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ANALYSE OF USED OIL IN ORDER TO EMIT DIAGNOSIS INTERPRETATIONS OF THE DIESEL ENGINE OPERATION

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Abstract: *An engine faulty operation may be the result of improper use of fuels and lubricants, fact that can conduct to a decrease in lubricating characteristics and oil protection. Technical inspection of an engine is important step that cannot be postponed and it is considered an obligatory process in the warranty period, but especially post - warranty. Also, speaking about wear, the wear nature and also the wear degree of engine lubricant could help in determining of possible engine malfunctions, like: combustion chamber with faults caused by leaks, incomplete burned diesel fuel, improper thickness of the oil film, etc. In this paper the authors are presenting the possible contaminants of used oil, which will help to emit diagnostic interpretations of locomotive diesel engine functioning.*

Key-Words: *lubricants, used oil, diesel engine, diagnosis interpretation, tests, dilution, flash point, water content*

1. INTRODUCTION

A fair comment could be the next one: if it is not damaged, then it is not appropriate to intervene, but if a system does not work optimally then it must be repaired. In this particular moment, the diagnostic skills are entering the scene. It is compulsory to realize that a process is functioning with faults, by using the fact of understanding the system and then by applying knowledge with diagnostic skills, to be able to find out what's working abnormal. A key control model can be composed of six investigation steps of faults (figure 1).

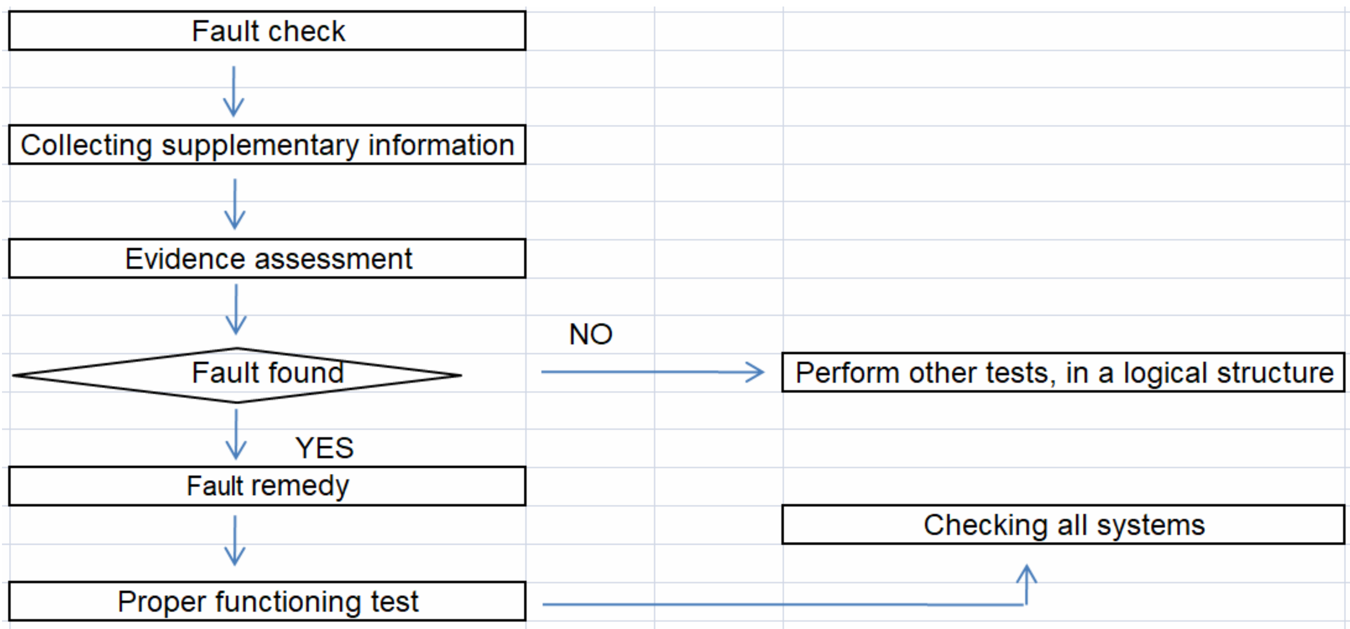

Figure 1. Faults investigation process

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Identifying the malfunction that appears when complex systems aren't functioning in parameters is kind of easy to do, if there is a certain amount of training. This involves the next steps: knowledge of the system where the fault appears, having the ability to compose and use of a diagnostic flowchart, determine their conditions of appearance: what the user (system) finds, identifying the malfunction or error that induced the symptom.
There is well known the fact that a lot of diesel engines have prematurely malfunctions caused by soot, water, glycol and diesel in engine lubricant. Each of contaminants is capable of producing premature damaging, sometimes unexpected, of the diesel engine. Problems become even more pronounced when there can be found combinations of contaminants, such as large soot loading oil and glycol, or dilution with diesel.

2. RESEARCH METHODOLOGY

A good lubricant analysis schema is composed by tests like Ferrography, or Scanning Electron Microscopy with Energy Dispersive X-Ray Analysis (SEM/EDX). The disadvantage is that both are time consuming and involves higher costs. The mentioned tests offer reliable information concerning the system wear, such as the composition of the wear particles, the emitting source and the degree of severity. Figure 2 shows typical oil analysis parameters by category and key analysis, in order to monitor machine wear, contamination and degradation [1].

Category		
Wear	Contamination	Degradation
- Fine or large wear metals	- Sand, dirt	- Oxidation, nitration, sulfation
- Distribution and particle amount	- Fuel dilution	- Viscosity
- Wear particle form	- Water	- Acid number / base number
	- Glycol or soot	
Key analysis		

Figure 2. Oil analysis parameters

Predictability in maintenance processes is a key factor and this requires real-time information and rapid feedback so reliability engineers can do maintenance practices on the spot. The actual state of the art in oil analysis instrumentation technologies also makes it possible and the result is performing on-site oil analysis saving years of training and testing or strong knowledge of oil chemistry.
The methodology consists in 3 steps (figure 3):
1. the first step: the methodology of collecting oil samples is important for obtaining good results for the diagnosing process; it is compulsory that the oil sample is collected while the diesel engine is functioning, not immediately, leaving to flow 2-3 liters; the oil sample contains only 1 liter of oil [4].
2. the second step is used oil analyze by specific equipments: flash point, dilution with diesel fuel, kinematic viscosity @ 40^{0}C, density, water content;
3. the third step involves diagnosing process.

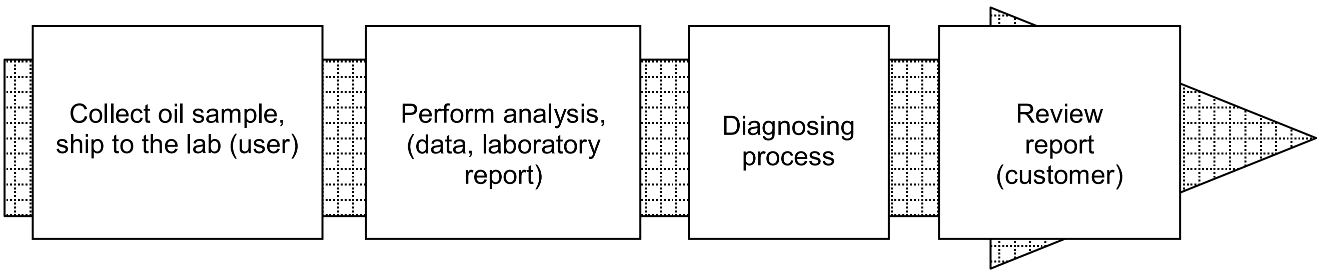

Figure 3. Off-site oil analysis scheme, [1]

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The case study is a middle rail mean of transport is produced in Romania and synthesizes some worldwide achievements in the construction of this type of locomotive. This locomotive is designed to handle heavy supply of services for passengers and freight hauling on secondary lines.
The diesel hydraulic locomotive is powered by a diesel engine with 6 cylinders, four-strokes, Sulzer type LDA 28 B, working with direct injection, turbocharged with a turbo 250 VTR, 1250 HP (figure 4) [2].

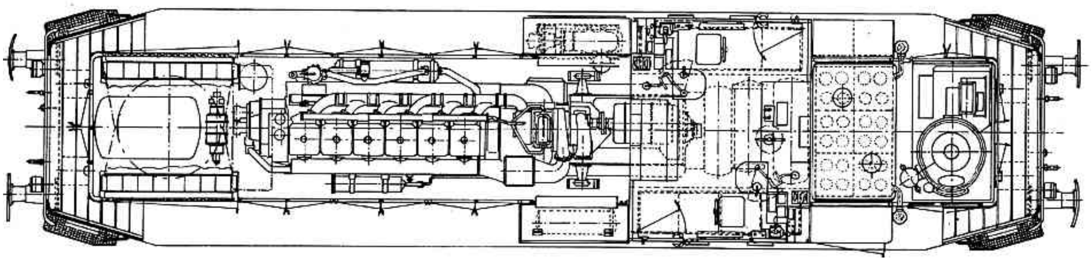

Figure 4. Diesel hydraulic locomotive

In the case of DHLe, in service oil analysis require the determination of oil change period of time and identifying the operational faults. The daily check of the oil or every two day in terms of diluting and water content is an important process because that prevents negative influences of the factors listed above. Aging and oil soiling are tracked monthly by performing oil analysis in a laboratory on samples taken from the diesel engine [3]. For a diesel engine operation, even the best oil is submitted to the process of aging and soiling. For that reason the entire amount of oil must be replaced with new oil. In our example, the oil change is programmed to be done at the end of 36 months or maximum 150.000 km. Oil change must be replaced faster in case the data obtained during regular analyzes reached values with allowable limits [8], such as presented in Table 1.

Table 1.
Admissible value for oil properties

Property	Admissible limit
Neutralizing number	2 mg KOH/g
Oil dilution (fuel contamination)	6 %
Flash point	180 ^{0}C
Water content	0.2 %

For the 6 LDA 28 B diesel engine is absolutely necessary to be used high-performance oils (75 – 80%) with additives antioxidants, anti-corrosion, anti-wear, alkaline detergent, dispersant and anti-foaming (15 – 25%) [3]. A key parameter in determining the condition of used oil or specification of new oil is viscosity measurements by oil analysis laboratories.
Thus, increase/decrease in viscosity of used oils could be a sign of oil degradation or oil dilution.
In paper [10] the authors presented the equipment used for tests. This is an automatic system which operates with glass capillary viscometers in order to determine the kinematic viscosity of Newtonian fluids. The equipment can be used for fresh and used mineral and synthetic oils and for any fluid but taking into account the viscosity range of the instrument. Standard Test Method for Kinematic Viscosity of Transparent and Opaque Liquids D 445 is under the jurisdiction of ASTM Committee D-2 on Petroleum Products and Lubricants. This test method specifies a procedure for the determination of the kinematic viscosity of liquid petroleum products, both transparent and opaque, by measuring the time for a volume of liquid to flow under gravity through a calibrated glass capillary viscometer. The time is measured for a fixed volume of liquid to flow under gravity through the capillary of a calibrated viscometer under a reproducible driving head and at a closely controlled and known temperature.
The kinematic viscosity is the product of the measured flow time and the calibration constant of the viscometer [5]. Flash point results can trace the presence of highly volatile and flammable material in a relatively non volatile or non flammable material; a good example is the presence of certain quantities of diesel fuel or gasoline in lubricating oils [6]. The authors used Eraflash apparatus, made by Eralytics, Austria in order to find the dilution degree and the oil flash point [10].
The concerned measurement is conducted under the latest and safest standards ASTM D6450 & D7094 and it is based on the Continuous Closed Cup method in order to find the Flash Point.

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It can be mentioned that this standard method is an versatile flash point method and it is appropriate for accurate flash point tests and also, to find the influence to the flash point of highly flammable fractions in samples like gasoline in diesel or diesel in oil or gas in oil.
An important characteristic of petroleum fluids being part of product specifications is density. Materials are usually commercialized on knowing the density or if on volume basis then converted to mass basis via density tests. The frequency oscillator is the work principle of the Density Meters DS7800 and allows finding density or relative density of crude oils with high vapour pressures in order to prevent vapour loss during transfer of the sample to the density analyzer. The principle of measurement it's based on the use of a small volume of liquid sample which is entered into an oscillating sample tube and the change process in oscillating frequency induced by the change in the mass of the tube is used in conjunction with calibration data to determine the density of the sample [7]. Water contamination conducts to emulsifying oil and creates an improper capacity of lubricating. The viscosity of the oil is increased and oil additives may be separated. This process leads to a higher reduction of oil lubricating capacity. Analytically the methods can be grouped as by distillation or by Karl Fischer titrations. Generally, trace amounts of water must be determined by Coulometric Karl Fischer methods. Combining the Coulometric technique with Karl Fischer titration, Aquamax KF titrators determined the water content of the sample by measuring the amount of electrolysis current necessary to produce the required iodine [10]. The well known oil stain method is based on oil analysis on filter paper which enables assessment of capacity detergent - dispersant, the degree of contamination of the oil with carbon particles and chemical degradation by oxidation engine oil increasing. The determination of debris in used lubricating oils is a key diagnostic method practiced in machine condition monitoring programs. The presence or increase in concentration of specific wear metals can be indicative of the early stages of wear.

