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THE CALCULUS OF THE TECHNICAL-ECONOMIC PARAMETERS OF A SPARK-IGNITED DIRECT INJECTION ENGINE

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Abstract: *The authors have proposed a model to simulate the processes that occur in a SI direct injection engine. The modelling of the engine's cycle consists of: the presentation of the initial data, the computer program, the calculus and the correlation between the expressions of the engine's parameters and the gasoline injection in order to realize the computational program. The proposed cycle is a helpful one for computer simulation of the gasoline direct injection. The computer simulation enables the determination of the proposed theoretical technical-economic parameters, the theoretical mechanical work, the coefficients corresponding to adjust the indicated diagram, the theoretical mean pressure, the theoretical efficiency and the theoretical specific fuel consumption. After calculating the mechanical losses, the effective theoretical technical-economic parameters are also determined.*

Key-Words: *technical-economic parameters, SI engine, simulation, direct injection*

1. INTRODUCTION

The gasoline direct injection was introduced in order to improve the engine's efficiency and to reduce the fuel consumption and the polluting emissions.
The improvement of this type of engine can lead to superior performances and, especially, to the compliance with the severe environmental standards regarding the emissions.
In this paper the authors have proposed an original model to calculate: the parameters of the thermodynamic cycle, the mean indicated pressure and the efficiency of the engine.
This model can be used to study the SI direct injection engines and for adopting solutions that should lead to increase their performances.

2. THE MODEL OF THE DIRECT INJECTION SI ENGINE

The engine for which the calculus was developed has the maximum effective power P=54 kW, the cylinder's displacement V_s=0,389 dm^3, the compression ratio ε =9,25, the bore D=77 mm and the piston's stroke S=83,6 mm.
The model proposed by the authors is an original one because the model of the injection system through which is realized the calculus of the pressure regulator, of the electromagnetic injector and the duration of the injection is original.
This model includes two subprograms:
- one to calculate the parameters of the thermodynamic cycle depending on the rotational speed, on the environmental temperature and on the air excess coefficient, at the environmental temperature p_0=0,1 MPa;
- one to calculate the parameters of the thermodynamic cycle depending on the rotational speed and on the environmental temperature for an air excess coefficient λ=1, at the environmental temperature p_0=0,1 MPa;
- an own model was designed for the calculus of the pressure in the intake pipe and of the pressure at the end of the intake stroke.

Structurally, a gasoline injection system for an ICE represents a succession of energy conversions (C1 to C5 in figure 1).

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In Figure 1: EM is the operating electric motor, FP is the fuel pump, RCE is the regulating and control equipment and SIE is the SI engine.

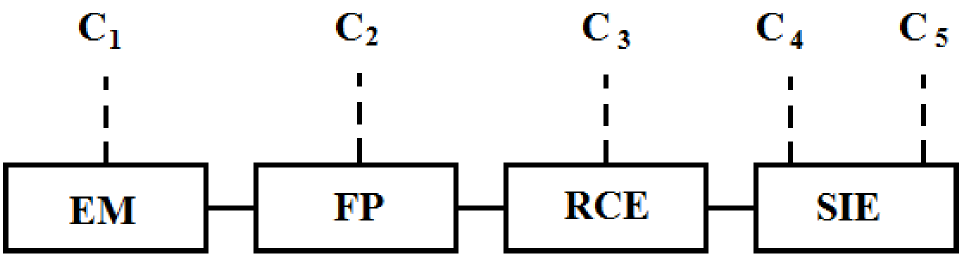

Figure 1. The scheme of the succession of energy conversion in an injection system

A series of laws, theorems, equations and basic relations from the theoretical mechanics, the fluid mechanics and mathematics are used to analyze and synthesize in dynamic regime the gasoline injection systems:

- from the theoretical mechanics:
- the fundamental laws of dynamics;
- the second Lagrange equation;
- the D'Alembert equation;
- from the fluid mechanics:
- the fluid's state equation;
- the characteristic equation;
- the law of viscose friction (Newton);
- the principle of mass conservation;
- the principle of energy conservation;
- the flow equation;
- the Kirchoff law in the hydraulic knots.

Most systems of differential equations that describe the phenomena occurring in an injection system are nonlinear. The method used by the authors to solve the system of differential equations is the linear analyze on parts [1]. The thermodynamic cycle on which the calculus was based is presented in figure 2.

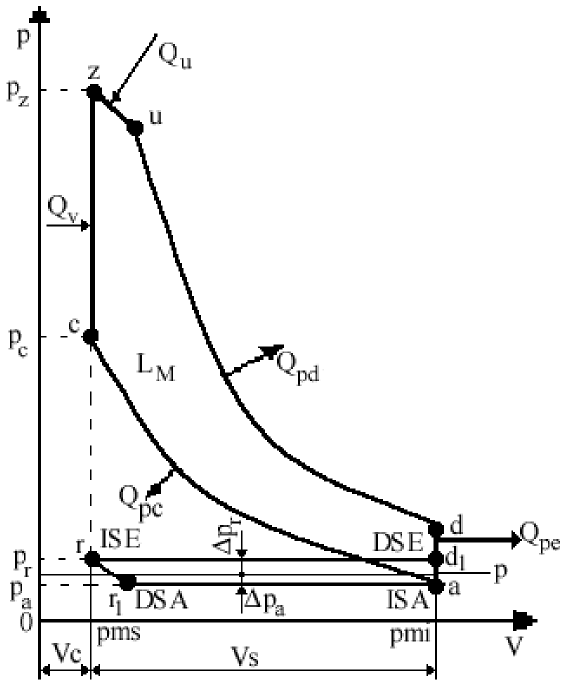

Figure 2. The thermodynamic cycle used for the calculus

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The simplifying assumptions on which the thermodynamic cycle was defined are the following:
- during the intake stroke (r_1-a) the pressure remains constant at the value p_a, smaller than the environmental one p_o;
- the exhaust process consists of two phases:
- one in which from the moment in which the exhaust valve opens the pressure drops sudden to the value p_r;
- one at constant pressure p_r during the exhaust stroke;
- the link between the exhaust valve and the intake valve is made through an adiabatic expansion of the residual burned gases, with constant isentropic exponent (r-r_1);
- the compression process (a-c) is considered to be a polytropic one, with constant exponent.
During this process, the working fluid transfers the heat Q_{pc} to the cylinder's walls.
- the burning process includes a constant volume phase (c-z) in which the heat input is Q_v and the polytropic faze (z-u) of exponent $n_u<1$, that defines the afterburning and in which the heat Q_u is transferred to the working fluid;
- the expansion process (u-d) is considered to be a polytropic one, with constant exponent (greater than the adiabatic exponent), in which the working fluid transfers heat to the cylinder's walls;
- the thermal agent is considered to be an ideal gas, with specific heat depending on the temperature.
The assimilation of the afterburning process with a polytropic one (with the sub unitary exponent n_u) is based on experimental data [2] showing that the temperature T_u at the end of the burning process is greater than the temperature at the end of the constant volume burning phase T_z as result of an intense heat release.
This heat release should lead to an increase of the pressure.
But the effect of the cylinder's instant volume increasing is greater so, during the afterburning, the pressure drops.
By modifying the exponent n_u one can highlight on the influence of the heat release speed during the afterburning.
On the other hand, one can use experimental data regarding the exponent n_d of the expansion process that characterize the general expansion of the gases (z-u-d).
In other words, by combining the afterburning (z-u) with a sub unitary exponent n_u with the actual expansion (u-d) with exponent n_d (greater than the adiabatic one k_d), one can obtain the actual expansion (z-d) characterized by a polytropic exponent $1<n_d<k_d$.
At the beginning all the temperatures and the pressures in the characteristic points of the thermodynamic cycle were calculated [3].
The burning efficiency, defined as the ratio between the heat transferred to the working fluid and the heat released by burning 1 kg of fuel, was also determined [3].

3. THE CALCULUS OF THE TECHNICAL-ECONOMIC PARAMETERS

The mechanical work developed by burning 1 kg of fuel:

$$L_M = L_{rr_1} + L_{r_1a} - \left|L_{ac}\right| + L_{zu} + L_{ud} - \left|L_{d_1r}\right|; \quad (1)$$

The mechanical work in the polytropic expansion of the residual burned gases, with constant isentropic exponent:

$$L_{rr_1} = \frac{p_o V_s}{1-\psi_e} \cdot \frac{1}{\varepsilon - 1} \cdot \frac{1-\psi^{\frac{k_e-1}{k_e}}}{k_e - 1}; \ [\mathrm{J}] \quad (2)$$

where:
- p_0 [Pa] – the environmental pressure;
- V_s [m^3] – the cylinder's displacement;
- $\psi_e=(p_r-p_0)/p_r$ – the coefficient of the exhaust pressure relative drop;
- ε – the compression ratio;
- $\psi=p_a/p_r$ – the global coefficient of the pressure losses;
- k_e – the mean adiabatic exponent of the exhaust process.

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The mechanical work in the intake stroke:

$$L_{r_1a} = p_o \cdot (1-\psi_a) \cdot V_s; \tag{3}$$

where:

$\psi_a=(p_0-p_a)/p_0$ – the coefficient of the intake pressure relative drop;
The mechanical work in the compression process:

$$|L_{ac}| = p_o V_s (1-\psi_a) \cdot \frac{\varepsilon^{n_c}}{\varepsilon - 1} \cdot \frac{1-\varepsilon^{1-n_c}}{n_c - 1}; \tag{4}$$

where:

- n_c – the polytropic exponent of the compression process;

The mechanical work in the afterburning process:

$$L_{zu} = \beta \cdot p_o V_s (1-\psi_a) \cdot \frac{\varepsilon^{n_c}}{\varepsilon - 1} \cdot \frac{1}{n_u - 1} \cdot \left[1 - \left(\frac{1}{\delta}\right)^{n_u - 1}\right]; \tag{5}$$

where:

- $\delta=V_u/V_z$ – the volume raise ratio in the afterburning process;
- n_c – the polytropic exponent in the afterburning process;
- $\beta=p_z/p_c$ – the pressure raise ratio in the constant volume burning;

The mechanical work in the expansion process:

$$L_{ud} = p_o V_s (1-\psi_a) \cdot \frac{\varepsilon^{n_c}}{\varepsilon - 1} \cdot \beta \cdot \delta \cdot \left(\frac{1}{\delta}\right)^{n_u} \cdot \frac{1 - \left(\frac{\delta}{\varepsilon}\right)^{n_d}}{n_d - 1}; \tag{6}$$

where:

- n_d –the polytropic exponent of the expansion process;

The mechanical work in the exhaust stroke:

$$|L_{d_1r}| = \frac{p_o}{1-\psi_e} \cdot V_s; \tag{7}$$

The theoretical mechanical work developed by burning 1 kg of fuel:

$$L_{Mt} = \eta_d \cdot L_M \tag{8}$$

where:

- η_d – a correction coefficient of the indicated diagram that takes into consideration the effects of the simplifying assumptions;

The theoretical indicated mean pressure:

$$p_{it} = L_{Mt} / V_s; \tag{9}$$

The theoretical efficiency [2]:

$$\eta_t = \frac{L_{Mt}}{\xi \cdot d \cdot \eta_v \cdot \rho_o \cdot V_s \cdot Q_i}; \tag{10}$$

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where:

- ξ – a coefficient that specifies the proportion of the heat released during the burning process that's transferred to the working gas;
- d – the air-fuel ratio;
- η_V – the volumetric efficiency [4];
- ρ_0 [kg/m^3] – the air density in environmental conditions;
- Q_i [J/kg] – the gross caloric power of the fuel;

The pressure losses due to the friction between the moving parts of the engine:

$$p_{mec}=a+b\cdot W_{pm} \tag{11}$$

where:

- a=0,21; b=0,037 – coefficients of which values are taken from tables with formulas for the approximate calculus of the mean pressure and of the mechanical losses depending on the piston's mean speed [2] [5];
- $W_{pm}=S\cdot n/30$ – the piston's mean speed (S is the stroke and n is the rotational speed).

The theoretical mean effective pressure:

$$p_{et} = p_{it} - p_{mec} \tag{12}$$

The engine's effective efficiency:

$$\eta_e = \frac{p_{et}}{\xi \cdot d \cdot \eta_v \cdot \rho_o \cdot Q_i}; \tag{13}$$

The theoretical effective power [5]:

$$P_t = \frac{p_{et} \cdot V_s \cdot i \cdot n}{30 \cdot \nu} \tag{14}$$

where:

- i=4 – the number of cylinders;
- v=4 – the number of the piston's strokes in one engine cycle;

The effective engine's torque:

$$M_e = 9550 \cdot P_t/n \tag{15}$$

The fuel mass injected in one engine cycle [6]:

$$m_{cb} = 10^3 \cdot \xi \cdot d \cdot \eta_V \cdot \rho_0 \cdot V_s \tag{16}$$

The hourly fuel consumption:

$$C_h = 120 \cdot \frac{m_{cb} \cdot i \cdot n}{\nu}; \tag{17}$$

The effective specific fuel consumption:

$$c_e = C_h/P_e \tag{18}$$

In Figure 3 is presented the variation of the fuel mass injected in one cycle m_{cb}, depending on the rotational speed and on the environmental temperature, for an air excess coefficient λ=1.

