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COMPRESSION TESTS METHODS ON CAR ENGINES

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Abstract: *A correct operation of the internal combustion engine requires the best possible sealing of the combustion chamber during the period when the valves are closed. A perfect sealing of the combustion chamber cannot be achieved in view, first of all, of the constructional features of the piston rings as well as the clearance between the piston and the cylinder. Along the use of the engine over time, the wearing of the moving parts leads to a continuous worsening of the sealing. To measure the combustion chamber's leakage, methods that differ in terms of both complexity and accuracy are used. In this article, the following methods of assessing the combustion chamber sealing are presented: measuring the pressure in the cylinder at the end of the compression process using a compression gauge; measuring the current drawn by the starter motor during cranking; comparing the engine speeds in the final part of the compression stroke of each cylinder when cranking; comparing the engine speeds at the beginning of the expansion process of each cylinder when idling. The paper presents a comparative analysis of the values obtained by different methods with the cold engine versus the hot engine and, also, a comparative analysis between the results obtained by using different methods of evaluation of the combustion chamber sealing. The paper demonstrates that pressure measurement using the compression gauge is the most accurate method for determining the sealing of the combustion chamber but it is not the most expeditious.*

Key-words: *test methods, engines, compression*

1. INTRODUCTION

A correct operation of the internal combustion engine requires the best possible seal of the combustion chamber during the period when the valves are closed. The way this goal is achieved influences the development of thermal processes, having direct repercussions on the engine's energy and economic performance, higher fuel and engine oil consumption, lower cold start capacity, faster engine oil degradation, as well as its behavior regarding its environmental pollution [1].
The level of the combustion chamber sealing depends on the technical state of the following engine components: piston - cylinder - piston rings group, valve - valve seat couple, cylinder head and cylinder head gasket.
Measurement of the level of the combustion chamber leakage may therefore be an indirect way of assessing the technical condition of these parts without the need for dismantling the engine.
A perfect sealing of the combustion chamber cannot be achieved in view, first of all, of the constructional features of the piston rings as well as the clearance between the piston and the cylinder.
Along the use of the engine over time, the wearing of the moving parts leads to a continuous worsening of the sealing.
Also, the sealing of the combustion chamber may also be compromised as a result of accidental causes: breaking or blocking of piston rings, damage of the cylinder head gasket, valves blocking, pinching of the valve head edges and/or of the valve seats, deterioration of the contact surface between the cylinder head and the engine block, cracking of the cylinder head or incorrect setting of the valve clearance to a low value [2].
Differences of values prove that not only the tightness of the combustion chamber has been deteriorated, but also its technical condition and correctness of adjustments have been affected [3].

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To measure the combustion chamber's leakage, methods that differ in terms of both complexity and accuracy are used. In this article, the following methods of assessing the combustion chamber tightness are presented:

a) quantitative methods: measuring the pressure in the cylinder at the end of the compression process during cranking;

b) qualitative methods:
- measuring the current drawn by the starter motor during cranking;
- comparing the engine speeds in the final part of the compression stroke of each cylinder when cranking;
- comparing the engine speeds at the beginning of the expansion process of each cylinder when idling.

2. OBJECTIVES

The research was conducted on a passenger car equipped with a compression ignition engine and pursued the following objectives:

a) comparative analysis of the results obtained using different methods;
b) comparative analysis of the values obtained with the cold engine compared to those obtained with the hot engine when applying the same method;
c) establishing a degree/mode of correlation between different methods of assessing the tightness of the combustion chamber.

3. PREPARING AND CONDUCTING THE EXPERIMENTAL INVESTIGATION

The paper refers to the evaluation of the combustion chamber tightness by four quantitative and qualitative methods. The tests were carried out on a 2008 Opel Astra H passenger car equipped with a compression ignition engine with Z19DT engine code, 1910 cm^3 piston displacement and compression ratio of 18.4 [4]. Specific preparatory operations were carried out according to the measurement method, prior to the testing. The battery charge level has been checked and the proper operation of the diesel fuel system has also been checked.

3.1. Measuring the pressure in the cylinder at the end of the compression process during cranking

This method involves measuring the pressure at the end of the compression process using a special compression gauge for diesel engine, while the engine is cranked by the electric motor.

In [4] and [5] it is recommended that the measurement should be done when the engine is hot (coolant temperature higher than 80^{o}C).

Measurement of the compression pressure can be done by replacing the glow plugs or injectors with special compression gauge adapters.

For technical reasons, it has been chosen to remove the injectors. All four diesel injectors have been removed after the engine has been warmed up, in accordance with the procedure indicated in [5].

The high-pressure pump has to be disabled.

For each cylinder, five measurements were made. Measurements have been made so that the tests ran as soon as possible and the engine temperature drop during the measurements was reduced at minimum possible.

The compression pressure was determined (figure 1) first with the cold engine (20°C coolant temperature) and then with the hot engine, using the Diesel Motometer compression gauge.

According to [4] and [6], for Opel Z19DT engine in hot condition (coolant temperature higher than 80^{o}C), the pressure at the end of the compression process must be between 24 bar and 32 bar.

3.2. Measuring the current drawn by the starter motor during cranking

The principle underlying this method is that a current of specific intensity is absorbed during the compression process in each cylinder: the better the tightness of the combustion chamber and cylinder, the higher is the resistance to the piston displacement in the compression stroke.

Therefore, the intensity of the current drawn by the starter in the cranking process will be higher.

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In [7], the worse tightness of a cylinder decreases the values of the instantaneous current during the compression stroke comparing to a faultless cylinder.
The measurement results in [8] confirm a strong relationship between the tightness of the engine combustion chamber and the current drawn by the starter.
The measurements in the present research were made with the Bosch FSA 740 diagnostic unit from Politehnica University of Bucharest, the Automotive Engineering Department.
Figure 1 shows the interface where the result of a measurement is displayed.

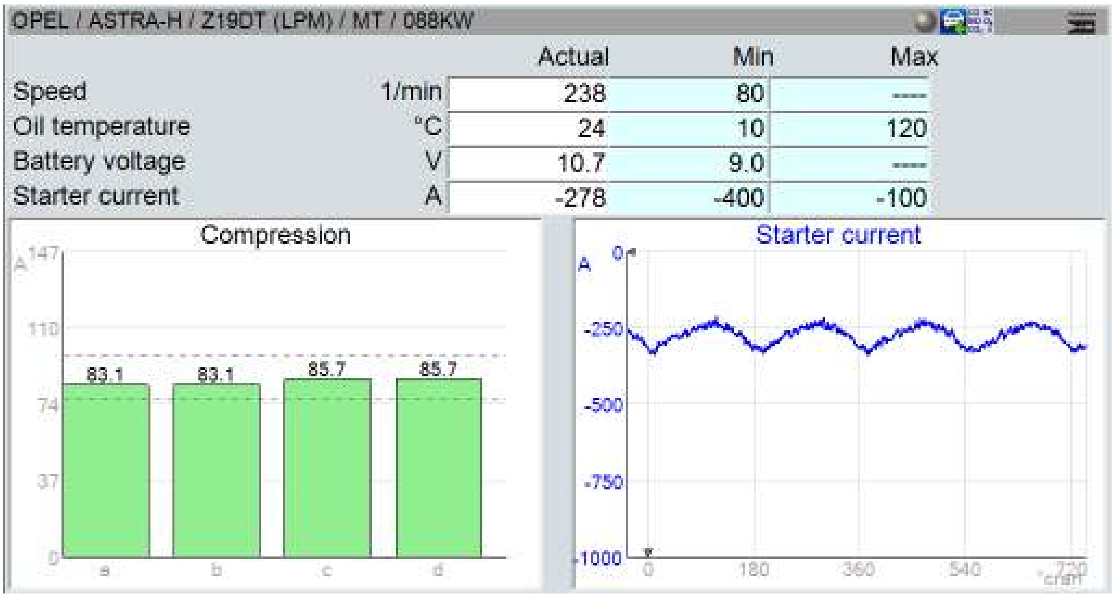

Figure 1. Measuring the current drawn by the starter motor during cranking

At the end of the measurement, the following measured parameters are displayed: speed, oil temperature, battery voltage during the cranking process, and the average value of the drawn current, corresponding to a complete cycle.
It also shows graphically the values of the currents required to overcome the resistance of each cylinder (left) and the variation of the current drawn by the starter (right) during one cycle.
The cylinders' denominations "a, b, c, d" do not correspond to the engine cylinders' positions.
They comply with the ignition sequence of the engine "1-3-4-2", but cylinder "a" is the one at which the first compression was recorded from the moment the contact was actuated.
Thus, this method cannot determine the location of possible leaks, but only their existence.
Figure 1 shows also the allowable range in which the drawn current has to be located for acceptable leakage.
This range depends on the vehicle and engine types, on the test conditions (motor temperature) and on the value of the drawn current.
The higher the current, the higher the admissible range. For example, in figure 1, the admissible range is between 76 A and 97 A.

3.3. Comparing the engine speeds in the final part of the compression stroke of each cylinder when cranking

The test is based on the analysis of the angular speed values obtained from the engine speed transducer. The engine speed is displayed for each cylinder, for the final part of the compression stroke, in the starting mode by the starter - figure 2.
The principle of the method is as follows: the better the tightness of the combustion chamber, the higher are the gas pressure and the resistance against the piston movement in that cylinder; therefore, the piston speed before the top dead center is smaller.
Diagnosis was done using the Bosch KTS 570 device which was connected to the car's OBD interface.
A specific software was used. The test was done both with cold and hot engine. The results of this test show the speed values of each piston during cranking, figure 2. According to [6], a difference of maximum 7 min^{-1} between any two values measured on the four cylinders is considered acceptable.

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Figure 2. Comparison of speeds in the last part of the compression stroke of each cylinder when cranking.

3.4. Comparing the engine speeds at the beginning of the expansion process of each cylinder when idling

The test is based on the analysis of the information received from the engine speed transducer when idling. The Engine Control Unit (ECU) sets an identical injection duration for all injectors for a certain speed and load regime.

The engine speed at the beginning of the expansion process is measured and displayed for each cylinder, as shown in figure 3.

The principle of the method is as follows: the better the tightness of the combustion chamber, the higher the gas pressure during the combustion process and therefore the higher the engine speed at the beginning of the expansion process in the respective cylinder.

Since the specified speeds are influenced by the injection quality and by the double-flywheel, this method is only applicable once it is known that the ignition, supply and distribution systems are in good working condition. Diagnosis is performed using the Bosch KTS 570 device. It is connected to the car's OBD interface and specific software is used.

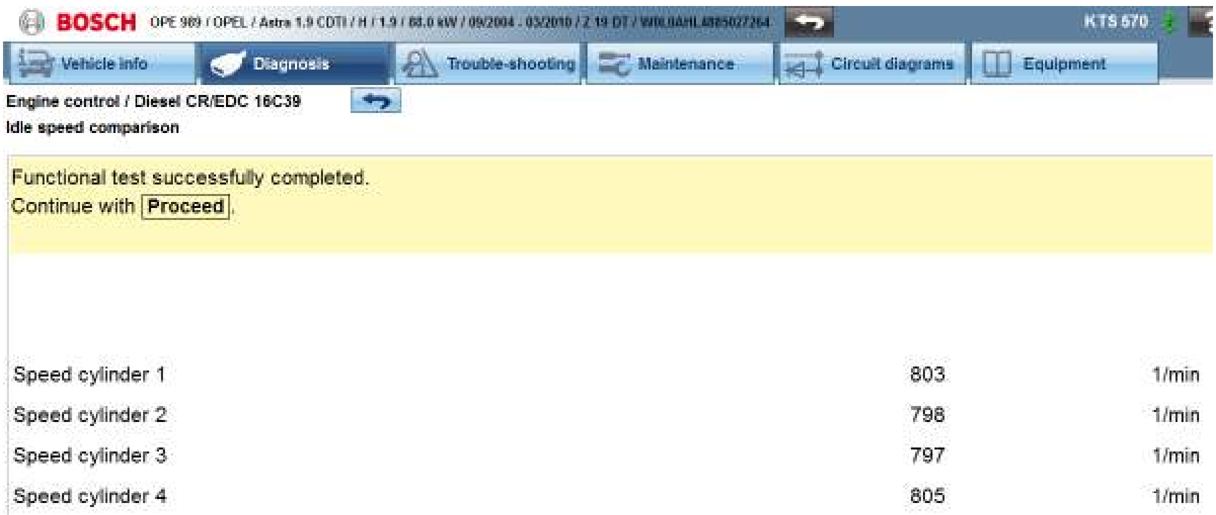

Figure 3. Comparison of engine speeds at the beginning of the expansion process of each cylinder when idling.

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According to [6], a difference of 21 min^{-1} between the highest and lowest speed measured for the four cylinders may be considered acceptable.

4. RESULTS AND THEIR INTERPRETATION

4.1. The pressure at the end of the compression process

Following the tests, the pressure values obtained with the cold and hot engine are shown in figure 4.