3. RESULTS

M25W40 is the type of oil used in our case study. The test consisted in dilution determination and for that was realized the dilution curve (figure 5). The other results obtained are shown in table 2.

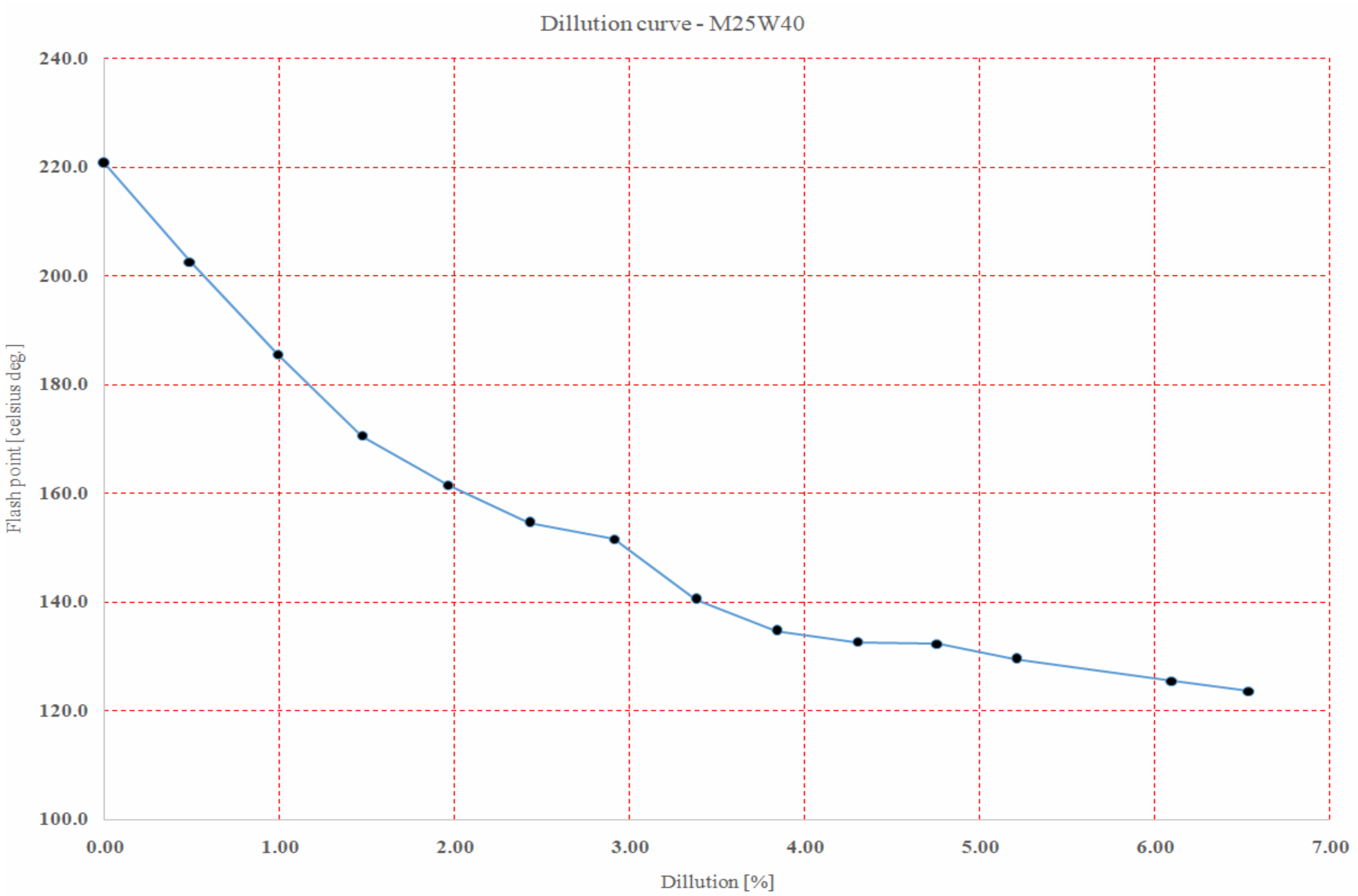

Figure 5. Dilution curve

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Table 2.
M25W40 oil - test results

	Used oil [10]	Fresh oil [10]	Admisible limit [11]
Flash point [^{0}C]	184.10	220.70	180
Kinematic viscosity@ 40^{0}C [cSt]	95.44	113.60	+/-25%
Dilution [%]	0.90	0	5
Density [g/cm3]	1.48	0.89	-
Water content [%]	0.26	0	0.20

Taking into account solid contamination degree it is clear that high density growth is a normal process. As expected, the dilution with diesel fuel caused the decrease of kinematic viscosity and flash point (table 2). Glycol is present in diesel engine oils if there are faulty seals, damaged head gaskets, cracked cylinder or corrosion. In the case study of diesel hydraulic locomotive, the subject of this paper, glycol contamination is removed from the discussion, taking into account that the engine cooling system uses only water, as a cooling technical fluid. Water contamination by small amounts is normal for engine oils. High water levels are a sign of attention and rarely can be corrected only by changing the engine oil. Long idling in wintertime it's a inappropriate operating engine mode and can induce water condensation in the oil sump, leading to corrosion of the surfaces, the oil oxidation, etc.
A drop in the flash point normally indicates contamination of the lubricant by distillate fuel, although with residual fuel no significant change may be apparent.
For guidance, it is advisable to check for fuel leakages when the flash point drops by 30°C or more.
A reduced Flash Point normally indicates that there is a fuel leakage into the lubrication system.
A more precise test method for measuring fuel contamination in distillate fueled engine oil, which is commonly utilized in used oil analysis laboratories, is gas chromatography [12].
Fuel dilution, in operating conditions at low temperatures, can cause paraffin deposits.
This may conduct, when starting the engine, at low oil pressure and no lubrication.
This dramatically reduces the oil film thickness and results in premature wear of the combustion zone (piston, piston rings and cylinder) and crankcase bearing wear.
Severe dilution (over 2 %) is associated with leaks, fuel injector problems and low efficiency combustion.
Fuel dilution may cause decrease in the viscosity of engine oil, for example, from 15W40 to 5W20.
The author compared the results of conducted tests with the limits values (table 1).
Thus, the conclusions are:
- concerning flash point, the obtained value of 184,1^{0}C it is very close to the limit value of 180^{0}C; for that reason it can be imposed the oil changing in order to eliminate explosion risk; the risk of explosion is bigger when high flash points are measured; the crankcase explosion appears when there are noticed a presence of a flammable vapor atmosphere and a heat source; the heat source can be a piston malfunctioning or a running bearing;
- 0,9% is the obtained value in terms of fuel dilution and is not to high, comparing this value with 6% which is the maximum value according to specific literature (table 1); in order to find the fault responsible for the appearance of fuel dilution, it seems that are sealing problems at the level of injection pump,
- the qualitative analysis EDXRF (Energy dispersive X-ray fluorescence) of under study oil samples is shown in the below spectra; the acquisition conditions were the following: direct excitation in the atmosphere of He, slot 1 mm, working voltage U = 44.70 kV, current intensity I = 0.30 mA, acquisition time 180 s, number of energy channels used 2048 (figure 6);
- in terms of density values there can be found a raise from 0.89 to 1.485 g/cm^3; the cause of this density variation is the presence of metal wear and contaminants; in internal combustion engines, iron can have like source the cylinders liners, rings, crankshaft, camshaft, oil pump gear; also, the fluorescence spectra indicates the presence of Cu, as well as a Fe in the used oil sample; this amount of Cu and Fe may the result of wear of bearings (rod, turbo, etc.), piston rings, cylinder liners, cams, tappets or dust presence;
- the source for zinc can be anti-wear additive, but it's not the case here because the additives from the used oil are not yet transformed in contaminants; also, the presence of Zn can be an indicator that the analyzed oil has been mixed of contaminated by another type of oil grade or brand;
- the obtained value of water content is higher than the limit and for this reason the used oil must be changed immediately; here the authors proposed a diagnosis process in terms of sealing matters of cooling system;

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- a improper combustion process could be the cause for the blackening of oil stain; like recommendation, the calibration must be done.

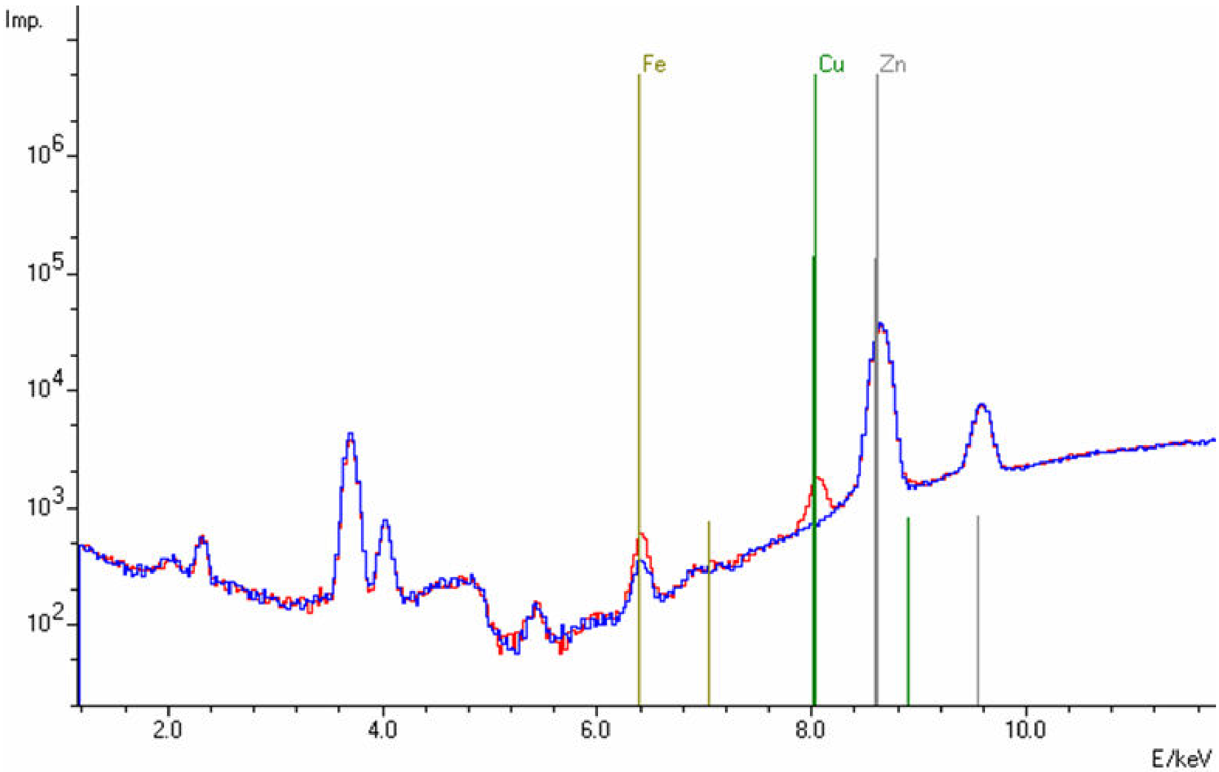

Figure 6. Amount of Fe, Cu and Zn in the used oil sample

4. CONCLUSIONS AND FUTURE WORKS

Test results conducted in this case study are presented below:
- first of all, making used oil analysis is important because the results assesses the condition of the oil and provides recommendations on its suitability for future utilization and optimization of the oil change intervals,
- the conducted study develops the state of the engine and enables the detection and thus prevention of issues which left unattended may impact the reliable operation of the engine.

The cumulative effect of oil contamination on engine reliability, fuel economy, emissions in exhaust gases and the maintenance cost are significant.
These days, it doesn't exists any motor oil additives that can control the damages produced by these contaminants, therefore, the processes of diagnostics, predictive maintenance and oil analysis are the main strategies to counteract the effects of these contaminants.
The advantage of this type of approach is that an off-site research laboratory will create a complete set of oil analysis instruments to run the tests and researchers to review the data.
The response time is faster and the up-front capital investment is relatively low.
Thus, the purpose of this study was to the find the possible malfunctions on the operation of a diesel engine locomotive and to identify their causes; this approach it's based on analyse of the oil used contaminants.
For future works the authors will try to develop a detailed methodology for engine diagnosing by used oil analysing, by taking into account of more physico - chemical characteristics oil.
Also, for the future, the research will help to elaborate specific set of procedures usable for any type of internal combustion engine.