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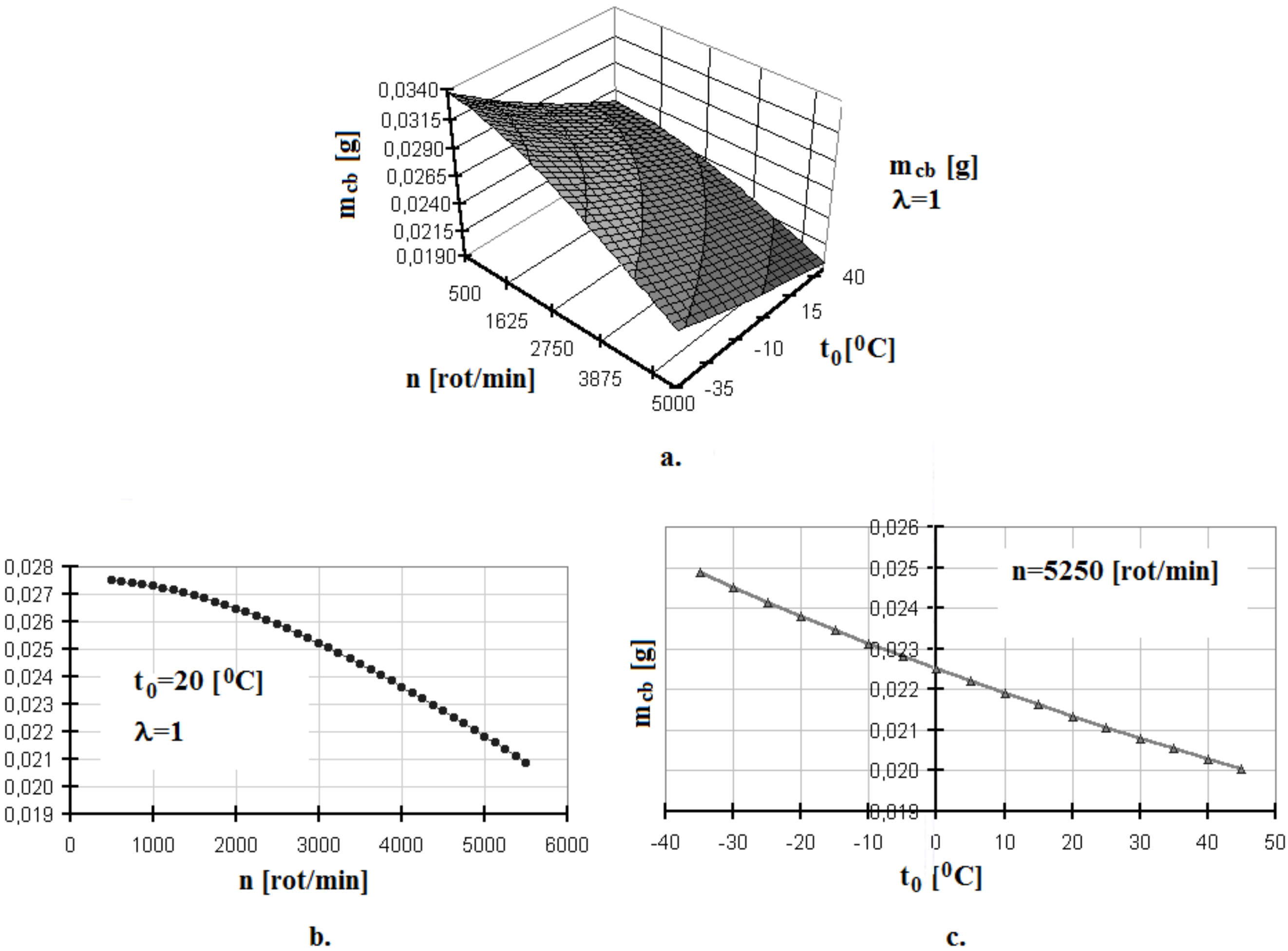

Figure 3. The variation of the fuel mass injected in one engine cycle: a) depending on the rotational speed and on the environmental temperature b) depending on the rotational speed c) depending on the environmental temperature

As shown in figure 3b, the fuel mass injected during an engine cycle decreases as the environmental temperature increases.
This is normal because, maintaining the same volume of injected fuel, the density decreases and so the mass also decreases.
Also (see figure 3c) the fuel mass injected during an engine cycle decreases as the rotational speed increases. This looks odd taking into consideration that the engine's power increases as the rotational speed increases. But again this is normal because also the number of engine's cycle in a given time increases with the rotational speed, so in the same time a greater amount of fuel is injected in the cylinder at higher rotational speeds.
In Figure 4 is presented the variation of the effective specific fuel consumption c_e, depending on the rotational speed and on the environmental temperature, for an air excess coefficient λ=1.
The calculated indicated diagram is presented in Figure 5.
These are all the engine's parameters that were determined with the proposed model.

4. CONCLUSION

In order to evaluate the accuracy of the proposed model, a comparison between calculated and experimental data for the effective specific consumption was made (Figure 6).
The maximum differences are registered in the rotational speed interval n=4000 – 5000 rot/min, with values between 18-21,5%. At low rotational speeds the differences are of 9,3-11,84%. Instead, at high rotational speeds the differences are only of 2,2-2,3%.

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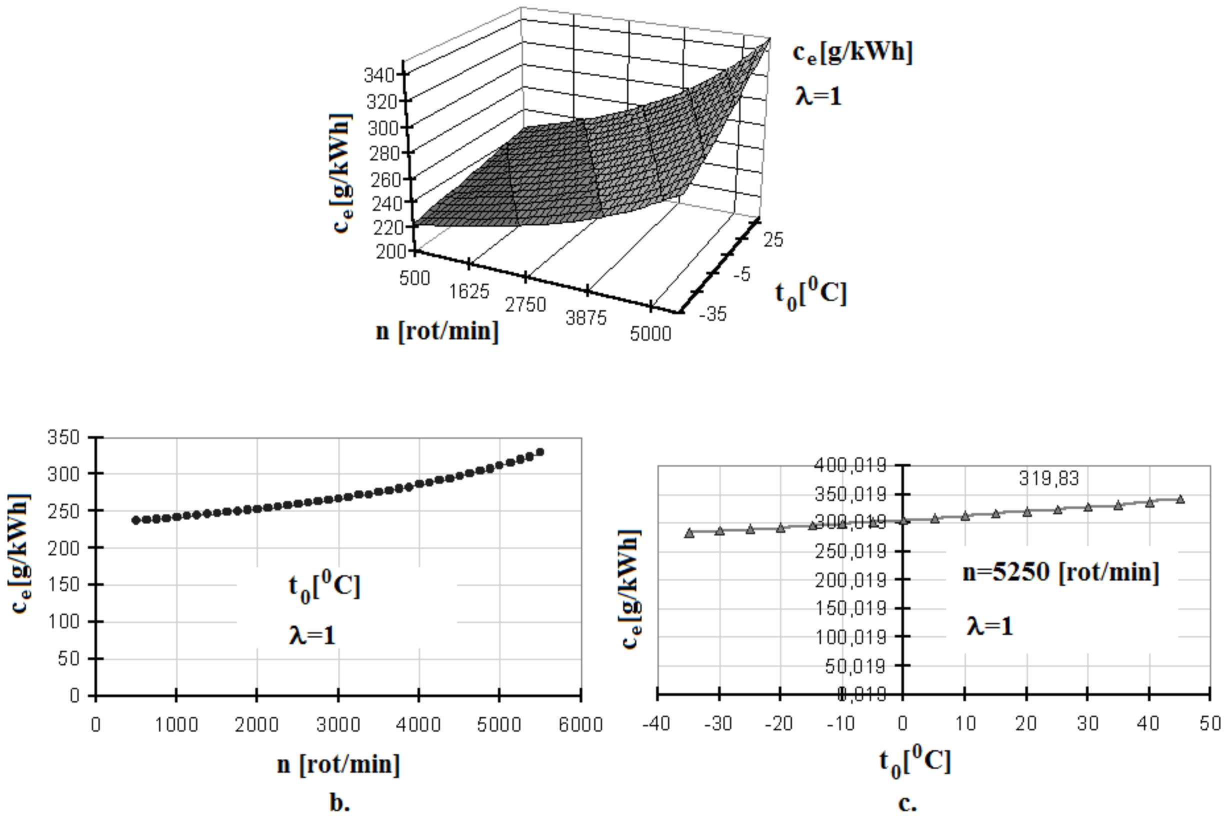

Figure 4. The variation of the effective specific fuel consumption: a) depending on the rotational speed and on the environmental temperature b) depending on the rotational speed c) depending on the environmental temperature

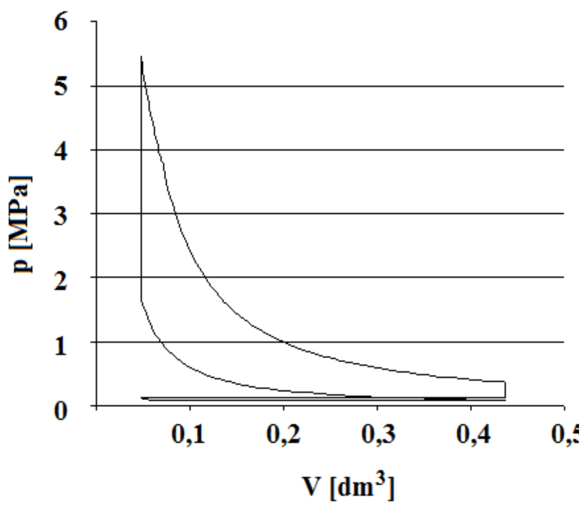

Figure 5. The calculated indicated diagram

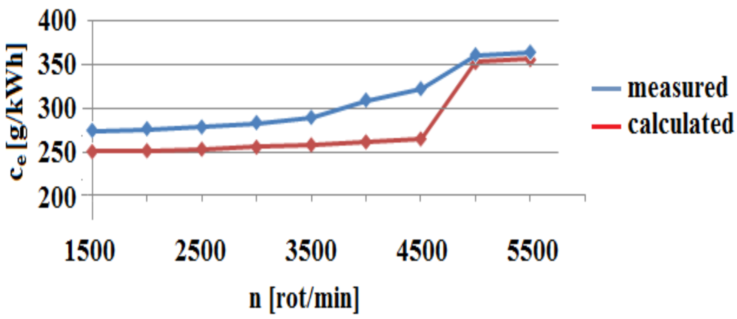

Figure 6. Comparison between calculated and experimental data for the effective specific fuel consumption

So, one can observe that the model is fully applicable for full loads at rotational speeds close to the one of maximum power.

For medium and low rotational speeds the differences between the calculated and the experimental data are relatively great. This means that the model of the injection system must be improved.

The next studies should be focused on the calibration of the injection system model at low and medium rotational speeds and at partial loads. At full load SI direct injection engines operate with stoichiometric mixtures and the fuel injection takes place during the intake stroke. It seems that the model developed by the authors covers very well this range of the engine's operation regime.

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In return, at partial loads and low-medium rotational speeds, SI direct injections are operating with ultra lean mixtures that lead to a reduction of the fuel consumption and of the polluting emissions.
In this case, after the closure of the intake valve, the injection can take place in multiple phases.
This is a phenomenon hard to model.
Maybe, after some corrections, the model proposed in this article could be used to design new types of direct injection SI engines or to improve the existing ones.