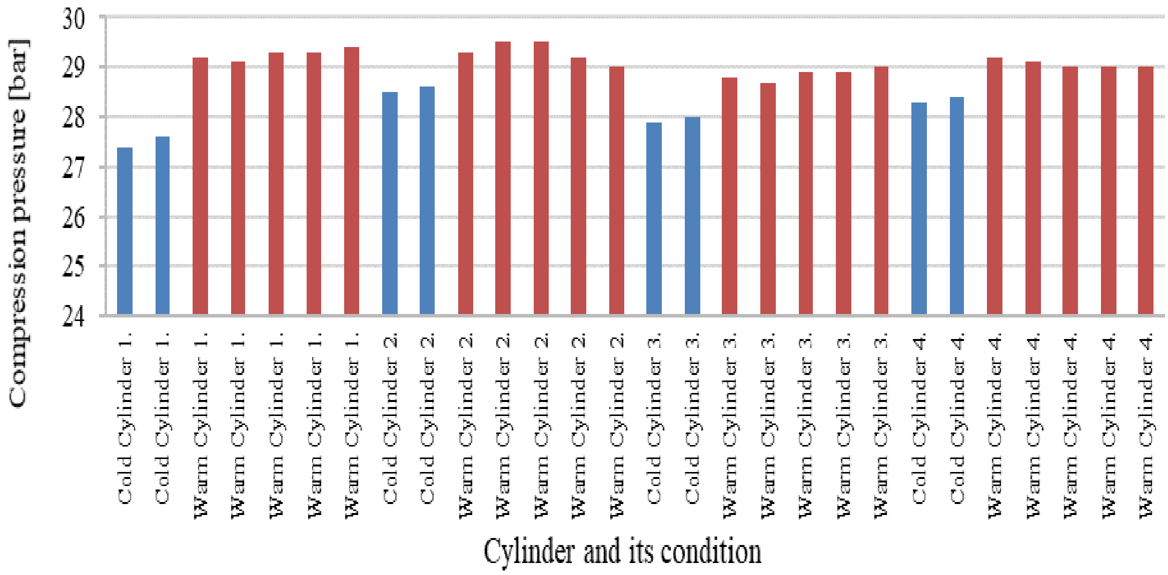

Figure 4. The pressure at the end of the compression process.

The compression pressure measured is within the tolerable range of 24 ÷ 32 bar in both thermal measurement modes. The average values of the measured pressures when the engine is cold are lower than the average values measured with the hot engine by 3% to 6%. In the paper [5] it is said that these differences between the two sets of values are around 10%. Important differences may occur because "cold engine" expression is relative, depending on the ambient temperature. All five measurements for each cylinder have very close values. In the order of cylinders from 1 to 4, the non-uniformity degrees of the compression pressure are: 1%, 1.7%, 1% and 0.7%. The third cylinder has the smallest pressure, with an average of 28.9 bar. Cylinders one and two have the highest average values of the compression pressure of 29.3 bar. The degree of non-uniformity of the average values measured on all cylinders is 1.4%, below the maximum admissible limit of 2 ÷ 4%, indicated in paper [2].
The compression pressure values measured according to [4] show that the engine is properly leakproof.

4.2. Results obtained when measuring the current drawn by the starter motor during cranking

For measuring the current drawn by the starter motor, five sets of measurements of cold engine (24^{o}C oil temperature) and five sets of measurements of hot engine (80^{o}C oil temperature) were made.
The results are shown in figure 5. The maximum values of the current intensities drawn by the starter during cranking corresponding to each cylinder are shown for each set of measurements.
The order of cylinders in the graphical representation is different at each measurement since the first result will correspond to the cylinder at which the first compression was recorded from the moment the starter was actuated.
In figure 5, the order of graphical representation corresponds to the order of the measurements.
The mean values of current measured with the cold engine are higher by 41% on average than the mean values of current measured with the hot engine.
The average value of the cranking engine speed is 244 min^{-1} and ranges from 216 min^{-1} to 258 min^{-1}. When the engine is cold, plays in kinematic couplings are bigger, and the lubrication is deficient because the higher viscosity of the oil increases the friction in the couplings of the mechanisms.
This explains the high values of the current measured at cold engine tests.
The first measurement was made after a 24 hour immobilization of the car.

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It was observed that the first recorded values were the highest and after that, they were stabilized. The non-uniformity of the values measured with the cold engine was between 3.1 ÷ 6.5%, below the 10% limit recommended in the paper [4].
The sets of measured values with hot engine have a non-uniformity of 4.4% (sets 1, 2, 4 and 5) while for the set number 3 it was 8.3%. Four of the five sets of measurements are identical as shown in figure 5. It was shown that the third cylinder has the lowest leak-tightness (4.1).
The method of measuring the current drawn by the starter when cranking the engine confirms this situation. Given the order of ignition, it is pointed out that there is the same cylinder that absorbs the smallest current. Although valuable information is obtained during the cold test, it is recommended that this type of measurement to be made with the warm engine for better accuracy of the results. Measuring the current drawn by the starter motor during cranking is very useful in evaluating the tightness of the combustion chamber because it is very fast and requires few preparatory operations.

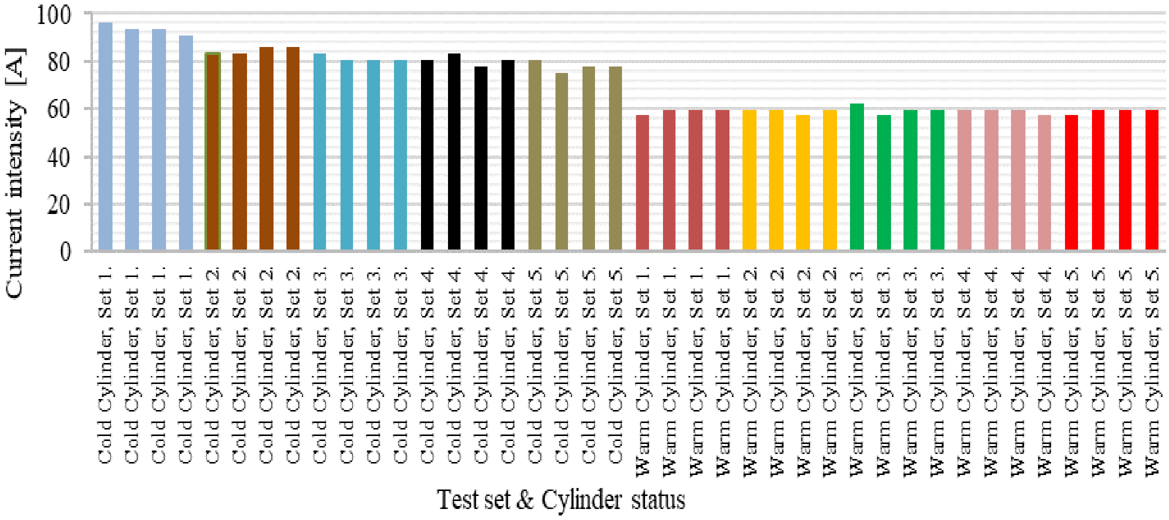

Figure 5. The current drawn by the starter motor during cranking.

4.3. Results obtained when comparing the engine speeds in the final part of the compression stroke of each cylinder during cranking

For each cylinder, five sets of speed measurement with cold engine (engine oil temperature 26^{o}C) and five sets of speed measurement with hot engine (oil temperature in the range 87 ÷ 90^{o}C) were made. Individual engine speeds corresponding to each cylinder are displayed, as shown in figure 6. The engine speeds are lower by an average of 16% when the engine is cold than when the engine is hot. This is due to the causes above mentioned in section 4.2.

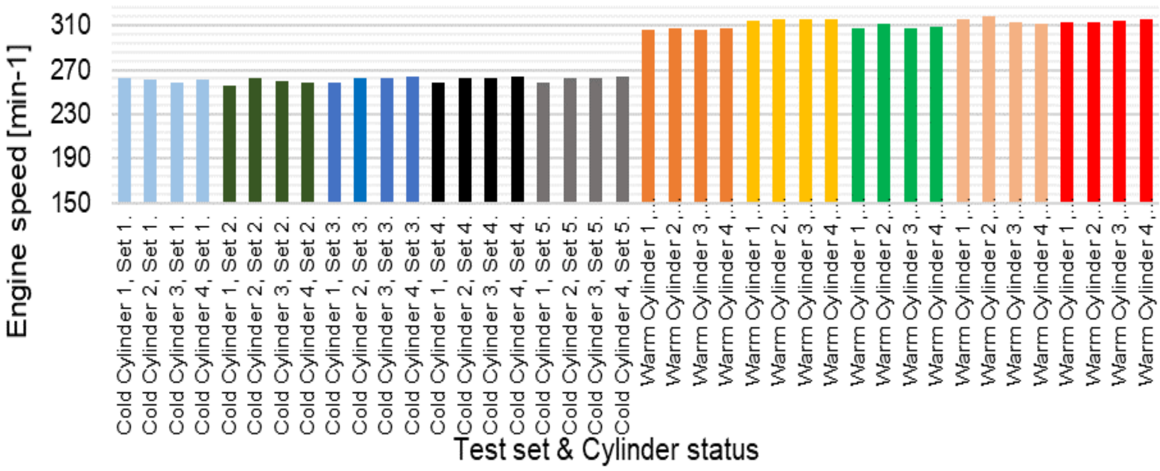

Figure 6. Engine speeds in the final part of the compression stroke of each cylinder during cranking.

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On average, at cold measurements, the engine cranking speed is 261 min^{-1} and varies between 256 ÷ 264 min^{-1}, which is 1.5 ÷ 2.3% below the 10% recommended in the [4].
Analyzing the results obtained with hot and cold engine, it is found that in both situations, the cylinder one has the lowest speed. That means the best sealing is shown for cylinder one in four of the five sets of tests. For cylinder one and two, the average pressure values measured with the compression gauge (4.1) were the highest. Although the information obtained during the cold test is quite relevant, the unevenness of the measured values is relatively high. Consequently, for better accuracy of results, it is recommended that this type of measurement to be done with warm engine.
Comparing the engine speeds in the final part of the compression stroke of each cylinder during cranking is quite a useful method in evaluating the tightness of the combustion chamber as it is quick and requires little preparatory work.

4.4. Results obtained when comparing the engine speeds at the beginning of the expansion process of each cylinder when idling

To measure the engine speeds when idling, five measurements sets of cold (t_{oil} = 50°C) and hot (t_{oil} = 80°C) engine were made. The results are shown in figure 7. The engine speeds corresponding to each investigated cylinder are displayed for each set of measurements.

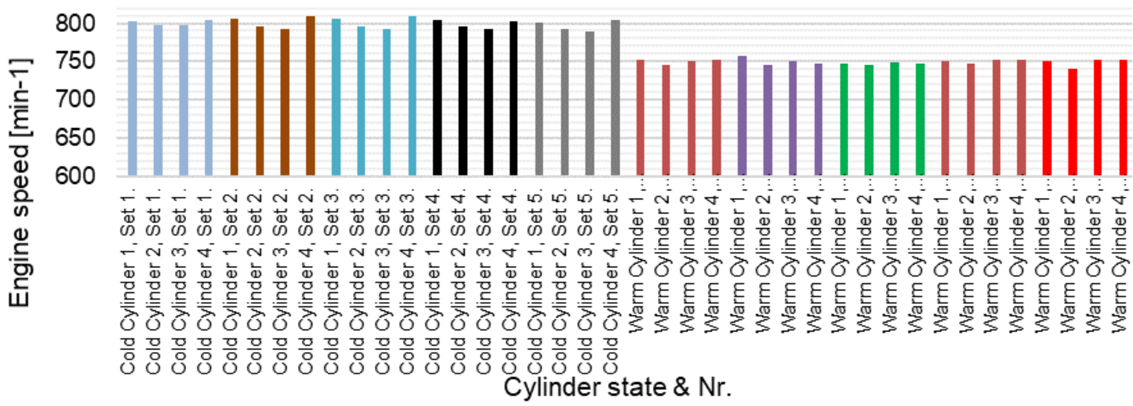

Figure 7. Engine speeds at the beginning of the expansion process of each cylinder when idling.

On cold start, the ECU commands the fuel injection system to ensure a higher idle engine speed for a steady ride in this regime. Then, the engine speed is dropped and stabilized to the normal value as the engine gets warm. The engine speeds are lower on average by 6.3% when they are measured on cold than they are measured on hot engine. For cold measurements, the idling speed varies between 793 ÷ 809 min^{-1} with an average value of 800 min^{-1}. The non-uniformity of the cold measured values was between 1 ÷ 2% below the 10% limit recommended in the paper [4]. With hot engine, the average value of idling speed is 750 min^{-1} and ranges from 741 to 757 min^{-1}. The non-uniformity of the measured values was between 0.4 ÷ 1.6%. Analyzing the results obtained with cold engine, it is found that, in all five sets of tests, the third cylinder corresponds to the lowest speed, so it has the lowest tightness.
The same result was achieved by measuring the pressure with the compression gauge. This method indicates that the fourth cylinder has the best tightness, but this is not confirmed by the pressure measurement using the compression gauge.
When analyzing these results, it is important to consider that the engine speed is changed after the cold start until it is stabilized. An important factor is the working temperature of cylinders. It is known that the middle cylinders warm up faster than the exterior cylinders, so their corresponding speed will decrease faster (will be stabilized faster). Figure 7 shows that at all five sets of measurements made with cold engine the lowest speed was measured on cylinders 3 and 2 which warm up faster.
This explains why the test program requires the measurement to be done with the warm engine.

5. CONCLUSION

Measuring the pressure by using the compression gauge is the most reliable method because it is a quantitative method and allows the identification of the engine cylinders.