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FINITE ELEMENT ANALYSIS OF OCCUPANT SAFETY IN THE CASE OF FAR SIDE IMPACT WITH A RIGID POLE

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Abstract. *Finite element method supports engineering development by creating a virtual environment in order to deliver useful information. Road crash analyses are nowadays based on this method which is used worldwide. A statistical study, based on a five year period, was done on Romanian roads and it was found out that fatal lateral impact between a passenger vehicle and rigid pole or similar objects are statistically significant. A simulation study was realized, to determine the safety level after such collisions on co-driver side. Results show that a five star car will not reach the same safety rating, for the next crash criteria implemented by authorities.*

Key-Words: *Finite Element Method, Occupant Safety, Far Side Impact*

1. INTRODUCTION

Computational tools help researchers to develop new technologies that are implemented in different areas of interest. Road crash studies are nowadays realized in virtual programs, which present many advantages in order to increase the traffic safety. Computational environment offers details about the causes and the effects produced during a crash. Nevertheless, the costs are significantly diminished and also it offers many research possibilities that in a real environment would be expensive or difficult to be implemented in a test phase. Finite element method represents a numerical computing technique which implies engineering knowledge and computational skills.
Authorities and other governmental organizations are implementing different crash scenarios that have to be fulfilled by the new produced cars. Otherwise, these could not be sold on the market. Examples of such organizations are: UNECE (United Nations Economic Commission for Europe), FMVSS (Federal Motor Vehicle Safety Standards), Euro NCAP (The European New Car Assessment Programme), IIHS (Insurance Institute for Highway Safety) and many others.
A recent statistical road crash study [7] was conducted based on a five year period database in order to evaluate fatal crashes occurred in curve and inclined plane on Romanian traffic roads. One of the many other statistically significant conclusions was found to be lateral impact between a vehicle and a rigid pole or similar objects. The main anatomical cause of death is not described in the database, but other papers like [5] concluded that thorax ribs fractures are the main fatal cause for frontal collisions.
Based on the findings from [7], in another research [8] it was simulated a sedan four door car in case of a lateral impact against a rigid pole. The car is a representative five star passenger vehicle which fulfilled the safety requirements of such an impact described by the UNECE new regulation from 2017.
Anthropomorphic test dummies are often used to predict the level of injury in motor vehicle collision [2], based on bio-dynamic characteristics which correspond to real situations.
In this paper was used the latest side crash dummy model WorldSID 50% Male.
In this paper the novelty consists in testing the safety of the co-driver in the same case of lateral impact between the same vehicle and a rigid pole.
Euro NCAP plans for 2020 to implement co-driver scenarios, named as far side collisions, involving the collision case between a passenger car and a movable barrier and also against a rigid pole.
The scenarios consist actually of two sled tests based on the pulse value determined on B-pillar after the whole car simulation tests are done.
The requirements described by the new Euro NCAP test were implemented in this paper for this type of crash scenario.

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2. METHODS

The road vehicle used for far side simulation was a five star Toyota Camry 2012 (Figure 1).
The finite element model (Figure 1) was realized by the NCAC (National Crash Analysis Center) and validated by the NHTSA (National Highway Traffic Safety Administration) [6].
The finite model is a free model made available for each researcher.

Figure 1. Toyota Camry 2012 (left) and the structure of the finite element method model (right)

The lateral impact scenario between the road vehicle and the rigid pole is based on the UNECE 2017 regulation (Figure 2), but implemented for the co-driver safety.
The actual UNECE 2017 side crash test is described as follows: near-side collision, with only one dummy (WorldSID 50% Male) on driver side; the static pole is ideally rigid and has a standard diameter of 254 mm; impact velocity is 32 km/h under an angle of 75° [9].

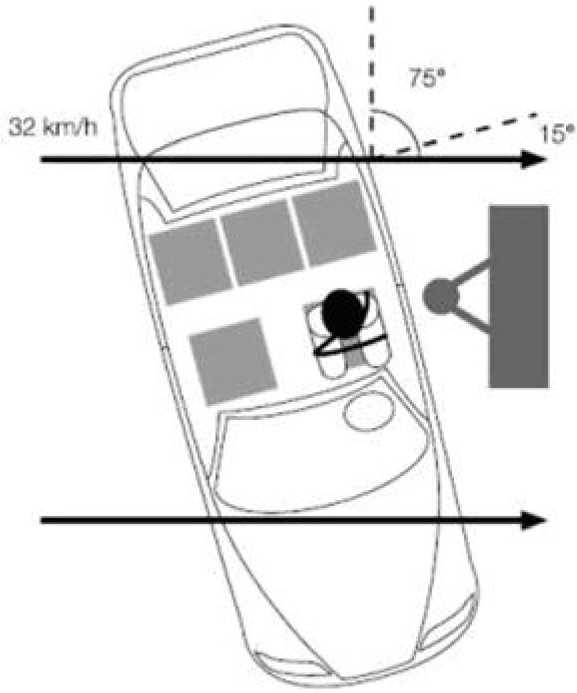

Figure 2. UNECE 2017 regulation of a lateral pole collision, with one dummy (WorldSID 50% male) on driver side [9]

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The finite element dummy model used in the paper is the latest version developed for lateral collisions. WorldSID 50% male dummy (Figure 3) was supplied by the LSTC (Livermore Software Technology Corporation) being compatible with the LS-DYNA software system.
This dummy model has a weight of 74.88 kg [1].

Figure 3. WorldSID 50% Male Dummy [4]

Due to the dummy kinematic during the far side test, four marking grids (Figure 4) were implemented in order to evaluate the dummy excursion.
The meaning of each parallel excursion line is described as follows:

- Red line – maximum intrusion – marking the maximum intruding point of the door panel after the crash;
- Yellow line – head excursion higher performance limit – marking the seat center line on the struck side;
- Green line – occupant interaction limit – marking 250 mm inboard from struck side seat centerline;
- Blue line – marking the vehicle center line [3].

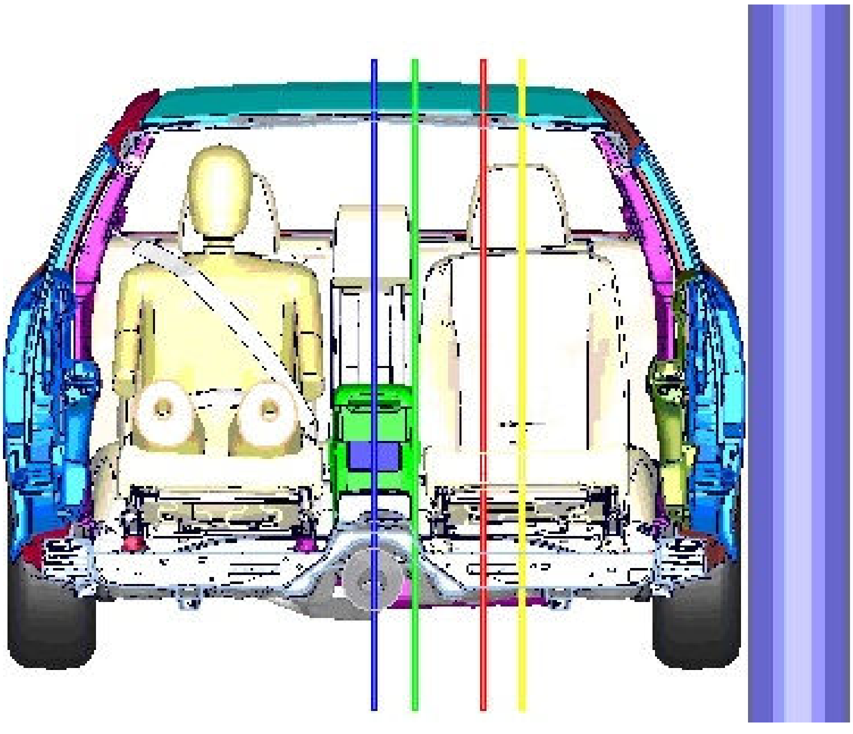

Figure 4. Vehicle markings

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The assessment criteria for the far side scenarios are listed in the following table. The evaluation consists of four body regions, the head, neck, chest and abdomen and pelvis and lumbar spine.
Capping is applied to head, chest and abdomen and the test will not be validated if the red line is exceeded. If the head exceeds the green line, the occupant interaction limit, the score of the test will be reduced by 50% [3].

Table 1.
Far side test – performance limits [3]

Criteria		Performance limits		
		Higher	Lower	Capping
Head	Head Injury Criteria$_{15}$	500	700	700
	Resultant 3ms acceleration	72g	80g	80g
	Excursion		yellow line	red line
Neck	Tension Fz		3.74 kN	
	Lateral flexion Mx		50 Nm	
	Extension negative My		50 Nm	
Chest & Abdomen	Chest lateral compression	28 mm	50 mm	50 mm
	Abdomen lateral compression	47 mm	65 mm	65 mm
Pelvis & Lumbar	Pubic symphysis		2.8 kN	
	Lumbar Fy		2.0 kN	
	Lumbar Fz		2.84 kN	
	Lumbar Mx		100 Nm	

3. DATA ANALYSIS

The simulation was done using the LS-DYNA solver and for post-processing the model was uploaded in GNS Animator4 to visualize the kinematic of the co-driver during the impact.
To capture the maximum values of the dummy's injury, the simulation time was set up to 200 ms.
In Figure 5 was captured the maximum head excursion around 115 ms which exceeds the green line.
This fact leads to a score diminution down to 50% which means that the already producible Toyota Camry 2012 will not reach five star rating regarding new procedures from 2020.

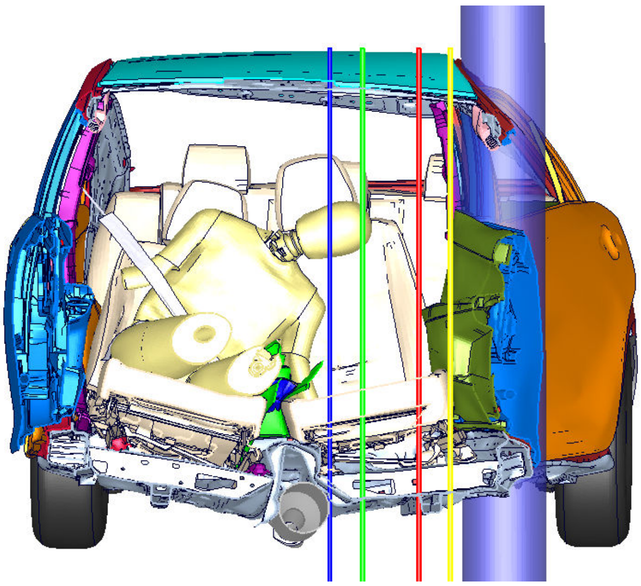

Figure 5. The head excursion exceeds the occupant interaction limit line at 115 ms

As it can be seen in Figure 5, the dummy interacts with the center console.

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Depending on the height and stiffness of the center console, the dummy head excursion could be directly influenced. But a higher stiffness of the center console could worsen the pelvis force.
It is essential to take into account that the belt buckle remains between dummy pelvis and center console, because it could have a negative influence on pelvis force too. Belt buckle could be redesigned or repositioned for the next car generations. The body in white floor has a direct impact of the dummy excursion. Better results could be obtained if the floor remains on a flat level. In this case the floor does not have a desired deformable behavior, which has a direct effect on the seats kinematic.
Without using a restraint system the injury level will be significantly increased.
The worse value captured during the simulation (Figure 6) was registered in the neck area, at 158 ms, representing the lateral flexion of the neck.
The value of 112.79 Nm is significantly higher than the limit value of 50 Nm which means that this test will not end with the maximum rating.