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APPLICATION OF ADVANCED ENGINEERING METHODS IN STUDYING A ROAD TRAFFIC EVENT BETWEEN A 12-WHEELER TRUCK AND A SMALL TOURISM IN A LOCAL JUNCTION FROM CLUJ-NAPOCA

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Abstract. *Road traffic and transportation are some of the most important factors that influence the social and economic contemporary life. Developing new systems and technologies for traffic control and transportation respectively all their auxiliary components is difficult without considering computer aided processing power in testing and evaluation. Main objective of the present paper is to point out the new and advanced method in car crash evaluation of two vehicles caught in a specific traffic event from Cluj-Napoca. Specific objectives are a theoretical study of the idea for advanced engineering method in car crash evaluation and an applied research developed in the Automotive Laboratory at Technical University from Cluj-Napoca with the motor vehicles implied in road event. Different distractions and inattention situations can transform in catastrophes for truck drivers, particularly with unexpected stoppages in traffic occur. Collisions warning with emergency brake for years now, helps road traffic participants in preventing rear-end collisions by regaining and refocusing their attention to the road and vehicles ahead of them. Thus, when a critical event is detected, a series of collision warnings and signs prompt the driver to react, and if the driver still fails to react, the emergency brakes make sure that a fatal collision is prevented anyway. Beside what already is on the spot present paper proposes an integrated complex system of communication and data transfer between multiple parts and layers involved in road traffic events. The concept and drawings are shown in the paper. Practical measurements were also considered.*

Key-words: road traffic event, truck, crossroad, advanced engineering methods

1. INTRODUCTION

Drivers may fail to sufficiently increase safety margins to allow time to respond to possible unpredictable events (e.g., lead vehicle braking). Advanced driver assistance systems should facilitate and possibly boost drivers' self-regulating behavior. For instance, they might recognize when appropriate adaptive behavior is missing and advice or alert accordingly.
The results from this study could also inspire training programs for novice drivers, or locally classify roads in terms of the risk associated with secondary task engagement while driving 0.
Established in 1997, the European New Car Assessment Programme (Euro NCAP) provides consumers with a safety performance assessment for the majority of the most popular cars in Europe.
Thanks to its rigorous crash tests, Euro NCAP has rapidly become an important driver safety improvement to new cars. After ten years of rating vehicles, Euro NCAP felt that a change was necessary to stay in tune with rapidly emerging driver assistance and crash avoidance systems and to respond to shifting priorities in road safety. A new overall rating system was introduced that combines the most important aspects of vehicle safety under a single star rating.
The overall rating system has allowed Euro NCAP to continue to push for better fitment and higher performance for vehicles sold on the European market. In the coming years, the safety rating is expected to play an important role in the support of the roll-out of highly automated vehicles 0.
The protection of children in motor vehicle crashes has improved since the introduction of child restraint systems. However, motor vehicle crashes remain one of the top leading causes of death for children. Today, computer-aided engineering is an essential part of vehicle development and it is anticipated that safety assessments will increasingly rely on simulations 0.

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In this paper is analyzed the concept of telecommunication feature which may be implemented in road traffic between vehicles and a headquarter in order to facilitate fast transferring information or event data to emergency units, police, firefighter, employer company, home and family members (or close relatives). In order to make this study a particular event was analyzed and proper system design was developed. Thus a Dacia Logan was under the research scope, a common vehicle in Romania and EU, which in this particular case was involved in a road traffic event with an 12 wheeler truck few weeks ago (April 2017). Before the crash the vehicle looked as in the Figure 1.

Figure 1. The studied vehicle type model Dacia Logan before the road-traffic event

In attention or distraction is a growing concern in the traffic environment and loss of forward road vision in combination with an unexpected event will normally result in an accident. Collision Warning with EMERGENCY BRAKING brings support to the driver in these critical situations by helping to focus the driver's attention towards the traffic in front of the vehicle. If a critical situation is detected, for example, slow or stopped traffic ahead, the Head Up Display on the windshield is be illuminated. If there is no reaction from the driver the Head Up Display starts to flash, supported by a loud audible alarm. If still no reaction from the driver the trucks brakes are applied with a low braking force. Finally if still no reaction from the driver is detected an emergency full-braking is performed. By supporting the truck driver in being aware of the traffic situation ahead, many accidents can be avoided. In order to alert other drivers of panic braking situations the brake lights on the truck will change from fixed to flashing, when extreme brake retardation is called for either by the driver or by the Collision Warning system 0. If the 12 wheeler truck would be equipped with the auxiliary system of “Collision Warning with Emergency Brake (CW-EB)” most probably the road event would not be taken place. Anyway, in this particular context there was not a CW-EB system installed on board and vehicles collided with each other. Figure 2 shows the end result of the road traffic event after the tourism was rear hit by a twelve wheeler truck. The event was generated when the tourism’s driver hit the brakes before cornering to the left toward a smaller street. The truck driver wrongly appreciated the situation and didn’t hit the brakes so the result was.

Figure 2. Vehicle after the road-traffic event (1-crashed area)

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There were young people and children involved in the car crash.
That was the main reason for sketching out a protocol for fast data transfer and proper analyzes in order to provide the interested parts (authorities and families) with adequate information about the road event.
In figure 3 is shown the Volvo concept of CW-EB in operation on a virtual model.

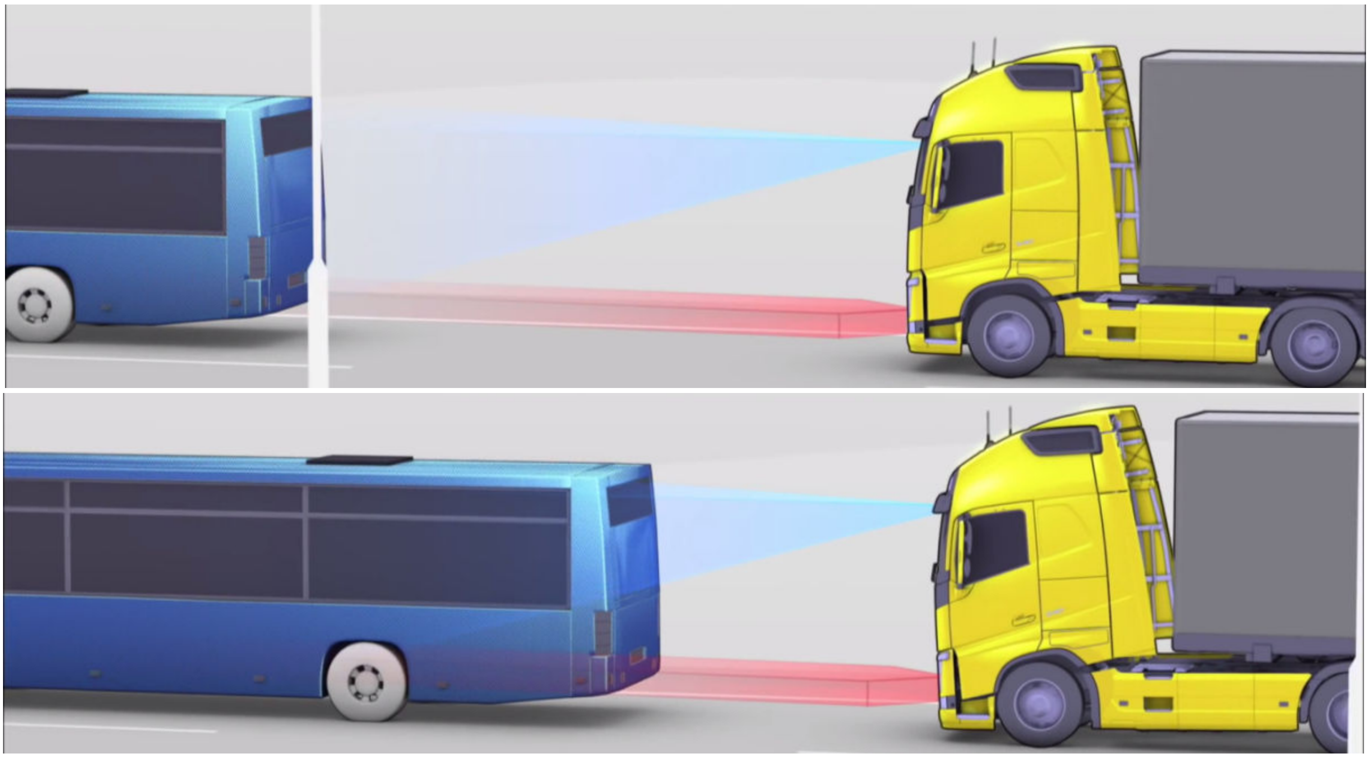

Figure 3. CW-EB from Volvo Trucks working in order to prevent a collision
a-start of detection and anticipation of the road traffic risk ahead;
b-emergency brake activation after the expected driver's response was not received by the management system in order to prevent the road hazard

The present paper shows the possibility of implementing a telecom concept in automotive products in order to improve the data transfer procedure and fast sending information to those that have an interest in the event.

2. METHODOLOGY AND MATERIALS

2.1. Methodology

By measuring the deformations, considering the materials and components that were crushed in the impact, respectively by using the computer aided evaluation, it may be assessed and interpreted the damage level. The present work highlights an engineering method for computer aided car crash evaluation and appreciation in the field of material damages concerning the road events in real time traffic. The studied and evaluated event is a real traffic accident which took place in Cluj-Napoca, and was analyzed with accurate means by use of latest technology. A new method of advanced engineering research was put into practice. The relation between computer processing power and car-crash evaluation effort is significantly underlined.

2.2. Materials

The present work makes use of two motored vehicles available and involved in road traffic event in order to analyze the causes of the problem and development possibilities for securing future life. Also there were used devices and equipment in order to improve data communication between the involved parties (road traffic participants, police, insurance companies, transport companies and owners) and authorities for a better solving of legal aspects, repairing process and financial aspects.
Figure 5 shows the rear part of the tourism involved in the road traffic event and odometer recorded data, which in some cases is important information.

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The materials used for making road traffic safer consists in mobile information storage and fast connections to a world wide web or a data transfer network, event documentation and damage appreciation. All of these and the specific components involved are considered in the process of implementing the fundamental idea of this work.

Figure 4. The motor vehicle crashed at the rear end by the 12 wheeler
(a-rear end of the vehicle after the crash; b-odometer actual value)

2.3. Mathematics and calculus

Inertia is the property of all material things especially that makes them to oppose any change in their change of motion-state. Galileo Galilei defined the concept of inertia in the late 1500 s. Almost a hundred years after Galileo introduced the concept of inertia, Newton used it to formulate his first law of motion – the Law of Inertia. The rapid transfer of kinetic energy is the cause of crash injuries, so managing kinetic energy is what keeping people safe in car crashes is all about. Crash worthiness is a term used to describe the protection a car offers an occupant in a crash and involves the structure, the restraint system, etc. Knowing the mass of the vehicle and the speed there may be determined the momentum and the force during the collision.
A more accurate definition is represented in the following relation:

$$F = \frac{m \Delta v}{t} \tag{1}$$

where:
F is force, in *N*;
m – mass, in *kg*;
Δ***v*** – change in velocity, in *m/s*;
t – time, in *sec*.
The impulse of the collision is expressed in the following relation:

$$Ft = m \Delta v \tag{2}$$

where:
Ft is impulse, in *N*sec*;
mΔv – change in momentum.

In the present case the truck travels with (v_{tr}) 50 km/h (which is 13.89 m/s) and has a mass (m_{tr}) of 30000 kg, while the tourism precedes the truck with (v_{to}) 5 km/h (1.389 m/s) and a mass (m_{to}) of 1.1 t:

$$Momentum \Leftrightarrow M_x = \begin{cases} M_{tr} = m_{tr} \cdot v_{tr} = 30000 \cdot 13.89 = 416700\,\text{kg} * \text{m/s} * 1\text{sec} \cong 4087827\ \text{Nm}, \\ M_{to} = m_{to} \cdot v_{to} = 1100 \cdot 1.39 = 1527.8\,\text{kg} * \text{m/s} * 1\text{sec} \cong 14987.72\ \text{Nm}. \end{cases} \tag{3}$$