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This method requires the disassembly of some components of the diesel fuel system, which represents a disadvantage of the method. The measurements are relevant for both situations: hot and cold engine. The five measured values for each cylinder have very close values. The degree of non-uniformity of the measurement sets for each cylinder varies between 0.7% and 1.7%. The experiments carried out showed that cylinders 1 and 2 have the best tightness, while cylinder 3 has the lowest one. By measuring the current drawn by the starter motor during cranking, it is determined whether the tightness is good or not. The method is a qualitative one. Although valuable information is obtained during the cold test, it is recommended that this type of measurement to be made with warm engine for better accuracy of the results. The method is very useful in evaluating the tightness of the combustion chamber because it is very fast and requires few preparatory operations. Comparing the engine speeds in the final part of the compression stroke of each cylinder during cranking is an indirect but quite useful method in evaluating the tightness of the combustion chamber as it is quick and requires little preparatory work.
Comparing the engine speeds at the beginning of the expansion process of each cylinder when idling is a rapid, indirect method for evaluating the sealing of the combustion chamber. The measured values are influenced by the fuel injection quality, the state of the ignition system and the dual mass flywheel.
These last two methods are convenient when using a complex diagnosis system which is able to perform a wide range of tests concerning several engine devices: injectors, transducers, EGR system, injection timing etc. The method of measuring the current drawn by the starter during cranking shows that the value of cylinder 3 has the weakest tightness. The pressure measured by using the compression gauge shows the same. Analyzing the results obtained when comparing the engine speeds in the last part of the compression stroke of each cylinder during cranking it was found that the cylinder 1 has the smallest leakage. Cylinders 1 and 2 presented the highest mean values of compression pressure.
The results obtained by comparing the engine speeds when idling confirm that the third cylinder had the lowest tightness, the same result being achieved by measuring the pressure with the compression gauge. This method indicates that the fourth cylinder has the best tightness, but this is not confirmed by the compression pressure measurement.
When interpreting these results, it is important to consider that the engine speed changes relatively fast after the cold start until it is stabilized.
The first three methods of measuring the combustion chamber's tightness have shown very close results using different measurement principles.
This conclusion recommends the use of alternative (qualitative) methods of evaluating the tightness of the combustion chamber because they are fast and reliable without requiring costly preparatory actions.

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CONSIDERATIONS CONCERNING THE POWER LOOPS WITHIN THE ALL-WHEEL DRIVEN TRANSMISSIONS OF THE AUTOMOBILE

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Abstract. *This paper aims at analyzing the ways the power loop magnitude varies within the 4x4, inter-axle transmission of and automobile, according to different parameters. More accurate, it aims at determining the way the power loop magnitude varies according to the variation of the tire radii difference between the front and the rear axle. To achieving the paper's goals, a set of specialized tests has been developed, using a military APC with a 4x4 driveline. The dedicated vehicle was the reconnaissance vehicle TAB C-79. To reveal the power loop between the axles, we needed to measure the torque and the angular speed within different relevant spots of the driveline. Specialized sensors have been used together with a GPS VBOX Mini equipment.*

The experimental results have been illustrated within a series of charts providing the power loop's magnitude time history in different situations. A complex analysis of these data led to useful scientific conclusions. As a novelty, this paper presents a set of conclusions concerning the power loops when the vehicle was rolling on a concrete surface. We cold say that this paper clarifies the way to determine the power loop occurrence and also its magnitude within the trans-axle transmission when taking into account various factors. Moreover, the conclusions can be used to identify the necessary measures to be taken to diminishing the power loop at the longitudinal level of the automobile's driveline.

Key-words: *power loops, all wheel driven transmission, rolling radius, inter-axle differential*

1. INTRODUCTION

The all-wheel driven transmission yet has a big problem: the self generated torque that occurs inside the driveline. The only way to override the wind-up torque, occurred inside of a normal driveline, is to increase the power delivered by the engine.

On the other hand, this power loss can't be used as a traction power for the automotive. Hence, it leads to extra fuel consumption, supplementary wear of the driveline's components and of the tires.

Nevertheless, decreased maneuverability should be taken into account [2][8][9].

This paper performs an analysis of the ways the power loops vary along the inter-axle driveline of a 4x4 automobile with respect to some certain parameters.

The loop power-flow is the power that loads a closed loop of the transmission driveline and it is not generated by the automobile's driving or by coasting processes.

Mathematically speaking, it is computed by gaining the self-generated torque on some component with that component's angular speed [6].

The cause generating the power loops between the vehicle's axles is the difference between the wheels' travel. At its turn, a major reason to have different wheels travel is the unevenness of the rolling radius. Within this paper, whenever the tire radii difference is called, we assumed the tire radii difference between axles, since the tires of the same axis have the same rolling radius.

Many papers have developed studies with respect to the analyzed problem [2][4][5].

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2. THEORETICAL NOTIONS WITH RESPECT TO THE POWER LOOPS BETWEEN AXLES

We hereby present some theoretical notions concerning the power loop between a all-driven transmission of a vehicle.
The balance of power in the situation described, according to the literature, is given by [6]:

$$P_{mot}\,\eta_{T_{M-S}} = \varphi Z_S\, r_{r_S}\,\omega_r - \left(\varphi Z_S - \sum R\right) r_{r_F}\,\omega_r\,\eta_{T_{F-CD}} \tag{1}$$

Equation 1 gives the power balance within the transmission, at the longitudinal level, and also makes a good analysis of the components of the power flow inside the driveline for the specific driving conditions. If considering the terms of this equation, following identifications are possible: $P_{mot}\,\eta_{T_{M-S}}$ is the engine's output as input for the transfer case; $\varphi Z_S\, r_{r_S}\,\omega_r$ is the grip power and $\left(\varphi Z_S - \sum R\right) r_{r_F}\,\omega_r\,\eta_{T_{F-CD}}$ is the self-generated power.

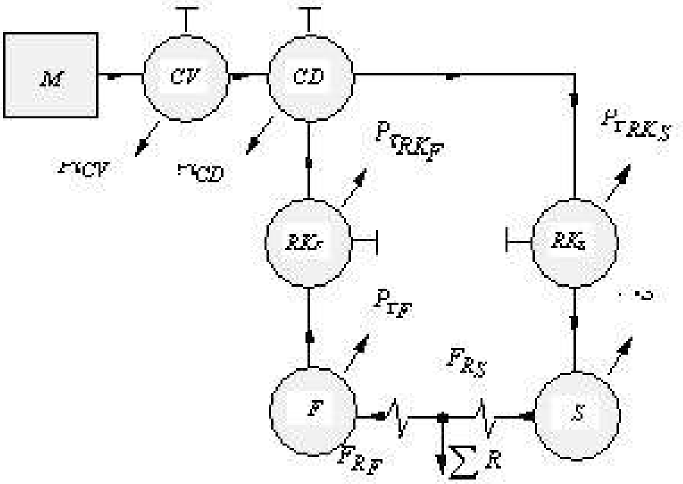

Figure 1. Front-to-back loop power-flow graph of 4x4 vehicle

In the previous formulae, P_{mot} is the power that comes from the vehicle's engine, $\eta_{T_{M-S}}$ is the engine-to-rear axle circuit's efficiency, $\eta_{T_{F-CD}}$ is the front axle-to-central differential circuit's efficiency, φ is the adherence limit, Z_S is the rear axle's vertical load and ω_r is the wheel's angular speed.
Should also be mentioned that r_{r_F} and r_{r_S} are the front and rear tire radii respectively.
The rear wheels radius can differ from the front ones but the wheels of the same axle are equal to each other. We also assumed that the angular speed is the same to the rear and front axle's wheels [2].
ΣR is total drag while.
On the other hand, the power flow driven to the front axle is given by equation 4, where, along the already defined terms, we yet have r_c as the cinematic tire radius and k_e as the tire's elasticity [2].

$$P_F = \frac{r_r \eta_{T_{F-S}}}{1+\cos\alpha}\left[\sum R - \frac{r_c}{k_e}(1-\cos\alpha)\right]\omega_r \tag{2}$$

These flows can be noticed in figure 1 [7]. In these figures M stands for engine, A is the clutch, CV stands for gearbox, CD stands for the transfer case with an open, lockable differential, RK stands for final drive's transmission ratio, ***F*** and ***S*** stand for front and rear wheels respectively while ***P*τ** represents the power losses (heat). Also, ***ΣR*** is total drag, ***v*** is the vehicle's speed, **P$_{mot}$** is the engine's output power, **F$_{RF}$, F$_{RS}$** are the driving forces at front and rear axle, respectively and ***P$_{F-S}$*** is the loop power-flow between axles [7].

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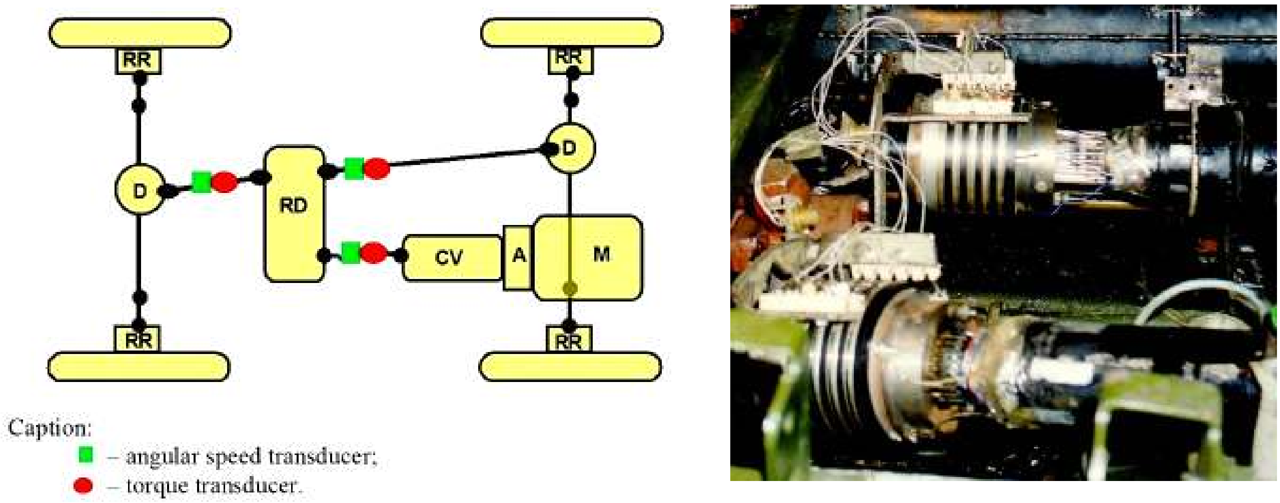
3. EXPERIMENTAL RESEARCH

To achieve the goals, some transducers were mounted on different components of the driveline (figure 2 [4]). So, on the shaft between the engine and the central differential were mounted a Wheatstone bridge to get the engine's torque and an inductive angular speed transducer.
On the shafts between the rear and the front axle, on one hand and the central differential on the other hand, were also mounted Wheatstone bridges to get the torque and optical angular speed transducers [1, 2].



a *b*

Figure 2. Mounting the transducers to reveal the loop power-flow.

Figure 2-a uses the following abbreviations: M – engine; A – clutch; CV – gearbox; RD – transfer case with open, lockable differential; D – transversal differential; RR – hub gear.
Figure 2-b [3] it is represents the torque acting on a transmission shaft of a military vehicle.
The tests have been developed under three circumstances, as follows:

- no tire rolling difference between axles
- 30 mm tire rolling difference between axles (rear axle having larger rolling radius)
- 60 mm tire rolling difference between axles (rear axle having larger rolling radius)

To experimentally assess the rolling radius we used:

$$P_F = \frac{S}{2\pi} \tag{3}$$

where *S* is the travel of a wheel when taking a complete rotation.
Technically, to experimentally determine the rolling radius of a tire ($\boldsymbol{r_r}$) we made a mark on the tire (with the wheel in working position) and traveled 5 complete turns of the wheel.
We measured the traveled distance then we used equation (3) to compute the rolling radius.
We used:

$$\Delta r = \left| r_{r_S} - r_{r_F} \right| \tag{4}$$

to determine the tire rolling difference between the front and rear axle's wheels.
The measurements performed to determining the variation of the self-generated power between the axles took into account the rolling resistance of the Armoured Personal Carrier.
To having a controlled rolling resistance we have created a convoy made of the tested vehicle towing a heavy military truck.
The rolling resistance variation has been simulated by progressively braking the towed truck.

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4. RESULTS

We studied all the charted plots when the vehicle was traveling on concrete to achieving a complete analysis of the self-generated power magnitude's variation when the rolling resistance varies. In this respect, we had a great interest in the following charts that provide information about the variation of the self-generated power:

- engine to transfer case;
- transfer case to front axle (self-generated power);
- transfer case to rear axle.

The objectives of the paper are achieved by identifying the shape and the slope of the curves that describe the evolution of the analyzed power flows.
When analyzing the figure 3 (up) one could notice that all the power fluxes are positive when traveling on concrete, normal drive and no tire radii difference. It can also be noticed that the power evolution on the input shaft of the transfer case consists of an increasing stage, followed by a constant development of the magnitude to about 14 kW. A steep evolution can be noticed as the towed vehicle is progressively braked (simulating an increase of the rolling resistance).

