Volume 37, Number 4, 2011.

Mobility & Vehicle Mechanics

International Journal for Vehicle Mechanics, Engines and Transportation Systems

ISSN 1450 – 5304

UDC 621+ 629(05)=802.0

UNIVERSITY OF KRAGUJEVAC – FACULTY OF MECHANICAL ENGINEERING

SERBIAN SOCIETY OF AUTOMOTIVE ENGINEERS

M V M Mobility Vehicle Mechanics

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For Publisher: Prof. dr Miroslav Babić, dean, Faculty of Mechanical Engineering, Kragujevac
Ordering: Vesna Maksimović
Printed by: SaTCIP Ltd., Tržni centar Pijaca 106, 36210 Vrnjačka Banja, Serbia
tel: + 381/36/622454

Publishing of this Journal is financially supported from:
Ministry of Science and Technological Development, Belgrade, Serbia
Center of Traffic Safety, Faculty of Mechanical Engineering, Kragujevac

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Mobility & **V**ehicle **M**echanics

Volume 37
Number 4
2011.

Motorna **V**ozila i **M**otori

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Number 4
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[1] REDUCTION OF STRUCTURAL NOISE OF VEHICLE ENGINE-ESSENTIAL FACTOR IN DECREASE THE ACOUSTIC POLLUTION OF ENVIRONMENT

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UDC: 534.833: 621.43

Abstract

The article discusses the new standards adopted in Russia (Russian Federation) on the level of structural noise of automobile Internal Combustion Engine (hereinafter referred to as ICE), the main factors determining the formation of the studied noise and ways to reduce structural noise on the example of diesel powered truck.

Key words: vehicles, engines, structural noise.

SMANJENJE STRUKTURNE BUKE MOTORA VOZILA - ZNAČAJAN FAKTOR U SMANJENJU AKUSTIČKOG ZAGAĐENJA OKOLINE

UDC: 534.833: 621.43

Rezime: U radu je izvršena analiza standarda pripremljenog u RF koji se odnosi na nivo strukturne buke motora automobila (po ugledu na ICE), glavnih faktora koji doprinose stvaranju buke, kao i načina za snižavanje strukturne buke na primeru jednog kamiona.

Ključne reči: vozila, motori, strukturna buka.

[1] Received: February 2011. Primljen: februar, 2011.god.
Accepted: March 2011. Prihvaćen: mart, 2011.god.

REDUCTION OF STRUCTURAL NOISE OF VEHICLE ENGINE-ESSENTIAL FACTOR IN DECREASE THE ACOUSTIC POLLUTION OF ENVIRONMENT

Vladimir Evgenevich Tolsky [1], A.D. Konev

UDC: 534.833: 621.43

1. INTRODUCTION

External noise of automobiles and buses (hereinafter referred to as Motor Vehicle -MV) along with the toxic properties of ICE exhaust gases characterize their environmental parameters.

Due to this fact as far back as 1968. UN ECE Regulations were introduced, which restricted the level of external noise (in dBA) of the different categories of automobiles and buses. Standards for noise levels in automobiles and buses were being tightened all the time. For example, maximum permissible noise levels in light vehicles so far have declined from 82 to 74dBA, and heavy trucks like KAMAZ decreased from 92 to 80dBA (The UN ECE Regulation № 51-02). In the Soviet Union, standards for external noise of MV, including motorcycles, have begun to act with 1975. and the standards were different, more liberal and only 2000. have been equated with the international ones.

Of course, the former higher noise levels did not always require to carry out special measures to reduce external noise of Soviet MV's. However, after the introduction of ECE Regulation № 51-02 since 2000. development departments of automobile and engine plants in Russia started to give greater attention to the solution of this problem.

In accordance with the methodology of the UN ECE Regulation № 51, MV test aimed to determine the external noise is carried out for a ride 20m long, with the intensive acceleration and at high engine speeds. For example, a light automobile of VAZ-class with manual transmission accelerates on the specified trajectory with 2 and 3 shifts. Acceleration with 2 gear shifts begins with 50km/h (approximately n=4500rpm), and ends with n=5200 rpm.

Ranking of sources of external noise according to the internationally accepted method of testing shows as a rule that the majority of MV's manufactured in Russia, express the most influential source on the overall level of external noise of vehicle as a structural engine noise (system and exterior surfaces vibration).

Figure 1 shows the example of 2 spectra of external noise in case of acceleration of a heavy truck with 5-speed and 8-speed transmission. The initial speed of diesel engine crankshaft

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during acceleration of the vehicle with 8 gears is much lower, and virtually does not increase, while the 5-speed gearbox causes a sharp acceleration the vehicle. This circumstance explains that high-frequency part of the spectrum of external noise at frequencies above 800 Hz, with the 8-speed transmission is reduced by 5-8 dB(A), which leads to a decrease in the overall level of external noise of the vehicle of 4 decibels.

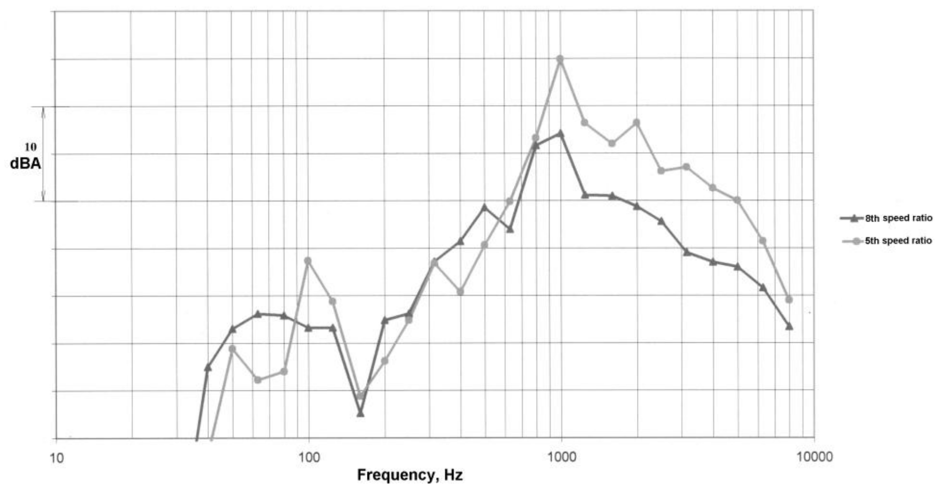

Figure 1: *The dependence of external noise spectra from the acceleration of 5-speed and 8-speed transmission*

By comparing the spectra of noise during acceleration of the vehicle with 5-speed transmission and freewheeling with the engine shut off, it followed that in the frequency range from 630 to 5000Hz noise decreased from an average of 78 to 69 dB (A), and the overall level of external noise, the car has decreased by 10 dBA. rpm.

2. ACCEPTABLE LEVELS OF STRUCTURAL NOISE OF AUTOMOBILE ENGINES IN THE USSR AND THE RUSSIAN FEDERATION

In NAMI there used to be developed OST 37.001.266-83 (Industry Standard), "The noise of automobile engines. Permissible levels and measurement methods ", which came into effect from 1985. It was based on the techniques of noise measurement of stationary machines. In this case, normalized to a distance of 1m, maximum structural noise at the point with the highest noise level) in dBA with the engine on the max.rpm .This document guided until 2010. motor factories of the Russian Federation in the preparation of technical specifications (TU) for ICE relative to its level of structural noise. In OST 37.001.266-83, the boundary levels of structural noise on the serial engine of Soviet manufacturers were adopted 103dBA, and for prospective engines 101dBA. Now, such noise levels of ICE should be considered as too high. These noise levels of ICE were caused primarily by the tougher flow of the working process in automotive diesel engines of Soviet manufacture, lack of turbocharging, increased frequency of the crankshaft of diesel engines vibration without turbocharging, insufficient rigidity of the crankcase details of individual ICE including through the use of aluminium alloys. In those years, the main objective was to reduce engine mass and reduction of fuel consumption of ICE, which did not help reduce the structural noise of ICE. In addition, as noted above, the rules to limit external noise of vehicles until 2000. were much more liberal then UN / ECE № 51.

The situation has now changed dramatically, the engine builders are required primarily to fulfil the modern environmental requirements for MVs: standards for emissions of automobile engines (requirement E-3: E-4) and standards for exterior noise of MVs (the UNECE Regulation № 51-02), but without reducing the structural noise of MVs, the problem of reducing engine external noise in the majority of MVs produced in Russia can not be solved.

Hence, the relevance of introduction of a new document for standardizing the noise of automobile engine produced in Russia was evident.

In addition, in recent years, new regulations on noise assessment, taking into account the metrological modern technology adopted in the measurement of noise of stationary machines. Based on these circumstances, in 2010. in NAMI, for the first time in Russia, was designed GOST (State Standard) R 53838 to limit the structural noise of ICE (1).

3. NOISE CHARACTERISTICS OF AUTOMOTIVE ENGINES

When rationing engine structural noise according to GOST 53838, the following will be taken into account:

- 2.1 Adjusted, for frequency response of A sound level meter, sound pressure level LrA, dBA;
- 2.2 Sound pressure levels in octave or third octave bands Lp, dB;
- 2.3 Sound power level in octave bands Lw with mean frequencies ranging from 125 to 8000 Hz, dB;
- 2.4 Adjusted, for frequency response of a sound level meter (hereinafter - corrected to A), sound power level L_{WA}, dBA;

Noise characteristics of clause 2.1 shall be determined in the certification and verification of engines (acceptance testing), and their values may be claimed by the manufacturer (supplier) in accordance with GOST 30691.

Noise characteristics of clause 2.2-2.4 define for research purposes in the acoustic design of engines and comparison of their structures by acoustic emission, for comparison with the noise standards of other ICEs, etc. In the acceptance testing of the new engine, all the noise characteristics listed in 2, should be determined and for control tests - only the sound pressure level - LrA. (2.1)

Noise characteristics are determined for single engine and release (types, models, brands), or serial deliveries and are included in the supporting documents in the application form in accordance with GOST 30691 and (or) in the operational documents (passport, label, manuals and others by choice of manufacturer or supplier). Manufacturer (supplier) guarantees the values of noise characteristics specified in the engine documents or in a contract for the supply.

Noise characteristics for the serial delivery engines are based on statistical analysis of measurements sampling machine according to GOST 27408 (2). In accordance with GOST 53838 LrA sound pressure levels for engines, production of which will begin before 01/01/2012, as well as engines upgraded during this period, the measurement at different rpm of crankshaft rotation ***n*** and the engine at full load, should not exceed the values shown in Table. 1.

Permissible sound pressure levels LrA at the measuring distance of d=1m from the engine, production of which began before and after 01/01/2012 was listed in Table 1.

Table 1: *Permitted values of sound pressure* L_{rA}

Engine type	Rated crankshaft speed min^{-1}	Standard value dBA		Type of transport vehicle
		Before 01.01.2012.	After 01.01.2012.	
V-8 Diesel	2100	98	96	M3, N3
V-6 Diesel	1900	97	06	M3, N3
V-8 Petrol	3200	94	94	M3, N3
L-6 Diesel	2500	97	95	M2, N2
L-4 Diesel	over 2500	98	96	M1, N1
L-4 Diesel	locked 2500	96	94	M2, N2
L-4 Petrol	over 4000	99	97	M1, N1
L-4 Petrol	locked 4000	96	94	M1, N1
Remark: For plants manufacturing engines for vehicles of its own, the rules on the permissible sound level values are not established. Level of external and internal noise of these vehicles (Motor Vehicles - MVs) shall comply with ECE Regulation № 51-02, and GOST R 51616 (inside noise Motor Vehicles). This provision also applies to cars and buses (MVs), manufactured by other companies, which apply the above-mentioned engines.				

As can be seen from Table 1, permitted values of sound pressure levels -LrA should not exceed 99, or 97dBA (as of 2112.).

As follows from the notes on the Table 1 for plants manufacturing engines for vehicles of its own production, the noise of vehicles is normalized in accordance with the requirements of ECE Regulation № 51-02 (GOST R41.51) and GOST 51616 (internal noise). The same goes for the other manufacturers where the above-mentioned engines are applied. The definition of conditions is due to the acoustic characteristics of the test room. Volume of the room without sound insulation coverings should not be less than 200 m^3. The coefficient of the acoustic conditions of the premises K_{2A} (dB) should be determined, its value should not exceed 2dBA. Methods of determination of K_{2A} are annexed to the GOST 53838 [1].

In the recent years, in the global engineering, it is common that engine manufacturers announce their own noise characteristic data [3].

In this regard, in the appendix to GOST 53838 there is a method of determining the declared noise of the engine. Engine manufacturer guarantees that in conditions of mass production, the sound pressure level Lra will not exceed the values quoted in Table 1 for more than 1dBA.

4. WAYS TO REDUCE THE STRUCTURAL NOISE OF VEHICLE ENGINE

There are two fundamentally different ways of reducing engine structural noise.

- 1. Reduction of the structural noise of the engine itself (*active* methods of noise reduction);

- 2. *Passive* methods of reducing structural noise due to application of vibro-acoustic material installed around the vehicle's powertrain (the screens and materials on the walls of the engine compartment of a bus, full insulation of the engine).

The first method is more complex, especially its not easy to implement under the current design of ICE, it requires specialized knowledge, necessary experimental equipment and instruments, required software, and most importantly- changing the design of ICE, and hence the acquisition of new equipment for the manufacture of quieter engine. Nevertheless, in the whole global automotive engine development in recent years suggests that the first method is preferred. In addition to reducing the external noise of automobiles and buses, the application of the first method reduces the cost of MV, while the application of the second way of reducing structural noise increases the cost of MV. For example, a full insulation of powertrain compartment of heavy vehicles increases the cost of MV for about 1000 $ U.S.

The second method is used in the current design of ICE and there is no guarantee that screens and elements of the capsule in the service conditions, after the maintenance and repair of MV will be set back, or will be reinstalled in such a way that the necessary sealing of the power unit is violated (there are gaps, openings, etc..) while increasing external noise of automobile or bus, and eventually leads to the conclusion that the use of passive methods of noise reduction is pointless. At the present time in Russian Federation in conditions of operation MVs, only the noise of engine exhaust system is controlled for the stationary position of ICE, by use of the method according to ECE Regulation № 51-02. [4].

5. FACTORS INFLUENCING THE FORMATION OF STRUCTURAL NOISE OF AUTOMOBILE ICE

5.1 The Organization Workflow

(Below there are data on the example of the results of experimental work carried out with the prototype V-type diesel engine, designed for heavy trucks). With the direct fuel injection, the degree of pressure increases over time in the cylinder at the time of the explosion of the working mixture is usually determined by the maximum value of structural noise rate.

This is shown by use of flow indicator diagrams obtained experimentally and the character of variation of noise in the first seconds after the explosion of the working mixture. The shorter the duration of this period, the higher the level of structural noise rate. The essential value of variation of flow indicator diagrams and noise over time relies on the fuel system design and characteristics of the fuel supply. As experience has shown in recent years, the use of Common Rail fuel system contributes both to reduce the toxicity of exhaust gases and a decrease structural noise rate of ICE. This is confirmed by the experimental work on samples of new diesel engines of Yaroslav motor plant -JAMaZ (P-4). and the report is given in [5].

5.2. Relining the piston

Reduction of structural noise, by reducing side clearance between the piston and the cylinder, can be carried out in a very limited extent.

5.3. The angle of fuel pre-injection

Earlier in the USSR, when there were not in power the conditions of performance standards for toxicity, one of the priorities was to reduce fuel consumption. The maximum value of pressure on the indicator diagram should be located as close to TDC (Top Dead Center), and in this case, the major impact on structural noise is done by even a slight change in the angle of fuel pre-injection.