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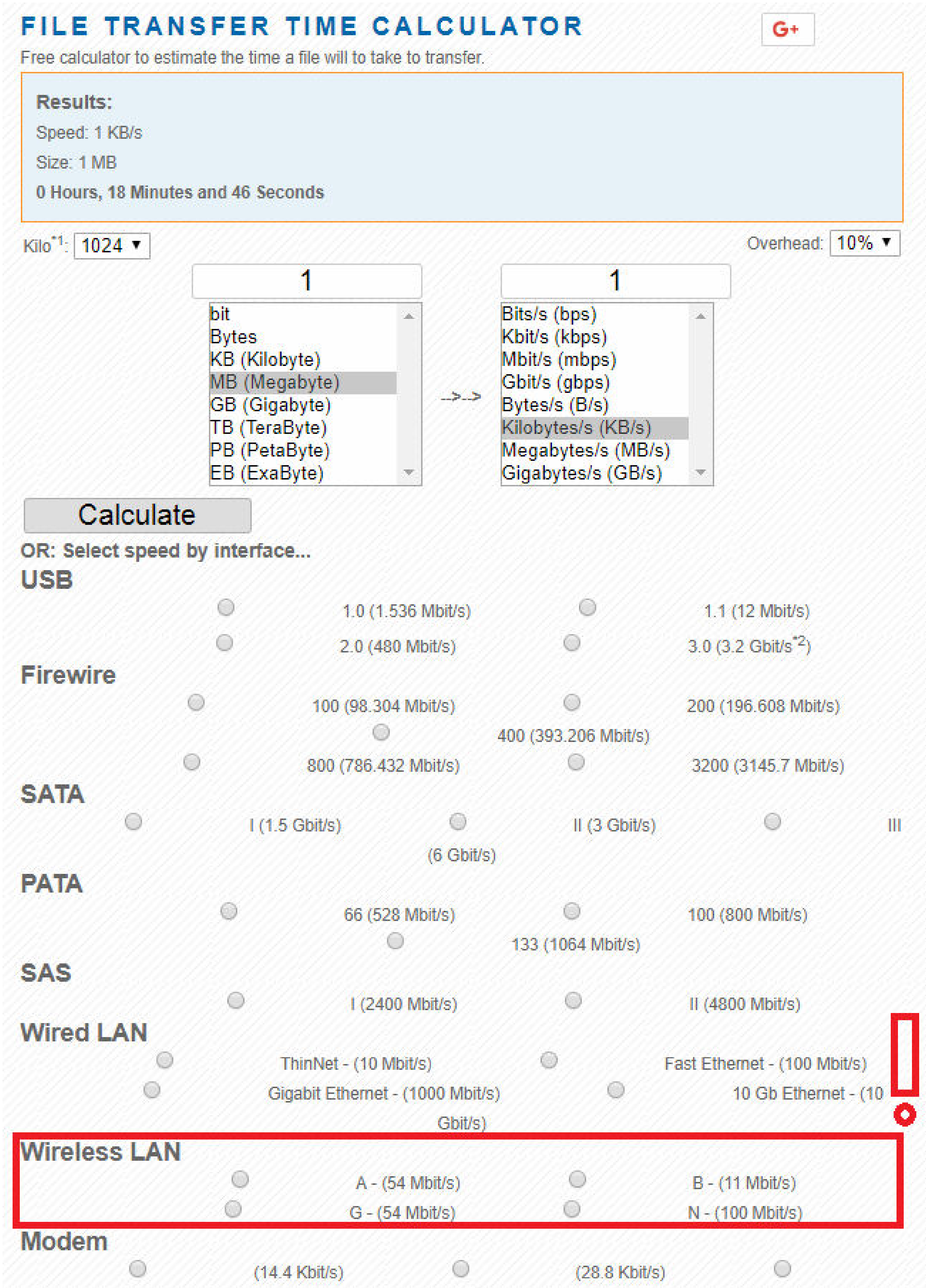

Figure 6. Calculus protocols of the data transfer rate - via different connections and speeds before/after the crash of the vehicles

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According to the theory of impulse of impact strength for vehicle 1 :

$$m_1\upsilon_1' - m_1\upsilon_1 = -\int_0^t F_R dt\,; \qquad (4)$$

for lorry (2):

$$m_2\upsilon_2' - m_2\upsilon_2 = \int_0^t F_R dt\,; \qquad (5)$$

When a collision is fulfilled law of conservation of linear momentum:

$$m_1\upsilon_1' + m_2\upsilon_2' = m_1\upsilon_1 + m_2\upsilon_2\,; \qquad (6)$$

Based on the previous equations we use decomposition into x and y coordinates, we get:

$$\begin{aligned} m_1\upsilon_{1.x}' + m_2\upsilon_{2.x}' &= m_1\upsilon_{1.x} + m_2\upsilon_{2.x}\,; \\ m_1\upsilon_{1.y}' + m_2\upsilon_{2.y}' &= m_1\upsilon_{1.y} + m_2\upsilon_{2.y}\,. \end{aligned} \qquad (7)$$

υ - velocity of vehicles before collision [m.s^{-1}]

υ' - velocity of vehicles after collision [m.s^{-1}]

m_1 - mass of vehicle 1 ; $m_1 = \frac{G_1}{g} = \frac{10791}{9{,}81} = 1100\,kg\,;$ (8)

$G_1 = 10791\text{N}$ – gravity of vehicle 1 including passengers, drivers and storage;

$g = 9.81 m.s^{-2}$ – gravitate acceleration;

$G_2 = 294300\,\text{N}$ – immediate weight of truck 2 including passengers, drivers and storage, the mass is 12782kg;

m_2 - truck mass 2 , $m_2 = \frac{G_2}{g} = \frac{294300}{9.81} = 30000\,kg\,;$ (9)

$\upsilon_{1.x}$ - component speed of vehicle 1 before the collision axis x;

$\upsilon_{1.y}$ - component speed of vehicle 1 before the collision axis y;

$\upsilon_{2.x}$ - component speed of truck 2 before the collision axis x;

$\upsilon_{2.y}$ - component speed of truck 2 before the collision axis y;

Velocity of vehicle after collision may be determined using energy conservation law. Comparing the kinetic energy and work consumed by sideways skid of the vehicle 1 and also work consumed in changing the amount of center of gravity after impact.

Figure 6. Calculus protocols of the data transfer rate - based on known existing models

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Making the transformations it may be observed that the truck momentum during the collision is around 4.08 MN·m/s.
These parameters may be calculated very easily and rapidly by the vehicle control unit (VCU) and initiate a data transfer when the distance is too little and the speed high or when the crash risk increases (or accident already took place).
Using mobile devices to get some access level on the Internet has become lately a common daily activity for many people. Information is transferred and consumed every time a mobile data user is connected to the Internet, whether for manipulating data, sending or receiving emails, downloading or simply browsing web pages. In this way the data transfer can be modelled and tested practically.

3. EXPERIMENTAL SETUP AND PROTOCOLS

3.1. Experimental setup for testing and measurements

In order to achieve the expected experimental results and to develop some interesting conclusions in relation with the application of advanced engineering methods in studying a road traffic event between a 12-wheeler truck and a small tourism in a local junction from Cluj-Napoca there were designed and implemented specialized devices as shown in figure 7 and figure 8.
Deformations and vehicle body displacements may be determined specifically in each particular place and case after the vehicles are stabilized as it is shown in figure 9.
Also a method of calculating the deformation taking into consideration the mass and energy conservation law in order to have an immediate estimation of the damage on site of the event and to send the data immediately to those who may have some concerns about it.

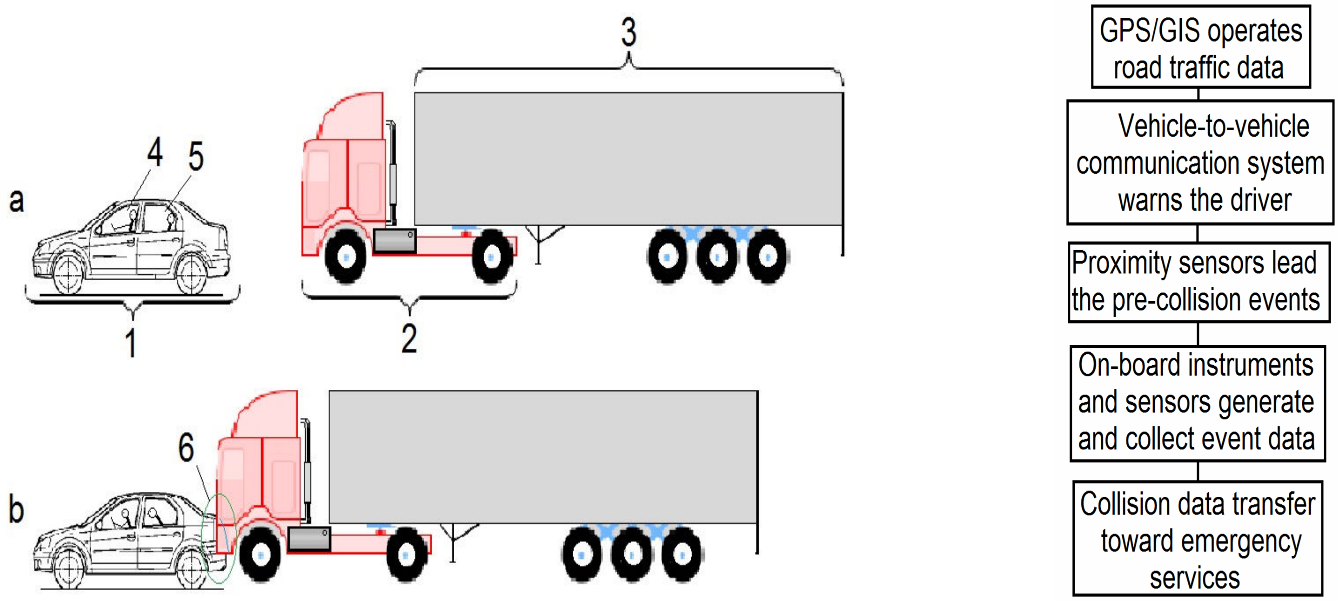

Figure 7. The sequences of the road traffic event
a-prior car crash; b-collision; 1-tourism; 2-semi-truck; 3-trailer; 4-driver (and probably the front occupants); 5-rear seat occupants (other passengers)

Deformations and vehicle body displacements may be determined specifically in each particular place and case after the vehicles are stabilized as it is shown in figure 9.
Also a method of calculating the deformation taking into consideration the mass and energy conservation law in order to have an immediate estimation of the damage on site of the event and to send the data immediately to those who may have some concerns about it.
The calculation of crush energy (and equivalent speed) is done by modelling the damage area into crush zones and then determining the energy needed to cause the damage in each zone.
The intrusion into each zone, called the crush depth, is measured by a strict protocol that is consistent with the measurements made during the original staged crash tests.

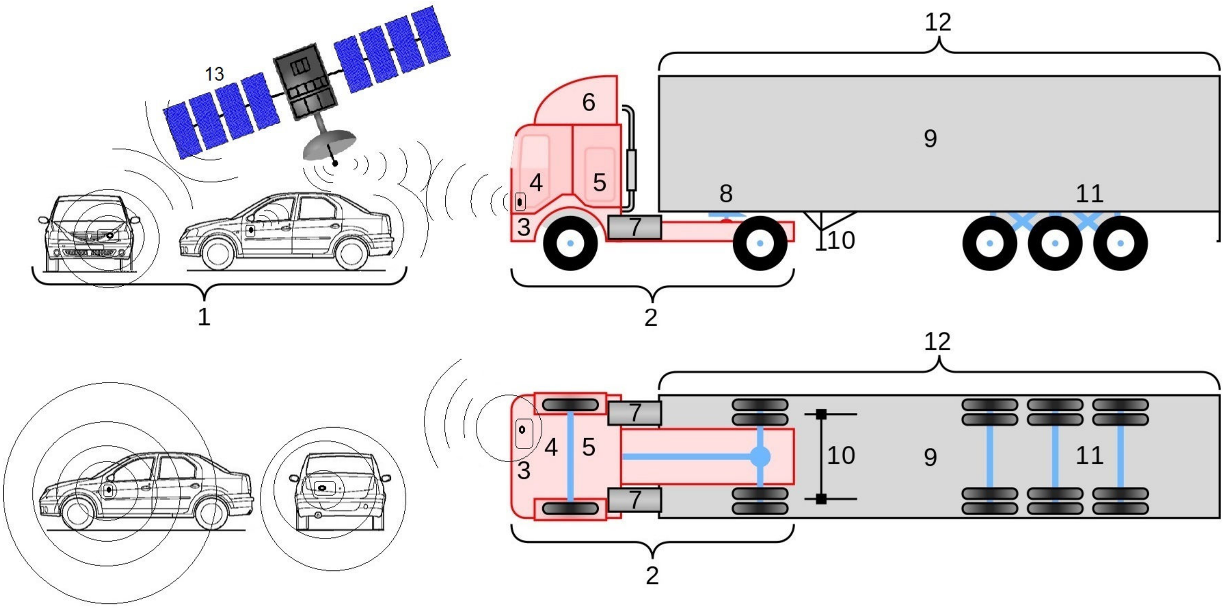

Figure 8. Experimental test equipment for improving the safety standards in road traffic 1-specialized equipped vehicle with VCU; 2-semi-truck with CW-EB and satellite communication capabilities; 3-receiver-transmitter via Wi-Fi connection; 4-driver location in the cabin; 5-resting space; 6-wind shield over the cabin; 7-fuel tank; 8-mechanical connection between the semi-truck and trailer; 9-storage space of the trailer; 10-supporting structure for trailer; 11-rear axles of trailer; 12-trailer; 13-satelit connection facility

3.2. Experimental testing protocol

By defining up a computer assisted communication model and determining the primary components which may give consistent signals of damage, as well as taking into consideration the cables and electric signals it may be generated the data encoding and transfer protocol.

In the present paper there is presented an experimental first stage of the system implementation realized till now by defining the necessary apparatus and communications protocols in order to gain access to information on crash site, as shown in figure 10.