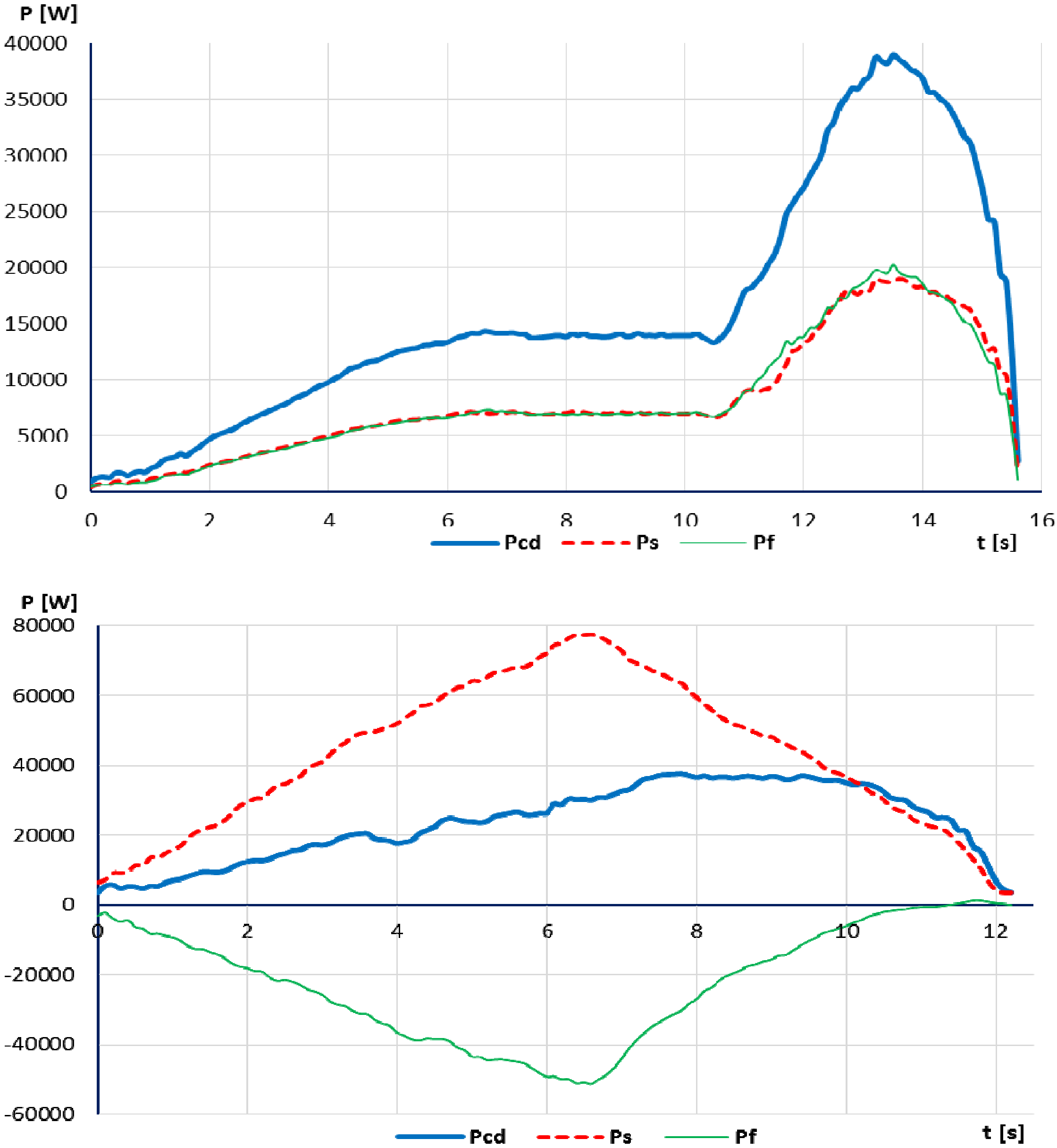

Figure 3. Power fluxes variation within the vehicle's driveline for $\Delta r = 0mm$ (up) and $\Delta r = 30mm$ (down) when rolling on concrete

On the other hand, the power flux evolution on the input shaft of the transfer case grows up to its maximum value of 39 kW and it is followed by a sudden decrease down to a complete disappearance, as the towed vehicle is braked.
The powers on the front and rear output shafts of the transfer case are similar to the one on the input shaft of the transfer case, yet their values are constant and closed to 7.5 kW during the taking off stage. Their maximum value is about 20 kW. They are reached during braking the towed vehicle and they rapidly decrease to null. Analyzing the chart of the power variation within the same conditions but considering a tire radii difference of 0.03 m (figure 3 - down), one could notice that the power on the rear axle (***Ps***) is permanently positive, as the input shaft's one is (***Pcd***).
Nevertheless, the power on the front output shaft (towards the front axle) is mostly negative.
To underlining the evolution of the studied phenomenon we should recall that the positive power makes the vehicle moving while the negative ones (closed loop power) don't.
Studying the shapes of the power curves delivered to the rear axle and the front one respectively, we could notice that they are similar but the main discrepancy is that one of them is positive while the other one is negative. From the magnitude point of view, the output power towards the front axle (***Pf***) continuously decrease during the take off stage down to a minimum of about -50 kW.
It is followed by an increase of its value during the towed vehicle braking stage until reaching positive values. It means that the front axle becomes, briefly, a driving axle.
As the theory predicts, a difference between the tire radii (between the axles) induces self-generated power. When the rolling resistance is high, the power towards the front axles briefly becomes positive, at the very end of the test.
Studying the researched parameters when the tire radius difference is of 0.06 m (figure 4 – a), we could notice a similar behavior to the previous case (when the difference was of 0.03 m). The only difference consists of the larger magnitude of the power towards the front axle, also negative: $P_{f\,\min} \approx 52$ kW.

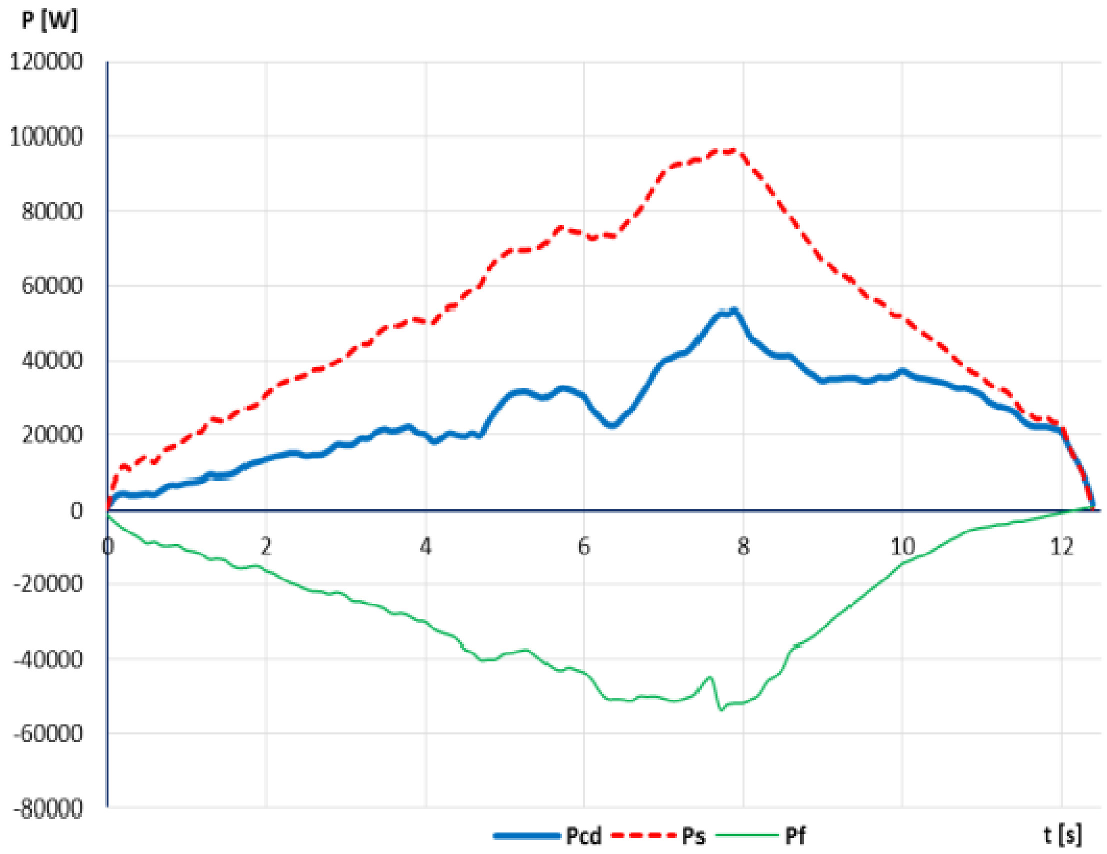

Figure 4. a Power fluxes variation within the vehicle's driveline for $\Delta r = 60mm$

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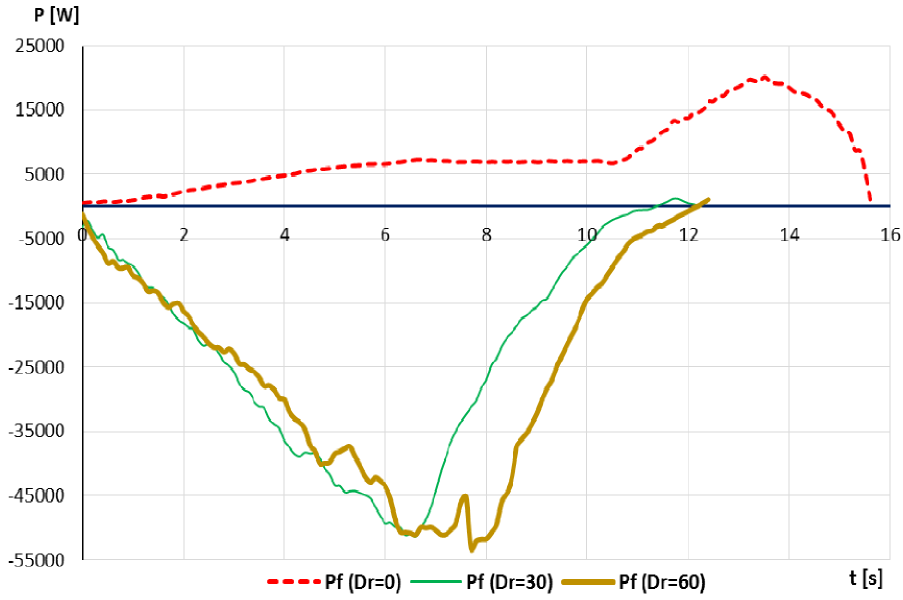

Figure 4. b Power fluxes variation within the vehicle's driveline for all the cases $\Delta r = 0mm$, $\Delta r = 30mm$, $\Delta r = 60mm$ when rolling on concrete

5. CONCLUSION

The mathematical model given in this paper was confirmed by the large amount of data achieved during the developed tests. The main problems within the 4WD drivetrains are the self-generated torque and consequently the looping power that occurs.
Examining the charts 3 şi 4 we could notice:
- the self-generated power didn't occur when the tire radii were the same;
- the self-generated power occurred when there was a difference between the tire radii;
- the self-generated power increases with the tire radii difference;
- when the tire rolling difference increases, the slope of the power on the output shaft towards the front axle increases.

Whenever a difference between the tire rolling radii occurs, a cinematic misfit occurs and it becomes responsible of the power loops occurrence at the longitudinal level of an all terrain vehicle driveline.

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ANALYSES AND STATISTICS ON THE FREQUENCY AND THE INCIDENCE OF TRAFFIC ACCIDENTS WITHIN DOLJ COUNTY

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Abstract. *Within the entire territory of the EU Member States, traffic accidents cause yearly over 37,000 deaths and about 1.2 million injured, as well as damages estimated, annually, at 145 billion Euros. Under the circumstances, a series of strategies and measures have been adopted over the past two decades in order to reduce the number of victims caused by traffic accidents, hence preventing some of the worst consequences. Aiming at improving road traffic safety, joint efforts have been carried out to estimate and identify those factors leading to the occurrence of accidents, in order to adopt and implement the most appropriate measures to reduce the risk of collisions. The dynamics of a traffic accident is different from one case to another; thus, safety measures can only be enforced following in-depth analyses on the occurrence of traffic accidents. The main objective of our research study is to design and develop an analysis model on the occurrence and the circumstances leading to traffic accidents, considering at the same time some main characteristics of the driver, such as age, gender, etc.*

Key-words: traffic accidents, statistical analysis,

1. INTRODUCTION

Throughout the territory of the European Union Member States, traffic accidents cause each year over 37,000 deaths and about 1.2 million injuries, as well as damages estimated, annually, at 145 billion Euros. Within this context, a series of strategies and measures have been adopted in the past decades to reduce the number of victims caused by traffic accidents, aimed also at preventing some of the worst consequences. The establishment of a European database on the incidence of traffic accidents has secured a higher flexibility and a maximum potential with regard to the analysis of the information contained within the system, and, opened up a whole range of new possibilities in the field of accidents' analysis [1]. As indicated in [6], a research study carried out by NHTSA (National Highway Traffic Safety Administration) set to feature general vehicle traffic crushes that occur at intersections.
Accordingly, special attention was paid to such instances triggered by the interdependence between the critical reason and some key crash factors such as drivers' gender and age, traffic control devices, critical pre-crash events, and weather conditions. Further research studies undertaken by NHTSA, as in [7], put forward an overview on traffic accidents incidents to highlight that over the past 10 years, there has been a reduction of nearly 25 percent in the number of fatalities.
In [3], Goswami et al. validates another method applied to the analysis and statistics of traffic accidents occurrence. Here the author departs from the premises that accidents are uniformly distributed over the year and over the seasons. How traffic accidents occur and their consequences stand as a major field of research, both at national and international level. Thus, at national level, various specialists have sought to determine the parameters of the crash tests between a vehicle and a pillar, at a speed of 50 km/h, as in [2]. Another research project, carried out in [4], was to study frontal vehicle collisions using high-speed videos, in order to assess the injury criteria of the occupant.
Vehicle-pedestrian collisions, as well as vehicle- bicycle driver collusions have been studied in [5], in order to determine the differences regarding the head injury risk.