In those years, this angle is equal to 16-18 deg. to TDC for a diesel KAMAZ (Kamsky Automobile Factory) and JMZ. Structural noise rate was very sensitive to a variation of the angle of fuel pre-injection and, accordingly, the rigidity of the flow of the working process in a diesel engine.

Currently, in striving to achieve modern standards for toxicity, the character of the flow of indicator diagrams in the Russian diesel engines has virtually changed, the fuel pre-injection angle was positioned closer to TDC, and as shown by recent experience, the variation of the angle of fuel pre-injection did not provoke a significant change of structural noise rate of KAMAZ-type engine.

5.4 Turbocharging

Turbocharging increases the energy performance of ICE, which in turn can lower the maximum speed of diesel engine crankshaft. Every drop of crankshaft rotation shaft for 100 rpm leads to a noise reduction for about 0,8-1,0 dBA. Turbocharging increases the maximum pressure in the cylinder, but at the same time, it reduces the rigidity of the workflow, which also contributes to a decrease of structural noise.

So, for example, diesel engine KAMAZ-740, manufactured before 1995., with n = 2600 rpm without turbocharging, produced noise level 4-6 dBA higher than noise produced now with the turbo diesel KAMAZ engines (n = 2100 rpm) of the same displacement.

5.5 Engine crankshaft speed

As the speed of rotation of the crankshaft varies, with the fixed values of load and angle of fuel pre-injection, the shape of the pressure curve in a cylinder fixed on the angle of rotation of the crankshaft (the indicator diagram) varies only slightly. Minor changes are associated with different values of turbocharging, due to the variation of crankshaft speed of rotation.

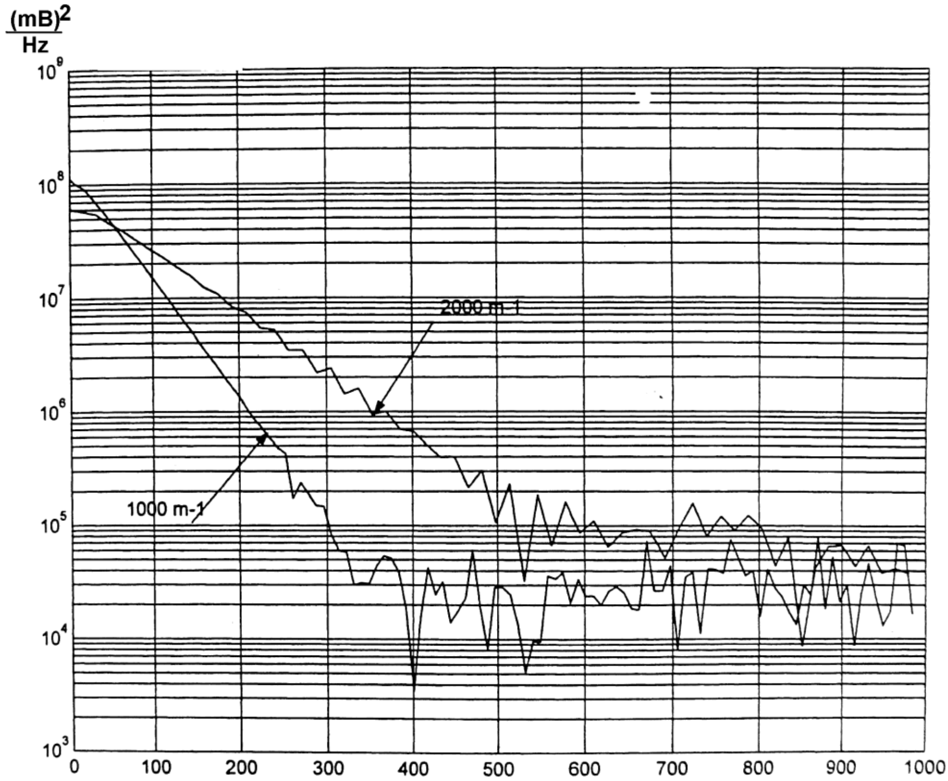

Figure 2: *Frequency content of the pressure pulse in cylinder for different crankshaft speeds*

Basic attention in the following text will be paid to the frequency analysis in time domain. Frequency of changes in pressure in the cylinder should be investigated in the absence of fuel, since maintaining the same cycle of fuel supply at different engine speeds presents certain difficulties. Performed studies (without fuel supply) have shown that the increase of the engine crankshaft speed from 1000 to 2000 rpm will *reduce* duration of pressure pulse. It is obvious that reducing of the duration of pulse leads to an *expansion of its frequency content* to the area of higher frequencies (pulse becomes more "sharp"). This phenomenon is confirmed by the analysis of the frequency of the pressure pulse in the cylinder. (Figure 2).

The above results explain the reason for increasing noise with the increase of engine crankshaft speed.

5.6 Load of diesel engine

Effect of engine load was investigated under the conditions of engine speed at 2200 rpm and the fuel supply from 25% to total 100%. The angle of the start of fuel injection was 18 degrees before TDC.

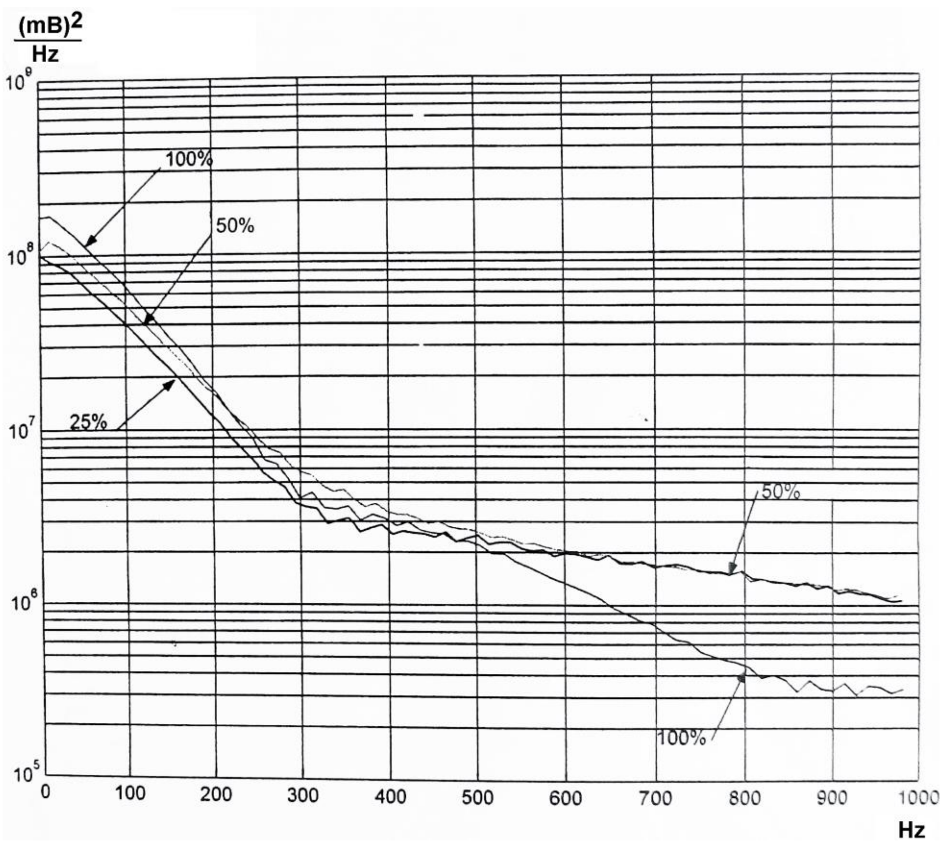

Figure 3: *The frequency content of the pressure pulse in the cylinder of a diesel engine with different fuel supply*

It turned out that an increase in fuel supply will lead to the *reduction* of gradient of the pressure front in cylinder, caused by the start of combustion. It is known that the gradient of combustion front determines the rigidity of the engine work, which affects the noise level of the combustion process. This phenomenon is also reflected in the spectral composition of the pressure pulse in the cylinder (Figure 3).

When the load increases the high-frequency components in the spectrum of influence of pressure decreases by about 70%, and in the low frequency zone the increase is less than 10%. This indicates a softening of the combustion process by increasing the fuel supply. It is also possible to notice that when the fuel supply is increasing, at the same time the pressure at the end of the compression stroke increases too, due to high boost pressure as a result of higher energy of exhaust gases. This explains the reason for reducing the rigidity of the combustion process. At the beginning of the combustion process, combustion of fuel-air mixture is quicker due to the lower duration of ignition delay. As a result, the accumulated unburned mixture is smaller and the intensity of the explosion in the initial period of combustion is lower. This explains the decrease in rigidity of the process with the increase of the fuel supply.

5.7 Transient regimes of diesel engine (acceleration without load)

Tests of engine transient regimes of operation of ICE are very important, since in this case vehicle acceleration is simulated up to some extent according to the ECE Regulation № 51 for external noise measurement. It should be noted that the noise level does not significantly change with the variation of engine load.

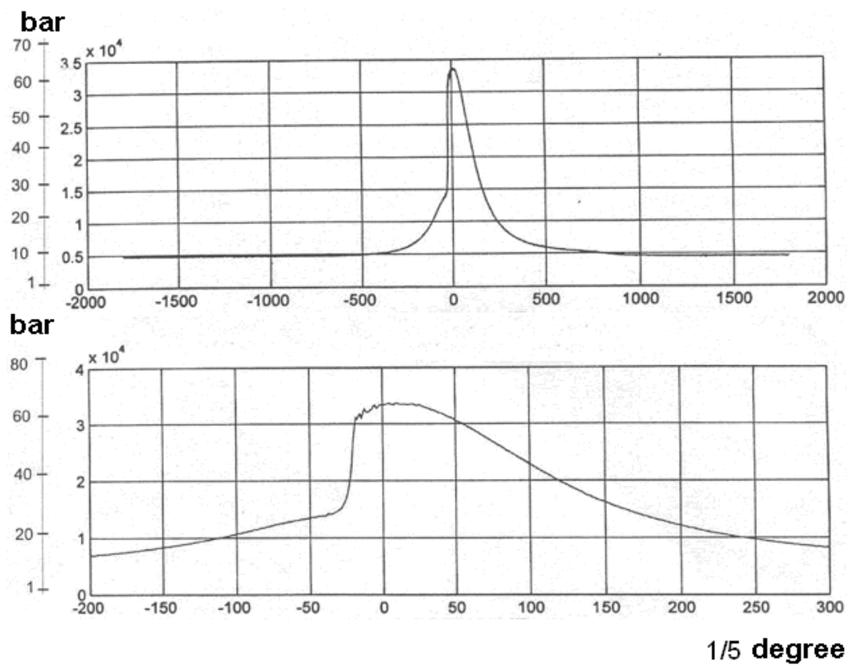

Figure 4: *The first cycle of the engine operation with a sharp increase of fuel at idle*

In our case, a sharp turn of the fuel pump lever is performed from zero (idle) to maximum with no load on the engine crankshaft. These tests simulate the most dramatic changes in the two main operating parameters of the engine: fuel supply cycle and the engine crankshaft speed. Given that the engine was equipped with a gas turbine supercharger, as an inertial element, this kind of test is important because for the automobile engine, such regimes are typical. In terms of noise emissions, the greatest attention should be paid to the indicator diagram of the first cycle after a sharp increase in fuel supply (Figure 4).

At this point, the engine speed and the magnitude of the boost pressure is the same as when idling and the fuel supply is maximal. The diagram shows that the leading edge of the combustion process is very steep and significant in magnitude. Frequency analysis of the process and the corresponding process on the defined regime (i.e. the regime in which the boost pressure reached its defined value) is shown in Figure 5.

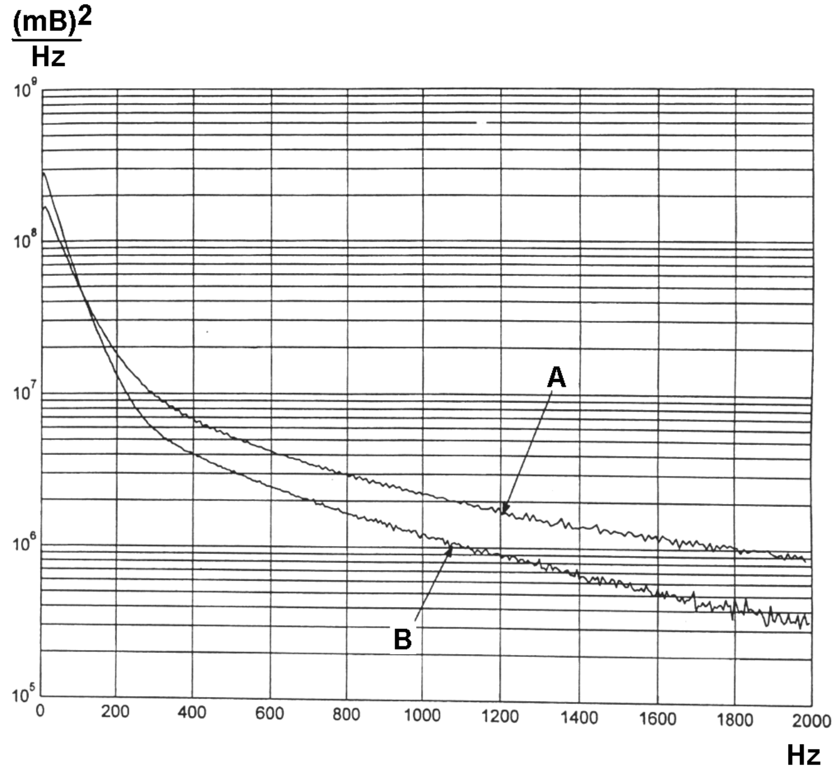

Figure 5: *The frequency content of the pressure pulse in the cylinder at work on the defined regime (n = 1000 rpm, the fuel supply 100% - the curve "B") and with increasing speed of the diesel engine crankshaft - the curve "A"*

It is evident that when there is a sharp increase in fuel supply, the frequency content of the influence of pressure pulse significantly increases in the high frequency domain. This corresponds to an increase in the leading edge of combustion. The reason for that is that the parameters of the pressure at the end of compression are small due to the low boost pressure.

5.8 The design of the external surfaces of ICE

The design of the external surfaces plays a significant role in the formation of structural noise of the engine. The presence of separate cylinder heads, the application of aluminium alloys. insufficient bending stiffness of the block increases the noise level of ICE.

Significant role in the formation of structural noise shows the character of the flow of bending vibrations of the block in the frequency of about 1000Hz, caused by shock loads during the early stage of the operation process. Tightening of the block by increasing its weight and with the application of modern fuel system helps reduce the noise of diesel engine. An example would be the last work JMZ cooperation with the firm AVL (Austria) to reduce the noise of prototype diesel engine P-4.

Currently, there is the increasing application of a calculation-experimental method for studying structural noise of ICE.

Its essence is as follows:

Calculation part consists of two phases

- Calculation of forms of bending vibrations of the external surfaces of ICE by use of Finite Element Method (FEM).

- Application of the Boundary Element Method (BEM) to evaluate directly the nature of structural noise of the engine caused by vibration [6].

CONCLUSION

The above-mentioned method of investigation requires the mandatory identification of computational models based on the results of the first experimental work, only after that the method should be modified, i.e. perform the search for ways to improve the design of ICE.

Such computational-experimental work was carried out with the participation of the authors of the article, which allowed a presentation of specific recommendations for change of the character of bending vibrations by modifying the design of the engine block of the tested diesel ICE, in order to increase its rigidity, as well as changing the design of some of its outer parts. These recommendations were made based on analysis of forms of elastic vibrations received directly on a running diesel engine with multi-channel equipment for vibration measurement.