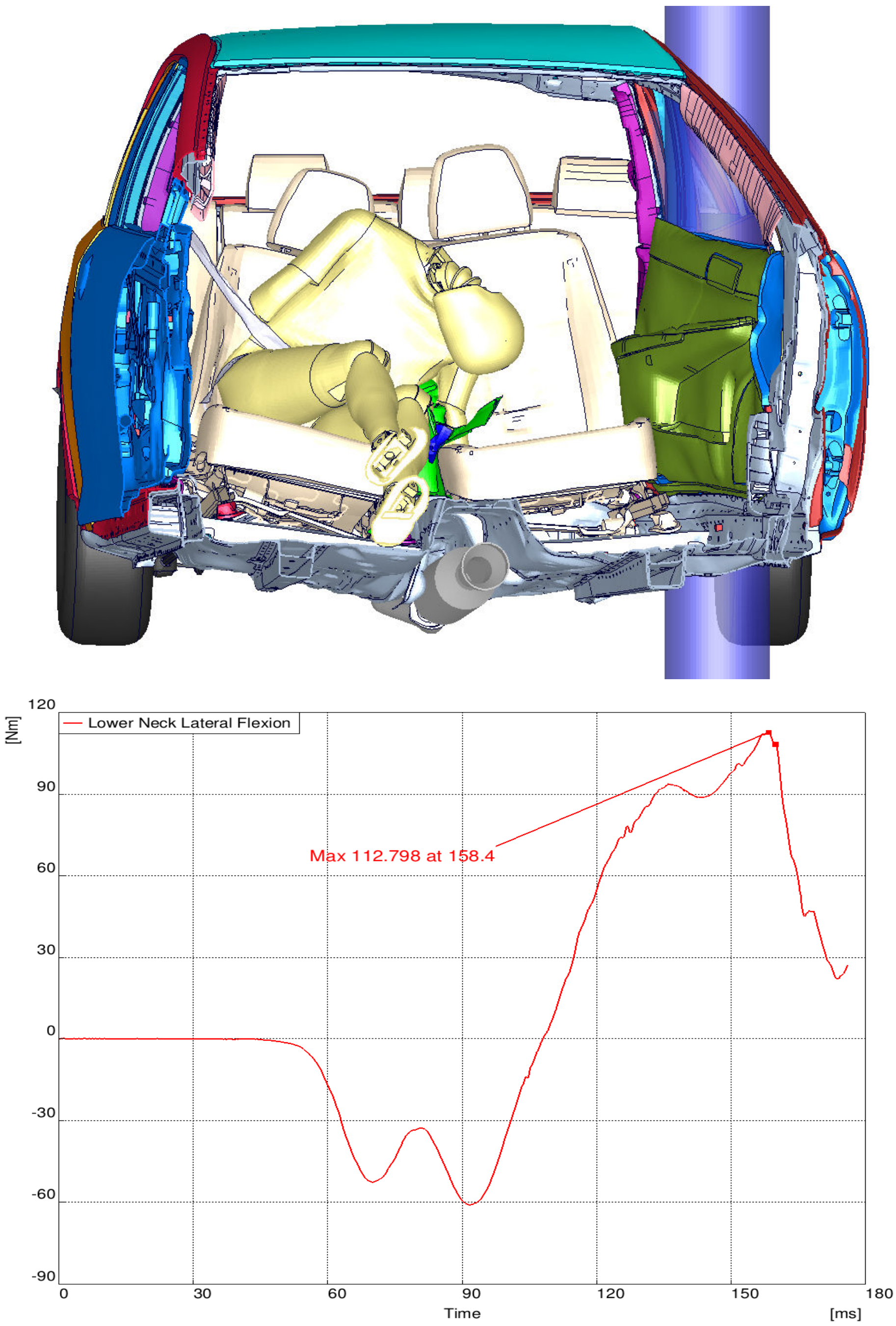

Figure 6. Lower neck lateral flexion – kinematic at maximum value

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For the actual manufactured cars, the main focus, regarding occupant safety measurements, was on the driver's side for the front and rear positions. Next car generation will implement auxiliary technologies in order to protect all passengers involved in car accidents.
For far side situations, in the driver's seat there has to be implemented an additional side airbag that will deploy between the seats, in order to reduce the dummy excursion and to reduce the interaction effects between the driver and the co-driver during the impact.
The driver's seat has to be robustly designed because its structure has to support two deployments of the side airbags in case of a far side situation.
Next studies will be focused on the situation when this additional airbag will be mounted in the driver's seat. Nevertheless, the injury level of the rear passengers in these cases and which solution could be then implemented in order to reach the safety level have to be taken into account.

3. CONCLUSION

In this paper is presented a new case of impact scenario that Euro NCAP wants to implement for the new car generation, released in 2020. A five star sedan Toyota Camry 2012 was tested after this new regulation and it was found out that additional measurements have to be implemented in order to reach the same rating as before. A center airbag mounted in the driver's seat represents the optimal solution to increase the passenger safety, for far side impacts.

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CONSIDERATIONS ON THE GAS EXCHANGE PROCESS IN INTERNAL COMBUSTION ENGINES WITH OPPOSITE PISTONS

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Abstract: *The internal combustion engines with opposite pistons present the possibility to create a dynamically balanced crank-piston mechanism with a lower number of cylinders in comparison with conventional engines. The gas exchange process is of utmost importance inasmuch as both the general development of the engine and the design of the combustion chamber are concerned. The motion conditions, the temperature and the dilution with burnt gases in which the cylinder charge is found at the closing of the scavenging lights, majorly influence the formation process of the air-fuel mixture and of the burning.*

Key-Words: *Internal combustion engines with opposite pistons, Gas exchange process, Combustion chamber.*

1. INTRODUCTION

One of the most efficient methods to reduce the polluting exhaust gases generated by an internal combustion engine is to reduce the effective specific fuel consumption. Apart from the reduction of the polluting exhaust gases, the reduction of the effective specific fuel consumption of the internal combustion engine is an important aspect also from the point of view of cost reduction in the exploitation of the vehicle or of the aggregate that the respective engine is attached to. The effective specific fuel consumption of an engine is influenced by a lot of parameters, out of which: the loss of power from the internal friction of the crank-piston mechanism, the loss due to the pressure variation phenomenon, the architecture of the combustion chamber and the ratio between the volume where the combustion and the expansion take place and the area of the surfaces that represent the boundaries of that volume, play a very important role [1][2].
The present tendencies in the design and development of internal combustion engines are, on the one hand, the reduction of the total cylinder capacity (downsizing) with maintaining and even the increasing of power through supercharging, and, on the other hand, the decrease in the number of cylinders, in order to obtain an as large as possible volume per cylinder, for a given cylinder capacity. By this, one considers the decrease in the loss of specific power due to friction and to the heat exchange process with the environment, which finally leads to the reduction in the effective specific fuel consumption.
For the conventional four stroke engines, the challenges involved consist in the high levels of pressure and temperatures in the thermodynamic processes, while the total cylinder capacity decreases and the power level remains constant. Also, the decrease in the number of cylinders is generally limited to four, out of the balancing condition of the crank-piston mechanism.
Another way to increase the specific volumetric power is to reconsider the two-stroke engines. Supposing that the problems regarding the gas exchange process and the polluting exhaust gases, the two-stroke engines present the advantage of a lower loss of specific power through friction, with respect to the four-stroke engines. Due to the fact that the thermodynamic cycle of the two-stroke engines requires a single revolution of the crankshaft, for the same engine speed and power one can obtain lower combustion pressures in comparison with the four-stroke engines.
The engines with opposite pistons present the possibility to create a dynamically balanced crank-piston mechanism with a lower number of cylinders in comparison with the conventional engines.

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Also, the total stroke of the crank-piston mechanism is divided between the two pistons which are displaced one against the other in a cylinder. Therefore, for the same engine speed, the mean velocity of the piston is considerably reduced, and also the engine friction is lower. On the other hand, if the mean velocity of the piston is kept constant, the engine speed is doubled, and, such, the engine power is doubled [3][4]. For the two-stroke engine with opposite pistons the gas exchange process with equal current scavenging can be adapted, one of the pistons controlling the exhaust light while the other piston controls the intake light. Also, this type of mechanism presents the possibility of a relatively simple implementation of an asymmetric distribution diaphragm with respect to the bottom dead center.
Considering the previously mentioned, the development of a compression ignition engine with opposite pistons is not only an interesting scientific research project, but also one with special practical implications, starting with the vehicle propulsion, and leading to the flight equipment propulsion, to stationary and marine applications.
The analysis of the gas exchange process will be accomplished on a compression ignition engine with opposite pistons, with two opposing cylinders. The concept of this engine has been patented in 1999, having been dubbed EM100D, by one of the authors of this paper, Prof. Dr. Eng. Peter Hofbauer: “Internal combustion engine with a single crankshaft and having opposed cylinders with opposed pistons”, U.S. patent n. US6,170,443, B1, Santa Barbara, 1999 [5][6].

2. CONSIDERATIONS ON THE GAS EXCHANGE PROCESS

The gas exchange process has a significant importance in both the general development of the engine and the design of the combustion chamber. The motion conditions, the temperature and the dilution with burnt gases that characterize the cylinder charge when closing the scavenging lights, majorly influence the formation process of the air-fuel mixture and the combustion process [7], [8].
In order to obtain an efficient burning throughout the whole range of engine speed and load, it is necessary to carefully consider the choice of architecture and the correct dimensioning of the components in the gas exchange system. Unlike the four-stroke engine, in the two-stroke engine, the gas exchange process lasts for a shorter time span and takes place around the bottom dead center. In order for the scavenging of the cylinder to happen it is necessary that the pressure in front of the inlet lights to be larger than the pressure in the cylinder, while the pressure in the cylinder needs to be larger than the pressure behind the exhaust lights. These pressures are not constant during the gas exchange process, but have a pronounced undulatory behaviour, a situation which can imply difficulties in the scavenging process. The pressure waves in front of and behind the intake and exhaust lights are greatly influenced by the engine speed and load, by the volumes of these areas, by the lengths of the inlet and exhaust manifolds, and also by the intake and exhaust lights timing.
The fresh charge enters the cylinder through the intake lights at the same time that the exhaust gases leave the cylinder through the exhaust lights. Thus, between the fresh charge and the exhaust gases a contact surface is created, with a shape and regularity that majorly influence the unwanted mixture between the two fluids. Should the ideal situation of a perfect scavenging be considered, this contact surface would have the minimum possible area given by the geometry of the cylinder and the chosen type of scavenging and it would act like an exhaust piston, stopping whichever mixture between the fresh air and the burnt gases. In reality, the shape of this contact surface is irregular and variable in time during the scavenging process, and depends on various parameters out of which there can be mentioned the pressure waves from the gas exchange system, the orientation and the shape of the intake and exhaust lights, the engine speed, the supercharging system, the shape of the piston head and the geometry of the combustion chamber, etc. Finally, the shape of this surface, and also the size and the velocity direction of the fluid molecules in its vicinity influence the quantity of residual gases that remain stuck in the cylinder at the end of scavenging, and also influence the fresh air quantity which slips through the exhaust lights, which short circuits the cylinder. In the case of a two-stroke engine, the important parameters in the development of the combustion chamber, such as the air excess coefficient “λ” and the compression ratio “ε” are closely related to the gas exchange system. At the two-stroke engine, the computation of the air excess coefficient “λ” is a much more complex issue than in the case of the four-stroke engine. On the one hand, it is needed to determine the quantity of fresh air trapped inside the cylinder, as well as the quantity of exhaust gases stuck inside the cylinder, at the closing of the intake lights.

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The burnt gases from the cylinder contain oxygen molecules which have not reacted with the fuel in the combustion phase of the previous cycle, as a consequence of the air excess and incomplete burning. The fresh air that short circuits the cylinder during scavenging does not participate in the combustion process, therefore, the measurement of the total flow of fresh air which passes through the engine and it being considered to the number of cycles, is insufficient to determine the air excess coefficient. Moreover, the use of oxygen sensors in the exhaust manifolds does not offer enough precision for the determination of the air excess coefficient "λ", because of the short circuiting of the cylinder by a part of the fresh charge. The real compression stroke for a two-stroke engine only starts after the closing of the intake lights, and this moment actually defines real cylinder capacity of the engine. Consequently, it is necessary to obtain an acceptable compromise between the need of an as large as possible surface of the intake and exhaust lights and the need of an as large as possible real cylinder capacity, for a given cylinder bore and engine stroke.

3. THE PARAMETERS OF THE GAS EXCHANGE PROCESS

For the analysis of the gas exchange process the following parameters are defined [9][10][11]:
The delivering coefficient "L" is a measure of the air quantity delivered to the engine and which crosses all the intake lights, divided to the engine volume:

$$L = \frac{V_{liv}}{V_{cil}} \tag{1}$$

In equation (1), "V_{liv}" represents the air quantity supplied to the engine's cylinders, while "V_{cil}" is the cylinder capacity of the engine. The air quantity that crosses the intake lights is divided in two.
One part short circuits the cylinder and is evacuated through the exhaust lights, while the second part of it is kept inside the cylinder and takes part in the next cycle's combustion phase.
The retaining coefficient "η_r" is the ratio between the air mass trapped inside the cylinder and the air quantity that passes through the intake lights:

$$\eta_{\mathrm{r}} = \frac{V_{ret}}{V_{liv}} \tag{2}$$

Where "V_{ret}", represents the fresh air quantity trapped inside the cylinder.
The retaining coefficient represents the air quantity delivered to the engine and retained in the cylinders, while the rest of it is lost through the exhaust lights.
The relative charging coefficient of the cylinder "C_{rel}" is the ratio between the charge volume in the cylinder and the cylinder volume:

$$C_{\mathrm{rel}} = \frac{V_{inc}}{V_{cil}} \tag{3}$$

Where "V_{inc}" is the charging of the cylinder and represents the sum between the air quantity retained in the cylinder "V_{ret}" and the residual burnt gases quantity stuck inside the cylinder after the evacuation phase of the gas exchange process ended "V_{rez}":

$$V_{inc} = V_{ret} + V_{rez} \tag{4}$$

The scavenging efficiency "η_{bal}" is the ratio between the air volume retained in the cylinder "V_{ret}" and the cylinder charge "V_{inc}":

$$\eta_{\mathrm{bal}} = \frac{V_{ret}}{V_{inc}} \tag{5}$$

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The mixture's purity coefficient "η_p" is the ratio between the air volume in the cylinder "V_{pur}" and the charge volume of the cylinder "V_{inc}". During combustion in Diesel engines there remains a certain quantity of oxygen, due to the air excess. Some of these oxygen molecules are also found in the residual burnt gases "V_{rez}" in the cylinder. Consequently, the gas mixture in the cylinder at the end of gas exchange process includes three parts: the fresh air quantity retained in the cylinder "V_{ret}" at which is added the quantity of the exhaust products "V_{cp}", which are also found in the residual gas volume "V_{rez}" and, additionally, the fresh air quantity from the residual gas volume "V_{rez}-V_{cp}":

$$\eta_p = \frac{V_{pur}}{V_{inc}} = \frac{V_{inc} - V_{cp}}{V_{inc}} = 1 - \frac{V_{cp}}{V_{inc}} \tag{6}$$