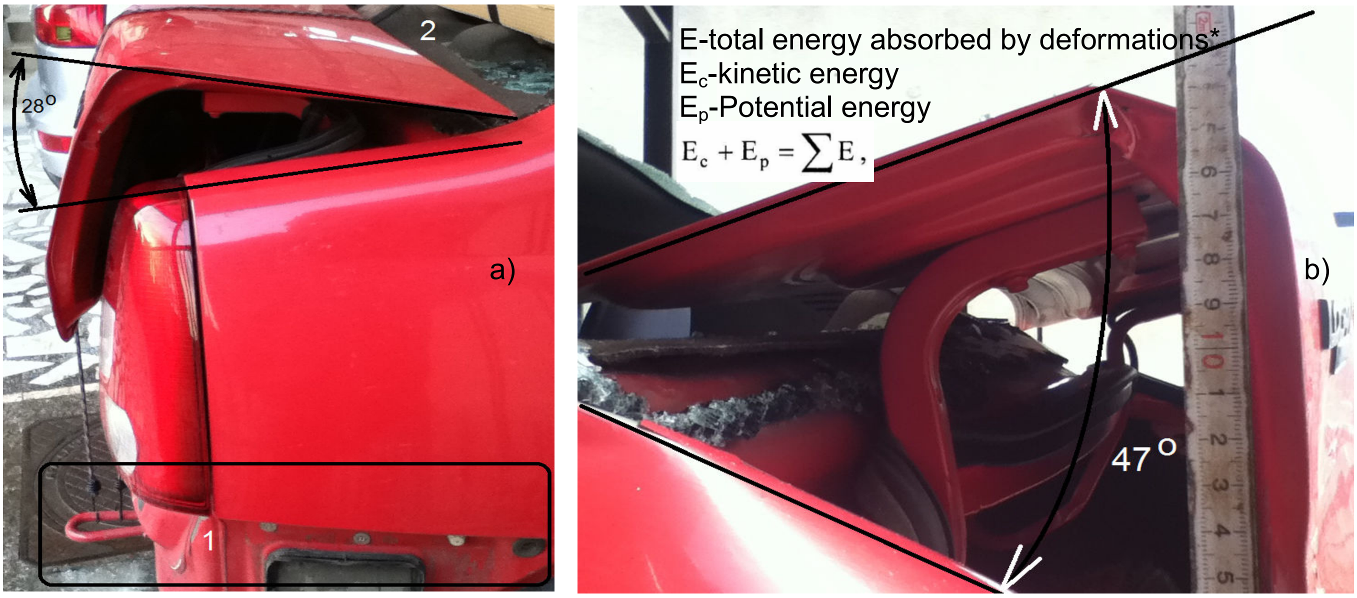

Figure 9. Measuring the deformations of the body components a-rear-right side of the crashed vehicle; b-rear-left side with a greater deformation of the components; 1-lower damaged area; 2-uper area (rear window)
**speed during collision is determined considering deformations*

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Figure 10. Crash site map and significant components that generate signals and data which may be considered in communication process
a-site map; b-crashed vehicle; y-tourism on site; z-truck on map; 1-light body destroyed; 2-cable interrupted; 3-cargo door displaced.

The vehicle considered is presented in table 1.
The light bulbs if deteriorated present some damaged signals when OBD2 protocol is applied so that signals, as well as the broken cables signals and door open/damaged signals may be encoded in a short message for emergency call or via Wi-Fi communication protocol in order to send the best and appropriate info to whom it may concern.

Table 1
Technical specs of tested tourism

Parameter	Actual Value
Manufacturer	Dacia
Model	Logan
Engine displacement	1500 cm^3
Fuel type	Diesel
Particular features	Engine Control Unit, ABS C.U., Central C.U.
Maximum speed	160 km/h
Number of standard control units / diagnosis method	3 / OBD 2

3.3. Limitations of the proposed study
The research is limited to the presentation and application of the new engineering method in computer aided car-crash evaluation for a specific road event with two motor vehicles, but it may be applied in complexes cases and specific situations also. The case study and method will be further improved.

3.4. The novelty of the achievement
The authors gained experience in manufacturing engineering, rapid prototyping, alternative technologies in manufacturing and computer aided technologies as well as in automotive engineering. The consideration of car crash evaluation aspects in road traffic conditions represents an extension of research capabilities and interdisciplinary experience. Even if there were also studied in the previous works some particular dynamic tests of vehicles, this is the first integrating research of so many important aspects that define road events evaluation and advanced engineering methods based on computer processing power available at Technical University from Cluj-Napoca.

4. CONCLUSIONS

Redefining the methods in safety and transportation is close related to the car-crash and road events research. Crash parameters are not manifesting alone as sole factors in conservative environment.

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A road event should be considered in relation with the complex traffic whole which is dynamic environment. The car crash aspects have a major influence in the vehicles and traffic development and their operation or correlation, but in the same time both of them (vehicle and the system) have a secondary financial impact when it comes to human society.
Development possibilities and trend lines are also shown in order to lead out the further studies on the proposed subject and to effectively create a safer road traffic environment.
The proposed idea transforms the motored vehicles in data storage devices and routers in a vast information network that has secure dynamic transfers as the main objective of its existence.
They even may determine by themselves degree of deformations and speeds.

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EMISSIONS CHARACTERISTIC OF DIESEL ENGINE FUELED BY BIODIESEL AT PARTIAL LOAD

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Abstract. *All This paper investigates the performance and emissions characteristics of a diesel engine fueled by biodiesel at partial load conditions. Experiments were conducted on a Renault K9K common-rail fuel injection diesel engine using mineral diesel and blends fuels of 6% and 10% with biodiesel obtained from corn oil and sunflower oil. The testes were made for 50%, 75% and 100% engine load. During experimental tests were measured the engine torque and the emissions of CO, CO2 and NOx. The engine torque for engine fueled with biodiesel blends is higher than for diesel fueled with mineral diesel in all tests. The CO and NOx, emissions decrease for all blends, and the CO2 increase. These are the results of better oxidation and of lower carbon content of biofuels.*

Key-words: *Biodiesel fuel, emissions, diesel egine, partial load*

1. INTRODUCTION

Generally, biofuels can provide an excellent opportunity to reduce some of the harmful emissions without costly engine changes. Because vegetable oils satisfy the main requirements of the diesel engine, in the latest years has been investigated their use as alternative fuels.

From all biofuel for diesel engines, biodiesel is a promising fuel, because it is a sulfur-free, non-toxic, oxygenated, renewable biofuel, and more than 90% can be biodegradable within 21 days [1].

Due to differences in chemical and thermal-physical properties (higher density, viscosity, cetane number, bulk modulus, and oxygen content) are differences in biodiesel's performance, combustion and emissions characteristics [2]. Biodiesel has a shorter ignition delay than diesel due to a higher cetane number and a better combustion due to additional oxygen content. Because of lower heating value of biodiesel the maximum power of the engine can be smaller. To compensate the loss of power a larger amount of biodiesel should be injected into the combustion chamber [3].

Diesel engines mostly operate under partial load conditions, especially for cars. Therefore, it would be more valuable and important to evaluate the performance of biodiesel during partial load conditions.

In this paper corn oil and sunflower oil biodiesel were tested at 50%, 75% and 100% engine load conditions.

2. EXPERIMENTAL SETUP

The experiments were made on a Renault K9K common rail fuel injection diesel engine. A schematic diagram of the engine test bed is shown in Figure 1.

The engine specifications are presented in Table 1.

The engine test bed is equipped with an electric Dynas3 LI250 dynamometer, which is designed for operated within a range of 0-8000 rotations per minute. It can measure engine power up to 250 kW with an accuracy of ±2%. For the measurement of CO and NOx emissions was used the HGA 400 Pierburg chemiluminescence analyser.

This analyzer can measure CO emission between 0 and 10% vol, with accuracy of 0.01%; for the NOx emission the range is between 0 and 5000 ppm, with an accuracy of 1 ppm.

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Table 1.
Engine properties

Engine type	Renault K9K four stroke
Number of cylinders	4
Bore (mm)	76
Stroke (mm)	80.5
Total displacement (cm^3)	1451
Compression ratio	15.3
Fueling	Common-rail direct injection

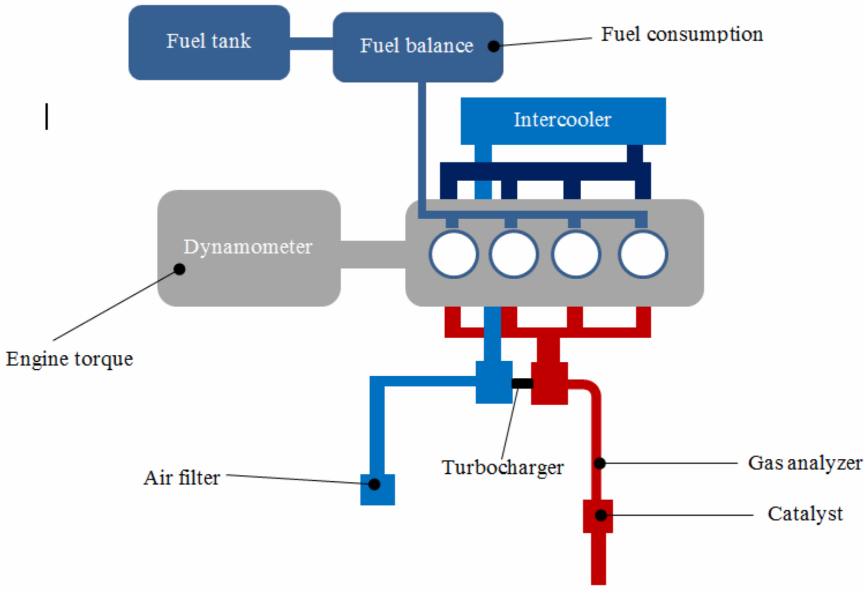

Figure 1. Schematic diagram of the engine test bed

In the present paper, biodiesel and its blends (Diesel, B6 and B10) were studied at different engine speed from 1200 rpm to 3700 rpm with an interval of 500 rpm and for three different engine loads 50%, 75% and 100 %. The biodiesel was produced from corn oil and sunflower oil.
The properties of fuels used in the tests are presented in table 2.

Table 2.
Tested fuels properties

Fuel	D	B6C	B10C	B6SF	B10SF
Density at 20°C (kg/m^3)	840.2	841.7	842.4	841.9	843.1
Viscosity at 20°C (mm^2/s)	5.34	5.04	4.99	5.27	5.10
Cetane number	51.1	57.6	62.1	54.5	57.6
Flash point (°C)	67	71.4	67.3	67.2	67.8
Caloric value (MJ/kg)	43.16	42.63	42.27	42.58	42.19

D – Diesel fuel; B6C – 6% biodiesel from corn oil; B10C – 10% biodiesel from corn oil;
B6SF – 6% biodiesel from sunflower oil; B10SF – 10% biodiesel from sunflower.

The engine was running at medium speed and load until the cooling water temperature stabilizes at 80 °C.

3. RESULTS AND DISCUSSIONS

3.1 Impact of biodiesel blends on engine torque

Figure 2 presents the variation of engine torque under full load for diesel and blends (6% and 10%) with biodiesel form corn oil and sunflower oil.

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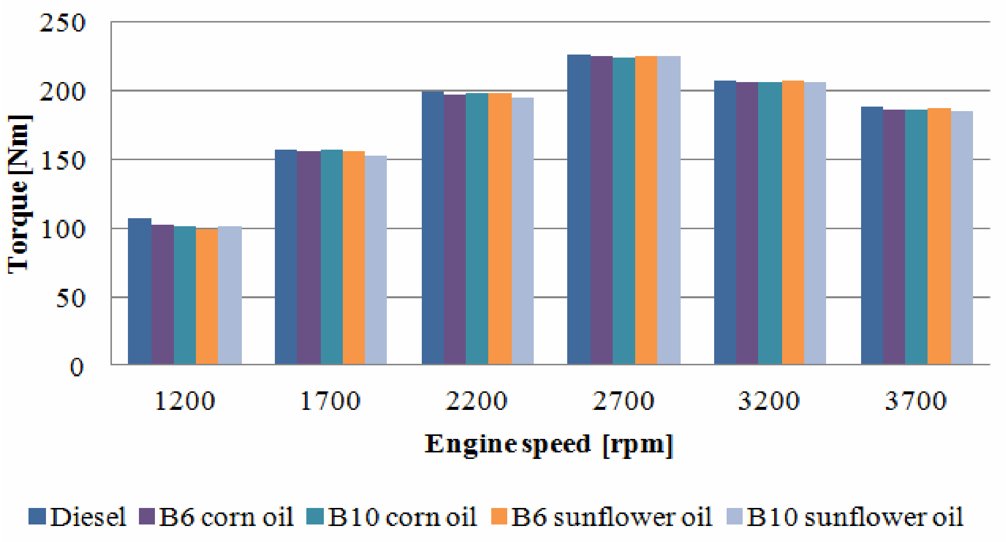

Figure 2. Variation of engine torque at full load.

The torque values for all tested fuels increase with the increase of engine speed until 2700 rpm, and then the torque decrease. For all blends tested was observed a small drop compared to diesel fuel.
The maximum reduction is for B6 form sunflower oil at 1200 rpm (7.8%) and the smaller reduction is for B6 for sunflower oil at 2700 rpm (0.1%).
These declines are due to the lower calorific value for the blends of biodiesel.