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[1] AERODYNAMIC ATTRACT OF VEHICLES IN REAL TRAFFIC

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UDC: 533.6.01:629.113.3/.5

Abstract

A vehicle, being defined structurally, contextually, shapely and dimensionally is used under real conditions and within its purpose, and as it is, it is in permanent interaction with the environment. The conclusion is logical that any vehicle, in its life time, has its functional and work capability and having allowed load and accepted work regime, it is adapted to be stable and driven to road conditions.

It is often happened that vehicles pass by closely or pass each other from different directions on the road. From such conditions and circumstances it is, in processes, initiated appearance of virtual Ventura's tubes and effect of vehicle aerodynamic side attracts. The produced forces speed and pressure fields influence further behavior of any vehicle disturbing its stability and driving.

In the paper this problem is underlined in order to contribute to theory and practice of aeromechanic being implemented in the field of vehicles.

Key words: aerodynamic, vehicle, attraction.

AERODINAMIČKO PRIVLAČENJE MOTORNIH VOZILA U REALNOM SAOBRAĆAJU

UDC: 533.6.01:629.113.3/.5

Rezime: Motorno vozilo definisano strukturno, sadržajno, oblikovno i dimenziono, koristi se u realnim uslovima a u okviru svoje namene, i kao takvo je sa okruženjem u stalnoj interakciji. Logično se nameće zaključak, da motorno vozilo u eksploataciji poseduje funkcinalnu i radnu sposobnost i da se pod dozvoljenim opterećenjem, a u prihvatljivom režimu rada, stabilno i upravljivo prilagođava putnim uslovima.

Nije redak slučaj da se motorna vozila na putu preteču ili mimoilaze, na veoma bliskom međusobnom rastojanju. Iz takvog stanja i okolnosti procesno se inicira pojava virtuelne Venturijeve cevi i efekta aerodinamičkog bočnog privlačenja motornih vozila. Nastale sile, polja brzina i pritiska, utiču na dalje ponašanje motornih vozila, narušavajući njihovu stabilnost i upravljivost.

[1] Received: December 2010. Primljen: decembar, 2010.god.
Accepted: December 2010. Prihvaćen: decembar, 2010.god.

U radu se akceptira ova problematika u cilju doprinosa teoriji i praksi aeromehanike implementirane u oblasti motonih vozila.

Ključne reči: aerodinamika, motorno vozilo, privlačenje.

AERODYNAMIC ATTRACT OF VEHICLES IN REAL TRAFFIC

Mile S. Šiljak [1]

UDC: 533.6.01:629.113.3/.5

1. INTRODUCTION

Motor vehicle, as technical means for mass transport has forever attracted attention and been an object of interest but also a subject of research of many researchers.

In the base, a vehicle has its functional and work capability that should be constantly and imperatively upgraded during its development from many reasons and mostly from safety, ergonomic, technical, technological, finance and energy reasons. Stability and driving of motor vehicles under real traffic conditions are also very important.

Personal road experience, quantitative, qualitative and cumulative, being reached contemplatitively under real traffic conditions as an active participant in it, pointed out the existence of phenomenon which is presented as a change of behavior of a motor vehicle at passing by or passing with another motor vehicle from opposite direction under real traffic conditions. Practically, three characteristic phases in process of passing by or passing with other motor vehicle from opposite direction are identified as follows:

- Phase of slowing a driven motor vehicle down, in the moment at approach of a motor vehicle that is passing by or coming from opposite direction;
- Phase of side attract of a driven motor vehicle, at parallel motion with another motor vehicle that is passed by or come from opposite direction;
- Phase of acceleration of a driven motor vehicle in the moment with ended parallel motion with another motor vehicle being passed by or come from opposite direction.

It is also noticed that these phases as per duration and intensity, besides others, depend on the following: type, shape and size of motor vehicles; mutual side distance, during their parallel motion; speed of motor vehicles; and besides others, if it is on parallel or from different directions passing by of motor vehicles. From such information and experience, the starting theoretical researches are initiated that are subject to this paper.

2. MOTOR VEHICLE

Any motor vehicle that may be legally present in real traffic is to be registered and it means it must be under standard qualification of road motor vehicles, being attested, having known origin and technically regular.

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For this accepted problem, the type, shape and size of a vehicle are very important. Using basic principle of modeling it is possible to show simply the motor vehicles by geometric primitive shape of parallelepiped from where all characteristic types of motor vehicles come from being important for this subject research (Figure 1).

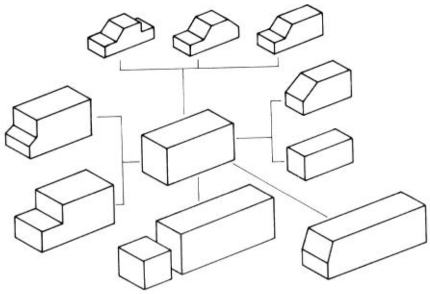

Figure 1: *Simplified view of a motor vehicle [5]*

From the aeromechanic point of view, a motor vehicle, at its motion through air, is flown by air. Around the motor vehicle, the current flow of air is formed with fricatives of defined shapes along with appropriate fields of speed and pressure on motor vehicle body surfaces. Simplified view of flowing of motor vehicle characteristic type is presented in the Figure 2, with assumption that the motor vehicles are alone in the flow of air, air is with no motion and there are no outside initiations that may disturb symmetric air flow of the motor vehicle.

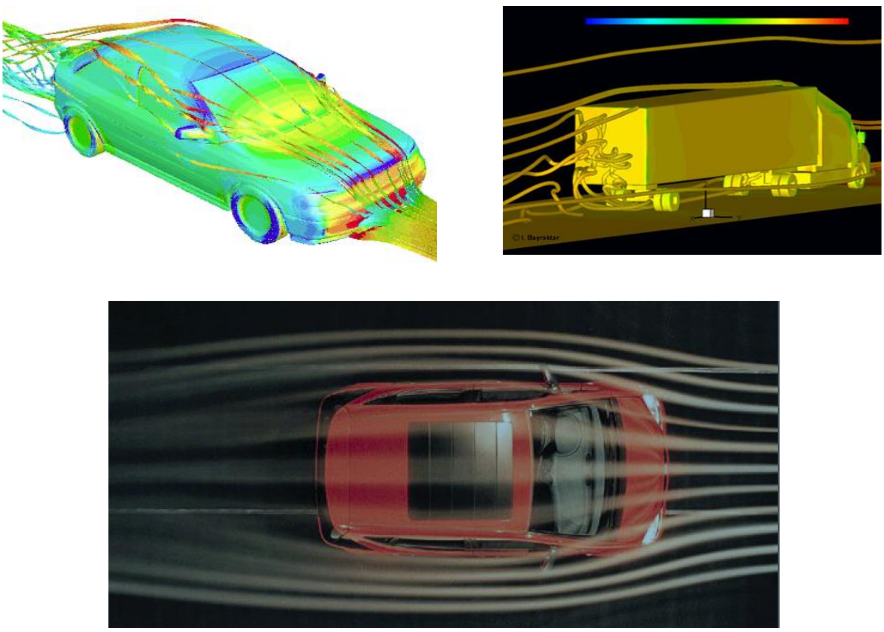

Figure 2: *Schematic view of air flow of some motor vehicle types*

Because of conditional assumptions in the process of air flow the following is noticed:

- Symmetrically longitudinal air flow of subject motor vehicle;
- Equalized field of speed on both sides of subject motor vehicle;
- Equalized field of pressure on both sides of subject motor vehicle;
- With no side forces that, due to pressure differences, may affect the subject motor vehicle.

It is important to note that also frontal surfaces of some motor vehicles are mutually different as presented in the schematic view in the Figure 3.

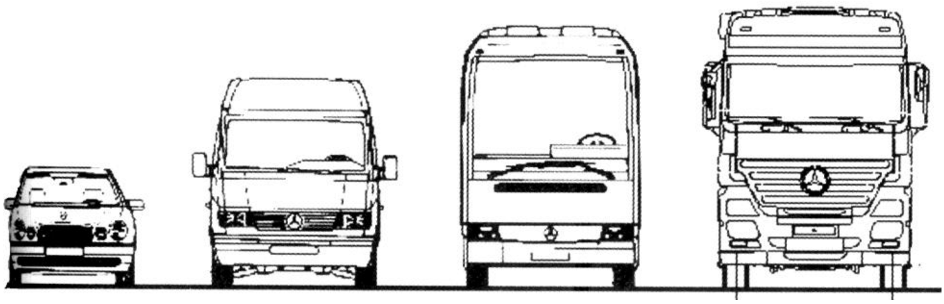

Figure 3: *Schematic view of some vehicle frontal surface [4]*

3. VENTURI'S TUBE

As known, the link between pressure and speed of fluid flow, in subject fluid flow of uncompressible and not viscous fluid, is done by Bernuli's equation.

$$p + \frac{1}{2}\rho v^2 + \rho g h = const. \qquad (1)$$

Where is:

p – Absolute pressure in fluid
ρ – Density of fluid
v – Fluid speed in fricative
g – Acceleration of earth's gravitation
h – Fricative height relating to referent level

For noticed path of fluid part and two points on it and between them, there is no size change of noticed fluid parts, the Bernuli's equitation for the subject fluid flow may be written.

$$p_1 + \frac{1}{2}\rho v_1^2 + \rho g h_1 = p_2 + \frac{1}{2}\rho v_2^2 + \rho g h_2 = const. \tag{2}$$

If Venturi's tube is considered, set horizontally relating to horizontal level (Figure 4), and if the fluid flow of uncompressible and not viscous fluid is in it, then using manometer of U-tube type one may measure the pressure difference in characteristic points of Venturi's tube.

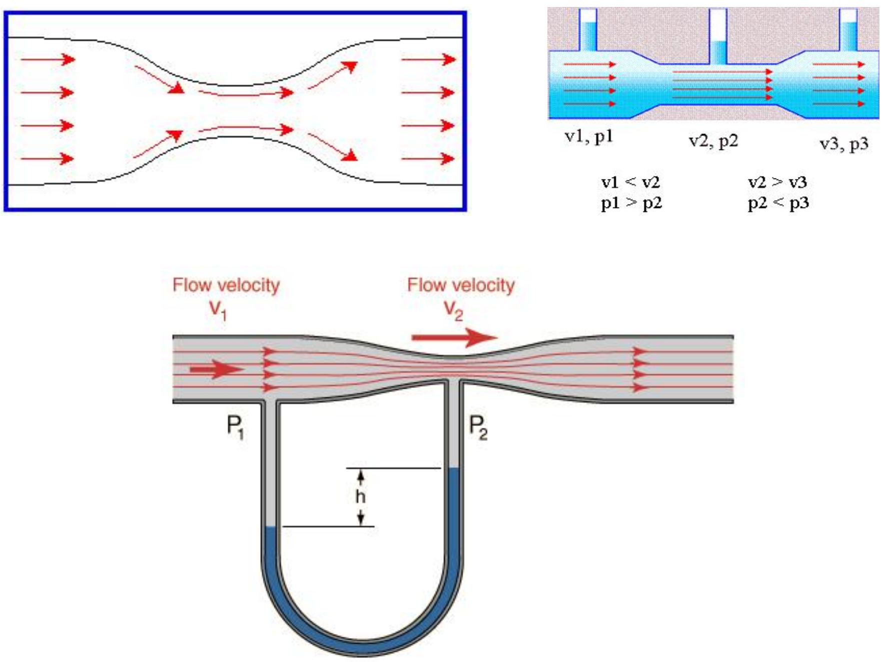

Figure 4: *Venturi's tube with manometer of U-tube type*

Appropriate equations for subject fluid flow:

- Equation of continuation,

$$\rho v_1 A_1 = \rho v_2 A_2 = \rho v_3 A_3 = const. \tag{3}$$

$$v_1 A_1 = v_2 A_2 = v_3 A_3 = const. \tag{4}$$

- Bernuli's equation,

$$p_1 + \frac{1}{2}\rho v_1^2 + \rho g h_1 = p_2 + \frac{1}{2}\rho v_2^2 + \rho g h_2 = const. \tag{5}$$

$$h_1 = h_2 = h_3 \tag{6}$$

$$\rho = const. \tag{7}$$

$$\rho g h_1 = \rho g h_2 \tag{8}$$

$$p_1 + \frac{1}{2}\rho v_1^2 = p_2 + \frac{1}{2}\rho v_2^2 \tag{9}$$

$$(p_1 - p_2) = \frac{1}{2}\rho\left(v_1^2 - v_2^2\right) \tag{10}$$

$$\Delta p = p_1 - p_2 \tag{11}$$

Manometer shows as follows, $\Delta p = \rho_o g h$ (12)

namely, $p_1 - p_2 = \rho_o g h$ (13)

Namely, it is

$$p_1 > p_2 \tag{14}$$

If the subject fluid flow through Venturi's tube is idealized and virtualized completely, the pressure change in the Venturi's tube (Figure 5) may also be conditionally shown.

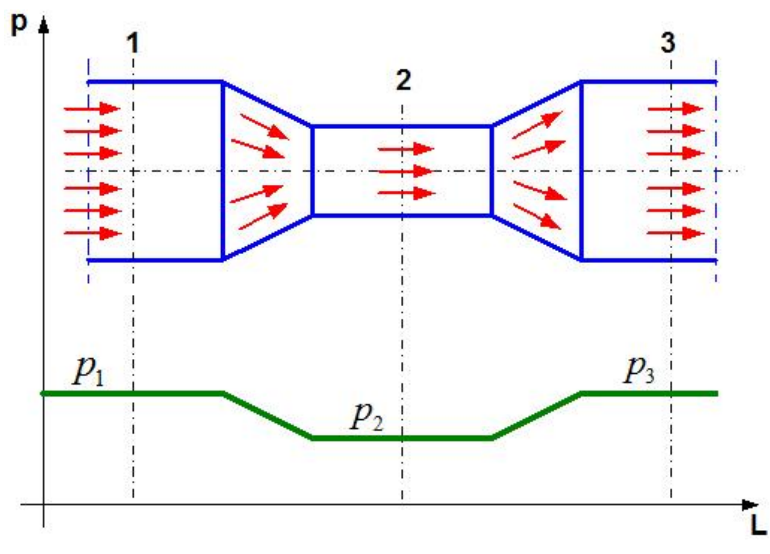

Figure 5: *Pressure change in Venturi's tube*

4. PHENOMENA OF AERODYNAMIC ATTRACT

A motor vehicle in real traffic conditions is used within its purposes and in accepted work regime, adapting to stable and driven road conditions, climate and methereologic conditions, static, kinematics and dynamic environment.

Still considering the accepted problem, two characteristic cases will be noticed as follows:

- Motor vehicle-1, passes another vehicle-2, Figure 6;
- Motor vehicle-1, passes another vehicle from opposite direction-2, Figure 8.

To show motor vehicles in appropriate combination, the simplified model of motor car, so called the geometric primitive parallelepiped shape, will be used. It will be supposed that air in a motor vehicle environment is in no motion, the road is flat and straight, no outside initiations disturbing the subject state and mutual interaction of subject motor vehicle.

Mutual side distance between motor vehicles is geometric value (a), which is always positive but higher than zero, to perform passing physically, namely passing of motor cars from different directions, and among others it depends on the following: road width; position of motor vehicle-2; position of motor vehicle-1; and motor vehicle width.

The subject motor vehicle's-1, speed is

$$(v_{MV})_1 \neq 0 \tag{15}$$

and motor vehicle's-2, speed is

$$(v_{MV})_2 \neq 0 \tag{16}$$

and both adapted to allowed speed on subject road part.