The cylinder charging efficiency "η_{inc}" is the ratio between the air volume retained in the cylinder and the cylinder volume:

$$\eta_{inc} = \frac{V_{ret}}{V_{cil}} \tag{7}$$

The cylinder charging efficiency is a measure for the successful filling up of the cylinder by fresh air. From equations (1), (2) and (7) there results:

$$\eta_{inc} = L \cdot \eta_r \tag{8}$$

The scavenging coefficient "R_s". In all of the above equations, the described volumes correspond to the normal atmospheric temperature and pressure conditions. In the case of supercharged engines it is preferred to make the reference between the delivered air mass "M_{liv}" and the air mass which might occupy the cylinder volume in the temperature and pressure conditions of the intake lights "M_{cil}". Considering this aspect, the scavenging coefficient is defined as the ratio between the two measures:

$$R_s = \frac{M_{liv}}{M_{cil}} \tag{9}$$

The cylinder residual gas coefficient "C_{crez}" is the ratio between the residual gas quantity found in the cylinder "M_{crez}" and the total quantity of gases in the cylinder "M_{cil}":

$$C_{crez} = \frac{M_{crez}}{M_{cil}} \tag{10}$$

The coefficient of residual gases found in the exhaust lights "C_{erez}" is the ratio between the exhaust gas quantity exiting the exhaust lights "M_{erez}" and the total quantity of gas mixture that exits the exhaust lights "M_{evac}":

$$C_{erez} = \frac{M_{erez}}{M_{evac}} \tag{11}$$

For the EM100D engine there has been chosen the equal currents scavenging with asymmetric advance and delay timing for the exhaust and intake lights, with respect to the bottom dead centers (BDC) of the cylinder BDC1 and BDC2.
The Figure 1 depicts the engine distribution diagram and the area of the intake and exhaust lights, where there can be observed that the exhaust lights are closed before the intake lights, after the bottom dead center of the cylinder.

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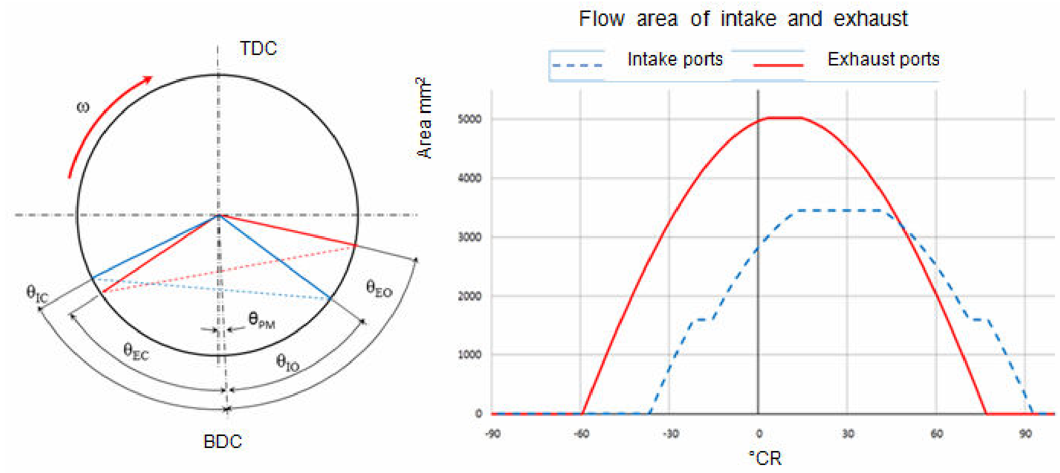

Figure 1. The distribution diagram and the area of the intake and exhaust lights of the EM100D engine

In Figure 1, the following notations have been made:
- θ_{EO} – the opening angle of the exhaust light;
- θ_{IO} – the opening angle of the intake light;
- θ_{EC} – the closing angle of the intake light;
- θ_{EC} – the closing angle of the exhaust light;
- θ_{PM} – the deviation angle of BDC relative to the 180º position with respect to TDC (top dead center);
- *CR* – crankshaft rotations.

In Figure 2 it is presented the chosen architecture for the gas exchange in the EM100D engine, in which one can observe that the fresh air is introduced at one end of the cylinder through the intake lights, while the exhaust gases are evacuated at the opposite end of the cylinder through the exhaust lights.

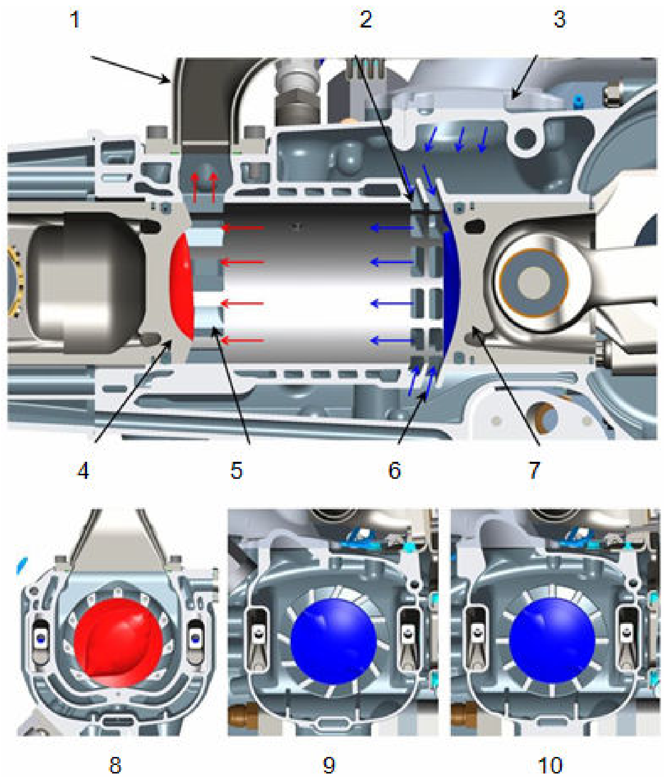

Figure 2. The architecture of the gas exchange process in the EM100D engine: 1 – exhaust manifold; 2 – swirl intake ports; 3 – intake manifold; 4 – exhaust piston; 5 – exhaust ports; 6 – straight intake ports; 7 – intake piston; 8 – section through exhaust ports; 9 – section through intake swirl ports; 10 – section through straight intake ports.

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In order to obtain an optimum scavenging it is necessary that the fresh air supply to replace the burnt gases in the whole volume of the cylinder.
The replacement of the burnt gases around the cylinder axis can be accomplished in the situation when the fresh air introduced through the intake lights has a high penetration, and the replacement of the burnt gases at the periphery of the cylinder, close to the walls, can be realized in the situation when the fresh air supply that enters the cylinder, has a reduced penetration.
In order to simultaneously fulfill the two conditions, is has been chosen the division of the intake lights into two columns: the turbulence intake light column and the column of the straight intake lights. Both columns are oriented in an opposite direction with respect to the top dead center of the cylinder and of the exhaust lights in order to favor the replacement of the exhaust gases from the center of the cylinder.
The first column generates a turbulent motion of the fresh charge which is maintained during the whole scavenging process. The turbulent motion generates a centrifugal force which pushes the fresh charge, which is denser than the residual gases, towards the periphery of the cylinder center.
The second column directs the fresh air in a radial direction towards the center of the cylinder.

4. CONCLUSION

The orientation of the two columns oppositely to the exhaust lights stops, on the one hand, the rapid short circuit of the cylinder by the fresh charge that enters through the turbulent flow lights column, while, on the other hand, it helps to enlarge the penetration of the fresh air supply which enters the cylinder through the straight lights column.
The intake and exhaust lights are placed radials on the whole circumference of the cylinder, in order to regularize as much as possible the contact surface between the fresh charge and the residual gases.
The uniformly radial placement is also desired from the point of view of the uniform thermal loading of the cylinder, especially in the case of the exhaust lights, around which the thermal deformation of the cylinder must be circular, in order not to deform the exhaust piston rings.

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STEADY STATE ENGINE EFFICIENCY SPECIFIC TO SERIES HYBRID ELECTRIC VEHICLES

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Abstract: *The European Commission regulatory proposed average CO_2 emissions of the EU fleet of new passenger cars and light commercial vehicles in 2030 will have to be 30% lower than in 2020. Hybrid electric vehicles (HEV) are becoming a more and more important vehicle segment in the European car park. In the case of HEV a more efficient energy management can be achieved compared to the conventional internal combustion engine powered vehicles. Due to this reason, HEV represent a viable solution throughout which a diminishing of the fuel consumption as well as vehicle emissions can be achieved. Conventional vehicles and parallel HEV frequently operate using transient state. Otherwise, the series HEV engines operate at steady state. The main objective of this paper is to emphasis the advantages of HEV-series compared to HEV-parallel, or in other words, the advantages steady state engines compared to transient state, throughout experimental researches. The transient regimes were obtained successively by the variation of following parameters: engine speed, engine load and both. For comparison, the experimental values obtained in the steady state presented. In the case of steady regimes, the engine's performance and environmental characteristics are higher than for the transient regimes. The increase of engine speed in transient regimes mostly affects the engine's performance and environmental performance. The difference between this values obtained in the stabilized and transient regimes is minimal when the load modified compared to the speed variation.*

Key-Words: *hybrid electric vehicle – HEV, engine, transient state, steady state.*

1. INTRODUCTION

At present are established EU fleet wide targets value for 2020 of 95 CO_2 g/km for passenger cars and 147 CO_2 g/km for light commercial vehicles.
This values are based on the New European Driving Cycle test procedure (NEDC).
The European Commission regulatory proposed for the period after 2020 CO_2 emission targets for passenger cars and light commercial vehicles. Average CO_2 emissions of the EU fleet of new passenger cars and light commercial vehicles in 2025 will have to be 15 % lower than in 2020 [1].
For 2030, CO_2 targets value for both type vehicles are 30 % lower than in 2020.
The values will be based on the Worldwide Harmonized Light Vehicle Test Procedure (WLTP). These CO_2 limits values are almost impossible to meet if unless used the hybrid electric vehicle powertrain.

1.1 The Hybrid Electric Vehicle

Hybrid electric vehicles (HEV) are becoming a more and more important vehicle segment in the European car park. Based on the statistics from domain, in the year 2017, in Europe, 431.504 HEV (full and mild hybrids) have been registered. Which represent a 54.8% increase of new HEV registered in 2017 compared to the 2016.
HEVs can be classified based on the degree of hybridization: Micro-Hybrid, Mild-Hybrid and Strong-Hybrid. Depending on the way in which the energy flow is transmitted, the Strong-Hybrid type systems can be divided into three types: serial, parallel and Power Split HEV (Figure 1) [3] [4].
In the case of HEV a more efficient energy management can be achieved compared to the conventional internal combustion engine powered vehicles (Figure 2).
Due to this reason, HEV represent a viable solution throughout which a diminishing of the fuel consumption as well as vehicle emissions can be achieved [5][6].

