the number of resonance frequencies decreases by one unit (identical to the case of using reduced masses) [2]. Figure 4 theoretically validates the need to use a dynamic absorber in engine construction. The theoretical study was designed by modeling the balanced system by assimilating one of the masses mounted at one end of the shaft, with a dynamic absorber.

- Following the simulations, it is found that, during the dynamic balancing using a dynamic absorber as one of the balancing masses, the new system's vibrations (A_i harmonics) are reduced, of course taking into account the simplifying assumptions.
- By solving the equations, the obtained solutions give us the values of the harmonics of the system.
- The next step was for the theoretical study to be simulated in Matlab, and the graphic form to provide a first validation of the theoretical modeling.

• The last part of the work dealt with a preliminary pre-calculation regarding the elements of an engine to which the theoretically studied absorbent could be applied, and the results validated the possibility of its use.

Thus, the objective of the work was achieved by the fact that the theoretical foundation was demonstrated through the two validations given by simulation and experimental measurements.

Regarding the last study in the paper, it starts from the well-known fact, [6], [13], that the vibrations of vehicles, especially of gasoline ones, increase when the number of engine pistons decreases. For this reason, to improve the engine consumption and to reduce the ambient noise in the vehicle as well as the external ambient noise, it was shown in a theoretical study in a previously published work, [2], that the static and dynamic balancing of an engine it is indisputable. The installation of a dynamic shock absorber considerably reduces the vibrations of the engine casing, especially in the points that are near other sources of vibration, such as the water pump, etc.

An extensive repair was performed on a three-cylinder, four-stroke engine that was already supported on the stand, with the stand taking the place of the vehicle's engine housing (Fig. 1). The repair included the proper segmentation of the standard pitch of the segments so that the engine could be said to be in good working order, therefore comparable results were obtained with the parametric and real life results. After the dynamic balancing of the crankshaft, together with the related mobile crews of the system (masses m_i, i = 1-5), vibration measurements were made. RMS acceleration values were recorded at the five points and measured cyclically at 10 second intervals. Out of these chosen points on the engine casing, only three points, namely Point 2, Point 3 and Point 4 correspond to the three cylinders with associated mobile crews.

Of course, the obtained measurement results are influenced by other external or internal factors, such as engine heating, etc., but the results validated the fact that these torsional vibrations exist and are transmitted from the engine through the studied points, but not. only.

The results obtained through measurements and presented in Figures 7-9 as well as in Table 1, show an improvement in engine vibrations by applying the dynamic damper if comparing the case of an unbalanced engine with a statically and dynamically balanced one. Regardless of the measurement point, these vibrations are anyway enhanced by the presence of the absorber on the flywheel. Thus, the paper achieved its proposed objective but emphasized the fact that engine vibrations could not be eliminated and will never be eliminated due to the thermal processes that occur in the operation of the engine. The most significant results are observed for Point 1 and Point 5 (according to Table 1), respectively points measured on the engine block and located near the alternator and on the engine block crankcase near the engine flywheel (neighborhoods that also produce additional excitations).

Practically, the realization of a dynamic shock absorber poses construction problems in the structure of a vehicle's engine, especially for reasons of size, although this paper effectively explained that the presence of such an absorber (screw and washer), otherwise a rudimentary and small-sized piece, that has a great influence in the transmission of vibrations from the engine. All this implicitly leads to the reduction of the vehicle's fuel consumption.

Since the paper still talks about the efficiency of using a dynamic absorber in the structure of an engine, in the following, some disadvantages of their use will be presented:

- It is necessary to dynamically balance the engine before the experiments;
- The need to verify experiments on several types of four-stroke engines is a certainty;

For these reasons, further future research in the field could consist of:

- Contributions to the study of dynamic shock absorbers in the balancing of internal combustion engines, by using two dynamic shock absorbers;
- Contributions to the study of dynamic pendulum shock absorbers in bifilar suspension for the balancing of crankshafts;
- Contributions to the reduction of torsional vibrations of crankshafts with the help of several dynamic shock absorbers or their verification on different types of engines.

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Aurora Felicia Cristea - simulated the measurements, and reviewed the paper.

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