4.1. Motor vehicle-1 passes another motor vehicle-2

$$(v_{MV})_1 > (v_{MV})_2 \tag{17}$$

It is indisputable that,

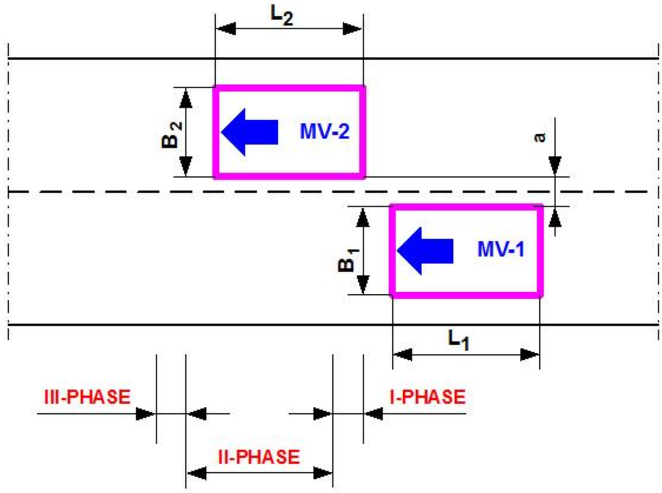

Figure 6: *Motor vehicle-1 passes the motor vehicle-2*

In the process of passing of motor vehicles, but in the phase two, the motor vehicles reach parallel position forming the virtual Venturi's tube (Figure 7).

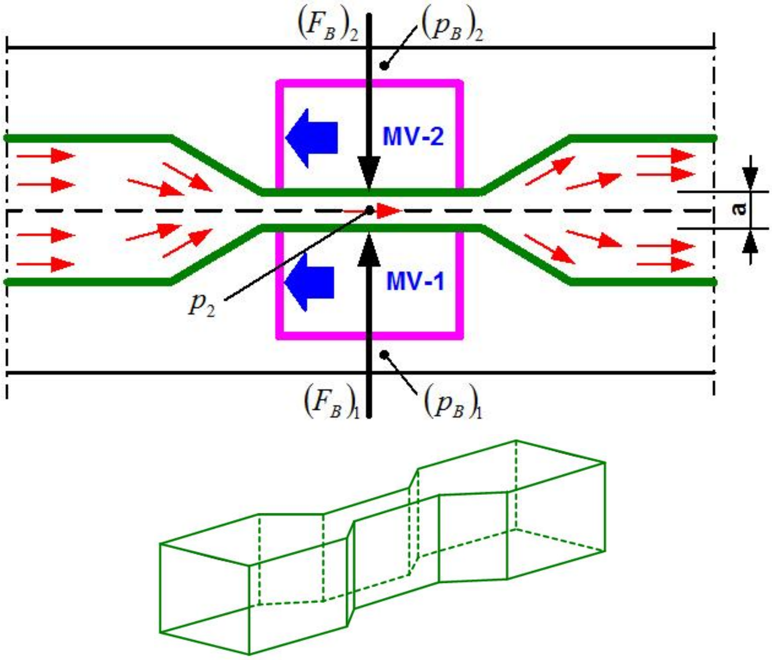

Figure 7: *Virtual Venturi's tube being formed by motor vehicles in passing*

Relative possible speed of air in area between subject motor vehicles is conditionally as follows:

$$v_2 = (v_{MV})_1 - (v_{MV})_2 \tag{18}$$

Side attract forces on subject motor vehicles,

$$(F_B)_1 = [(p_B)_1 - p_2](A_{MV})_{B1} \tag{19}$$

$$(F_B)_2 = [(p_B)_2 - p_2](A_{MV})_{B2} \tag{20}$$

where is:

$(p_B)_1$	–	Pressure in side free zone of the motor vehicle-1
$(p_B)_2$	–	Pressure in side free zone of the motor vehicle-2
p_2	–	Pressure in the zone between subject motor vehicles
$(A_{MV})_{B1}$	–	Side surface of the motor vehicle-1
$(A_{MV})_{B2}$	–	Side surface of the motor vehicle-2

4.2. Motor vehicle-1 passes the motor vehicle-2 from opposite direction

It is indisputable that,

$$(v_{MV})_1 \neq 0 \;;\; (v_{MV})_2 \neq 0 \;;\; (v_{MV})_1 >/=/< (v_{MV})_2 \tag{21}$$

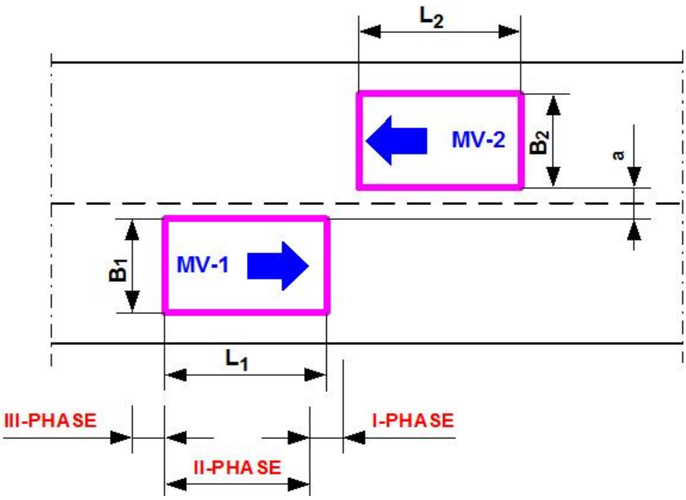

Figure 8: *Motor vehicles pass from different directions*

In the process of passing of motor vehicles from different directions in the phase two, the motor vehicles reach parallel position keeping it for some time, forming virtual Venturi's tube (Figure 9).

Relative possible speed of air in area between subject motor vehicles is conditionally as follows:

$$v_2 = (v_{MV})_1 + (v_{MV})_2 \tag{22}$$

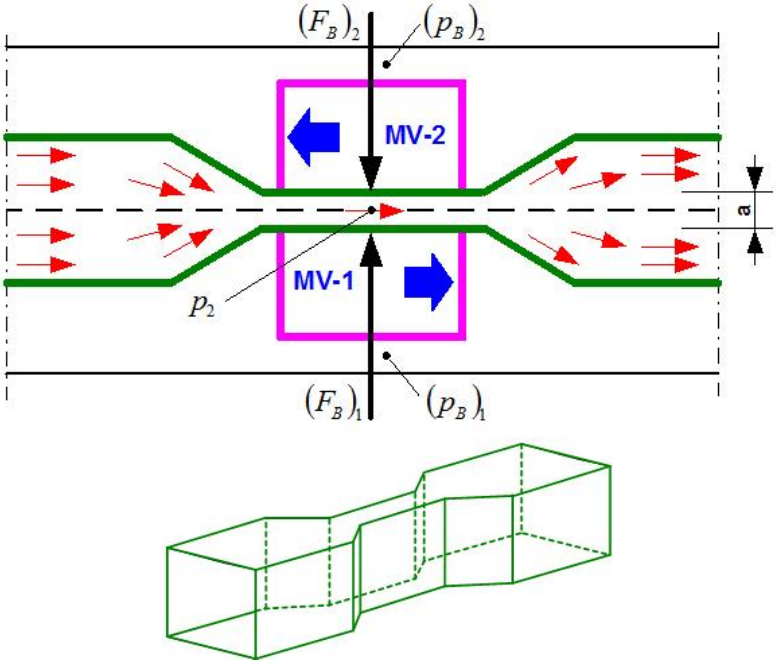

Figure 9: *Virtual Venturi's tube formed by motor vehicles in passing of motor vehicless from different directions*

In this case, side attract forces on subject motor vehicles are calculated by the same expressions as in previously described characteristic case, but with changed appropriate values only.

5. CONCLUSION

It is indisputable proved that the side attract forces also really exist between motor vehicles that pass by or pass from different directions during the process second phase when the subject motor vehicles are in parallel position but in real traffic. Considering their nature, it is justified to name these side attract forces the aerodynamically attracted forces since they are indeed. Their direction is known, but their intensity is changeable between limit values, i.e. between zero and the maximal value.

Existing of side aerodynamic attract forces indisputable influences the stability and driving of motor vehicles when they are in dangerous situations.

The traffic accidents are known where the participating motor cars come to side contacts, from "unknown reasons", and afterwards the unwanted circumstances occurred.

In existing domestic and foreign professional literature, being available to this paper author, no description on accepted problem was noticed.

An imperative obligation comes out from above presented and thus, in analyses on stability and driving of motor vehicles, the aerodynamically attracted forces must be also included equally, and a reason more for that it is surely a fact that in real traffic there are more and more motor vehicles having large side surfaces (long motor vehicles).

The presented problem should be firstly accepted as a contribution to upgrade the theory and practice of motor vehicles along with implementation of aeromechanics, but also as positive initiation and stimulus to further theoretic and practice research. It is to expect that this should be happened after publishing of the paper.

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[1] A PHENOMENOLOGICAL MODEL OF TWO-PHASE (AIR/FUEL) DROPLET OF CUMMINS SPRAY

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UDC: 621.436.21

Abstract

Effervescent atomization namely the air-filled liquid atomization comprehends certain complex two-phase phenomenon that are difficult to be modeled. Just a few researchers have found the mathematical expressions for description of the complex atomization model of the two-phase air/diesel fuel mixture. In the following review, developing model of two-phase (air/fuel) droplet of Cummins spray pump-injector is shown. The assumption of the same diameters of the droplet and the opening of the atomizer is made, while the air/fuel mass ratio inside the droplet varies.

Key words: effervescent atomization, two-phase, droplet, diesel injector.

FENOMENOLOŠKI MODEL DVOFAZNE (VAZDUH/VODA) KAPLJICE CUMMINS-OVOG MLAZA

UDC: 621.436.21

Rezime: Penušavo raspršivanje, odnosno raspršivanje tečnosti ispunjeno vazduhom sadrži neke složene dvofazne fenomene koje je teško modelirati. Samo nekolicina istraživača otkrila je matematičke izraze za opis složenog modela raspršivanja dvofazne mešavine vazduha i dizel goriva. U ovom pregledu, prikazan je razvoj modela dvofazne (vazduh/gorivo) kapljice iz Cummins-ove pumpe-brizgaljke. Uvedena je pretpostavka o istim prečnicima kapljice i otvora raspršivača, pri promenljivom masenom odnosu vazduh/gorivo unutar kapljice.

Ključne reči: penušavo raspršivanje, dvofazni, kapljica, dizel brizgaljka.

[1] Received: October 2010. Primljen: oktobar, 2010.god.
Accepted: December 2010. Prihvaćen: decembar, 2010.god.

A PHENOMENOLOGICAL MODEL OF TWO-PHASE (AIR/FUEL) DROPLET OF CUMMINS SPRAY

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UDC: 621.436.21

INTRODUCTION

In order to fulfill regulations for the emission of the exhausting products, Euro V and VI, as same as in order to reduce fuel consumption, manufacturers of diesel engines are using more often technologies with high-pressured fuel injection. Since 1980's a procedure, which atomizes liquid fuel (diesel), filled with the gas (air), under the pressure was known, this procedure is known in the newest editions [1] as the effervescent atomization – the atomization of the gas/liquid mixture. In the aerator, by means of special supplying system, gas is slowly injected in a liquid current, and on the exit of the atomizer a two-phase mixture – a liquid current mixed with air bubbles (air fractions) is created. The atomization process is stimulated with the gas expansion at very high speed on the atomizer exit, and this disintegrates fuel current on ligaments, lamellas and droplets [1]. Cummins's pump-injector under the high pressure injects two-phase of the diesel fuel mixture, air and the combustion products – and this is very interesting characteristic in the aspect of injection.

The aim of the present paper is to describe the recent research in the area of two-phase mixture air/diesel fuels atomization and to present our own a phenomenological model of two-phase (air/fuel) droplet developing and breakup on the Cummins's pump-injector example.

This paper is resumption and extending at a [9].

PREVIOUS RESEARCH

Roesler and Lefebvre [1, 2, 3] were investigated fluctuation of gas/liquid two-phase mixture. Through the atomizer's outlet, flowing can be in a form of bubbles, in a form of cylindrical gas tampons or in a form of the annular flow. On the outlet of the atomizer, gas fractions are getting into the phase of the relaxed pressure, they spreading very fast and disintegrate liquid into the droplets.

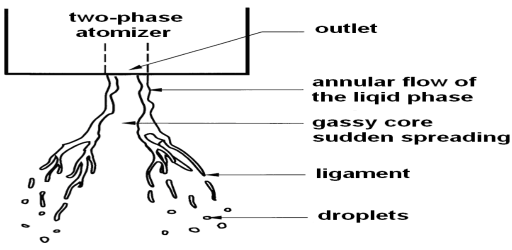

Figure 1: *Scheme of disintegration of two-phase annular flow after the atomizer's outlet [1]*

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The experiments that Sojke and coauthor has performed [1, 4] have shown the similar mechanism, as the air-filled fluid is getting out of the atomizer, the gas bubbles are spreading very quickly and thus they disintegrates fuel current on a thin ligaments. Under the aerodynamic influence of the gas, which is inside the space where the mixture is injected, they are rupturing, in such a way they are forming a droplets, figure 1.

In the paper [5] the results of performed examinations over the EDI injector - the atomizer of the air-filled gas are shown. The effects of injection pressure, gas/liquid mass ratio, atomizer outlet also, the influence of aerator parameters on atomization quality (the size of the droplet, and the spray conical angle) were examined.

MODEL OF TWO-PHASE DROPLET DEVELOPING AND DEFAULT ASSUMPTIONS

Introduction notes

In the chamber under the top of the needle of the Cummins's pump-injector, a liquid phase, fuel, is being mixed with the gas phase (air and the combustion products) and thus a two-phase mixture is created. A separate system for injector air supplying is not needed. A gas phase that is used for two-phase production, and which is consisting of air and the combustion products, comes into the Cummins's pump-injector out of the cylinders of the diesel engine, it is actually, being aspirated by the needle of the pump-injector, during its lifting of the seat in the atomizer.

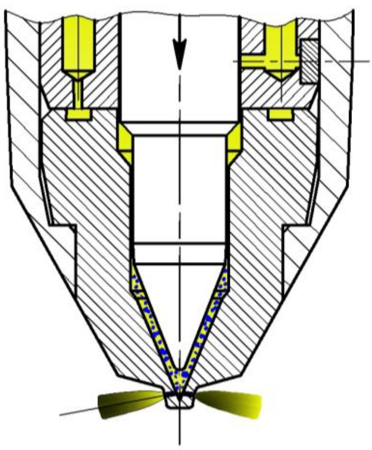

Figure 2: *A mixture of gas phase and diesel fuel under the needle of Cummins's pump-injector*

A liquid phase, fuel, comes into the chamber, under the injector needle; it comes under the fuel pressure in the inlet channel which is approximately same as the fuel pressure inside the fuel supply pump p_{gnp}. The fuel pressure in the supplying pump is significantly lower comparing to injection pressure, i.e. the pressure in the chamber under the top of the needle of the pump-injector p_b that can reach the values higher then 1500 bar [6]. A two-phase mixture produced in this way is a characteristic feature of Cummins's PT injection system. A needle of Cummins's pump-injector is pushing out, through the openings of the atomizer, a two-phase mixture, figure 2. A gas phase of the mixture (air), affects to liquid (diesel fuel) to flow through the openings of the atomizer in a form of a small fragments**.** Sudden spreading of the gas fragments, just after it came out of the atomizer outlet, stimulates liquid disintegration into the droplets.

The observation of the fluctuation process that occurs through the Cummins's atomizer outlet and the influence of the compressed air onto disintegration of the liquid flow can be significantly simplified in the first approximation, by means of the following assumption. The air-filled liquid fragments (a chain of primary droplets) of diesel fuel in the shape of the sphere, and inside the sphere's core is a bubble of compressed air, figure 3, are being injected.