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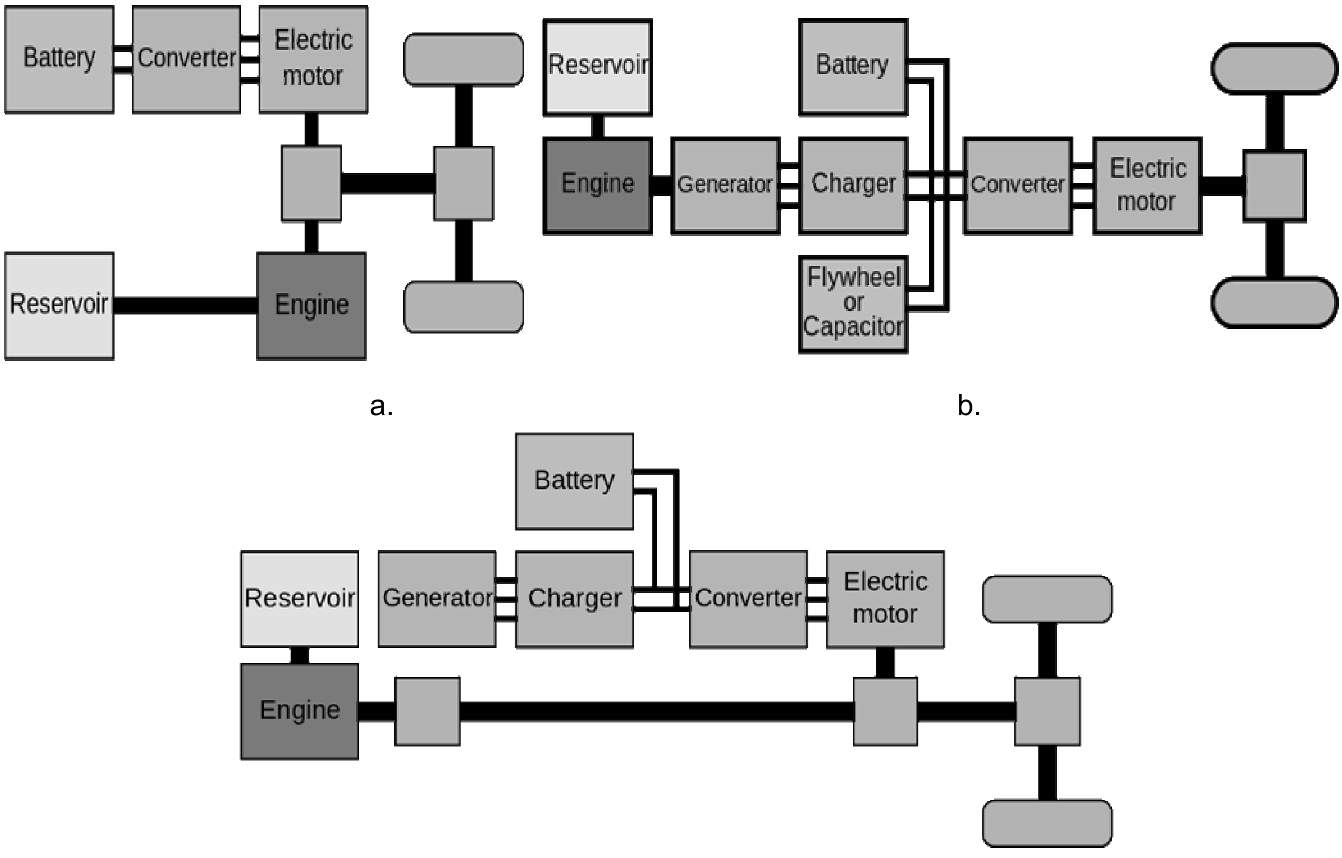

Figure 1. Configuration of a hybrid electric vehicle: a. Parallel HEV, b. Series HEV, c. Split (Series-Parallel) HEV [2].

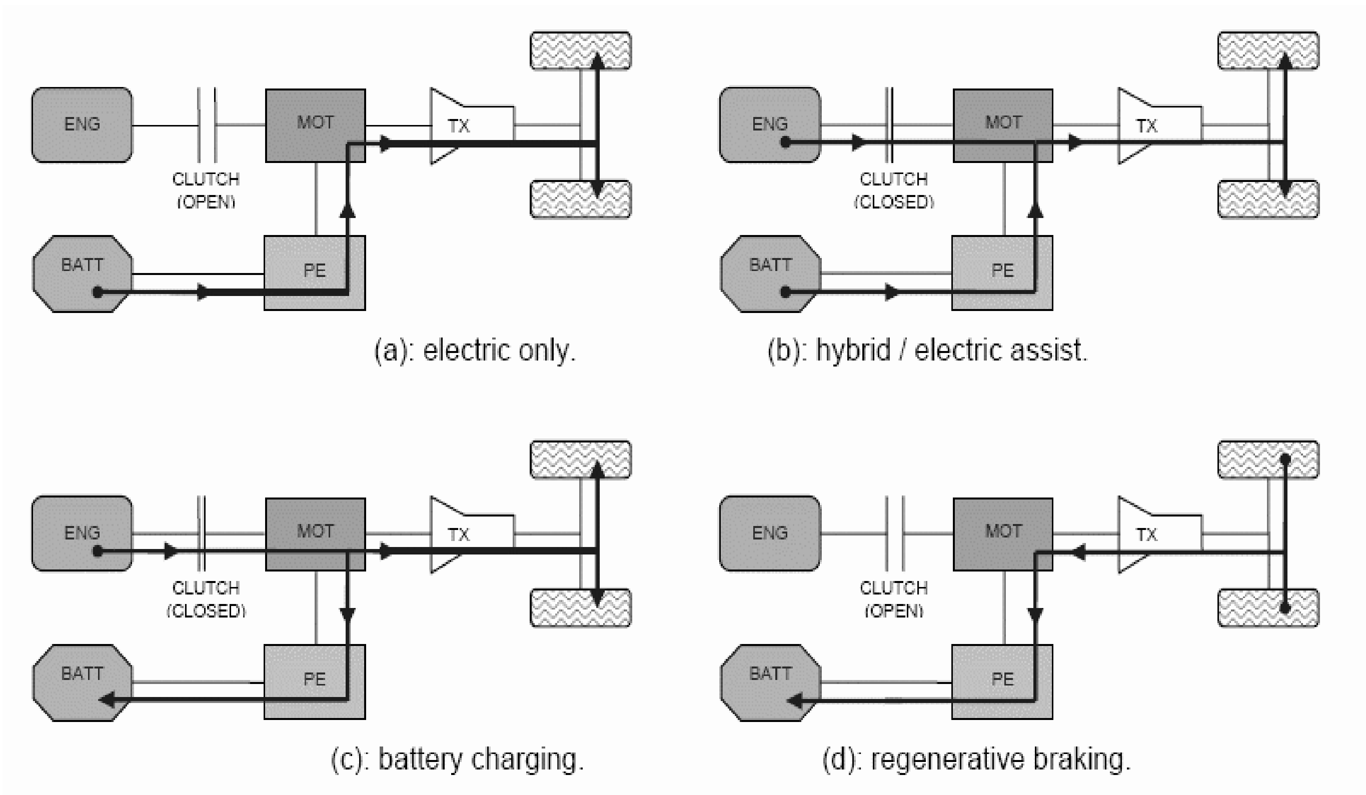

Figure 2. Some representative modes for a parallel hybrid configuration [2].

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Energy management in case of HEV that use a parallel drivetrain is based on the fact that the mechanical energy of the engine is transmitted, based on requirements, to the vehicle's wheels and/or to the generator, and afterwards stored in batteries. Moreover, during vehicle deceleration, the generator creates electricity, which is then stored in batteries for future vehicle propulsion purposes.
Series HEV, are usually powered by electric traction motors. The mechanical energy produced by the engine is converted into electrical energy used to power the traction motors and eventually the wheels. The main advantage of series drivetrain HEV is that the engine operates in a steady regime compared to parallel HEV or even conventional internal combustion engine powered vehicles, where the engine operates frequently in transitory regimes [7].
The NEDC, is supposed to generally represent the typical usage of a vehicle in Europe.
The performances of engines during transitory regime functioning are significantly superior compared to steady state regimes. The main objective of this paper is to emphasis this advantages of steady state engines throughout experimental researches compared to transitory regimes.

2. METHODOLOGY AND RESULTS OF EXPERIMENTAL RESEARCH

The experimental research was carried out at Transilvania University of Brasov, ICDT - Research & Development Institute. The engine used for experimental investigation is an AVL Single Cylinder Compression Ignition Engine type 5402. For experimental the AVL engine, operated in two modes: at steady state (similar to the series HEV) and transient state (similar to the conventional vehicles and parallel HEV). In stabilized regimes, the engine was run at three engine speeds: 1500, 1750, 2000 rpm's and three different engine loads: 20%, 30% and 40%.
As shown in Table 1, the transient regimes were achieved with the following three scenarios: constant engine speed and variable engine load; constant engine load and variable engine speed; both, the load and the speed are variable. In fact, three situations were simulated during the operation of the vehicle, namely:
1. Vehicle's resistances are increasing, but the engine acceleration remains unchanged, which means that the engine speed is reduced;
2. Vehicle's resistances are increasing, but engine speed remains quasi-constant due to engine acceleration;
3. The power surplus obtained through the engine acceleration is higher than the resistance level, so an engine speed increase occurs.

Table 1.
Experimental research strategy

Comparison 1: constant speed and variable load		**Comparison 2: variable speed and constant load**		**Comparison 3: variable speeds and loads**
Speeds [rpm] / Loads % - Steady state				
1500 / 20	2000 / 20	1500 / 20	1500 / 40	1500 / 20
1500 / 30	2000 / 30	1750 / 20	1750 / 40	1750 / 30
1500 / 40	2000 / 40	2000 / 20	2000 / 40	2000 / 40
Speeds [rpm] / Loads % - Transient state				
1500 / 20-40	2000 / 20-40	1500-2000 / 20	1500-2000 / 20	1500-2000 / 20-40

The experimental results obtained in the regimens expressed in Table 1 are shown in Figures 3-5.
For example, Figure 3a, shows that the engine running successively at steady state: 1500 rpm / 20%, 1750 rpm / 20% and 2000 rpm / 20% developed a power of 1.97 kW, 0.84 kW and 1.3 kW (the blue dots).
Average power is considered to be 1.37 kW (the continuous blue line). Instead, if the engine is running at transient speed rising from 1500 rpm to 2000 rpm then the engine develops a power of 0.94 kW (the discontinuous blue line). At the same time, the engine has a fuel consumption of 203 g/kWh, 497 g/kWh and 424 g/kWh (the red dots) at steady state with a mean value of 408 g/kWh (the continuous red line). In transient state the consumption value is 460 g/kWh (the discontinuous red line).
These experimentally obtained values highlight the advantages of series HEV engines compared to classic engines. A 46 % higher power and a lower specific fuel consumption (SFC) of 11% were obtained if the engine functioning to steady state at series HEV.

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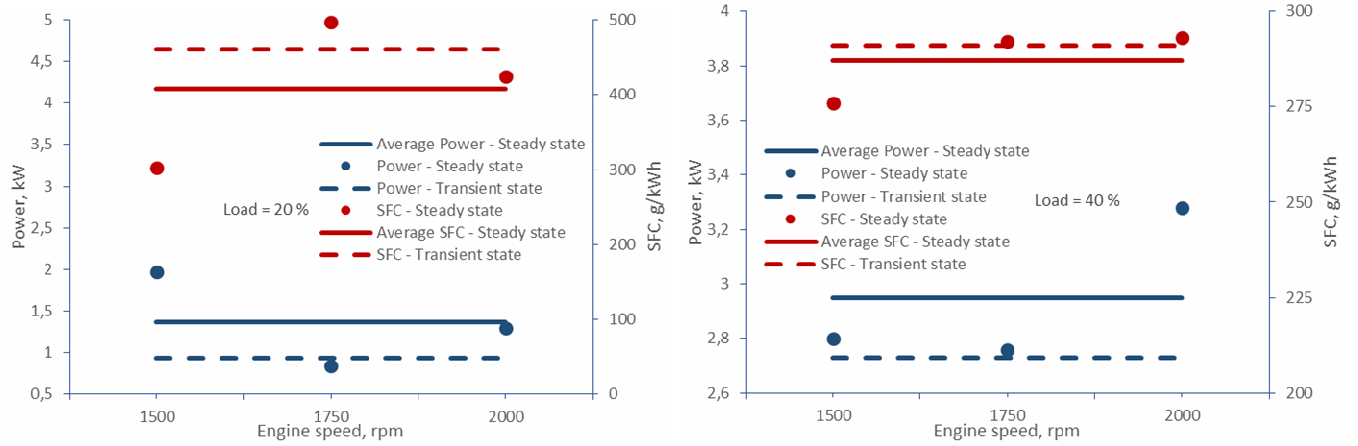
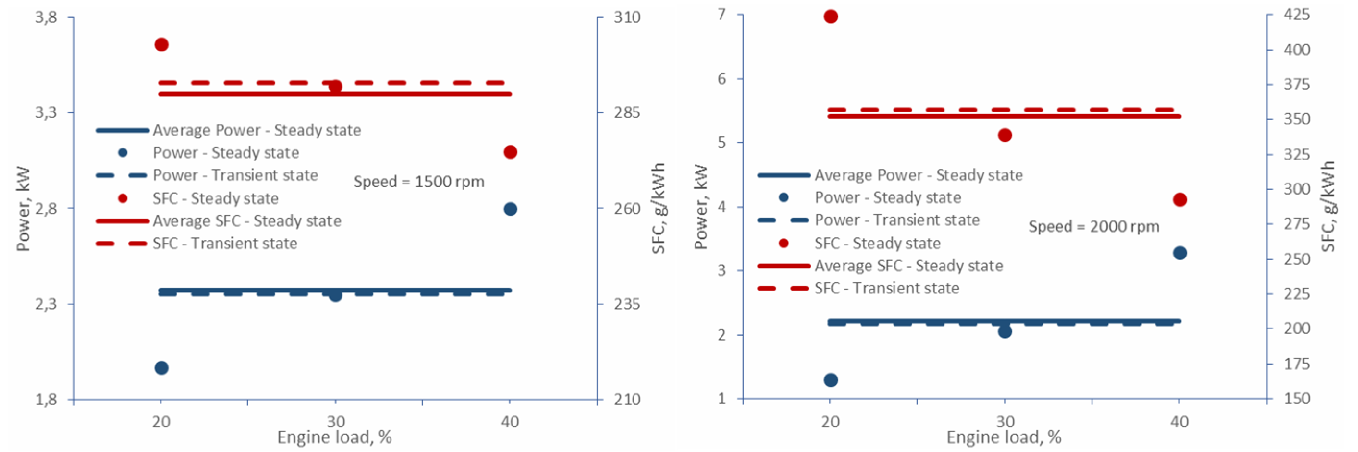


a. b.