3.2 Impact on carbon monoxide emissions.
Figures 3, 4 and 5 present the CO emissions for 50%, 75% and 100% engine load. It is observed that the emissions of CO decreases with the increasing of the engine speed.
The CO emissions for blends are bigger for the 6% blends versus 10% blends because the extra oxygen atoms contained into the biodiesel reduce the fuel-air ratio and provide a better fuel burning.

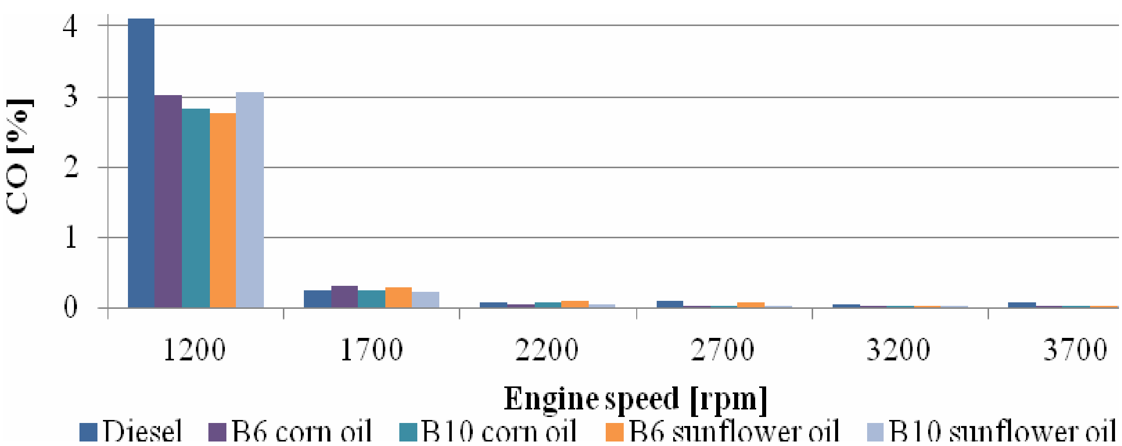

Figure 3. Carbon monoxide (CO) emission at 50% engine load.

Also the CO emission decreases with the increasing engine load. At lower load combustion conditions, the air fuel ration becomes too lean for a complete combustion, especially at lower engine speeds.
The greater viscosity of the blends plays a role on combustion process during partial load conditions, increasing the tendency for an incomplete combustion. With the increasing of engine load the CO emission decrease. This is because with the increase of engine loads, the fuel injection pressure increase and cancels the effect of higher viscosity of biofuels. Because of higher injection pressure the fuel spray is mixing better with the air from the cylinder and implicitly a more complete combustion.
And the oxygenated nature of biodiesel becomes advantageous which tends to result in more complete combustions and reduces the CO emissions.

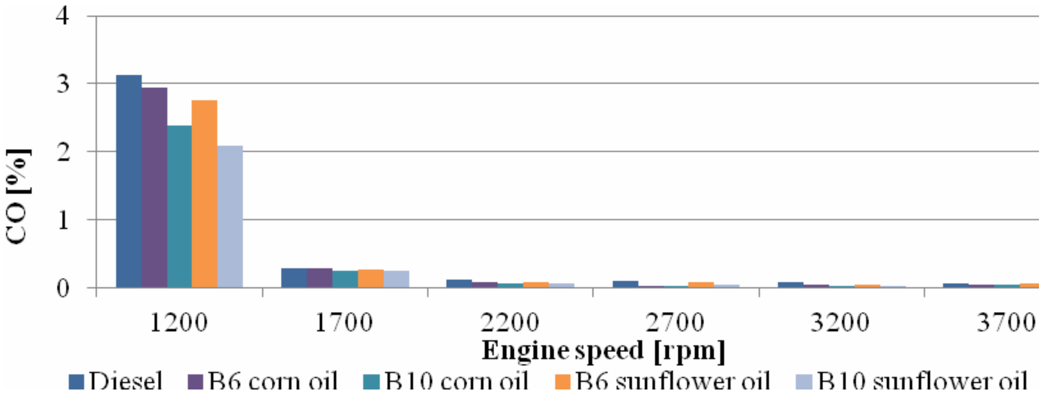

Figure 4. Carbon monoxide (CO) emission at 75% engine load

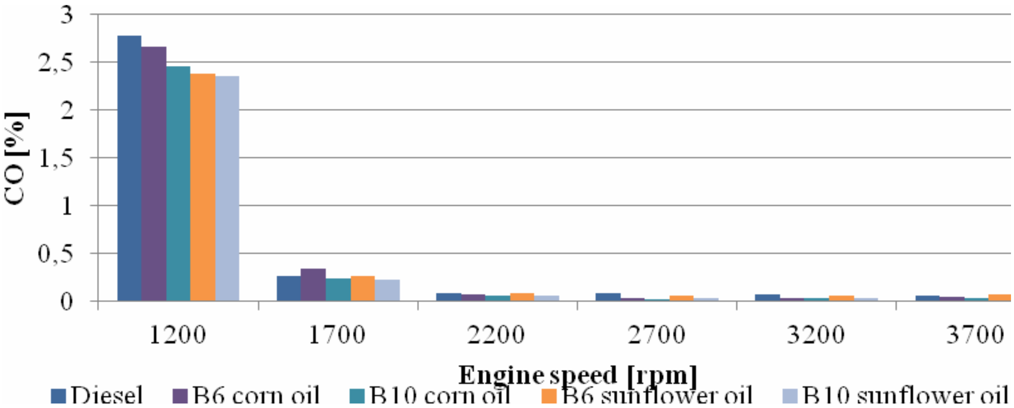

Figure 5. Carbon monoxide (CO) at 100% load.

3.3 Impact on carbon dioxide emissions

Figures 6, 7 and 8 present the CO_2 emissions for 50%, 75% and 100% engine load. For all partial load and full load the emission of carbon dioxide is greater when the engine is fueled with biodiesel blend. This increase of emission could be due to the higher density of biodiesel. The fuel injection is made volumetric. Due to higher density of biodiesel is injected a larger mass of fuel under complete injection. Another explanation is that the biodiesel give a more complete combustion, and the CO is transformed in CO_2. For the biodiesel from corn oil the emissions of CO_2 is higher for 6% blends and for biodiesel from sunflower oil the emissions is higher for 10% blends.

The flash point for 6% biodiesel form corn oil blend is higher than the flash point of 10% biodiesel form corn oil blend. Due of this the B6 from corn oil had a less complete combustion than B10 from corn oil.

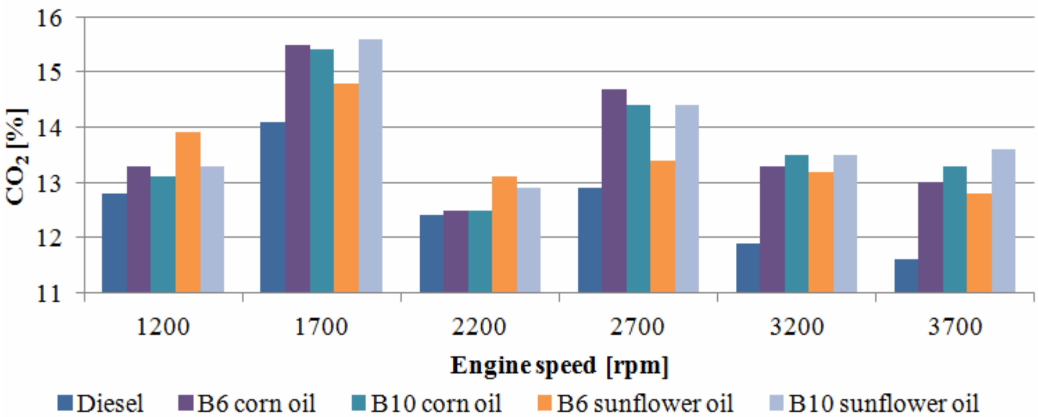

Figure 6. Carbon dioxide (CO_2) at 50% load

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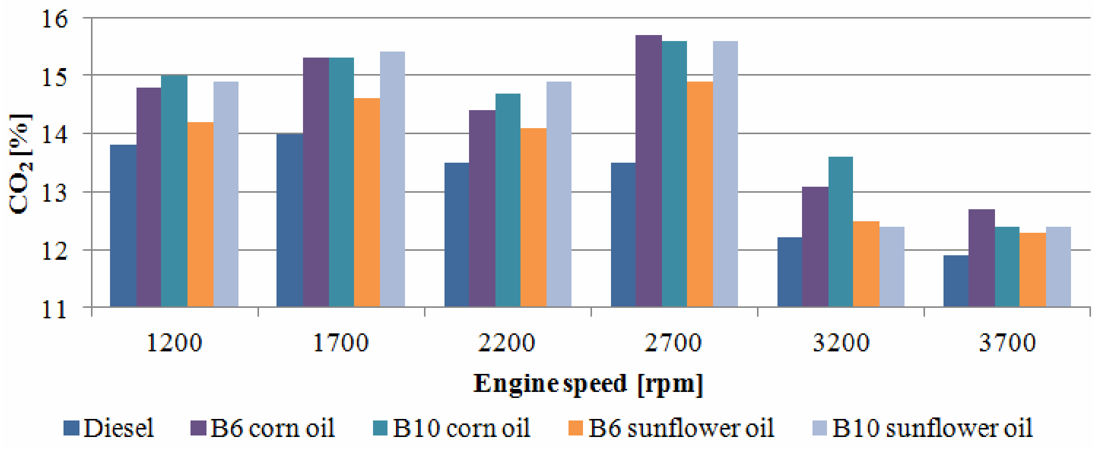

Figure 7. Carbon dioxide (CO_2) at 75% load

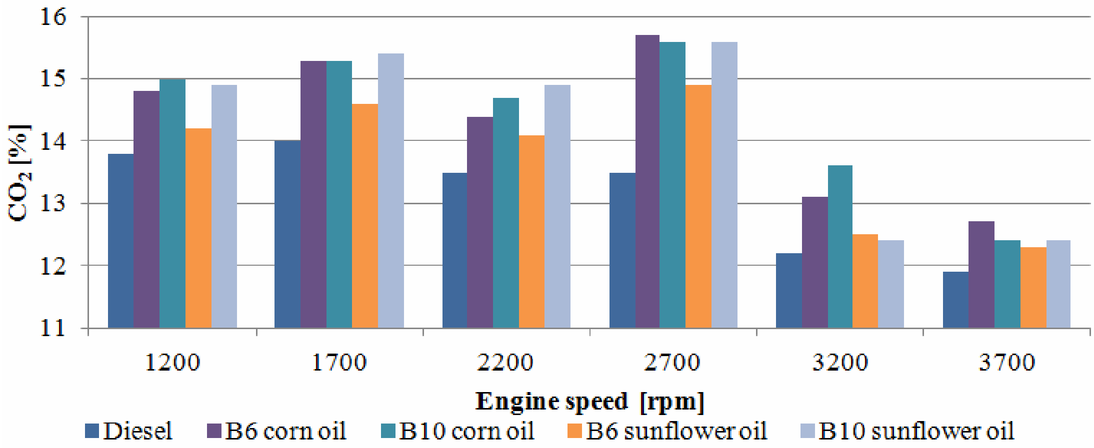

Figure 8. Carbon dioxide (CO_2) at 100% load

Another reason for the higher value of CO_2 emissions of blends is the supplementary oxygen contents of biodiesel that transform the CO emissions in CO_2 emissions.
Compared with the CO emissions, the impact of partial loads on CO_2 emissions is less pronounced.

3.4 Impact on nitrogen oxide emissions

Figures 9, 10 and 11 present the NO_x emissions for 50%, 75% and 100% engine load.
For all tested fuels the NO_x emissions increase with the increases of engine speed and decreases with the increases of the engine load.