The size of the primary spherical droplet of diesel fuel, a droplet with the gassy core, in a great deal depends on pressures. Pressure variations along the Cummins's pump-injector outlet, during the two-phase fluctuation, are significantly lower comparing to the values in the injectors with the one-phase fluctuation [1]. At the exit of the atomizer's outlet a two-phase pressure has a significant drop of the pressure. Two-phase fluctuation has a drastically dropping of the pressure at the exit of the atomization outlet. Diameter of the primary spherical droplet D_{kapl} *f(p)*, depending on the pressure, varies in the range of D_{kaplp_b} - a primary droplet diameter under the injection pressure p_b, up to the "rupturing" droplet diameter itself D_{kaplp_z} - a primary droplet diameter under the pressure in the engine cylinder p_z. The gas (air) is in a form of the bubble, which is compressed in the core of the spherical liquid droplet (diesel fuel), under the pressure over which a liquid is exposed. D_{vp} is a diameter of the air bubble in the spherical droplet under the pressure p.

According to this model, an initial calculation diameter of the primary spherical droplet $D_{kaplpoc}$ at the maximum pressure that can be achieved at the atomizer's outlet, and this value is approximate to the injection pressure - p_b, is equal to the diameter of the injector outlet - d_o (figure 3 a).

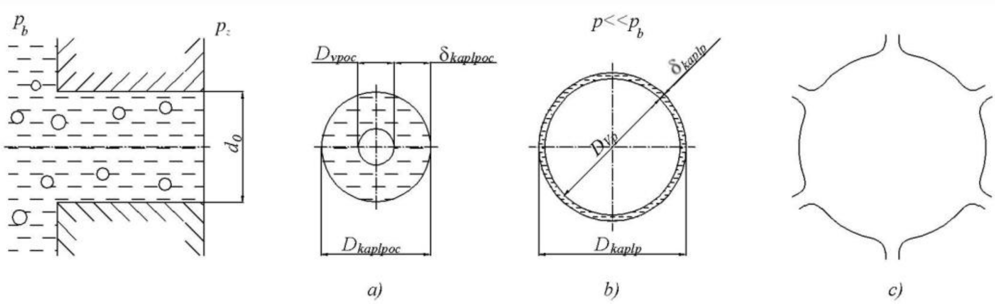

Figure 3: *Injecting scheme of air-filled liquid fragment of diesel fuel (a primary droplet), with the spherical shape and the air-compressed core*

$$D_{kaplpoc} = D_{kaplp_b} = d_o \tag{1}$$

Under the pressure of the engine cylinder - p_z, which is considerably lower then the injection pressure p_b, a droplet is spreading very fast**,** until the moment when it breakup (figure 3 b and c). Since a diesel fuel is the incompressible fluid**,** its volume is slightly increased, but due to that, the air volume inside the droplet is considerably increased. The volume of the air – that is compressible fluid is depending on the pressure in a great deal. While the droplet is spreading, a thickness of the liquid phase has been reduced more and more until the moment when a critical thickness of the liquid phase δ_{kaplkr} is reached, and then a droplet has been breakup- it decomposes. In that moment a droplet diameter is D_{kaplkr}. In order to perform a droplet volume calculation it is necessary to know air and diesel fuel specific density depending on the pressure, as same as the volume variations of the air and diesel fuel depending on the pressure.

Diesel fuel density ρ_g and elastic modulus e_g depending on the pressure p

Values of diesel fuel density and elastic modulus, depending on the pressure, which are experimentally determined, are given analytically in a form of the second-degree polynomial [6]. A diesel fuel density and elastic modulus at 20 ^oC:

$$\rho_{g20C} f(p) = 835{,}88 + 5{,}017x10^{-7} p - 6{,}974x10^{-16} p^2 \tag{2}$$

$$e_{g20C} f(p) = 1{,}4609x10^9 + 11{,}598p - 1{,}125x10^{-8} p^2 \tag{3}$$

at 80 ^oC:

$$\rho_{g80C} f(p) = 791{,}43 + 6{,}520x10^{-7} p - 1{,}036x10^{-15} p^2 \tag{4}$$

$$e_{g80C} f(p) = 9{,}906x10^8 + 11{,}497p - 1{,}0492x10^{-8} p^2 \tag{5}$$

In a above equations pressure is given in the units of Pa. Calculation results, which are presented, are performed under following assumption:

$$\rho_g f(p) = \rho_{g20C} f(p) \tag{6}$$

Air density ρ_v and elastic modulus e_v depending on the pressure p

In this model, a gassy part of the primary droplet – the air – is observed as the real gas, which is being very fast compressed at the pressure *p*. In a case of adiabatic air compression – (without heat exchanging with the environment) the air density at the pressure *p* is:

$$\rho_v f(p) = \rho_{vp} = \frac{p}{z(p,T) R_v T_{vo} \left(\frac{p}{p_o} \right)^{\frac{\chi - 1}{\chi}}} \tag{7}$$

where:

z(p,T) - function that shows deviations of the real gases in regard to the ideal gas at specific pressure and temperature, in analytical form [6]:

$$z(p,T) = 1 + \frac{9}{128} \frac{p_r}{T_r} (1 - 6T_r^{-2}) \tag{8}$$

where:

$p_r = \frac{p}{p_c}$ - reduced pressure, $T_r = \frac{T}{T_c}$ - reduced temperature,

p_c = 37,7 bar - critical pressure and T_c =132,2 K - critical temperature, at those values the air has been transformed into the liquid phase.

The air temperature T_v at the pressure p in a case of adiabatic change of the state:

$$T_v f(p) = T_{vp} = T_{vo} \left(\frac{p}{p_o} \right)^{\frac{\chi - 1}{\chi}} \tag{9}$$

where:

T_{vp} - the absolute air temperature at the pressure p,

T_{vo} - the absolute air temperature at the pressure p_o = 1 bar, T_{vo} = 20 oC ,

R_v - air gas constant R_v = 287 J·kg-1·K-1,

χ - adiabatic exponent, for air χ = 1,4.

Air compression is being performed very quickly, therefore there is no time for heat exchanging with the environment. The process of the air compression is considered as adiabatic change of the state, and that is the cause of the high temperatures of compressed air.

In a case of adiabatic air compression, while the system's entropy remains constant, a corresponding elastic modulus is called the adiabatic elastic modulus E_s .

$$E_s = -V\left(\frac{\partial p}{\partial V}\right)_{s=cons} \tag{10}$$

For the ideal gas:

$$E_s = \chi p \tag{11}$$

Air/ fuel mass and volume ratio in a two-phase droplet

It is difficult to evaluate the precise quantity of air presence in a two-phase fuel droplet. In this model is made an assumption that the same mass quotum is in every primary droplet and then the calculation is performed. During the droplet expansion, a mass quotum in a droplet remains the same. Ratio in between the air mass M_v and the fuel mass M_g inside the droplet is called air/fuel mass ratio glr .

$$glr = \frac{M_v}{M_g} \tag{12}$$

Ratio of air volume V_v and fuel volume V_g is called air/fuel volume ratio $vglr$:

$$vglr = \frac{V_v}{V_g} \tag{13}$$

There is an correlation In between air/fuel mass ratio glr and air/fuel volume ratio $vglr$:

$$vglrf(p) = vglr_p = \frac{V_v f(p)}{V_g f(p)} = \frac{V_{vp}}{V_{gp}} = glr\,\frac{\rho_{gp}}{\rho_{vp}} \tag{14}$$

where:

$vglr_p$ - gassy and liquid volume ratio at the pressure p ,

V_{vp} - gassy phase volume (air) at the pressure p ,

V_{gp} - liquid phase volume (diesel fuel) at the pressure p,

ρ_{gp} - fuel density at the pressure p,

ρ_{vp} - air density at the pressure p.

According to this - a volume ratio of gassy and liquid phase in the mixture (droplet) is depending on phase's densities, which are depending on the pressure, i.e. it has changing depending on the pressure that the droplet is exposed.

The air bubble diameter in a two-phase droplet at the injection pressure

A volume of the two-phase droplet mixture of fuel and air, at the injection pressure p_b is:

$$V_{kaplpoc} = V_{kaplp_b} = \frac{d_o^3 \pi}{6} \tag{15}$$

where: $V_{kaplpoc}$ - a starting calculation volume of the two-phase droplet.

Since the droplet is consisting of the fuel and the air then follows:

$$V_{kaplpoc} = V_{gpoc} + V_{vpoc} \tag{16}$$

or:

$$V_{kaplp_b} = V_{gp_b} + V_{vp_b} \tag{16'}$$

'

where:

V_{gpoc} - initial calculation volume of the liquid phase (fuel) of two-phase droplet.

V_{vpoc} - initial calculation volume of the gassy phase (air) of two-phase droplet.

According to equations 14, 16 and 16' it follows:

$$V_{kaplp_b} = V_{gp_b} + vglr_{p_b}, \Rightarrow V_{gpoc} = V_{gp_b} = \frac{V_{kaplp_b}}{1 + vglr_{p_b}} = \frac{V_{kaplp_b}}{1 + glr\,\dfrac{\rho_{gp_b}}{\rho_{vp_b}}} \tag{17}$$

Initial calculation volume of the droplet liquid phase is determined according to the following equation:

$$V_{vpoc} = V_{kaplpoc} - V_{gpoc} \tag{18}$$

Initial calculation diameter of the air bubble D_{vpoc} can be obtained according to the following equation:

$$D_{vpoc} = D_{vp_b} = \sqrt[3]{\frac{6V_{vpoc}}{\pi}} \tag{19}$$

Thickness of the liquid "film" of two-phase droplet at the injection pressure p_b:

$$\delta_{kaplp_b} = \frac{1}{2}\left(D_{kaplp_b} - D_{vp_b}\right) \tag{20}$$

Expansion of a two-phase droplet

When a droplet of compressed fuel and air, at the injection pressure p_b, get into the space of pressure $p \langle\langle p_b$, i.e. in the cylinder of diesel engine, an air-filled bubble inside of the core starts spreading very fast. Pressure alteration is very quick. If an assumption is made that air is actually an ideal gas, for the adiabatic change of the state, a volume of the two-phase bubble V_v at the pressure p is:

$$V_v f(p) = V_{vp} = \left(\frac{p_b}{p}\right)^{\frac{1}{\chi}} V_{vp_b} \tag{21}$$

for $p = p_z \Rightarrow V_{vp_z} = \left(\frac{p_b}{p_z}\right)^{\frac{1}{\chi}} V_{vp_b}$ (22)

In a case of pressure change from $p_b = 1000$ bar to $p_z = 1$ bar, a volume of the air-filled bubble has considerably changed:

$$V_{vp=1} = \left(\frac{1000}{1}\right)^{\frac{1}{\chi}} V_{vp=1000} = 138 V_{vp=1000} \tag{23}$$

Diameter of an air-filled bubble at the pressure p :

$$D_v f(p) = D_{vp} = \sqrt[3]{\frac{6V_{vp}}{\pi}} \tag{24}$$

for $p = p_z \Rightarrow D_{vp_z} = \sqrt[3]{\frac{6V_{vp_z}}{\pi}}$ (25)

Diesel fuel is a hardly compressed liquid. Fuel volume depending on the pressure has changed according to following definition:

$$\Delta V_{gp} = V_{gp} - V_{gp_b} = -V_{gp_b}\left(\frac{\Delta p}{e_g}\right) = -V_{gp_b}\,\frac{p - p_b}{e_g} \tag{26}$$

namely:

$$V_{gp} = V_{gp_b}\left(1 + \frac{p_b - p_z}{e_g}\right) \tag{27}$$

where: ΔV_{gp} - diesel fuel volume alteration due to pressure alteration. In the equation 26 there is minus sign because a volume alteration has an opposite sign comparing to the pressure alteration, $V_{gp} = V_g\, f(p)$ - fuel volume inside the droplet depending on the pressure.

At the pressure p a volume of the mixture consisting of diesel fuel and air is:

$$V_{gp} = \frac{1}{6}\pi\left[\left(D_{kaplp}\right)^3 - \left(D_{vp}\right)^3\right] \tag{28}$$

According to above equation a droplet diameter at the pressure p is:

$$D_{kaplp} = \sqrt[3]{\frac{6}{\pi}V_{gp} + \left(D_{vp}\right)^3} \tag{29}$$

for $p = p_z \Rightarrow D_{kaplp_z} = \sqrt[3]{\frac{6}{\pi}V_{gp_z} + \left(D_{vp_z}\right)^3}$ (30)

Thickness of the liquid “film” in a droplet at the pressure *p:*

$$\delta_{kaplp} = \frac{1}{2}(D_{kaplp} - D_{vp}) \tag{31}$$

for $p = p_z \Rightarrow \delta_{kaplp_z} = \frac{1}{2}(D_{kaplp_z} - D_{vp_z})$ (32)

Perhaps this will never be reached – if the droplet breakup before. That will happen whenever is $\delta_{kaplkr} \rangle \delta_{kaplp_z}$.

In order to analyze the model such as this one here presented, the following estimation is accepted; when due to air bubble expansion within the two-phase droplet, the thickness of the liquid “film” has been reduced up to its critical value δ_{kaplkr}, a two-phase droplet will disintegrate. In the moment when a two-phase droplet is being disintegrated, its diameter D_{kaplkr} - is critical diameter, diameter of two-phase droplet “disintegration”. The pressure inside the air bubble, during two-phase droplet disintegration, is a critical pressure p_{kr} .
If disintegration pressure – i.e. critical pressure of two-phase droplet is:

$$p_{kr} \rangle p_z \tag{33}$$

Then critical thickness of the liquid “film” is:

$$\delta_{kaplkr} \rangle \delta_{kaplp_z} \tag{34}$$

A two-phase droplet will disintegrate before the thickness of the liquid “film” reduce to critical value δ_{kaplp_z} , i.e. before its expansion process, defined by the pressure inside of the engine cylinder, is finished.

If the assumed value of liquid “film” thickness of two-phase droplet during its disintegration is:

$$\delta_{kaplp} = \delta_{kaplp} f(p_b, d_o, glr, p) = \delta_{kaplkr} \tag{35}$$

Then critical pressure, depending of gas phase and liquid phase mass ratio is determined as glr :

$$p_{kr} = pf(glr) = p\,/.FindRoot\left\lfloor \delta_{kaplp} f(p_b, d_o, glr, p) = \delta_{kaplkr} \right\rfloor \tag{36}$$

For the assumed value of liquid "film" critical thickness δ_{kaplkr}, a minimal and real value of gas phase and liquid phase mass ratio has been calculated glr_{p_z}, when under certain pressure value inside engine's cylinder p_z a two phase droplet has been disintegrated:

$$glr_{p_z} \rightarrow FindRoot\left[pf\left(glr\right)\right] = p_z \tag{37}$$

Critical pressure which is depending on gas/liquid phase mass ratio, is determined by equation 36 and has been graphically presented within the interval glr_{p_z} to $glr = 0{,}1$. Within the same interval a critical diameter of two-phase droplet D_{kaplkr} has been shown, which has been calculated for the state of critical pressure.

ANALYSIS OF OBTAINED RESULTS FROM TWP-PHASE DROPLET MODEL

Volume and diameter of the air-filled bubble inside the two-phase droplet

In figure 4 is shown a calculated volume alternation of the air-filled bubble V_{vp} inside the two-phase droplet of the diesel fuel (a liquid phase) and air (a gassy phase) mixture, depending on droplet gassy/fuel phase mass ratio *glr*, and depending on the pressure in a area close to the droplet *p*. It can be noticed that volume of the air-filled bubble considerably depends on the pressure in the area close to the droplet *p*.