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In compliance with both global and European-level organizational principles, the databases designed to provide statistics on the incidence of road accidents can be classified according to a number of various factors such as: type of victim, type of road, atmospheric conditions, and type of collision.
A statistical analysis with regard to the number, the type of injury and the frequency rates of traffic accidents was carried out between 2012 and 2016, within Dolj County. Hence, in cooperation with the Dolj County Police Inspectorate, we set up to design and develop a statistical analysis on the incidence of traffic accidents. Special attention was paid both to the steadily increasing incidence rate and the causes that led to traffic accidents within Dolj County during 01.01.2012 - 31.12.2016.

2. STATISTICAL ANALYSIS

In line with the official data provided by the Road Traffic Department of the Dolj County Police Inspectorate, during the previous 5 years, an average of more than 800 accidents was recorded per year, of which, approximately 260 accidents had serious consequences. To further develop our statistical analysis, we took into consideration a series of highly influential factors as we aimed to establish a series of traffic accidents particularities.

Table 1.
Total number of traffic accidents between 2012 and 2016

	2012	2013	2014	2015	2016
Total	**882**	**845**	**804**	**810**	**802**
Serious	269	268	265	262	259
Mild	613	577	539	548	543

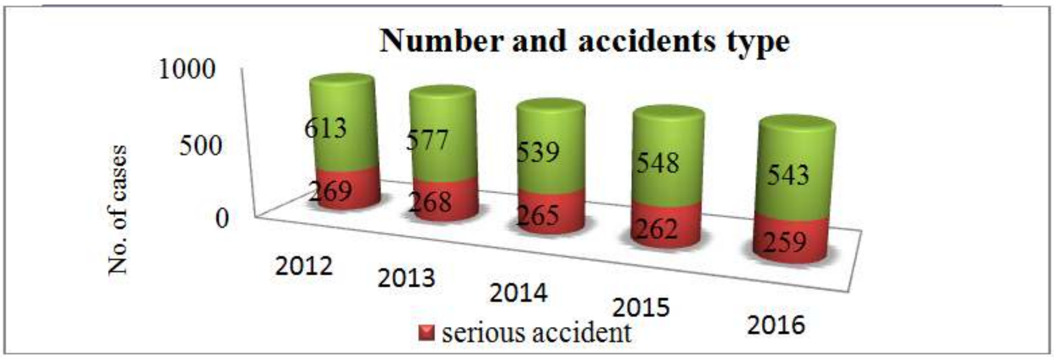

Figure 1. Total number of traffic accidents between 2012 and 2016

1.1. Type of injuries

The first step of our statistical analysis was to outline the total number of the individuals and of the vehicles involved in traffic accidents within the time span under investigation. Then, we established a classification of the two major factors. Also, we developed a categorization of the type of injuries caused, on an incident scale from minor injuries to deaths. An overview of the data provided by the Road Traffic Department, Dolj County Police Inspectorate, indicated that the lowest number of deaths caused by road accidents produced within Dolj County, i.e. 55, was registered in 2013. In 2012, the Road Traffic Department registered 70 deaths, in 2014 another 73 victims were the result of road accidents, in 2015 the number of victims went up to 75, and, last year there were registered 67 deaths.

Table 2.
The number and the type of injuries caused in 2014

2014	Vehicle	Dead	Serious injuries	Mild injuries
mild	779	0	0	669
serious	378	73	244	81
Total	**1157**	**73**	**244**	**750**

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Of a total number of 804 traffic accidents recorded in 2014, the authorities recorded 1067 injuries. Hence, we could establish that approximately 70% of the individuals involved in a traffic accident suffered minor injuries, 23% were seriously injured, and 7% of the traffic participants involved, died following a traffic accident.

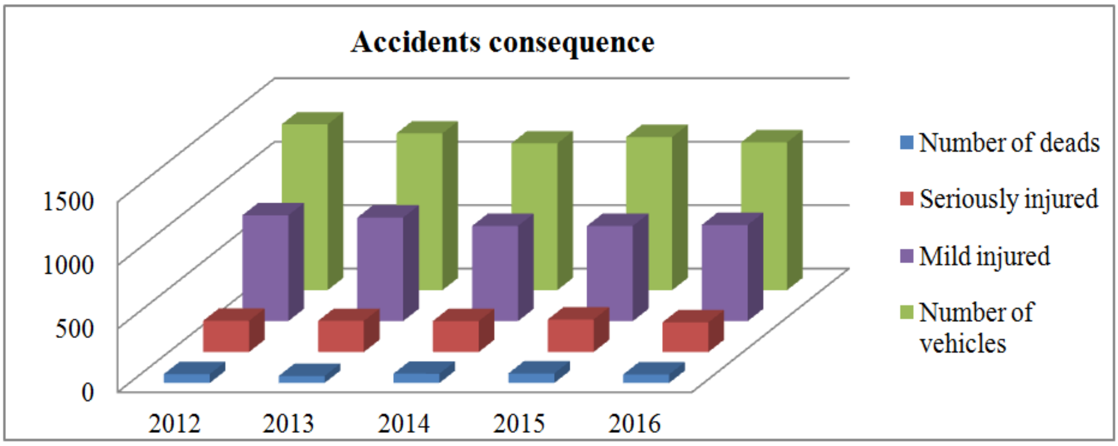

Figure 2. Consequences of traffic accidents

1.2. Areas with higher risks of accidents

By developing a model applied to classify the road accidents recorded within Dolj County between 2012 and 2016, we also aimed at identifying the area where each accident occurred. Hence, we could map those areas with higher risks of accidents, the so-called black spots. Following this pattern, out of the total number of events, we first sought to identify and classify those accidents occurring within localities, and, then, the ones occurring outside localities. The number of the traffic accidents registered was established in accordance with the type of the road where they occurred, i.e. communal, county and national roads, streets, and other types of roads. Also, within this classification we considered the characteristics of the road, as follows: road in curve, intersection, straight roads, and bridges. Weather conditions at the time of the collision have a particular influence on the occurrence of traffic accidents, as well as some specific characteristics concerning the adhesion of the road at the moment of collision.

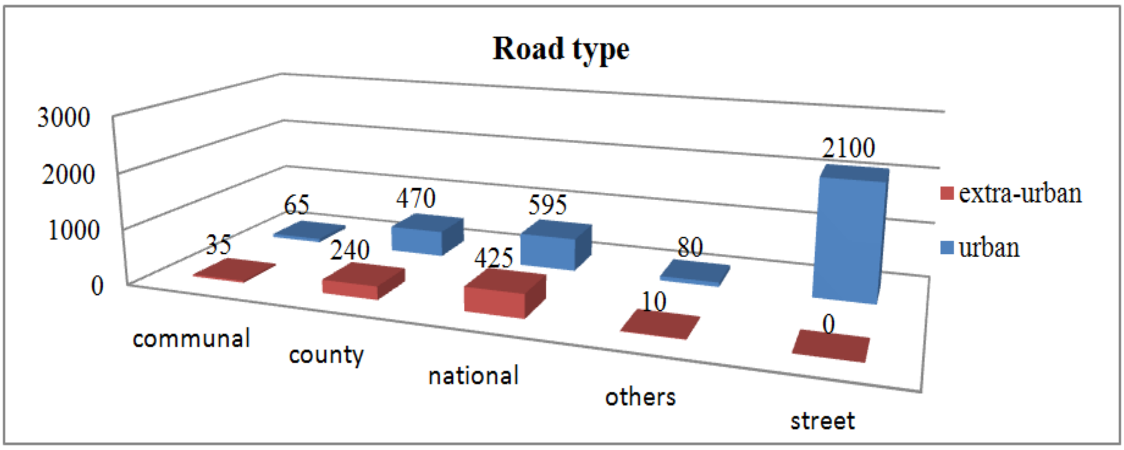

Figure 3. Road type - traffic events from 2012 to 2016

Our statistical analysis indicates that out of the total number of traffic accidents recorded over a calendar year, more than two thirds represent mild accidents. Thus, a first conclusion following our research study shows that the impact speed of the vehicles has been reduced in recent years, being adapted to the conditions of travel within localities. Figures 3 and 4 below validate our results, taking into account the fact that traffic accidents occur in the highest percentage within localities.
As far as road characteristics are concerned, we reached the conclusion that most traffic accidents occur on straight roads, without bumps, both within and outside localities.

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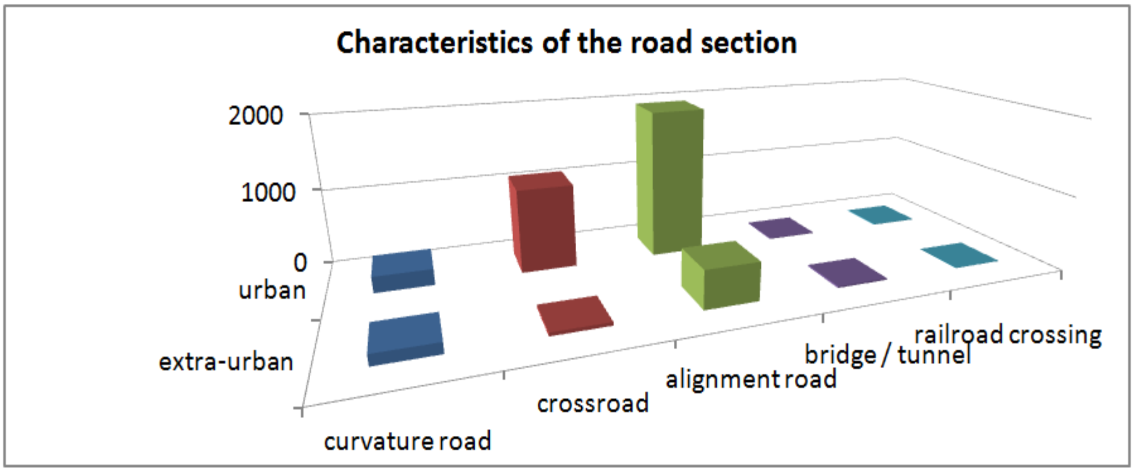

Figure 4. Traffic accidents localization within 2012 – 2016

2.3. Driver characteristics influence

Road traffic safety is influenced by three major categories of factors: the technical condition of the vehicle, the behaviour and the characteristics of the driver, road and visibility conditions. The data provided by the Road Traffic Department - Dolj County Police Inspectorate did not contain information about the technical condition of the vehicles involved in the accidents, thus we could not input and consider this criterion to our analysis. However, following the results of our analysis, we could establish the influence of the driver's characteristics and the road conditions on the risk of accidents incidence.

Concerning the type of traffic participants involved in traffic accidents within Dolj County, our statistical analysis indicated that a considerable frequency rate of victims was registered among vehicle drivers. The age and gender group of the participants in relation to the occurrence of the traffic accidents registered in Dolj County between 2012 and 2016 was also a key criterion input in our statistical analysis. Thus, we classified and further investigated 8 age groups.

Table 3.
Traffic accidents victims on age groups

2012	**0->10**	**11->20**	**21->30**	**31->40**	**41->50**	**51->60**	**61->70**	**over 70**	**total**
mild	*53*	*213*	*376*	*263*	*189*	*146*	*79*	*48*	*1367*
serious	*20*	*90*	*154*	*124*	*78*	*83*	*38*	*37*	*624*
total	**73**	**303**	**530**	**387**	**267**	**229**	**117**	**85**	**1991**
2013	**0->10**	**11->20**	**21->30**	**31->40**	**41->50**	**51->60**	**61->70**	**over 70**	**total**
mild	*58*	*152*	*328*	*286*	*185*	*136*	*79*	*64*	*1288*
serious	*20*	*89*	*157*	*106*	*86*	*77*	*57*	*34*	*626*
total	**78**	**241**	**485**	**392**	**271**	**213**	**136**	**98**	**1914**
2014	**0->10**	**11->20**	**21->30**	**31->40**	**41->50**	**51->60**	**61->70**	**over 70**	**total**
mild	*54*	*171*	*314*	*217*	*184*	*132*	*95*	*43*	*1210*
serious	*16*	*57*	*145*	*100*	*116*	*83*	*59*	*41*	*617*
total	**70**	**228**	**459**	**317**	**300**	**215**	**154**	**84**	**1827**
2015	**0->10**	**11->20**	**21->30**	**31->40**	**41->50**	**51->60**	**61->70**	**over 70**	**total**
mild	*51*	*189*	*335*	*237*	*188*	*141*	*87*	*45*	*1273*
serious	*15*	*63*	*158*	*98*	*107*	*87*	*64*	*42*	*634*
total	**66**	**252**	**493**	**335**	**295**	**228**	**151**	**87**	**1907**
2016	**0->10**	**11->20**	**21->30**	**31->40**	**41->50**	**51->60**	**61->70**	**over 70**	**total**
mild	*63*	*183*	*329*	*226*	*191*	*137*	*93*	*39*	*1261*
serious	*11*	*53*	*143*	*97*	*113*	*88*	*53*	*43*	*601*
total	**74**	**236**	**472**	**323**	**304**	**225**	**146**	**82**	**1862**

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The most vulnerable age group registered following the traffic accidents that occurred within the period 2012-2016 in Dolj County ranges from 21 to 30 years. The result of our statistical analysis in relation to the distribution of the number of victims by age can be explained by the lack of experience of the drivers.