Figure 3. The average and absolute value of the power and SFC of the engine operated at steady and transient state (variation speed: 1500-2000 rpm): a. 20 % load; b. 40 % load.



a. b.

Figure 4. The average and absolute value of the power and SFC of the engine operated at steady and transient state (variation load: 20-40 %): a. 1500 rpm engine speed; b. 2000 rpm speed.

If the Figures 3-5 are further analyzed is possible to identify the steady state the engine with the highest power and the lowest fuel consumption.

Or in other words, the state at which series HEV would have maximum performance compared to conventional vehicles.

It can be noticed that the engine speed change has the greatest influence on engine power and SFC during the transient regimes. In particular, by increasing the engine speed, the inertia forces of the moving parts increases.

The value of the inertia forces is higher during engine acceleration. It also increases the intake air speed entering the engine cylinders and reduces the intake time.

Thus, fuel burning in cylinders is affected during transient regimes [8].

By increasing only the engine load, it will change the amount of diesel fuel injected into the cylinders. Thus, the ratio from the amount of intake air and the amount of fuel in the cylinders is different during the transitory regimes relative to the stable regimes, which may affect the quality of the combustion.

The difference between this values obtained in the stabilized and transient regimes (as it can see in Figure 4) is minimal.

The series HEV equipped with a diesel engine that operates at steady state prove to be a viable solution that highest power, reduces the fuel consumption and, implicitly, the emission of CO_2.

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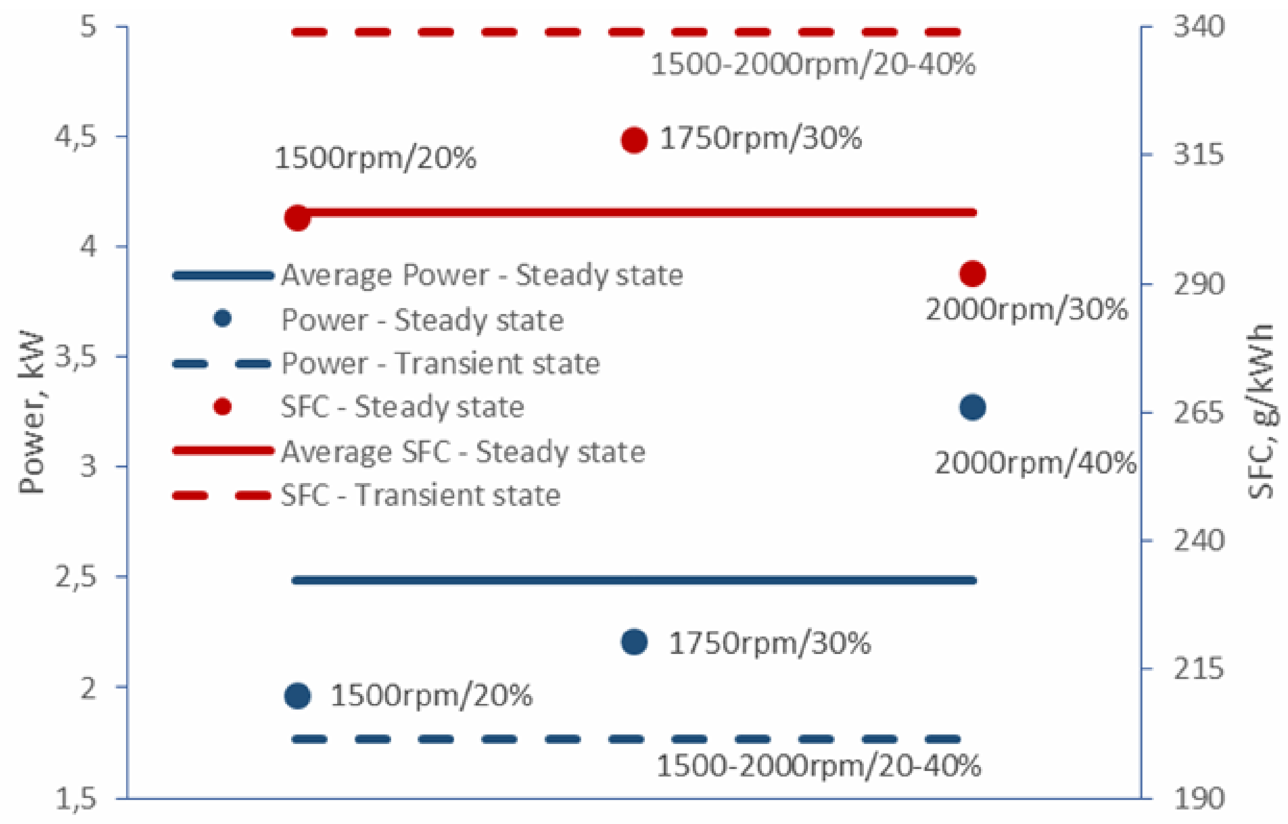

Figure 5. The average and absolute value of the power and SFC of the engine operated at steady and transient state (20-40 % variation load and 1500-2000 rpm variation speed).

5. CONCLUSION

In the case of steady regimes, the engine's performance and environmental characteristics are higher than for the transient regimes.
The increase of engine speed in transient regimes mostly affects the engine's performance and environmental performance.
One of the most important advantage of series drivetrain HEV is that the engine operates in a steady regime compared to parallel HEV or even conventional internal combustion engine powered vehicles, where the engine operates frequently in transitory regimes.

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ALGORITHM AND NUMERICAL CALCULATION OF UNDETERMINED STATIC REACTIONS TO THE PLAN ARTICULATED QUADRILATERAL MECHANISM WITH STRAIGHT BARS

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Abstract*. This paper represents the numerical calculation of the undetermined static reactions to the planar plane quadrilateral mechanism with straight bars and kinematic rotation couplers.The algorithm used was obtained using the relative displacement method and was presented in a previous paper.The method allows linear elastic calculation to determine undetermined static reactions. For a numerical application a computational program was used, the Matlab software, with which the variation diagrams for the determined static reactions were obtained for the undetermined static reactions as well as the variations of the small angles of rotation in the kinematic couplers.The results obtained, we hope to be the ones they are looking for, and they are being developed. The work can be considered a novelty. The research conducted on the literature at both international and national levels has demonstrated the lack of such research concerns. In conclusion, this work can be considered useful to researchers, manufacturers and users of such mechanisms. The quadrilateral mechanism is quite used, the results obtained in this paper will be useful to the researchers in this field, and the designers will more efficiently dimension the components of such a mechanism.The method may be useful in sizing calculations of existing vehicle mechanisms.*

Key-Words: *numerical calculus, undetermined static reactions, quadrilateral mechanism, straight bars*

1. INTRODUCTION

In the previous work [4] a method for the calculation of undetermined static reactions was developed for the planar straight quadrilateral mechanism and kinematic rotation couplers. In the present paper, starting from the obtained analytical results, the algorithm and the calculation program for a numerical application will be elaborated.

2. THEORETICAL ASPECTS. REMARKS

The articulated quadrilateral mechanism is considered $ABCD$ from figure. 1, formed by the straight bars AB, BC, CD indexed by numbers $1,2,3$ and positioned through the angles $\varphi_1, \varphi_2, \varphi_3$ compared to the fixed reference system AXY and notations:

- R_{AX}, R_{AY} forces of reaction, statically determined from the joint A;
- M_{AZ} - momentum, statically determined from the joint A, moment that maintains equilibrium;
- R_{AZ} - reaction force, statically indeterminate from the joint A;
- M_{AX}, M_{AY} - moments of reaction, statically undetermined from the joint A;
- $\{R_A\}$ - column matrix of plűckeriene coordinates of the torsors reaction from A;
- $\{P_B, P_C, P_D\}$ - column matrices of plűckeriene coordinates torso's external forces acting in B, C, D;

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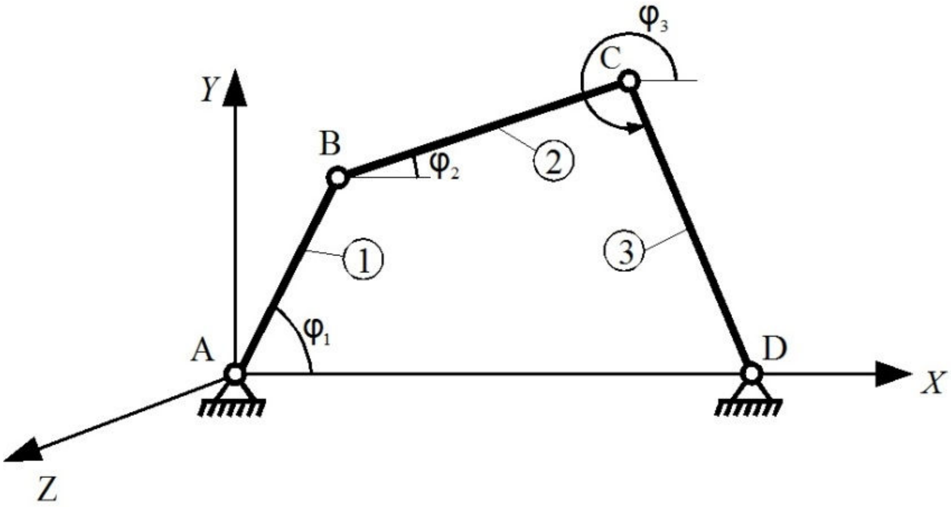

Figure 1. The articulated quadrilateral mechanism

- $\{U_B, U_C, U_D\}$ column matrices $[6],[8]$, attached to kinematic joints

$$\begin{aligned}\{U_B\} &= [0 \quad 0 \quad 1 \quad Y_B \quad -Y_C \quad 0]^T;\\ \{U_C\} &= [0 \quad 0 \quad 1 \quad Y_C \quad -Y_C \quad 0]^T;\\ \{U_D\} &= [0 \quad 0 \quad 1 \quad Y_B \quad -Y_D \quad 0]^T\end{aligned} \tag{1}$$

- $\{\widetilde{U}_B\}$, $\{\widetilde{U}_C\}$, $\{\widetilde{U}_D\}$ column matrices given by relationships:

$$\begin{aligned}\{\widetilde{U}_B\} &= [Y_B \quad -X_B \quad 0 \quad 0 \quad 0 \quad 1]^T;\\ \{\widetilde{U}_C\} &= [Y_C \quad -X_C \quad 0 \quad 0 \quad 0 \quad 1]^T;\\ \{\widetilde{U}_D\} &= [Y_D \quad -X_C \quad 0 \quad 0 \quad 0 \quad 1]^T\end{aligned} \tag{2}$$

- $[H_{AB}],[H_{BC}],[H_{CD}]$ the flexibility matrices of the bars AB, BC, CD;
- $[H_{AD}]$- the matrix given by relationship:

$$[H_{AD}] = [H_{AB}] + [H_{BC}] + [H_{CD}] \tag{3}$$

- $[K_{AD}]$- the stiffness matrix defined by the relationships

$$[K_{AD}] = [H_{AD}]^{-1} = \begin{bmatrix} 0 & 0 & K_{13} & K_{14} & K_{15} & 0 \\ 0 & 0 & K_{23} & K_{24} & K_{25} & 0 \\ K_{31} & K_{32} & 0 & 0 & 0 & K_{36} \\ K_{41} & K_{42} & 0 & 0 & 0 & K_{46} \\ K_{51} & K_{52} & 0 & 0 & 0 & K_{56} \\ 0 & 0 & K_{63} & K_{64} & K_{65} & 0 \end{bmatrix} \tag{4}$$

- $[K^1{}_{AD}],[K^2{}_{AD}]$ - matrices given by equality:

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$$\left[K_{AD}^{1}\right]=\begin{bmatrix} K_{13} & K_{14} & K_{15} \\ K_{23} & K_{24} & K_{25} \\ K_{33} & K_{34} & K_{35} \end{bmatrix},\left[K_{AD}^{2}\right]=\begin{bmatrix} K_{31} & K_{32} & K_{36} \\ K_{41} & K_{42} & K_{46} \\ K_{51} & K_{52} & K_{56} \end{bmatrix} \tag{5}$$