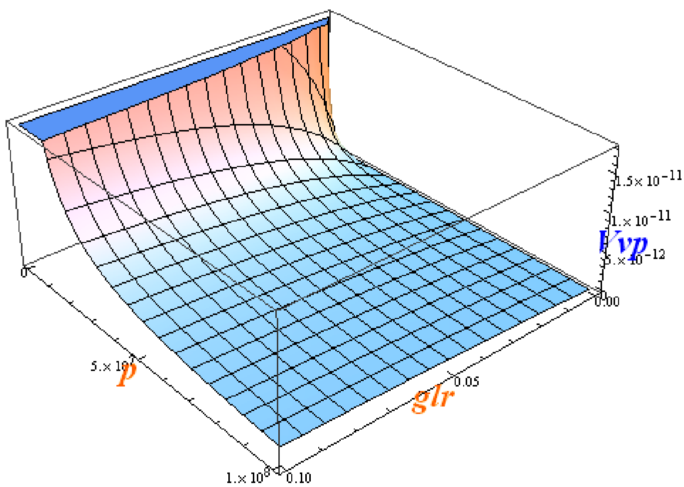

Figure 4: *Air-filled bubble volume alteration V_{vp} (m^3), inside the two-phase droplet of diesel fuel and air mixture, depending on droplet gassy/fuel phase mass ratio glr (–) and the pressure in the area close to the droplet p (Pa).*
Atomizer's outlet diameter d_o= 0,215 mm

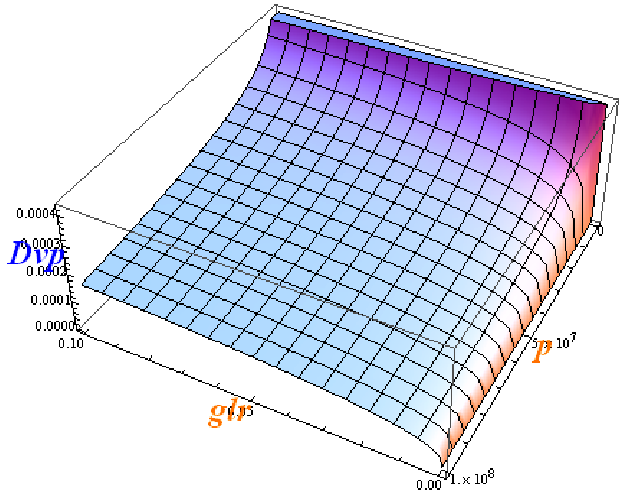

Figure 5: *Diameter alteration of the air-filled bubble* D_{vp} *(m) inside the two-phase droplet a mixture of diesel fuel and air, depending on the droplet gassy/fuel phase mass ratio glr (–) and the pressure in the area close to the droplet p (Pa). Atomizer's outlet diameter* d_o *= 0,215 mm*

Volume alteration intensity of the air-filled bubble increases as the pressure that reacts on the droplet decreases, and the same occurs even for the lower *glr* ratio values. Presence of small air quantity inside the diesel fuel droplet that is spreading very fast after injection has a devastating effect onto droplet.

Character of the air-filled bubble diameter alteration $D_{vp} = D_v\, f(p, glr)$ is very same to air-filled bubble volume alteration inside the droplet, and that can be noticed according to figures 4 and 5.

A diesel fuel volume inside the two-phase droplet depending on the environmental pressure

In figure 6, a diesel volume alteration V_{gp} is presented, inside the droplet mixture of diesel fuel (a liquid phase) and air (a gassy phase) depending on the droplet gassy/fuel phase mass ratio *glr*, and the pressure in the area close to the droplet *p* Pa. Those values are calculated with the following assumption:

$$e_g = e_{g20C} \tag{38}$$

It can be noticed that diesel fuel volume – i.e. liquid "film" of air-filled bubble inside the droplet – very little, insignificantly depends on the pressure *p* that effects to the droplet. Influence that is more important has a droplet gassy/fuel phase mass ratio *glr*. There is a significant difference in the character of the gassy and the liquid phase volume alteration, depending on the pressure.

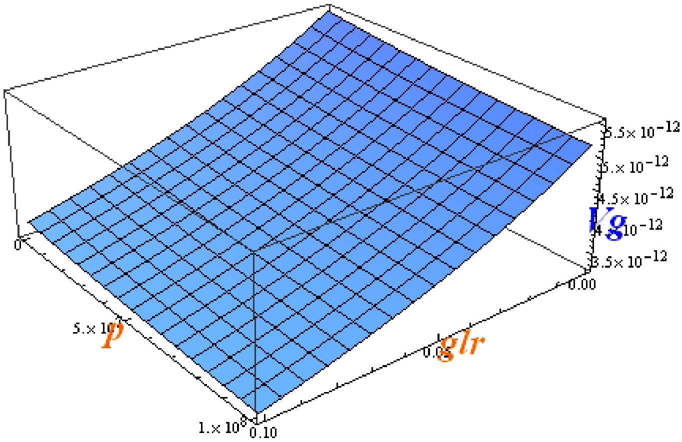

Figure 6: *Diesel fuel volume* V_{gp} *(m^3) in a two-phase droplet – mixture of diesel fuel and air, depending on the droplet gassy/fuel phase mass ratio glr (–) and the pressure in the area close to the droplet p (Pa)*

Outer diameter of the two-phase droplet depending on the environmental pressure

At the figure 7, a calculated correlation in between the outer diameter of the two-phase droplet of diesel fuel and air, depending on the environmental pressure and the droplet gassy/fuel phase mass ratio, is presented. Outer diameter of the two-phase droplet **considerably** depends on pressure *p* that effects to the droplet, and **not considerably** of gassy/fuel phase mass ratio. On the other hand, a presence of even small air quantities inside the mixture, at the low environmental pressure, increases very fast the outer diameter of the droplet – until it breakup, in that moment a droplet diameter is D_{kaplkr}.

Thickness of the liquid "film" of two-phase droplet depending on the environmental pressure

In figure 8 a calculated correlation in between the thickness of the liquid "film" of two-phase droplet is shown, depending on the droplet gassy/fuel phase mass ratio and the pressure in the area close to the droplet. Thickness of the liquid "film" of two-phase droplet is depending on the pressure in the area close to the droplet, it decreases as the pressure decreases. With the increase of the droplet gassy/fuel phase mass ratio, *glr,* the thickness of liquid "film" is reduced, a droplet become thinner. At lower pressures in the droplet environment, the thickness alternation intensity of droplet "film" becomes higher.

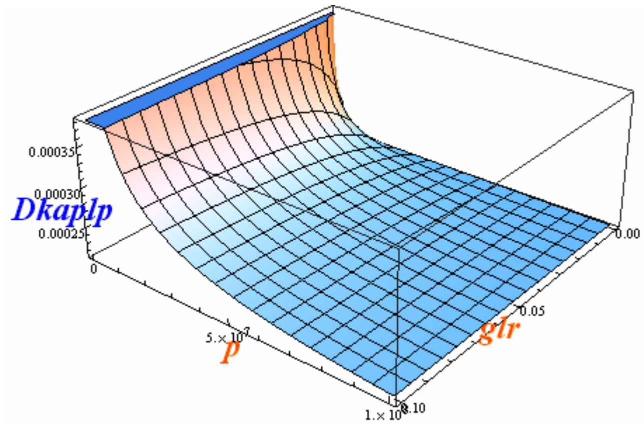

Figure 7: *Outer diameter D_{kaplp} (m) of the two-phase droplet of diesel fuel and air, depending on the droplet gassy/fuel phase mass ratio glr (–) and the pressure in the area close to the droplet p (Pa). Atomizer's outlet diameter d_o= 0,215 mm*

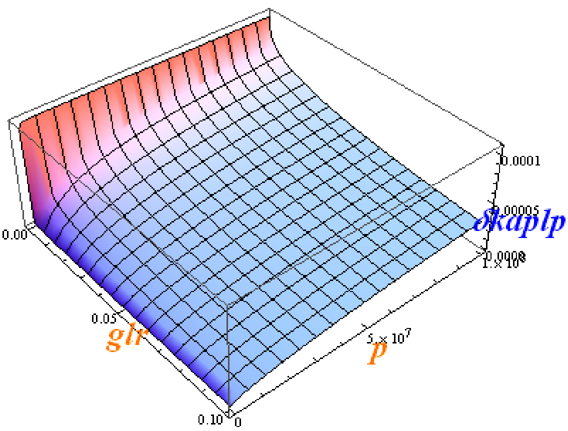

Figure 8: *Thickness of the liquid "film" of two-phase droplet – mixture of diesel fuel and air δ_{kaplp}(m) depending on the droplet gassy/fuel phase mass ratio glr (–) and the pressure in the area close to the droplet p (Pa). Atomizer's outlet diameter d_o= 0,215 mm*

It is very hard to distinguish criteria when the droplet is actually breakup. Inside the droplet, a spreading air is effecting, and from the outside an aerodynamic force, which is formed due to high values of relatives speed in between the droplets and the nearby air in a combustion area. In a reference [7] are given Mayer's criteria for the one-phase droplet breakup (one-phase droplet is consisting from the liquid phase – only) and those criteria are experimentally defined.

When the thickness of liquid "film" is reduced to it's critical value δ_{kaplkr}, a two-phase droplet is breakup. A critical thickness value of the liquid "film" can be experimentally determined. In figures 9 and 12 a values of environmental pressure and gassy/fuel phase mass ratio, in the moment of the two-phase droplet breakup are shown. A thickness of liquid "film" in the moment of two-phase droplet breakup is assumed to be δ_{kaplkr} = 0,00001 m (fig. 9) and δ_{kaplkr} = 0,000015 m (fig. 12). According to previously shown calculated values, it can be concluded following: if the two-phase droplet (air/fuel) has plenty of air (a gassy phase), then it will breakup at higher pressures. In that moment a droplet diameter is D_{kaplkr}.

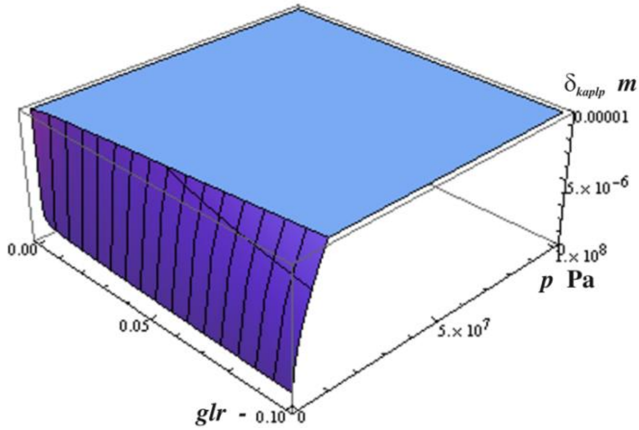

Figure 9: *The pressure in the area close to the droplet p (Pa) gassy/fuel phase mass ratio glr (–) in the moment of two-phase droplet breakup, with the liquid "film" thickness δ_{kaplp} = 0,00001 m (0,01 mm), d_o=0,215 mm*

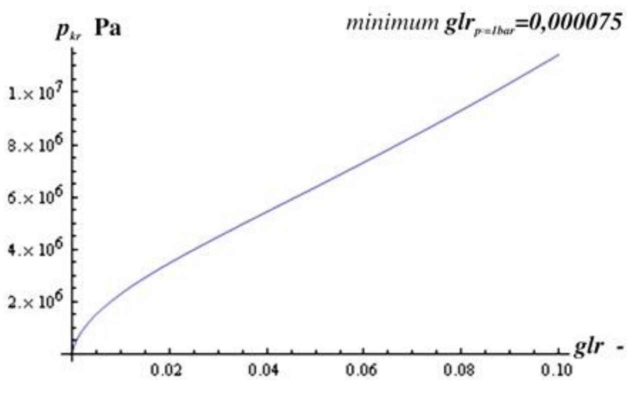

Figure 10: *Dependency of critical pressure p_{kr} (Pa) on gas/liquid mass ratio glr (–) during the two-phase droplet disintegration, liquid "film" critical thickens is δ_{kaplkr} = 0,00001 m, d_o= 0,215 mm. Injection pressure $p_b = 1x10^8$ Pa. The environmental pressure, where the two-phase droplet has been injected is p_z = 1bar = $1x10^5$ Pa. Minimal value of gas/liquid phase mass ratio $glr_{p=1bar}$ = 0,000075 which will cause disintegration of the droplet under the pressure p_z = 1bar*

In figures 10 and 11 is shown dependency, among the pressure and diameter during the disintegration of two-phase droplet, dependency of critical pressure p_{kr} and critical diameter D_{kaplkr} on gas/liquid phase mass ratio *glr*,, which is calculated based on here presented model under the assumed value of liquid "film" critical thickness δ_{kaplkr} = 0,00001 m (0,01 mm). In figures 13 and 14 dependency of the values of critical pressure and critical diameter on gas/liquid phase mass ratio are shown, calculated based on assumed value of liquid "film" critical thickness of two-phase droplet δ_{kaplkr} = 0,000015 m (0,015 mm).

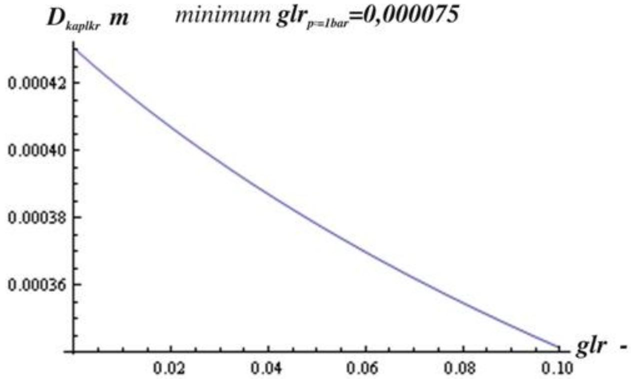

Figure 11: *Dependency of critical diameter D_{kaplkr} (m) on air/fuel mass ratio glr(–) during the two-phase droplet disintegration, critical thickness of liquid "film" is δ_{kaplkr} = 0,00001 m. Diameter of nozzle's outlet is d_o= 0,215 mm. Injection diameter $p_b = 1x10^8$ Pa. The environmental pressure, where the two-phase droplet has been injected is p_z = 1bar = $1x10^5$ Pa. Minimal value of gas/liquid phase mass ratio $glr_{p=1bar}$ = 0,000075 which will cause disintegration of the droplet under the pressure p_z = 1bar*

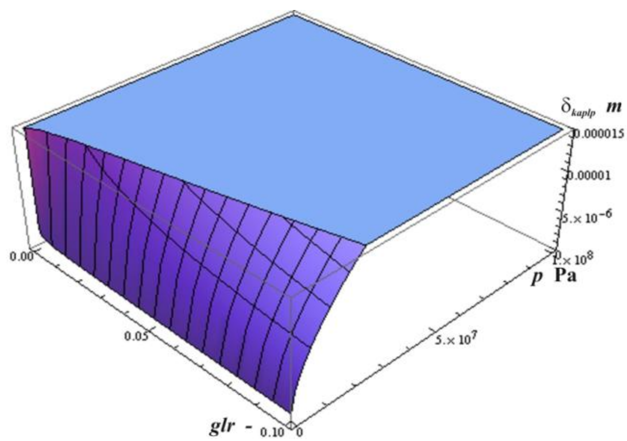

Figure 12: *The pressure in the area close to the droplet p (Pa), gassy/fuel phase mass ratio glr (–) in the moment of two-phase droplet breakup, with the liquid "film" thickness δ_{kaplp} = 0,000015 m, d_o=0,215 mm*