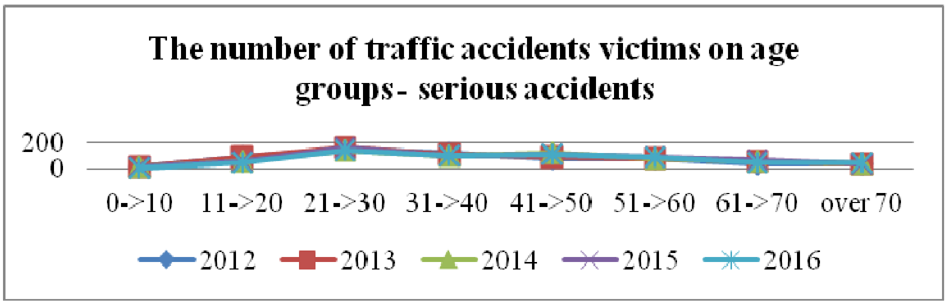

Figure 5. The age group of the victims caused by serious traffic accidents

Also, following the statistical analysis carried out we could note that the percentage of male victims was over 70% of the total number of injured persons, within the time interval under investigation. Thus, the results of our statistical analysis with regard to the victims' frequency rates in relation to the gender of the participants indicate the higher frequency of larger number of male drivers compared to a relatively low number of female drivers.

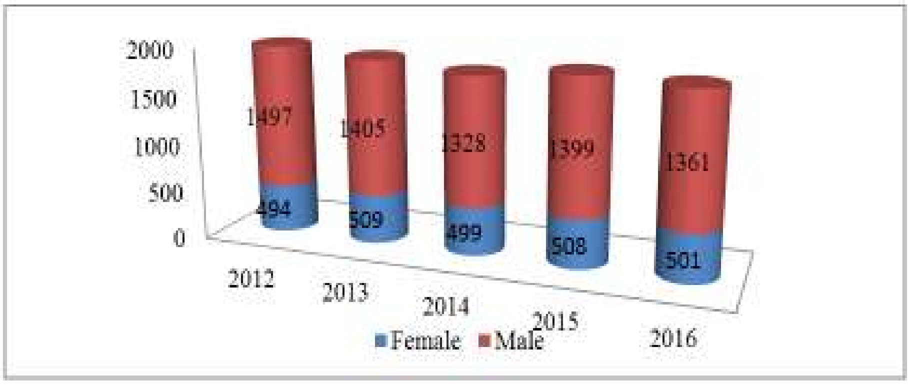

Figure 6. Traffic accident participants distributed on gender within 2012-2016

2.4. Type of collision

Furthermore, another important criterion that was considered in the classification of the traffic accidents was the type of collision. The most common types of collisions we have identified are: front-rear collisions, front collisions, lateral collisions, skidding, vehicle-pedestrian collisions, overturning, vehicle-obstacle collisions, and collision on straight-roads. Then, we sought to establish the frequency rate for each type of collision.

Table 4.
Type of collision for the mild accidents occurred between 2012 and 2016

	Collision type	2012	2013	2014	2015	2016
Mild accidents	front - rear	68	61	39	65	57
	frontal	18	30	18	23	34
	lateral	152	132	127	129	133
	skidding	22	10	6	15	17
	pedestrian	216	218	221	193	201
	rollover	44	45	32	34	23
	obstacle	48	46	54	51	36
	snap	15	9	7	13	19
	others	30	26	35	25	23

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As indicated by the diagram illustrated in Figure 7, we reach the conclusion that car-pedestrian collisions represent a considerable share of the total number of accidents occurring within the period 2012 - 2016 in Dolj County. The second place is occupied by lateral collisions, followed by front - rear collisions.

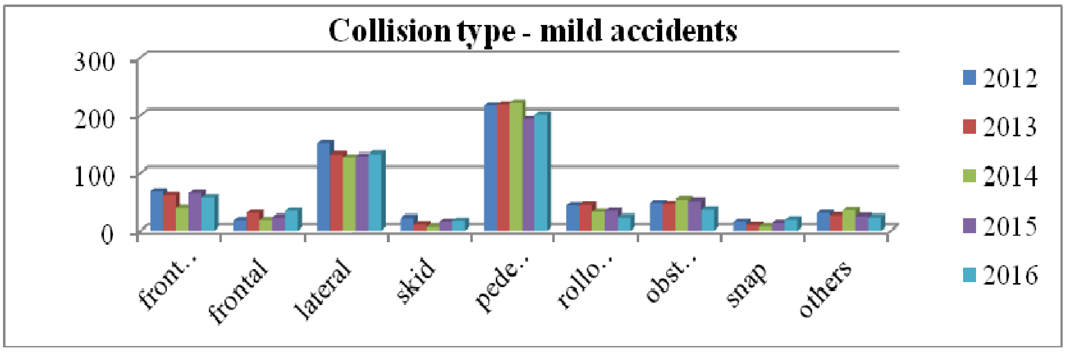

Figure 7. Type of collision for the mild accidents occurred between 2012 and 2016

Table 5.
Type of collision for the serious accidents occurred in 2013

2013	Type of collision	No. of accidents	Serious injuries and deaths
SERIOUS	front - rear	17	16
	frontal	19	23
	lateral	41	51
	skidding	4	4
	pedestrian	115	117
	rollover	25	31
	off-road obstacles	35	43
	snap	3	3
	others	9	10

We can see that the distribution of severe injuries and deaths is directly proportional to the number and type of collision. The less frequent accidents are caused due to skidding, thus, generating a considerably reduced number of server injuries or deaths. An account of serious bodily injuries and deaths in 2013 indicates that most accidents, i.e. 115 cases, were due to car-pedestrian-type collisions.
A high number of serious injuries occurred due to lateral-type collisions - 51, followed by collisions caused mainly by crushing against an obstacle outside the roadway - 43.
For 2013, the distribution of serious injuries and deaths is similar to the one registered in 2012, though with some variations.
Thus, in 2013 the lowest number of deaths and serious injuries occurred in the case of collisions caused due snapping and skidding.
For 2014, the distribution of the number of serious injuries and deaths among car-pedestrian-type collisions, i.e. 119, is maintained.
This time, however, the second place is occupied by collisions caused mainly due to crushes against off-road obstacles - 48, followed by injuries and deaths following frontal collisions - 31.
In 2015 there were registered 122 serious injuries and deaths due to car-pedestrian collisions, 45 collision injuries occurred due to vehicles crushing against off-road obstacles and 41 collisions due to lateral collisions.
For the year 2016, there were 113 serious injuries and deaths due to car-pedestrian collisions.
The second place, in terms of injury number, with 45 cases, is occupied by collisions following vehicles overturning, and, 38 cases of serious injuries and deaths were the result of lateral collisions.
Aiming to provide a highly reliable comparative analysis, we input in the diagram illustrated in figure 8 additional data with regard to the type of collision, the number of serious injuries and the deaths according to the type of collision for the entire period of time investigated, i.e. the period 2012-2016.

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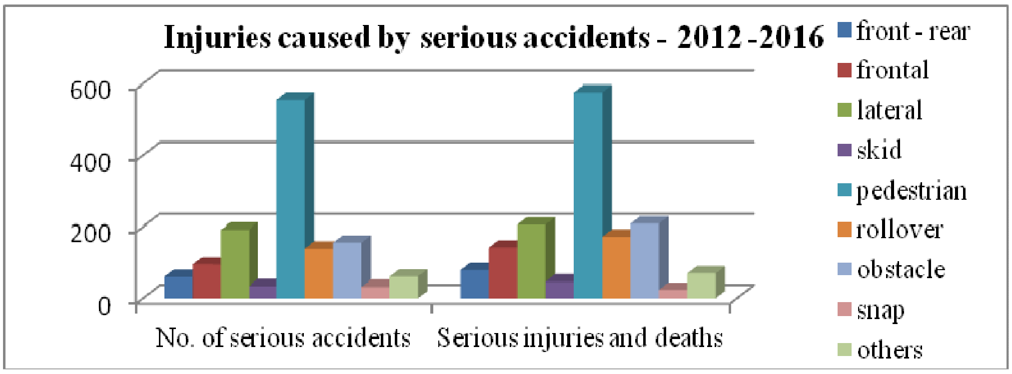

Figure 8. Time of collision and serious injuries produced during 2012-2016

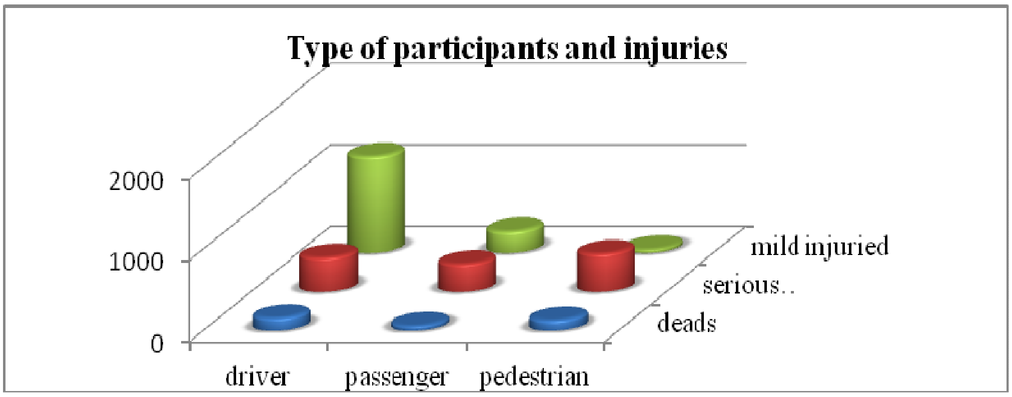

Figure 9. Type of participants and injuries produced during 2012-2016

The rate indicating the most frequently mild injuries, i.e. 1187 cases, as well as the highest number of deaths (160), within the time period under investigation, was recorded among vehicle drivers, while the most serious injuries occurred among pedestrians, i.e. 473 cases of serious injuries.

3. RESULTS

Embarking upon the analysis of traffic accident incidence we carried out a statistical analysis regarding the number, injury types and frequency rates of the traffic accidents occurred between 2012 and 2016, in Dolj County. According to our investigation, we highlight the following main conclusions:

✓ during the period analysed, i.e. 2012-2016, we registered an average of more than 800 accidents per year, of which approximately 260 accidents had serious consequences;

✓ most traffic accidents occur, especially, in localities (highest rates), on straight roads, with no bumps;

✓ a survey concerning serious injuries and deaths registered indicates that most accidents occurred due to vehicle-pedestrian-type collisions;

✓ an average of more than 90% of road accidents within a year occur under normal weather conditions;

✓ more than 1100 victims registered annually are drivers;

✓ the most vulnerable age group involved in mild traffic accidents, registered between 2012-2016 in Dolj County, ranges from 21 to 30 years;

✓ the victims rate according to the participants gender indicates a percentage of over 70% among male participants of the total number of injured persons

4. CONCLUSION

Identifying and assessing the factors that contribute to traffic accident occurrence is of particular importance in drafting and implementing effective measures to increase road safety.

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Based on the statistical analysis carried out within Dolj County, we can further determine the areas prone to increased accident-risks, and, also, we can identify and feature those categories of drivers that exhibit higher risks to traffic accidents.

This work was presented at the International Congress of Automotive and Transport Engineering CAR 2017, Pitesti, Romania and it was published in Scientific Bulletin of University of Pitesti, Automotive Series, Issue 27 (ISSN 1453-1100).

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THE INFLUENCE OF THE TECHNOLOGICAL DEVIATIONS OVER THE VIBRATION INHERENT FREQUENCIES AT BENDING FROM THE THREE-SHAFTS TRANSMISSION

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Abstract: *The technological deviations of manufacturing and assembly , lead to the change of the kinematic parameters and at further efforts in the kinematic pairs of the cardan joint mechanism from the component of the poly-cardan transmissions.The variation of these forces is constant in the local systems of reference of the kinematic pairs and harmonic varies in the general system of reference in which one of the axes coincides with the rotation axis.The component of the forces that acts perpendiculary on the direction this axis is a permanent source of excitation, leading to the change of their own frequencies and vibration modes at bending. In this paper are deducted the connections between technological deviations ,the excitation forces and own frequencies and vibration modes at bending and based on the results of numerical application conclusions will be drawn.*

Key-words: *vibration, manufacturing technology, shafts, transmission*

1. INTRODUCTION

Based on the dynamic model with distributed mass, in the previous papers [1][4] were studied the free vibrations of the two-shafts transmission with elastic frame, with and without technological deviations, were determined the own frequencies and were represented the vibration modes at bending. Starting from the same model and the papers results [5][7], the excitations that appear because of technical deviations will be stimulated and by using the mathematical model that I will establish in this paper will be determined the frequencies and inherent modes of vibration at bending of the three-shafts transmission.

2. TECHNOLOGICAL DEVIATIONS IN PLÜCKER COORDINATES

The cardan shaft joint mechanism is a particular case of the 4r Symmetrical Spherical Quadrilateral Mechanism which being of third family it is multiple statically undetermined.

Determination of the reactions from rotation kinematic pairs A, B, C, D is done by means of the linear elastic calculus using the method of the relative displacements with the expression of displacements in Plücker coordinates [8].

The kinematic pair from A, in the case of elastically linear calculation, it is considered fixed and the technological deviation of the AB element, represented in Figure 1, is given in the local reference system *Bxyz*, by the small rotation angle $\overline{\theta}_B^l$ and by the small displacement $\overline{\delta}_B^l = B\overline{B}'$.