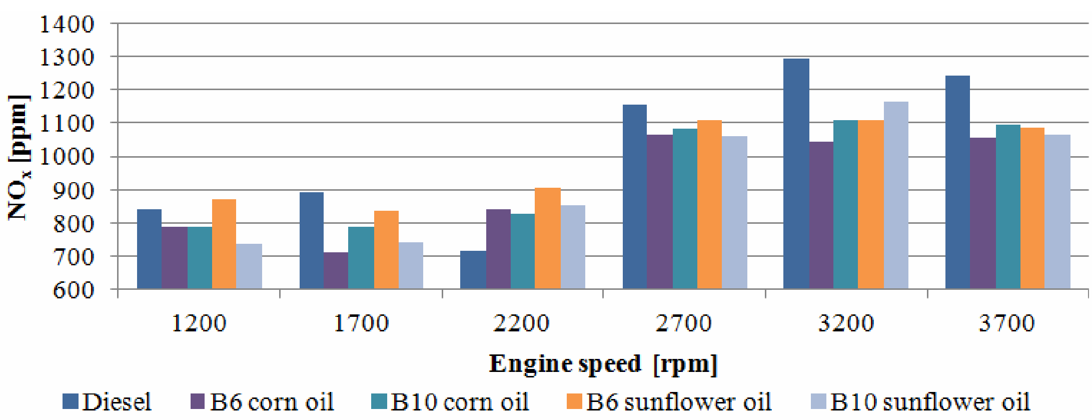

Figure 9. Nitrogen oxide (NO_x) at 50% load

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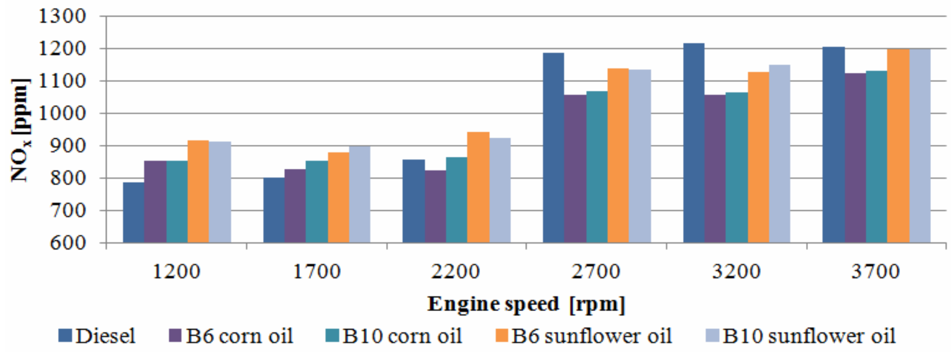

Figure 10. Nitrogen oxide (NO_x) at 75% load

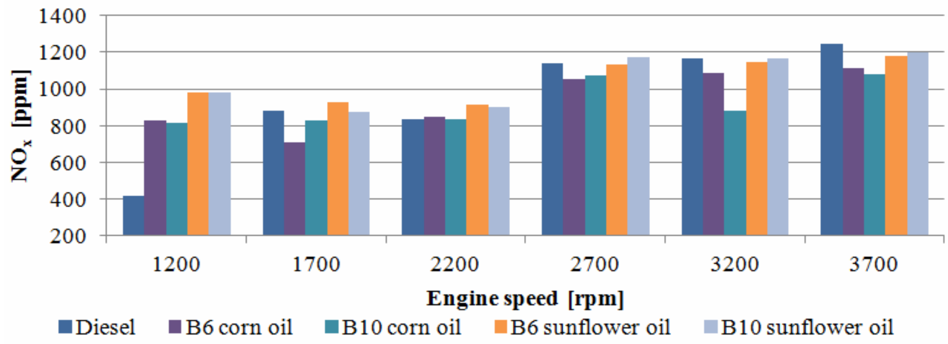

Figure 11. Nitrogen oxide (NO_x) at 100% load

The NO_x emission is sensitive to the combustion temperature. It is agreed that a higher temperature in the cylinder generate a higher NO_x emission. Because the cylinder temperature can not be measured, the exhaust temperature has been used as a comparison factor. The exhaust temperature was smaller when the engine was fueled with biodiesel blends. The exhaust temperature increases with the increasing of load and speed. Generally the NO_x emission was smaller when the engine was fueled with biodiesel blends. For all partial and full loads, at small and medium engine speed (1200 rpm – 2200 rpm) the NO_x emission was greater for biodiesel blends. At higher engine speed (2700 rpm – 3700 rpm) the NO_x emission was greater for diesel fuel.

4. CONCLUSIONS

In this study, the experiments were made using mineral diesel fuel and four blends (6% and 10%) with biodiesel obtained from corn oil and sunflower oil to investigate the impact of biodiesel on performance and emission characteristics of diesel engine at partial and full load. Due to the lower calorific value of biodiesel blends, the engine torque was smaller for all biodiesel blends. The CO emission decreases with the increasing of the engine speed and decreases with the biodiesel blend ratio. The CO emission decreases with the increasing of the engine loads because the fuel spray is mixing better due to the increases of the fuel pressure injection. The supplemental oxygen helps to a better combustion and reduces the CO emission. The CO_2 emission is higher for all biodiesel blends. One of the reasons is the greater density of the biodiesel blends. The injection of fuel is made volumetric, so the mass of fuel injected is greater for biodiesel blends. Another reason is the supplemental oxygen that helps to transform the CO in CO_2 emission. For NO_x the variation is random.

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THE SIMULATION OF THE DYNAMIC BEHAVIOUR FOR AN ELASTIC MECHANICAL TRANSMISSION OF PASSENGER CAR

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Abstract. *In the last years the engine-transmission system was optimized substantially with the purpose to obtain advanced dynamic performances. The transmission must cooperate with the propulsion system in a perfect way to obtain maximum acceleration performances and a low fuel consumption. This paper presents an extensive study of the acceleration performances of an automobile equipped with a mechanical transmission, the study is accomplished through numerical simulation. The numerical simulations were made with the Matlab-Simulink program. The article analyses the dynamic behaviour of an elastic mechanical transmission. The obtained results are similar with the results given by the constructor. The conclusion of the study is that the simulation model presented in this paper can be successfully used for all the automobiles equipped with a mechanical transmission for a fast analyse of the acceleration performances, as well can be used to optimize these performances.*

Key-words: *simulation, mechanical transmission, car, dynamic behaviour*

1. INTRODUCTION

The functioning of the passenger car in operating conditions takes place in wide limits for the vehicle speed, the payload and for the quality of the roads [7].
For the study of the acceleration performances an important factor is the engine speed characteristic of the engine.
The maximum acceleration performances are obtained when the engine is working at the total load [6]. Similar researches of the dynamic performances simulation are presented in the papers [2][8].
The friction modelling constitutes the base of all clutch models. Examples of friction models (hyperbolic tangent model) are presented in paper [1][4].
A hard task is solving the model equations including friction elements, because the adaptive time step methods integrating zero crossing detection are not achievable for real-time fruition [3].
The automobile acceleration performances can be estimated with the acceleration characteristic.
The acceleration characteristic can be assessed through the variation of the acceleration time reported to the passenger car speed at total load, as well can be assessed using the variation of the acceleration distance reported to the passenger car speed at total load. An automobile with a very good acceleration it is capable to increase his speed in very short space of time.
The importance of the acceleration results from the fact that the time of travelling with uniform speeds are relatively low. The acceleration characteristics can be settled by way of using the software programs for modeling and simulation (Matlab-Simulink, AMESim, etc), as well by experimental way using the dynamometric stand, and also the special tracks for researches can be used.
Obtaining the acceleration performances through the experimental method implies a direct measuring of the parameters, that involves costs for the materials and some special conditions for researches, especially in case of operating on a track.
The benefits of using modelling and simulation programs are: low costs in comparison with the experimental method, does not need the presence of the vehicle, neither the special measuring devices, and the time for obtaining the results of the acceleration performances is significantly reduced.

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2. THE MODEL PRESENTATION

For the modelling and simulation process of the acceleration performances there were implemented mathematical equations in the Matlab-Simulink program.
To implement the mathematical equations in an easier way and further to insert the numbers, the global model was divided in more sub models, every sub model representing a subsystem from the engine-transmission system of the vehicle subjected to the simulation (Figure 1).

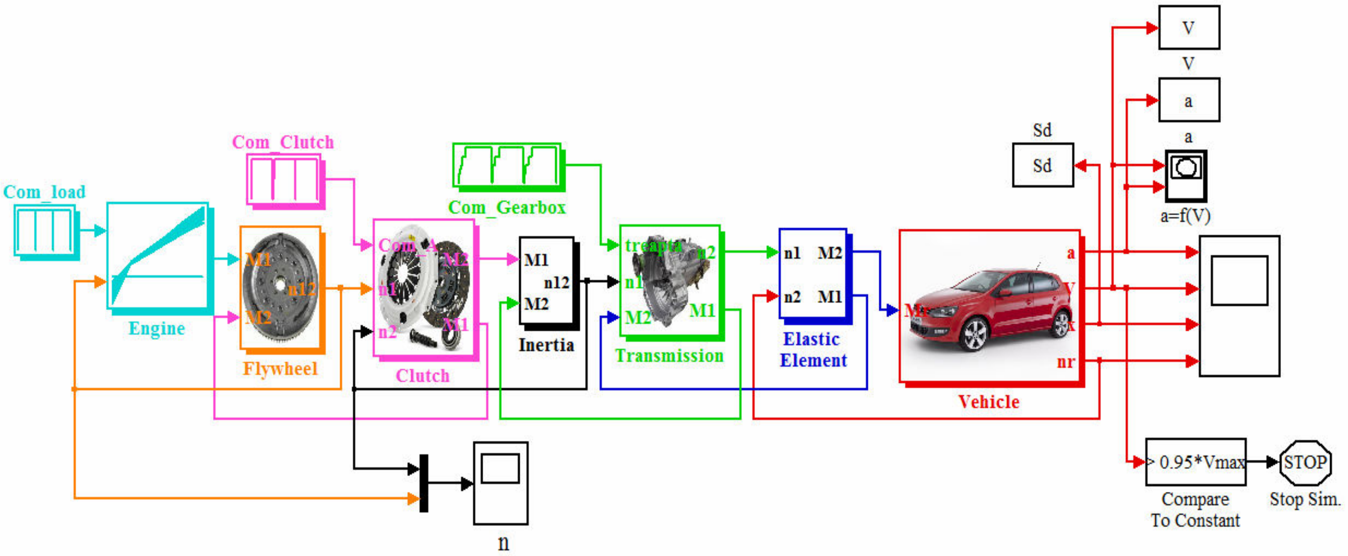

Figure 1. The global model simulated for the acceleration study

The engine was implemented through a mathematic function of two variables, with the load and engine speed variables starting from the maximum load characteristic given by the constructor that was further modelled for partial loads and for the engine brake regimes (Figure 2).
The torque at partial load was obtained from the full load torque, correlated with the throttle valve angle. When the accelerator pedal is complete depressed (100%), this corresponds to the engine full-load curve, and when the accelerator pedal isn't depressed (0%), this corresponds to the engine thrust characteristic curve [6].

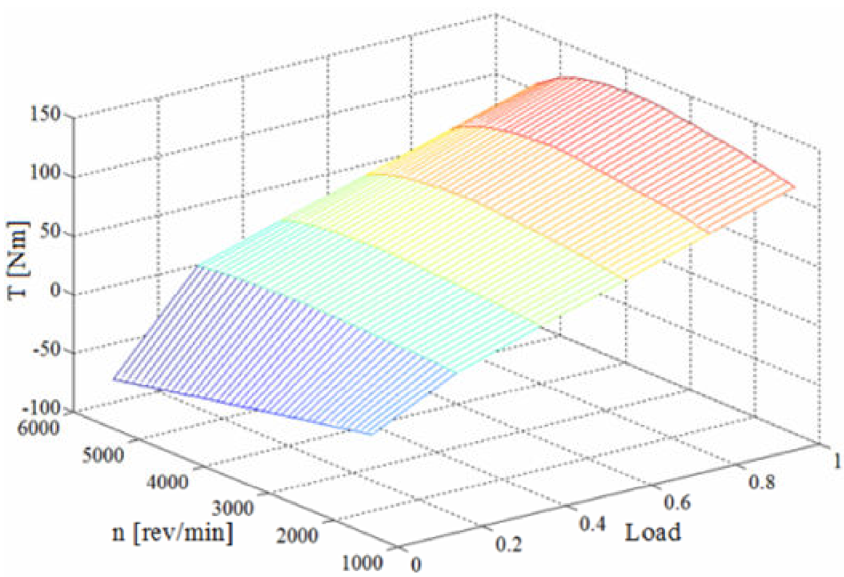

Figure 2. The engine speed characteristic of the engine for partial loads and engine braking digitized for the vehicle subjected to simulation

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The acceleration performances study for the simulated vehicle were used a series of mathematical equations presented in the papers [5, 9]. Starting from the wheel traction balance:

$$\frac{dv}{dt} = \frac{F_t - (R_{rul} + R_a)}{\delta \cdot m} \text{ [m/s}^2\text{]} \tag{1}$$

where:

$\frac{dv}{dt}$ – the vehicle acceleration;

δ – influence coefficient for the rotating masses;

m – vehicle mass;

F_t – traction force;

R_{rul} – rolling resistance;

R_a – air resistance.

The acceleration time and distance have been solved by using the following equations:

$$t_d = \int_{v_0}^{v} \frac{dv}{a} \text{ [s]}$$

and

$$S_d = \int_{v_0}^{v} \frac{v \cdot dv}{a(v)} \text{ [m]} \tag{2}$$

The yield of the gearbox was considered variable in each gear, with a slight increase in superior gears. For determine the variation of the velocity according to the acceleration time, the duration of the change was assumed constant and equal to one second.

3. SIMULATION RESULTS

In the figures 3, 4, 5 and 6 the simulation results obtained with the Matlab-Simulink program are presented. In figure 3 the vehicle speed variation reported to the acceleration time is presented. Following the simulation, the vehicle gets to 100 km/h in approximately 12.4 s, and gets to 80 km/h in about 8.6 s (Figure 3).