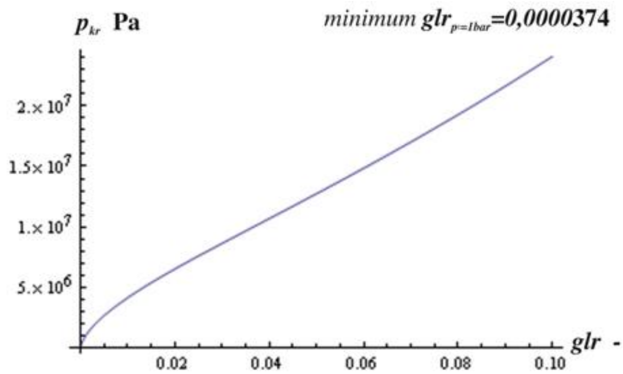

Figure 13: *Critical pressure dependency p_{kr} (Pa) on gas/liquid phase mass ratio glr (–) during two-phase droplet breakup, critical thickness of liquid "film" is δ_{kaplkr} = 0,000015m. Diameter of nozzle's outlet is d_o= 0,215 mm. Injection pressure $p_b = 1x10^8$ Pa. The environmental pressure, where the two-phase droplet has been injected is $p_z = 1bar = 1x10^5$ Pa. Minimal value of gas/liquid phase mass ratio $glr_{p=1bar}$ = 0,0000374 which will cause disintegration of the droplet under the pressure p_z = 1bar*

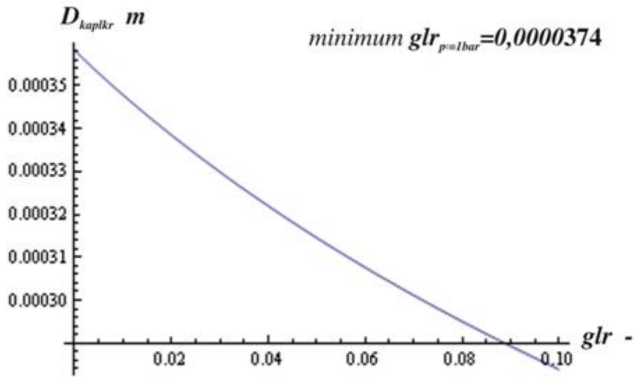

Figure 14: *Dependency of critical diameter D_{kaplkr} (m) on gas/liquid phase mass ratio glr (–) during two-phase droplet breakup, critical thickness of liquid "film" is δ_{kaplkr} = 0,000015 m. Diameter of nozzle's outlet is d_o= 0,215 mm. Injection pressure $p_b = 1x10^8$ Pa. The environmental pressure, where the two-phase droplet has been injected is $p_z = 1bar = 1x10^5$ Pa. Minimal value of gas/liquid phase mass ratio $glr_{p=1bar}$ =0,0000374 which will cause disintegration of the droplet under the pressure p_z = 1bar*

Under specified calculation conditions a minimal value of gas/fluid phase mass ratio has been calculated, under which a two-phase droplet will breakup (disintegrate) by the pressure p_z = 1bar and assumed critical thickness of liquid "film" minimum $glr_{p_z=1bar}$= 0,000075 for the thickness δ_{kaplkr} = 0,00001 m (0,01 mm), i.e. minimum $glr_{p_z=1bar}$= 0,0000374 for the thickness δ_{kaplkr} = 0,000015 m (0,015 mm). Two-phased droplet, according to the presented model and specified calculation assumptions, should contain a considerable small amount of air in order to breakup – i.e. to disintegrate, under the pressure p_z = 1bar. Considerable small amount of air, specified by *glr* = 0,000075, which is compressed inside the droplet under the injection pressure $p_b = 1x10^8$ Pa, is spreading and make the liquid "film" thinner up to δ_{kaplkr} = 0,00001 m (0,01 mm), and then a two–phase droplet will breakup.

For the constant value of the environmental pressure p_z under which a two-phase droplet has been injected, if the assumed value of liquid "film" thickness δ_{kaplkr} is being increased then a minimal value of droplet's gas and liquid phase (minimum glr_{p_z}), which will cause the breakup of a droplet under the pressure p_z, will be reduced, same as the diameter of two-phase droplet during the disintegration.

$$\begin{aligned}&\delta_{kaplkr} \Uparrow \Rightarrow minimum\,\delta_{p_z} \Downarrow za\ p_z = constans\\&\delta_{kaplkr} \Uparrow \Rightarrow D_{kaplkr} \Downarrow za\ p_z = constans\end{aligned} \tag{39}$$

Based on here presented model it might be concluded, if inside the droplet the air is not present, *glr* = 0, then the droplet will not breakup – it will not disintegrate at all. In that case liquid "film" thickness is equal to droplet's diameter and practically it's not depending on pressure.

It's not so easy to determinate the thickness of a two-phase droplet during its disintegration. In the literature, for the time being, such a data do not exist. So in that sense, in order to make the comparison, the experimental results of some other researchers might be used, "dispersion gas which is injected, inserts much more energy than it is really needed for the droplets forming. Even under low injection pressures 10 MPa and at 1% air/fuel mass ratio, Sovani [5, 8] estimates that the energy consisted inside dispersion gas is almost 30 times higher than it is necessary to disperse the mixture."

In figure 15. by coordinate axes *p, glr* a critical thickness of two-phase droplet liquid "film' has been shown δ_{kaplkr} calculated under the injection pressure p_b =10 MPa, all other calculation terms and conditions are shown in the figure. Based on here presented results, under specified calculation terms, it can be concluded following: inside the nozzle's outlet if *glr*>0,065 =6,5%, where the pressure is $p = p_b$, a two-phase droplet, consisted of air and diesel fuel mixture, with the thickness of the liquid "film" δ_{kapl} = 0,000015 m, cannot be formed. In that case, through the nozzle's outlet gas and liquid phase "spills" has been frequently injected.

Based on presented results of mathematical two-phase droplet disintegration, it can be concluded that during the injection under high pressure $p_b = 1x10^8$ Pa, compared to injection under considerable lower pressure $p_b = 1x10^7$ Pa = 10 MPa inside the nozzle's droplet will occur bubbled flow regime, in that case a gas phase will flow in the form of small bubbles. This conclusion is in the compliance with the results of the experimental researches performed for two-phase flow. By the experimental researches it is estimated that if the injection pressure is been increased then *glr* values, which will preserve bubbled flow regime, will be increased as well [1]. It should be emphasized that here mentioned experimental researches of two-phase flow are presented in literature [1], performed under injection pressures within the scope 0,1 MPa to 1,1 MPa.

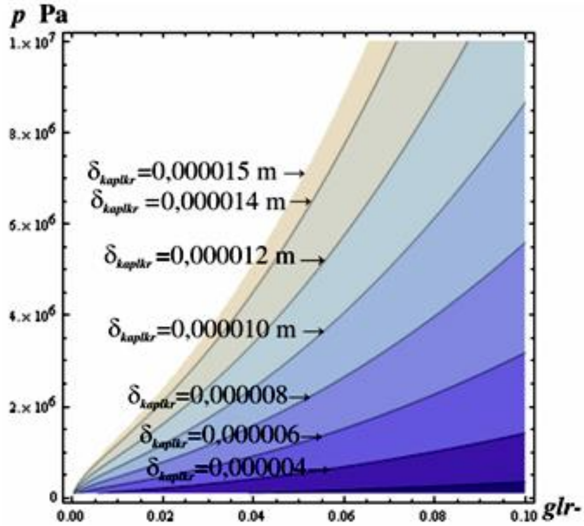

Figure 15: *Dependency of critical thickness of two-phase droplet liquid "film" δ_{kaplkr} (m) on gas/liquid phase mass ratio glr (-). Diameter of nozzle's outlet is d_o= 0,215 mm. Injection pressure p_b = $1x10^7$ Pa = 10 MPa. The environmental pressure, where the two-phase droplet has been injected is p_z = 1bar = $1x10^5$ Pa*

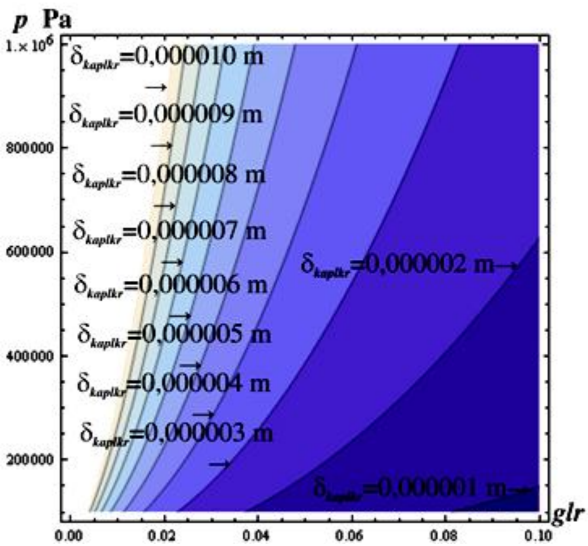

Figure 16: *Dependency of critical thickness of two-phase droplet liquid "film" δ_{kaplkr} (m) on gas/liquid phase mass ratio glr (-). Diameter of nozzle's outlet is d_o= 0,215 mm. Injection pressure p_b = $1x10^6$ Pa = 1 MPa. The environmental pressure, where the two-phase droplet has been injected is p_z = 1bar = $1x10^5$ Pa*

In figure 16 is shown a segment of two-phase droplet breakup calculation, injected under the injection pressure p_b = 1 MPa. Based on the presented results of two-phase droplet breakup, when the injection pressure is p_b = 1 MPa, if the gas/liquid phase mass ratio is

glr > 0,021 inside the nozzle's outlet, then two-phase droplet made of air and diesel fuel mixture, with the thickness δ_{kapl} = 0,00001 m of the liquid "film" cannot be formed. So, in that case, under the assumption that the thickness of the liquid "film" is δ_{kaplkr} = 0,00001 m a two-phase droplet will disintegrate – breakup, inside the nozzle's outlet and through the outlet a "spills" of gas and liquid phase will be injected. Obtained result is in the compliance with the experimental researches. In the experiment described in work-paper [1], researchers Santangelo and Sojka made the examination based on the injection pressures of two-phase air/water mixture in the scope 0,1 MPa do 1,1 MPa and *glr*<0,02 – and they have noticed that some gas bubbles fluently flow through the nozzle's outlet, i.e. the bubbled flow boundary is *glr*<0,02 - .

CONCLUSIONS

Here presented computer model shows the influence of the gassy phase (i.e. air) onto the development of the two-phase droplet – a mixture of diesel fuel (i.e. a liquid phase) and the air, depending on the gassy/fuel phase mass ratio *glr*, and the pressure in the area close to the droplet *p*. Initial calculation droplet diameter $D_{kaplpoc}$ is equal to the atomizer's outlet diameter d_o. A process of air compression is considered as adiabatic change of the state, and that's the reason of high temperatures of compressed air. A relative speed in between the droplet and a nearby air is not taken into consideration. A volume of the liquid phase (a fuel) practically is not depending on the environmental pressure. Intensity of volume alteration of the gassy phase (i.e. the air) and the alteration of the outer two-phase droplet diameter are higher at lower environmental pressures. Compressed air inside the two-phase droplet has a devastated effect on it. If there is more air inside the droplet, then it will disintegrate at higher environmental pressures, actually it will disintegrate faster. Based on the calculation it was established that increased injection pressure will also increase *glr* up to the value which will preserve bubbled flow regime inside of the nozzle's outlet, what is in the compliance with the experimental researches. Though this model is made with the variety of assumptions, it is indeed "rough", the obtained results are giving the essential basic information regarding a two-phase flow inside the engine cylinder behavior.

Any form of the liquid phase, with a certain gas quantity, can be identified to "equivalent" spherical liquid-phase droplet with the spherical gassy volume inside it. It is necessary to upgrade this calculation model and by means of experimental testing to determinate the criteria of droplet decomposing.

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CNG BUSES FOR CLEAN AND ECONOMICAL CITY TRANSPORT[1]

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UDC: 656.132:662.767.2]:504.3

Abstract

Global warming, increasing air pollution and diminishing oil reserves have made the need for alternative fuel use imperative around the world. Use of compressed natural gas as an alternative fuel is an effective, currently available way to help solve pressing environmental and fuel – resource problems. Because of its lower carbon content (H/C ratio close to 4), natural gas causes about 25% less CO_2 – emissions than diesel fuel for same amount of energy and thus makes an important contribution to the reduction of CO_2 and pollutants.

This paper presents the technical characteristics of urban buses powered on compressed natural gas, as well as environmental and economic benefits of the introduction of these vehicles in local public transport, primarily in conditions of exploitation in the larger towns. In first time, according to our project "KRAGUJ", diesel buses in urban traffic in the city of Kragujevac, will very quickly be replaced with new CNG powered buses. The future is dedicated to biogas buses, in an effort to reduce emissions of CO_2, particles and noise.

Key words: CNG buses, clean city transport, greenhouse gas emissions, ECE R 110.

AUTOBUSI SA POGONOM NA KOMPRIMOVANI PRIRODNI GAS ZA ČISTIJI I EKONOMIČNIJI GRADSKI PREVOZ

UDC: 656.132:662.767.2]:504.3

Rezime: Globalno zagrevanje, povećano zagađenje vazduha i smanjenje naftnih rezervi, daju sve veći imperativ primeni alternativnih goriva u svetu. Primena komprimovanog prirodnog gasa kao alternativnog goriva predstavlja efektivan i trenutno dostupan način za rešavanje problema zaštite okoline i resursa goriva. Zbog niskog sadržaja ugljenika (odnos H/C=4), prirodni gas prouzrokuje oko 25% nižu emisiju CO_2 u odnosu na dizel gorivo za istu količinu energije i prema tome, značajno doprinosi smanjenju emisije CO_2 i drugih polutanata.

[1] Received: October 2010. Primljen: oktobar, 2010.god.
Accepted: December 2010. Prihvaćen: decembar, 2010.god.

U ovom radu su prikazane tehničke karakteristike gradskih autobusa sa pogonom na komprimovani prirodni gas, kao i ekološke i ekonomske prednosti uvođenja ovih autobusa u javni gradski prevoz putnika, primarno za uslove eksploatacije u većim gradovima.

Saglasno našim predlozima u okviru projekta „KRAGUJ", dizel autobusi u javnom gradskom prevozu putnika grada Kragujevca bi trebalo vrlo brzo da budu zamenjeni novim autobusima sa pogonom na komprimovani metan CNG. U budućnosti se ide korak dalje sa primenom biogasa za pogon autobusa, saglasno redukciji emisije CO_2, čestica i buke.

Ključne reči: autobusi, čistiji saobraćaj, emisija gasova staklene bašte, ECE R110.

CNG BUSES FOR CLEAN AND ECONOMICAL CITY TRANSPORT

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UDC: 656.132:662.767.2]:504.3

INTRODUCTION

The goal of the European Union is that alternative sources of energy will represent 20% of total consumption by 2020, confirming the strong political will amongst European countries to reduce global pollution and improve living standards. The use of alternative sources of energy can significantly increase a country's energy independence from international political and economic issues, since they need not fear the threat of energy supply interruption. Geographical features of the country affect the final decision as to the type of alternative source of energy to use.

Whatever climate policies are introduced, natural gas – a special focus in WEO-2009 *(*The World Energy Outlook 2009) – is also set to continue to play a bridging role in meeting the world's sustainable energy needs. In the Reference Scenario, gas demand rises by 41% from 3.0 **t**rillion **c**ubic **m**etres (tcm) in 2007 to 4.3 tcm in 2030. Gas demand also continues to expand in the "450 Scenario" but is 17% lower in 2030 than in the Reference Scenario thanks to more efficient use, lower electricity demand and increased switching to non-fossil energy sources [1].

WEO-2009 demonstrates that containing climate change is possible but will require a profound transformation of the energy sector. The "450 Scenario" sets out an aggressive timetable of actions needed to limit the long-term concentration of greenhouse gases in the atmosphere to 450 parts per million of carbon-dioxide equivalent and keep the global temperature rise to around 2 ^{o}C above pre-industrial levels. To achieve this scenario, fossil-fuel demand would need to peak by 2020 and energy-related carbon dioxide emissions to fall to 26.4 gigatonne (Gt) in 2030 from 28.8 Gt in 2007 [1].