In Plücker coordinates [4][7][11] the deviations Δ_B^l , in the *Bxyz* local reference system is written:

$$\Delta_B^l = \left(\theta_{Bx}^l, \theta_{By}^l, \theta_{Bz}^l, \delta_{Bx}^l, \delta_{By}^l, \delta_{Bz}^l\right)^T . \tag{1}$$

where $\left(\theta_{Bx}^l, \theta_{By}^l, \theta_{Bz}^l, \delta_{Bx}^l, \delta_{By}^l, \delta_{Bz}^l\right)^T$ are the projections on the local axis of the vectors $\overline{\theta}_B^l, \overline{\delta}_B^l$.

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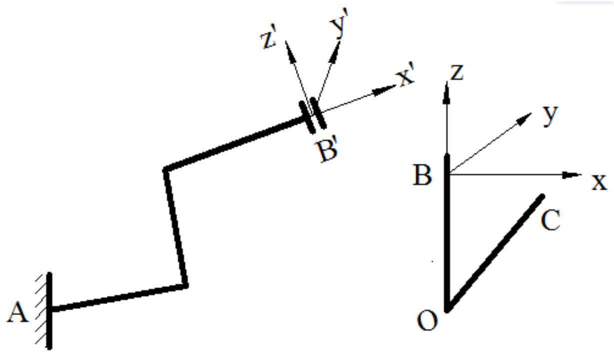

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Figure 1. The technological deviation of the element AB

3. THE MODEL WITH DISTRIBUTED MASSES FOR THREE-SHAFTS TRANSMISSION

The dynamic model is carried out on a three-shafts transmission used in the SUV field, whose construction model is shown in Figure 2.

Figure 2. The Constructive model

The constructive solution of the three-shafts transmission is associated with the mechanical model presented in Figure 3.
In the sections A, E and H are situated the elastic bearings with the elastic constants k_A, k_B, k_C, and the harmonic excitation forces R_A, R_B, R_C of amplitude $\widetilde{R}_A, \widetilde{R}_E, \widetilde{R}_H$ activating in the sections A, E, H.

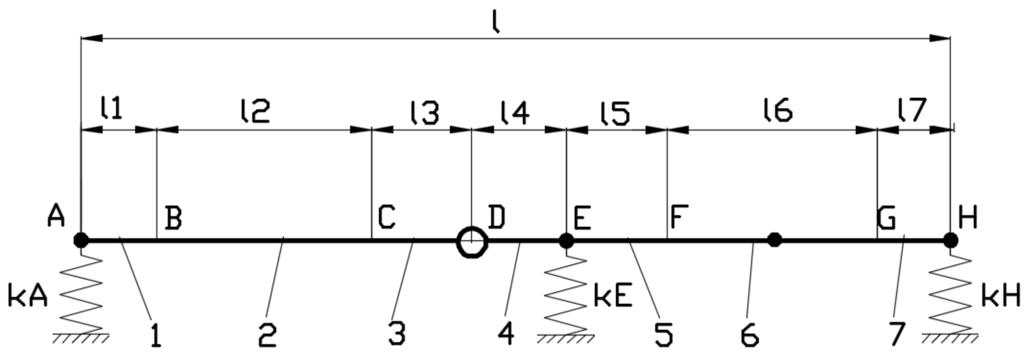

Figure 3. The equivalent mechanical model.

The dynamic modeling with distributed masses of the three-shafts transmission as well as the representation of the first two vibration modes at bending is presented in Figure 4.

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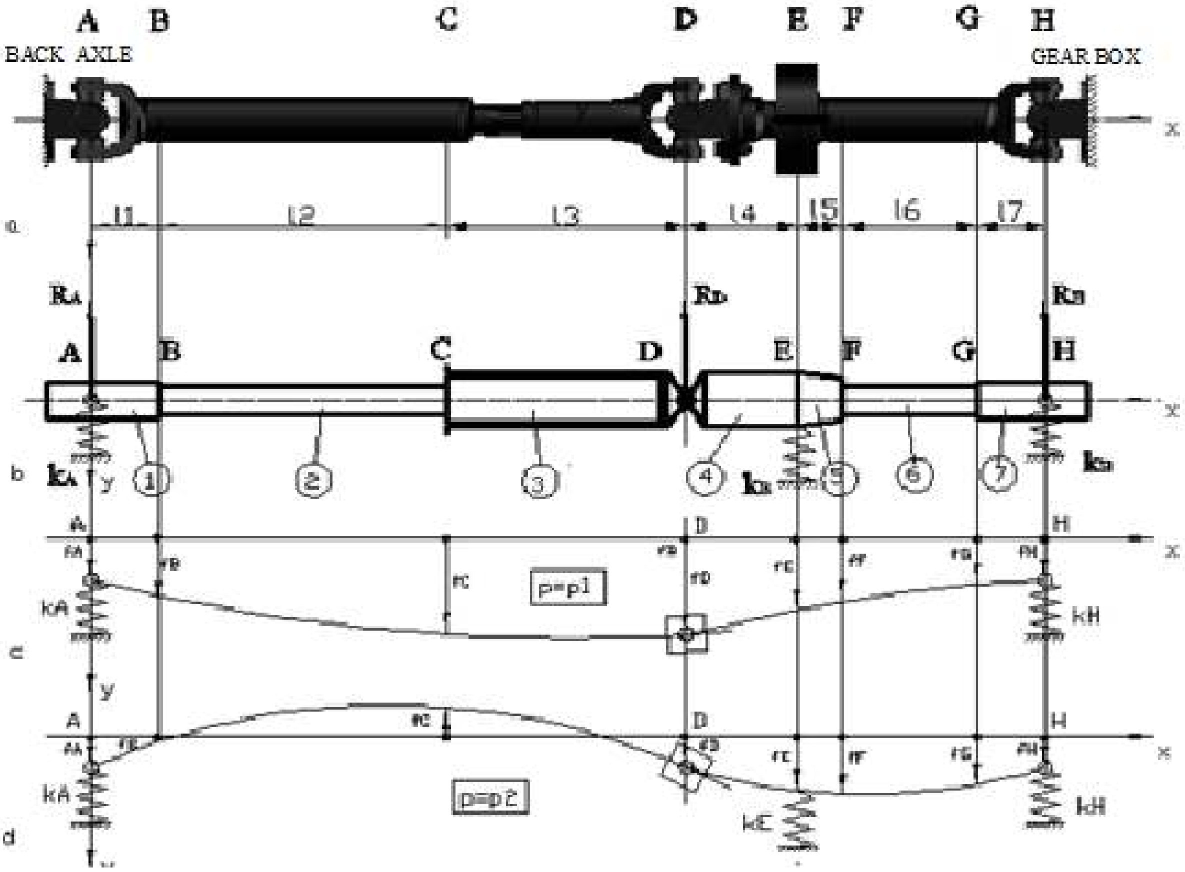

Figure 4. The dynamic modeling with distributed masses and first two vibration modes at bending

Next the notations are used [9][11]:

a) f_i, θ_i, M_i, F_i - as the deflection, rotation, deflection moment and the cutting force from the current section

b) $\Delta_i \; ;\Delta_A \; ;\Delta_B \; ;\Delta_C \; ;\Delta_D \; ;\Delta_E \; ;\Delta_F \; ;\Delta_G \; ;\Delta_H$ - state vectors, defined by the relations:

$$\begin{aligned}
&\Delta_i = (f_i, \theta_i, M_i, F_i)^T \; \Delta_A = (f_A, \theta_A, M_A, F_A)^T \; \Delta_B = (f_B, \theta_B, M_B, F_B)^T \\
&\Delta_C = (f_C, \theta_C, M_C, F_C)^T \; \Delta_D = (f_D, \theta_D, M_D, F_D)^T . \Delta_E = (f_E, \theta_E, M_E, F_E)^T \\
&\Delta_F = (f_F, \theta_F, M_F, F_F)^T \; \Delta_G = (f_G, \theta_G, M_G, F_G)^T . \Delta_H = (f_H, \theta_H, M_H, F_H)^T
\end{aligned} \tag{2}$$

c) $x_i, \rho_i, A_i, E_i, I_{yi}$ - respectivelly, the length, density, area, transverse modulus of elasticity and geometrical moment of inertia mainly for the section corresponding to the index *i=1,2,3,…7* .

d) The parameters $\alpha_i ; z_i$ defined by the relations:

$$\alpha_i = \sqrt[4]{p^2 \frac{\rho_i A_i}{E_i Iyi}} ; z_i = \alpha_i \cdot x_i . \tag{3}$$

where *p* is the vibration inherent pulse.

e) *chz, shz* - sin and cos the hyperbolic functions

$$ch(z_i) = \frac{e^{z_i} + e^{-z_i}}{2} \quad ; sh(z_i) = \frac{e^{z_i} - e^{-z_i}}{2} \quad . \tag{4}$$

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f) $f_j(z_i)$, j=1,2,3,4 the Krâlov functions defined by relations:

$$f_1(z_i)=\frac{ch(z_i)+cos(z_i)}{2};\ f_2(z_i)=\frac{sh(z_i)+sin(z_i)}{2}$$
$$f_3(z_i)=\frac{ch(z_i)-cos(z_i)}{2};\ f_4(z_i)=\frac{sh(z_i)-sin(z_i)}{2}. \tag{5}$$

g) $F(z_i)$ – the Krâlov matrixes defined by relations:

$$F(z_i)=\begin{pmatrix} f_1(z_i) & f_2(z_i) & f_3(z_i) & f_4(z_i) \\ f_4(z_i) & f_1(z_i) & f_2(z_i) & f_3(z_i) \\ f_3(z_i) & f_4(z_i) & f_1(z_i) & f_2(z_i) \\ f_2(z_i) & f_3(z_i) & f_4(z_i) & f_1(z_i) \end{pmatrix}. \tag{6}$$

h) α , α^{-1} – for diagonal matrixes:

$$\alpha=\begin{pmatrix} 1 & 0 & 0 & 0 \\ 0 & \frac{1}{\alpha} & 0 & 0 \\ 0 & 0 & -\frac{1}{\alpha^2 EI_w} & 0 \\ 0 & 0 & 0 & -\frac{1}{\alpha^3 EI_w} \end{pmatrix};\alpha^{-1}=\begin{pmatrix} 1 & 0 & 0 & 0 \\ 0 & \alpha & 0 & 0 \\ 0 & 0 & -\alpha^2 EI_w & 0 \\ 0 & 0 & 0 & -\alpha^3 EI_w \end{pmatrix}. \tag{7}$$

$R_i, i-1,2,3,...,7$, the field matrixes of sections, [7]:

$$R_i=\alpha_i^{-1}\cdot F(\alpha_i x_i)\cdot\alpha_i. \tag{8}$$

By assimilating the bearings from sections A, D, H with the joints, results that

$$M_A=M_D=M_H=0 \tag{9}$$

and by also taking into account the elastic brackets results

$$F_A=k_A f_A; F_E^d=F_E^s+k_E f_E; F_H=-k_H f_H \tag{10}$$

In the absence of exciting force the obtained state vectors are:

$$\{\Delta_A\}=\begin{bmatrix}\tilde{f}_A\\ \tilde{\theta}_A\\ 0\\ k_A f_A\end{bmatrix};\{\Delta_D^s\}=\begin{bmatrix}\tilde{f}_D\\ \tilde{\theta}_D^s\\ 0\\ F_D\end{bmatrix};\{\Delta_D^d\}=\begin{bmatrix}\tilde{f}_D\\ \tilde{\theta}_D^d\\ 0\\ F_D\end{bmatrix};\{\Delta_E^s\}=\begin{bmatrix}\tilde{f}_E\\ \tilde{\theta}_E\\ 0\\ F_E^s\end{bmatrix};\{\Delta_E^d\}=\begin{bmatrix}\tilde{f}_E\\ \tilde{\theta}_E\\ 0\\ F_E^s+k_E f_E\end{bmatrix};\{\Delta_H\}=\begin{bmatrix}\tilde{f}_H\\ \tilde{\theta}_H\\ 0\\ -k_H f_H\end{bmatrix} \tag{11}$$

where the indices s,d refers to the left and right sections.