- $\{P_T\}$ - matrix column

$$\{P_T\}=-\begin{bmatrix} \{\widetilde{U}_B\}^T \cdot \{P_B\} \\ \{\widetilde{U}_C\}^T \cdot \{\{P_B\}+\{P_C\}\} \\ \{\widetilde{U}_D\}^T \cdot \{\{P_B\}+\{P_C\}+\{P_D\}\} \end{bmatrix} \tag{6}$$

- $\{\widetilde{\Delta}\}$ - matrix column

$$\{\widetilde{\Delta}\}=[H_{BC}]\{P_B\}+[H_{CD}]\{\{P_B\}+\{P_C\}\}=\begin{bmatrix}\widetilde{\theta}_x & \widetilde{\theta}_y & \widetilde{\theta}_z & \widetilde{\Delta}_x & \widetilde{\Delta}_y & \widetilde{\Delta}_z\end{bmatrix} \tag{7}$$

- $[A],[B]$ - matrices

$$[A]=\begin{bmatrix} Y_B & -X_B & 1 \\ Y_C & -X_C & 1 \\ Y_D & -X_D & 1 \end{bmatrix};[B]=\begin{bmatrix} 1 & 1 & 1 \\ Y_B & Y_C & Y_D \\ -X_B & -X_C & -X_D \end{bmatrix} \tag{8}$$

- ζ_B,ζ_C,ζ_D - small angles of rotation in kinematic joins;
- $\{\zeta\}$ - matrix column

$$\{\zeta\}=\{\zeta_B \quad \zeta_C \quad \zeta_D\}^T \tag{9}$$

With these notations [5], deduct the determined static reactions

$$\begin{bmatrix} R_{AX} \\ R_{AY} \\ M_{AZ} \end{bmatrix}=[A]^{-1}\{P_T\} \tag{10}$$

statically undetermined reactions

$$\begin{bmatrix} R_{AZ} \\ M_{AX} \\ M_{AY} \end{bmatrix}=-\left[K_{AD}^{2}\right]\begin{bmatrix} \widetilde{\theta}_X \\ \widetilde{\theta}_Y \\ \widetilde{\Delta}_Z \end{bmatrix} \tag{11}$$

and the angles of rotation in the kinematic joints

$$\begin{bmatrix} \zeta_B \\ \zeta_C \\ \zeta_D \end{bmatrix}=[B]^{-1}\left[K_{AD}^{1}\right]^{-1}[A]^{-1}\{P_T\}+[B]^{-1}\begin{bmatrix} \widetilde{\theta}_Z \\ \widetilde{\theta}_X \\ \widetilde{\Delta}_Y \end{bmatrix} \tag{12}$$

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3. CALCULATION ALGORITHM

It is considered as for bars indexed with $1,2,3$ the lengths are known l_i, areas A_i, of the normal sections, geometrical moments of inertia $I_{yi}, I_{zi}, I_{xi}\left(I_{xi} = I_{yi} + I_{zi}\right)$, elasticity modules E_i, G_i the flexibility matrices [7],

$$[h_i] = \begin{bmatrix} 0 & 0 & 0_2 & \frac{l_i}{GI_{xi}} & 0 & 0 \\ 0 & 0 & \frac{l_i^2}{2E_iI_{yi}} & 0 & \frac{l_i}{E_iI_{yi}} & 0 \\ 0 & -\frac{l_i^2}{2EI_{zi}} & 0 & 0 & 0 & \frac{l_i}{E_iI_{zi}} \\ \frac{l_i}{EA_i} & 0 & 0 & 0 & 0 & 0 \\ 0 & \frac{l_i^3}{3E_iI_{zi}} & 0 & 0 & 0 & -\frac{l^2{}_i}{2E_iI_{zi}} \\ 0 & 0 & \frac{l_i^3}{3EI_{yi}} & 0 & \frac{l_i^2}{2E_iI_{yi}} & 0 \end{bmatrix} \tag{13}$$

$i = 1,2,3$ and distance $AD = X_D$

Calculates for angle values φ_1 ranging from grade to grade:

a) the angles φ_3, φ_2 with relationships

$$A = 2l_3\left(X_D - l_! \cos\varphi_1\right); B = 2l_1l_3 \sin\varphi_1; C = l_2^2 - X_D^2 - l_1^2 - l_3^2 + 2X_Dl_1\cos\varphi_1 \tag{14}$$

$$\varphi_3 = 2\pi + 2arctg\frac{B + \sqrt{A^2 + B^2 - C^2}}{C - A}; \varphi_2 = \arcsin\frac{l_3\sin\varphi_3 - l_1\sin\varphi_1}{l_2} \tag{15}$$

b) the coordinates of the points B, C, D;

$$X_B = l_1\cos\varphi_1; Y_B = l_1\sin\varphi_1; X_C = X_B + l_2\cos\varphi_2; Y_C = Y_B + l_2\sin\varphi_2; Y_D = 0 \tag{16}$$

c) translation matrices;

$$[G_1] = [0]; [G_2] = \begin{bmatrix} 0 & 0 & Y_B \\ 0 & 0 & -X_B \\ -Y_B & X_B & 0 \end{bmatrix}; [G_3] = \begin{bmatrix} 0 & 0 & Y_C \\ 0 & 0 & -X_C \\ -Y_C & X_C & 0 \end{bmatrix} \tag{17}$$

d) rotation matrices;

$$[R_i] = \begin{bmatrix} \cos\varphi_i & -\sin\varphi_i & 0 \\ \sin\varphi_i & \cos\varphi_i & 0 \\ 0 & 0 & 1 \end{bmatrix}; i = 1,2,3 \tag{18}$$

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e) position matrices;

$$[T_i]=\begin{bmatrix}[R_i] & [0]\\ [G_i][R_i] & [R_i]\end{bmatrix};[T_i]^{-1}=\begin{bmatrix}[R_i]^T & [0]\\ [R_i]^T[G_i]^T & [R_i]^T\end{bmatrix};i=1,2,3 \tag{19}$$

f) the flexibility matrices;

$$[H_i]=[T_i][k_i][T_i]^{-1};[H_{AB}]=[H_1];[H_{BC}]=[H_2];[H_{CD}]=[H_3] \tag{20}$$

g) the total flexibility matrix;

$$[H_{AD}]=[H_{AB}]+[H_{BC}]+[H_{CD}] \tag{21}$$

h) matrices $[K_{AD}],[K_{AD}^1],[K_{AD}^2]$ with relationships (3), (4), (5);
i) column matrices $[P_T],[\widetilde{\Delta}]$ with relationships (6), (7);
j) the matrices $[A],[B]$ with relationships(8);
k) static reactions determined by the relationship (10);
l) static reactions not determined with the relationship(11);
m) the rotation of the kinematic couple with the relationship(12).

4. NUMERICAL APPLICATION

It determines statically the reactions, determined and indetermined and then the angular rotation of the couplings for the mechanism of Figure 1, knowing that the element 3 is actuated in D in the direction of the axis AZ, for a moment $\widetilde{M}$ and knowing it as punctual C acting force P in the direction of the axis AZ. It is considered that the bars have equal, circular cross sections of diameter d.

Numerical data:

$$l_1=l=0.2\,m;l_2=l_3=3*l;X_D=3*l;d=0.01\,m;E_i=E=2.1\cdot10^{11}N\cdot m^{-2};G_1=G=0.8\cdot10^{11}N\cdot m^{-2};$$

$$I_{zi}=I_{yi}=\frac{\pi d^4}{64};I_{xi}=\frac{\pi d^4}{32};\{P_B\}=\{0\};\{P_C\}=[0\quad 0\quad P\quad PY_c\quad -PX_c\quad 0]^T;\{P_D\}=[0\quad 0\quad 0\quad 0\quad 0\quad \widetilde{M}]^T;$$

$$P=100\,N;\widetilde{M}=20N\cdot m$$

In the numerical case the calculation program is elaborated using the Matlab code, on the basis of which are represented the following diagrams

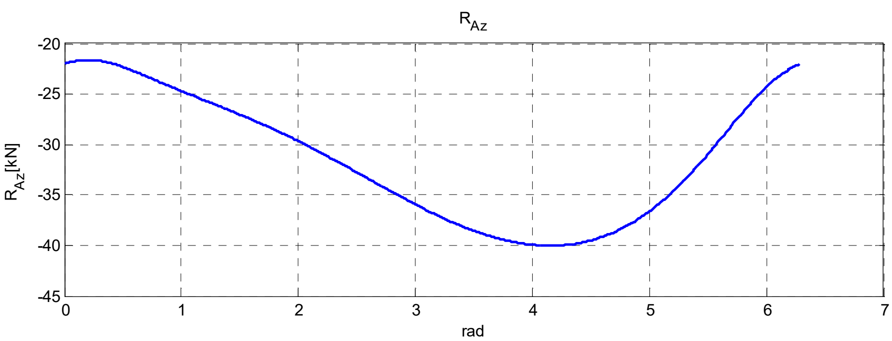

Figure 2. The statically undetermined reactions R_{AZ} vs. φ_1

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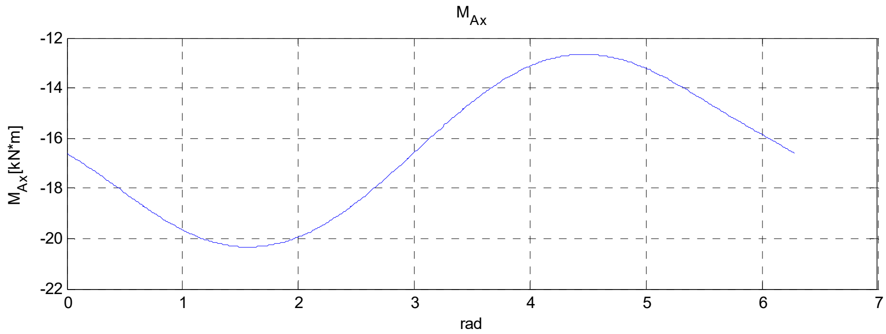

Figure 3. The statically undetermined reactions M_{AX} *vs.* φ_1

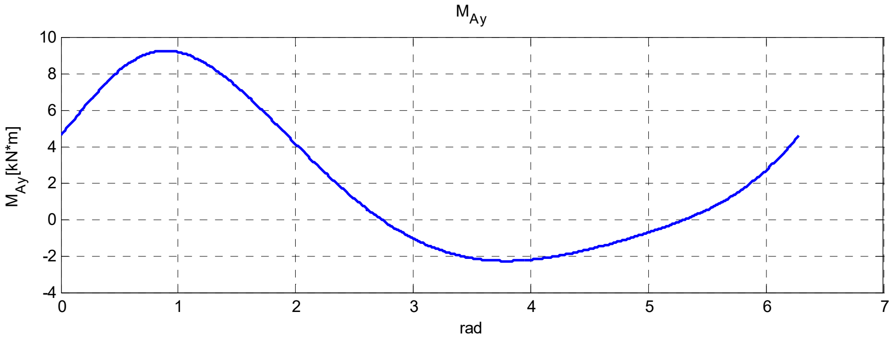

Figure 4. The statically undetermined reactions M_{AY} *vs.* φ_1

5. CONCLUSION

Under these conditions, based on the calculations described in the paper, a computational program was drawn using the Matlab code, and the results obtained for the undetermined static reactions are transcribed in the diagrams of Figures 2-4. In the absence of motion-compatible external forces (forces in the plane of motion and moment perpendicular to the plane of motion), the components R_{AX}, R_{AY}, M_{AZ} compatible with the movement of the reaction from A are null. The obtained variation diagrams represent periodical variations. For R_{Az} the extreme, minimum and maximum values are between; -40.0089, -21.6020. For component M_{AX}, values are between -20.3390 and -12.6537 and for M_{AY} between -2.2803 and 9.2389. Static undetermined reaction values are significant and can influence the movement of the mechanism. The minimum reaction moment, M_{AX}, in the absolute value, is greater than the momentum of the drive.

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2. Ingineria automobilului (in English: ***Automotive Engineering***)
ISSN 1842 – 4074
Period of publication: as of 2006 — Format: print and online, Romanian
Frequency: Quarterly — Electronic publication on: www.ingineria-automobilului.ro
Total number of issues: 50 — Type: Open Access
(including the March 2019 issue)
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3. Ingineria automobilului (in English: ***Automotive Engineering***)
ISSN 2284 – 5690
Period of publication: 2011 – 2014 — Format: online, English
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4. *Romanian Journal of Automotive Engineering*
ISSN 2457 – 5275
Period of publication: from 2015 — Format: online, English
Frequency: Quarterly — Electronic publication on: www.ro-jae.ro
Total number of issues: 17 (March 2019) — Type: Open Access
The above constitutes series nr. 4 of SIAR scientific magazine (English version).

Summary – on March 31, 2019

Total of series:	*4*
Total years of publication:	*25 (11: 1990 – 2000; 14: 2006 – 2019)*
Publication frequency:	*Quarterly*
Total issues published:	*80 (Romanian), out of which, the last 33 were also published in English*