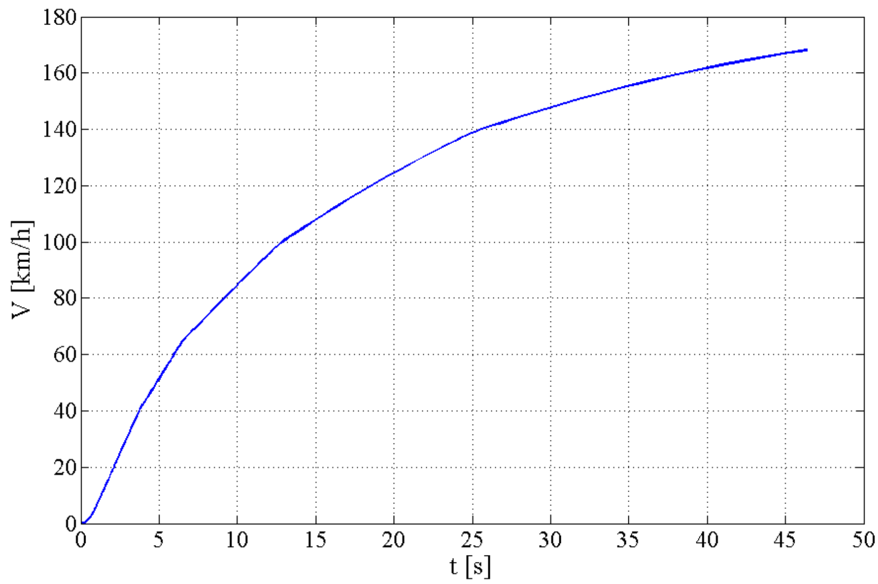

Figure 3. The vehicle speed variation reported to the acceleration time

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It is noted that as when the car's speed increases and the start-up time is increasing. So a passenger car has a good start when it is arriving as high a speed as possible. It is noticed that when the starting space increases, the start time is increasing.
So a car has a good start up space when it is covering a large space in a short time.
The acceleration performances given by the constructor are in table 1, and the performances obtained following the simulation in table 2.

Table 1.
Acceleration parameters [10]

Parameter	Time (s)
Acceleration 0 - 80/100 km/h	8/12.1
Acceleration 0 - 400/1000 m/s	18.1/33.4
Acceleration 80 - 120 km/h	18

Table 2.
Acceleration parameters

Parameter	Time (s)
Acceleration 0 - 80/100 km/h	8.6/12.4
Acceleration 0 - 400/1000 m/s	18.6/33.7
Acceleration 80 - 120 km/h	16.3

Figure 5 presents the partition for the fifth gear obtained following the simulation.
The increase of the speed from 80 km/h up to 120 km/h in the fifth gear, has been made with a partition time of 16.3 s. Following the simulation, the vehicle covers 400 m with 120 km/h, in 18.6 s and 1000 m, with 148 km/h, in 33.7 s (Figure 4 and Figure 6).
Figure 7 presents the moment of the gear shifting for an elastic mechanical transmission.
With yellow colour is represented transmission speed and with violet colour the engine speed (Figure 7).
Figure 8 presents the acceleration characteristic for a mechanical transmission (a) and for an elastic mechanical transmission (b).
The conclusion is that the maximum value obtained for the acceleration is in the first gear, then the acceleration decreases as we shift to higher gears.

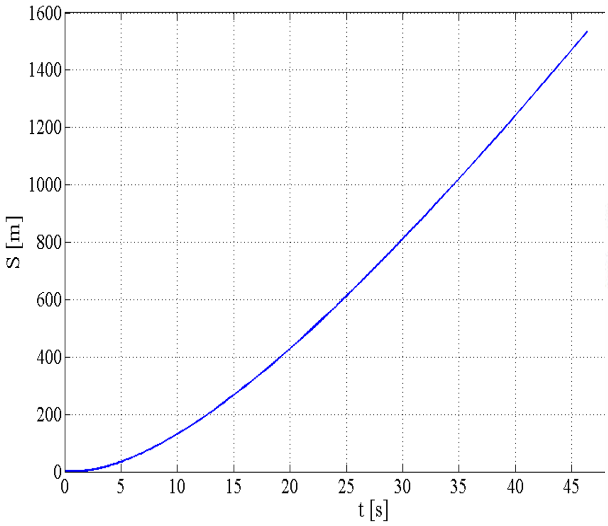

Figure 4. The acceleration distance reported to the acceleration time

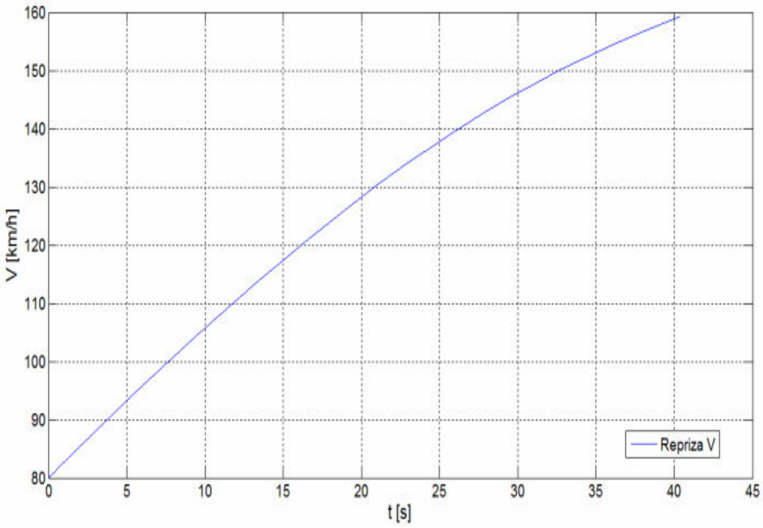

Figure 5. Partition in the fifth gear

In the fifth gear, we get to the maximum speed, but the acceleration is zero.
In our case the acceleration is annulled at the speed of approximately 180 km/h (Figure 8).
Because in the first gear the transmission ratio has the greatest value, the traction force in the tire contact patch increases, so we obtain an increased acceleration, accordingly a limited adherence is needed.
For the elastic transmission (Figure 8.b), the elastic oscillations are reduced as we shift to higher gears, the most oscillations are registered in the first gear.
The elastic oscillations appear following the elasticity decreasing of the drive shafts, through the introduction of an elastic element.
As the rigidity of the drive shafts is lower (meaning that the elasticity of the axles grows) the more oscillations appear. The acceleration characteristics depends on the clutch engagement mode.

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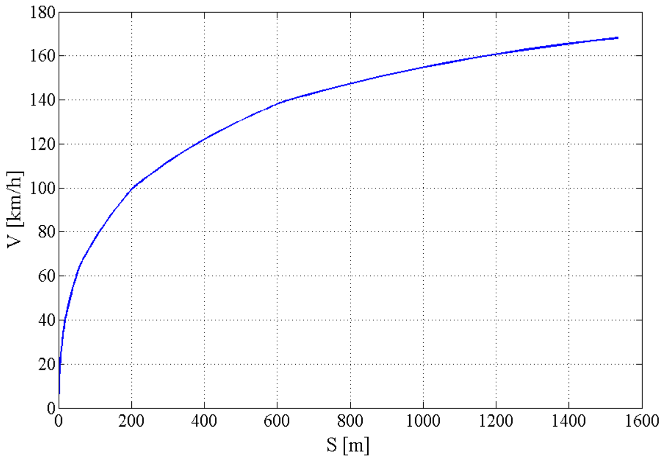

Figure 6. Speed variation reported to the acceleration distance

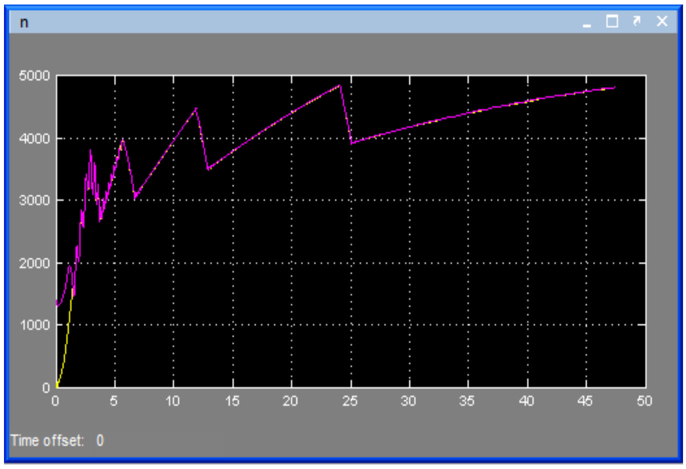

Figure 7. The moment of the gear shifting for an elastic mechanical transmission

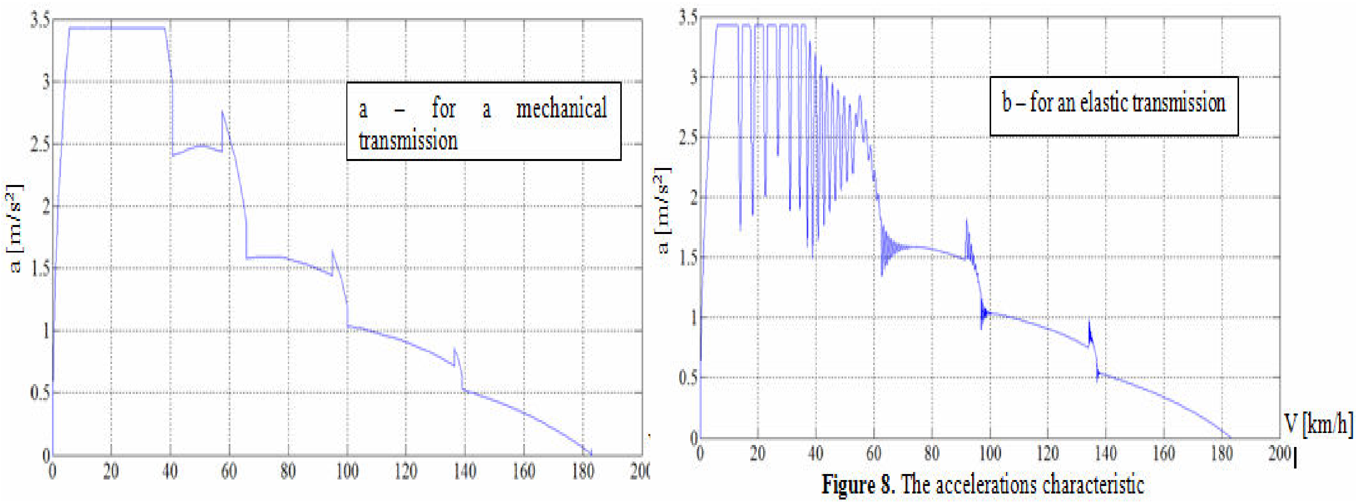

Figure 8. The accelerations characteristic

4. CONCLUSIONS

The model made for the acceleration performances study in Simulink can be used for all the vehicles with mechanical transmission and can be easily adapted for the CVT, DCT or automated transmissions. This model can be considered valid because the acceleration performances obtained following the simulation are satisfactory compared with the acceleration performances given by the constructor (for example the relative error $S_{d\ 1000}$ is approximately 0,8%).

The acceleration of the passenger car, generally characterizes its dynamic qualities.

High acceleration involves the increasing of the average speed of exploitation.

The acceleration qualities are the main performances of the vehicles, having a direct influence on traffic safety, average speed and market success, especially in the case of cars and motorcycles.

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The emphasis is placed on contribution that appears to be of permanent interest to research workers and engineers in the field. If furthering knowledge in the area of principal concern of the Journal, papers of primary interest to the innovative disciplines of „Automotive Technology, Science and Engineering” may be published.
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The Scientific Journal of SIAR
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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.
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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 | Format: print, Romanian
Frequency: Quarterly | Electronic publication on: www.ro-jae.ro
Total number of issues: 30 | 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 | Format: print and online, Romanian
Frequency: Quarterly | Electronic publication on: www.ingineria-automobilului.ro
Total number of issues: 47 | Type: Open Access
(including the June 2018 issue)
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 | Format: online, English
Frequency: Quarterly | Electronic publication on: www.ingineria-automobilului.ro
Total number of issues: 16 | Type: Open Access
(including the December 2014 issue)
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 | Format: online, English
Frequency: Quarterly | Electronic publication on: www.ro-jae.ro
Total number of issues: 15 (September 2018) | Type: Open Access
The above constitutes series nr. 4 of SIAR scientific magazine (English version).

Summary – on September 30, 2018

Total of series:	*4*
Total years of publication:	*24 (11=1990 – 2000; 13=2006-2018)*
Publication frequency:	*Quarterly*
Total issues published:	*78 (Romanian), out of which, the last 31 were also published in English*