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The equalities results

$$\begin{bmatrix}\tilde{f}_D\\ \tilde{\theta}_D^s\\ 0\\ F_D\end{bmatrix}=[R_3][R_2][R_1]\begin{bmatrix}\tilde{f}_A\\ \tilde{\theta}_A\\ 0\\ k_A f_A\end{bmatrix};\begin{bmatrix}\tilde{f}_E\\ \tilde{\theta}_E\\ M_E\\ F_E^s\end{bmatrix}=[R_4]\begin{bmatrix}\tilde{f}_D\\ \tilde{\theta}_D^d\\ 0\\ F_D\end{bmatrix};\begin{bmatrix}\tilde{f}_H\\ \tilde{\theta}_H\\ 0\\ -k_H f_H\end{bmatrix}=[R_7][R_6][R_5]\begin{bmatrix}\tilde{f}_E\\ \tilde{\theta}_E\\ M_E\\ F_E^s+k_E f_E\end{bmatrix} \tag{12}$$

that can be written under the form

$$[R_3][R_2][R_1]\begin{bmatrix}1&0\\0&1\\0&0\\k_A&1\end{bmatrix}\begin{bmatrix}f_A\\ \theta_A\end{bmatrix}-\begin{bmatrix}1&0\\0&0\\0&0\\0&1\end{bmatrix}\begin{bmatrix}f_D\\ F_D\end{bmatrix}-\begin{bmatrix}0\\1\\0\\0\end{bmatrix}\theta_D^d=\{0\}$$
$$[R_4]\begin{bmatrix}1&0\\0&0\\0&0\\0&1\end{bmatrix}\begin{bmatrix}f_D\\ F_D\end{bmatrix}-\begin{bmatrix}1&0&0&0\\0&1&0&0\\0&0&1&0\\0&0&0&1\end{bmatrix}\begin{bmatrix}\tilde{f}_E\\ \tilde{\theta}_E\\ M_E\\ F_E^s\end{bmatrix}-\begin{bmatrix}0\\1\\0\\0\end{bmatrix}\theta_D^d=\{0\}$$
$$[R_7][R_6][R_5]\begin{bmatrix}1&0&0&0\\0&1&0&0\\0&0&1&0\\0&0&0&1\end{bmatrix}\begin{bmatrix}\tilde{f}_E\\ \tilde{\theta}_E\\ M_E\\ F_E^s\end{bmatrix}-\begin{bmatrix}1&0\\0&1\\0&0\\-k_H&0\end{bmatrix}\begin{bmatrix}f_H\\ \theta_H\end{bmatrix}=\{0\} \tag{13}$$

By using the notations:

$$[E_1]=[R_3][R_2][R_1]\begin{bmatrix}1&0\\0&1\\0&0\\k_A&0\end{bmatrix};[E_2]=\begin{bmatrix}1&0\\0&0\\0&0\\0&1\end{bmatrix};[E_3]=\begin{bmatrix}0\\1\\0\\0\end{bmatrix};[E_4]=[R_4]\begin{bmatrix}1&0\\0&0\\0&0\\0&1\end{bmatrix}$$
$$[I_4]=\begin{bmatrix}1&0&0&0\\0&1&0&0\\0&0&1&0\\0&0&0&1\end{bmatrix};[E_5]=[R_7][R_6][R_5]\begin{bmatrix}1&0&0&0\\0&1&0&0\\0&0&1&0\\k_E&0&0&1\end{bmatrix};[E_6]=\begin{bmatrix}1&0\\0&1\\0&0\\-k_H&0\end{bmatrix} \tag{14}$$

the equations (13), can be written under the form:

$$[E_1]\begin{bmatrix}f_A\\ \theta_A\end{bmatrix}-[E_2]\begin{bmatrix}f_D\\ F_D\end{bmatrix}-[E_3]\theta_D^s=\{0\}$$
$$[E_4]\begin{bmatrix}f_D\\ F_D\end{bmatrix}-[I_4]\begin{bmatrix}\tilde{f}_E\\ \tilde{\theta}_E\\ M_E\\ F_D^s\end{bmatrix}-[E_6]\theta_D^d=\{0\}$$
$$[E_5]\begin{bmatrix}\tilde{f}_E\\ \tilde{\theta}_E\\ M_E\\ F_D^s\end{bmatrix}-[E_6]\begin{bmatrix}f_H\\ \theta_H\end{bmatrix}=\{0\} \tag{15}$$

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and with the notations:

$$\{\tilde{\Delta}\}=\left[f_A,\theta_A,f_D,\theta_D,\theta_D^s,\theta_D^d,f_E,\theta_E,M_E,F_D^s,f_H,\theta_H\right]^T \tag{16}$$

$$[E]=\begin{bmatrix} [E_1] & -[E_2] & -\{E_3\} & \{Q_{41}\} & [Q_{44}] & [Q_{42}] \\ [Q_{42}] & [E_4] & \{Q_{41}\} & -\{E_3\} & -[I_4] & [Q_{42}] \\ [Q_{12}] & [Q_{42}] & \{Q_{41}\} & \{Q_{41}\} & [E_5] & -[E_6] \end{bmatrix} \tag{17}$$

where by $[Q_{mn}]$ was noted the zero matrix with cu m lines and n columns, is obtained the homogeneous equation:

$$[E][\tilde{\Delta}]=\{0\} \tag{18}$$

that accepts the solution different of zero if:

$$det[E]=\{0\} \tag{19}$$

equation from which are determined the own pulsations.

In the case of harmonic excitations from A, D, H with the amplitudes $\tilde{R}_A,\tilde{R}_D,\tilde{R}_H$ in the equations (11) at the elements $k_A f_A$ is added $\tilde{R}_A$ at the element F_D added $\tilde{R}_D$ and at the element $-k_H f_H$ is added $-\tilde{R}_H$, and so the notations:

$$\{F_1\}=\begin{bmatrix} 0 \\ 0 \\ 0 \\ \tilde{R}_D \end{bmatrix}-[R_3][R_2][R_1]\begin{bmatrix} 0 \\ 0 \\ 0 \\ \tilde{R}_A \end{bmatrix};\{F_2\}=-[R_4]\begin{bmatrix} 0 \\ 0 \\ 0 \\ \tilde{R}_D \end{bmatrix};\{F_3\}=\begin{bmatrix} 0 \\ 0 \\ 0 \\ -\tilde{R}_H \end{bmatrix}$$

$$\{\tilde{F}\}=\left[\{F_1\}^T,\{F_2\}^T,\{F_3\}^T\right]^T \tag{20}$$

it is obtained the matrix equation:

$$[E]\{\tilde{\Delta}\}=\{F\} \tag{21}$$

from which results the column matrix of amplitudes:

$$\{\tilde{\Delta}\}=[E]^{-1}\{\tilde{F}\} \tag{22}$$

4. NUMERICAL APPLICATION

Consider a mobile three-shafts transmission equipping a SUV vehicle whose construction model is shown in Figure 3, for the following construction features that are known and mechanical:

$$k_A=k_H=85\cdot 10^6(N/m);k_E=2\cdot 10^6(N/m);$$
$$l_1=0{,}07(m);l_2=0{,}59(m);l_3=0{,}25(m);l_4=0{,}12(m);l_5=0{,}05(m);l_6=0{,}36(m);l_7=0{,}07(m);$$
$$A_1=A_3=A_4=A_5=A_7=19{,}6\cdot 10^{-4}(m^2);A_2=A_6=4\cdot 10^{-4}(m^2)$$
$$\rho_1=\rho_2=\rho_3=\rho_4=\rho_5=\rho_6=\rho_7=7800(kg/m^3). \tag{23}$$

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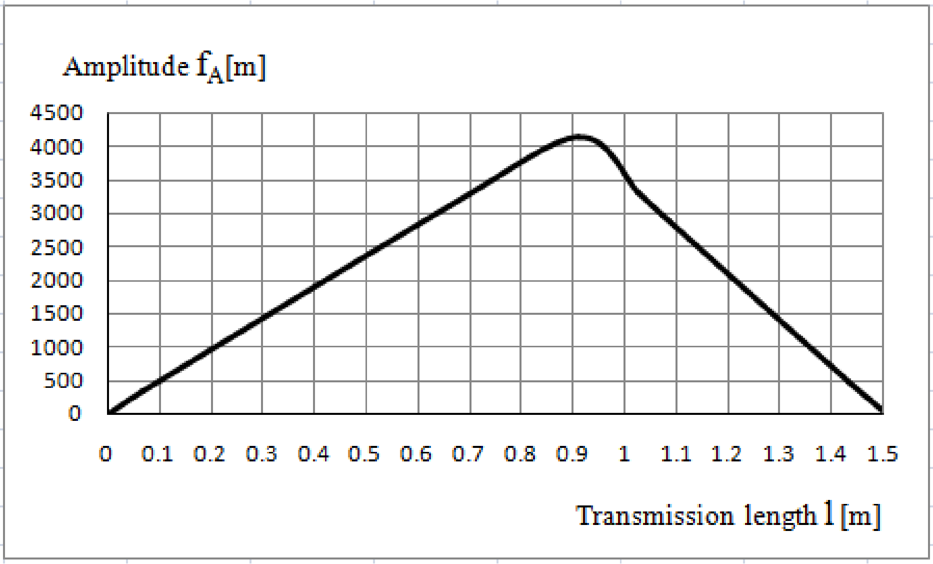
Based on an algorithm that I will present in future work and a computer program developed in Excel on obtained first and second pulsation own value p_1=155(s^{-1}), p_2=1072(s^{-1}). Corresponding to this pulse graphs were drawn at the bending vibration inherent modes shown in Figure 5.



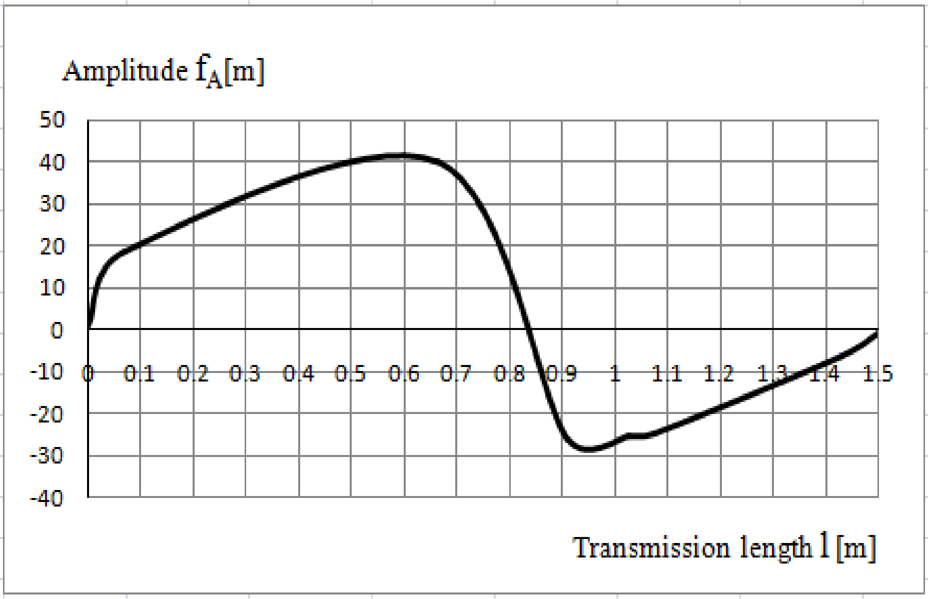


Figure 5. First and second pulsation own

5. CONCLUSION

The mathematical model presented, of the algorithm and of the program developed can be determinated the influence of each type of technological deviation an their own frequencies and vibration modes at bending of three-shafts transmissions.
The matrix approach allows us to directly determine the own pulsations and ways of vibration of the three-cardan transmissions, the method could be extended also to poli-cardan transmission.

This work was presented at the International Congress of Automotive and Transport Engineering CAR 2017, Pitesti, Romania and it was published in Scientific Bulletin of University of Pitesti, Automotive Series, Issue 28 (ISSN 1453-1100).

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The engineering of vehicles represents the engine of the global development of the economy.
SIAR tracks the progress of the automotive engineering in Romania by: the development of automotive engineering, the development of technologies, and road transport services; supporting the work of the haulers, supporting the technical inspection and of the garage; encouraging young people to have a career in the automotive engineering and road haulage; stimulation and coordination of activities that promote an environment that is suitable for continuous education and improving of knowledge of the engineers; active exchange of ideas and experience, in particular for students, master students, PhD students, and young engineers, and dissemination of knowledge in the field of automotive engineering; cooperation with other technical and scientific organizations, employers' and socio-professional associations through organization of joint actions, of mutual interest.
By the accession to FISITA (International Federation of Automotive Engineering Societies) since its establishment, SIAR has been involved in achieving an overall professional community that is homogeneous in competence and performance, interactive, dynamic, and competitive at the same time, oriented towards a balanced and friendly relationship between people and the environment; this action will be constituted as a challenge worthy of effort and recognition.
The insurance of a favorable framework for the initiation and the development of cooperation of the specialists in this field of activity allows for an efficient and easy exchange of information, specific knowledge and experience; it supports the cooperation between universities and between research centers and industry; it speeds up the process of implementing the new technologies, it simplifies the identification of training and specialization needs of the personnel involved in the engineering of motor vehicles, transport, and road safety.
In order to succeed, ever since its founding, SIAR has considered that the stress should be put on the production and distribution, at national and international level, of a publication of scientific quality.

Under these circumstances, the development of the scientific magazine of SIAR had the following evolution:

1. RIA – Revista inginerilor de automobile (in English: ***Journal of Automotive Engineers***)
ISSN 1222 – 5142
Period of publication: 1990 – 2000
Frequency: Quarterly
Total number of issues: 30
Format: print, Romanian
Electronic publication on: www.ro-jae.ro
Type: Open Access
The above constitutes series nr. 1 of SIAR scientific magazine.

2. Ingineria automobilului (in English: ***Automotive Engineering***)
ISSN 1842 – 4074
Period of publication: as of 2006
Frequency: Quarterly
Total number of issues: 46
(including the March 2018 issue)
Format: print and online, Romanian
Electronic publication on: www.ingineria-automobilului.ro
Type: Open Access
The above constitutes series nr. 2 of SIAR scientific magazine (Romanian version).

3. Ingineria automobilului (in English: ***Automotive Engineering***)
ISSN 2284 – 5690
Period of publication: 2011 – 2014
Frequency: Quarterly
Total number of issues: 16
(including the December 2014 issue)
Format: online, English
Electronic publication on: www.ingineria-automobilului.ro
Type: Open Access
The above constitutes series nr. 3 of SIAR scientific magazine (English version).

4. *Romanian Journal of Automotive Engineering*
ISSN 2457 – 5275
Period of publication: from 2015
Frequency: Quarterly
Total number of issues: 13 (March 2018)
Format: online, English
Electronic publication on: www.ro-jae.ro
Type: Open Access
The above constitutes series nr. 4 of SIAR scientific magazine (English version).

Summary – on March 31, 2018

Total of series:	*4*
Total years of publication:	*24 (11=1990 – 2000; 13=2006-2018)*
Publication frequency:	*Quarterly*
Total issues published:	*76 (Romanian), out of which, the last 29 were also published in English*