Today, there are two key benefits of operating a Compressed Natural Gas (CNG) powered vehicle:

- Lower harmful emissions, and
- Reduced fuel costs (CNG fuel cost is approximately 50% less than diesel).

The greatest environmental benefit of using CNG as fuel is the dramatic reduction of exhaust gases; for example, compared to the equivalent Turbo Diesel Engine carbon

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monoxide (CO) emissions are over 50% lower, nitrogen oxide (NO_x) emissions are 97% lower, and there are zero particulates.

The engine also can operate on biofuels, a sustainable fuel that is now becoming commercially available from renewable sources, or CNG, which is the same gas we use for our heating and cooking at home.

As concern road transport vehicles featuring following properties seems to be good candidate for shift from consumption of liquid hydrocarbons to use of natural gas as a fuel:

- Fleet vehicles whose are able to be supplied by fuel from one refuelling station,
- Vehicles performing limited amount of kilometres per day not demanding storage of high amount of fuel in vehicle tank,
- Vehicles which do not fully utilize their loading capacity during typical operations. Weight penalty due to high mass of fuel storage tank does not influence payload capacity of vehicle in this case, and
- Vehicles, whose makes possible by proper design and arrangement of their power unit to exploit the environmental advantages of natural gas use. In this way the still more strict environmental regulation demands can be fulfilled or even lower emission of pollutants can be obtained.

Usually the daily performance of city bus is about 350 km for our city conditions. Nevertheless the city bus is many times favourite subject for use of natural gas as a fuel.

Natural Gas is pressurized to 22 MPa in vehicular storage tanks, so that it has about 1/3 of the volumetric energy density of gasoline. The storage pressure is about 20 times that of propane. Combustion of methane is different from that of liquid hydrocarbon combustion since only carbon-hydrogen bonds are involved and no carbon-carbon bond, so the combustion process is more likely to be complete, producing less non-methane hydrocarbons. Operations under lean conditions will also lower the peak combustion temperatures. The lower combustion temperatures lower the NO_x formation rate and produce less engine-out NO_x.

Natural gas (from the chemical point of view mainly methane CH_4) contains 75% of carbon and 25% of hydrogen by mass. Diesel oil, which is composed from heavy hydrocarbons, contains about 87% of carbon and 12% of hydrogen by mass. Complete oxidization of 1 kg methane produces approx. 2.75 kg of CO_2, while complete oxidization of 1 kg diesel oil produces about 3.15 kg of CO_2. At the same time heating value of methane is higher than those of diesel oil in ratio about 50/43 (MJ/kg). Even if slightly lower efficiency of gas engine (compared to diesel one) is taken into account the production of CO_2 from natural gas powered vehicle is significantly lower than those from vehicle equipped with diesel engine. Natural gas fuelled engine practically does not produce any particles. Premixed combustion which takes place in combustion chamber of gas engine avoid the reactions which caused in diesel engine combustion chamber to fuel thermal decomposition and formation of soot [2,3,4,5,6].

THE ECO-CNG POWERED CITY BUS PROJECT

Take in mind the experience of leading manufacturers of buses, we propose the technical specifications and new design for a prototype of fully low floor city bus with CNG drive. The prototype bus is implemented with the original gas engine mark CUMMINS according to Euro IV legislations, while the vehicle serial production continued with engines to meet the Enchanced Enviroment friendly Vehicle (EEV/EEV+) norm. Assembling the original engine is provided the maximum efficiency of the applied fuel, since the same is designed for operation exclusively on CNG, so that their structural characteristics and projected operating cycle, ensures maximum dynamics and efficiency. In combination with the automatic gearbox mark ALLISON, achieved good performance of movement and maximum use of engine output parameters, which have a positive effect on passengers comfort and fuel economy [3].

Figure 1 shows the parts of the CNG fuel line equipment from on the bus roof mounted gas cylinders trade mark DYNETEK to the engine that is applied to bus. All parts of the installations are designed according to regulation UN ECE R 110 and the same labels are printed on the material.

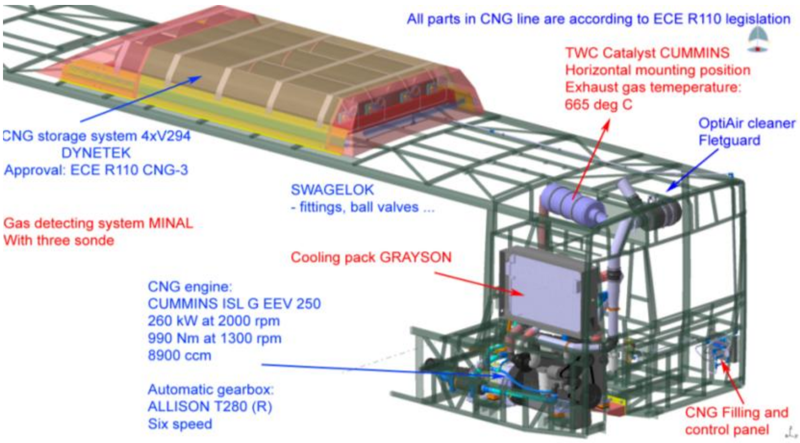

Figure 1: *The CNG fuel line equipment in the bus*

Implementation of CNG as fuel, first of all, request mounting of gas cylinders with the appropriate support. The mounting position on the bus roof has been selected according to the projected mass distribution on the front axle and rear – driving axle, all on the basis of primary diesel vehicle, in order to save vehicle performances and brake forces distribution [3].

On the vehicle roof are mounted four cylinders mark DYNETEK "V294", with a total water capacity of 1.176 l, diameter 386 mm and length 3128 mm. Weight of one tank was about 92.4 kg (0.308 kg/l). The composite DyneCell® cylinders are particularly lightweight cylinders for the storage of CNG. They consist of a thin-walled, seamless aluminium internal vessel whose entire surface is wrapped with a high-strength carbon fibre reinforcement (CNG Type III = “fully wrapped metal liner” in accordance with ECE R 110 and ISO 11439) [3].

By using tank-type CNG-3, achieved the better bus performance, lower weight, has up to 8 seats more for passengers, the lower number of failures and regular vehicle services, the friction on the wheels of the front axle is less for about 30%, and gas consumptions is lower for about 0.5 to 1 kg/100 km. As a comparison, for the same radius with one filling, if used gas tanks-type CNG-2, bus had about 40% more weight [7].

Low Emissions, Improved Performance, Lower Costs

This bus package contains the Cummins Westport 2007 ISL G spark ignited, natural gas engine. These engines are designed to meet the proposed 2010 U.S. Environmental Protection Agency (EPA) and California Air Resources Board (CARB) emission standards at launch in mid 2007, fig. 2. The engine is based on the 8.9 L diesel automotive (ISL) platform and shares many installation options with the diesel counterpart.

The 8.9 L 2007 ISL G engines will use stoichiometric combustion with Cooled Exhaust Gas Recirculation (CEGR) technology to enable a three way catalyst after treatment method, leveraging Cummins proven EGR technology to create a high-performance natural gas engine. This replaces the lean-burn technology of C Gas Plus and L Gas Plus engines. The cooled EGR system takes a measured quantity of exhaust gas and passes it trough a cooler to reduce temperatures before mixing it with fuel and the incoming air charge to the cylinder. Cooled EGR, in combination with stochiometric combustion (the theoretical or ideal combustion process in which fuel and oxygen are completely consumed, with no unburned fuel or free oxygen in the exhaust), provides significant benefits [3,4,8].

The use of cooled EGR (in place of large amounts of excess air used in lean burn technology) lowers combustion temperatures and knock tendency. Use stoichiometric combustion with CEGR technology also improves power density and fuel economy compared to lean and alone stoichiometric technologies. Compared to previous Cummins Westport Inc. (CWI) lean burn natural gas engines, ISL G torque at idle is improved over 30% and fuel economy is improved by up to 5% [8].

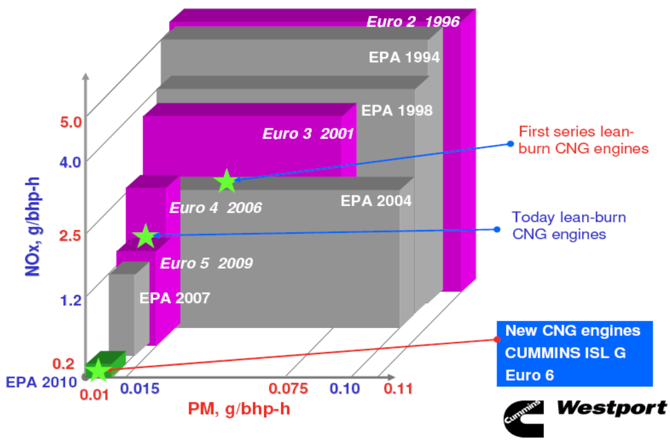

Figure 2: *Exhaust gas emission standards and ISL G position*

Another benefit of the ISL G`s advanced combustion technology is CEGR combustion creates an oxygen-free exhaust, which in turn allows for the use of Three-Way Catalyst (TWC) after treatment. TWC-s is effective, simple passive devices packaged as part of the muffler that provide consistent performance and are maintenance-free. The ISL G does not require active after treatment such as a diesel particulate filter (DPF) or selective catalytic reduction (SCR).

Air/Fuel regulation in Cummins closed-loop electronic control system based on Cummins Interact™ System. Sensors for engine parameters, including intake manifold pressure and temperature, fuel inlet pressure, knock detection, air/fuel ratio, and fuel mass flow. Electronically controlled waste gate turbocharger is also integrated. The ISL G engine is capable of operating on compressed or liquid natural gas (CNG, LNG). The ISL G can also operate on up to 100% biomethane – renewable natural gas made from biogas or landfill gas that has been upgraded to pipeline and vehicle fuel quality. Biomethane fuel is carbon dioxide (CO_2) neutral and using it as fuel reduces vehicle greenhouse gas emissions by up to 90% [8].

New CNG Powered Bus Safety Guidelines

Some relevant properties of natural gas like vehicle fuel include:

- Natural gas is invisible but odorized so its presence can be detected,
- CNG fuel systems store fuel at approximately 20 MPa, and as high as 26 MPa,
- Unlike gasoline vapors, natural gas and hydrogen are both lighter-than-air and in gaseous form at atmospheric conditions. This property allows these fuels to quickly rise and disperse in the unlikely event of leak,
- Both CNG and H_2 have an ignition temperature of around 480 ^oC to 650 ^oC – whereas gasoline is approximately 260 ^oC to 430 ^oC and diesel is less than 260 ^oC. This relatively high ignition temperature for CNG and H_2 is an additional safety feature of these fuels. To ensure a safe environment in the maintenance garage, the surface temperature of equipment that could contact a gas leak is usually limited to 400 ^oC [9], and
- Natural gas has a very selective and narrow range of flammability–that is, the mixture of gas in air that will support combustion (between 5% and 15% natural gas in air by volume–ratios outside of this range will not support combustion) [9].

The main safety concern regarding CNG vehicles in a repair facility is the possibility of fuel releases and their consequences. Natural gas leaks can be either fast or slow. A fast leak usually involves the release of a safety valve or complete severing of a fuel line. In the case of the safety valve, all the gas in the fuel tank will be vented to the atmosphere. Other major fuel releases can be controlled by closing appropriate valves. Slow releases are those caused by fuel escaping through a loose fitting or an abraded line or hose.

Both types of leaks can cause flammable mixtures to form in the work area. These flammable mixtures will disperse and over time will dissipate to safe levels. Workplace safety can be maintained by reducing fuel release volumes, keeping ignition sources away from areas where flammable mixtures may travel and using proper ventilation to control how long these flammable mixtures exist and where they will be present.

If a slow fuel release occurs, natural mixing of the released fuel with the surrounding air will cause **most** such mixtures to become too lean to support combustion. Methane's relatively narrow flammability range means that the diluting of the mixture occurs quickly and the only flammable mixture will be near the release site.

Fast releases have the potential to generate large clouds of fuel that can form flammable mixtures some distance from the release site. Since methane is lighter than air, released fuel will tend to rise from the release site. This contrasts with conventional fuels which puddle and form vapors that travel along the floor. In facilities where CNG vehicles are being serviced, ignition sources above the vehicles are of primary concern. These ignition sources can include electric equipment that generate sparks or high surface temperatures and open flame heaters. Ventilation systems should be designed to remove fuel from above vehicles or to promote mixing of the air in the space above the vehicles.

Figure 3 shows garage model for CNG vehicles with a flat roof and overhead lighting and infra-red heaters.

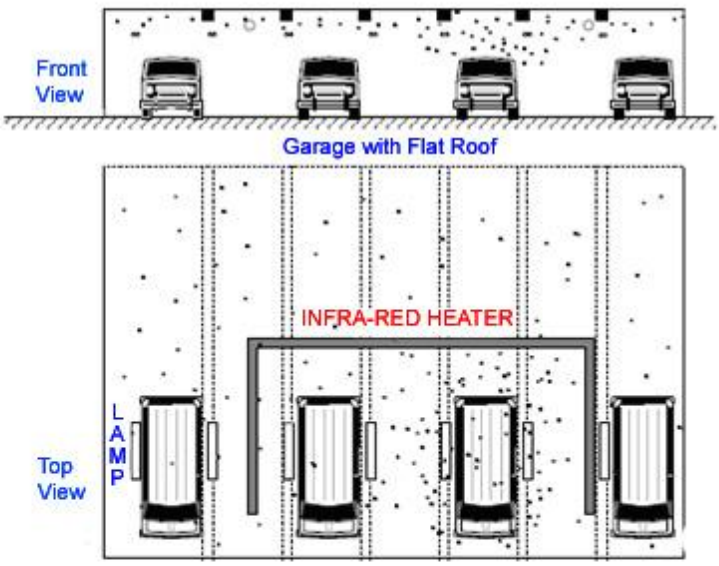

Figure 3: *Example of CNG Vehicle Fuel Release – Garage with Flat Roof*

A fast fuel release from one of the vehicles will cause a flammable cloud to travel across the ceiling. While the flammable cloud is rising it will pass over the lighting and the heating system. With the large flat ceiling, a flammable layer can form below the ceiling. Without proper ventilation removing fuel from near the ceiling, this layer can be ignited by a spark from the lighting or heating system.

According to previous analysis, lighter-than-air fuels have safety advantages, roofs and ceilings of these facilities must be designed without any unventilated “pockets” in the ceiling space that could trap gas. Ventilation systems for CNG – fuelled buses garages must be designed that typically provides between 5 and 6 Air Changes per Hour (ACH) (the requirement is for 425 L/min per 1 m^2 of ventilated area) [9]. There is no additional airflow requirement and cost, according to existing diesel facilities designed for a baseline ventilation rate of 4 to 6 ACH.

Used CNG tanks have been tested under a pressure of 30 MPa and for fire protection all cylinders fitted with Pressure Relief Devices (PRD), approved according to the relevant standard in connection with the cylinder type. Cylinders are equipped with electric shut-off valves to stop and open the CNG flow in fuel line. In the valve is integrated thermal switch that quickly respond to increasing temperatures (in 110 ^oC + - 3.5% according to ECE R 110). That is so called Temperature triggered Pressure Relief Device (TPRD) system that is integrated in the brass device, which are placed in the middle and at the end of the cylinders. Its working pressure is 26 MPa and this is also a protective device, pressure regulator that is thermal activated at the temperature of 110 ^oC in cylinders for CNG [3,7].

In figure 4 is shown one of the stickers on one of the mounted tanks. On the sticker there are useful data.

Figure 4: *The photo of gas tank sticker*

The cylinders have a maximum Service Life of 15 to 20 years [7] from the final manufacturing inspection date, depending on the number of cycles per year specified in the standard for the country where the cylinder is operated. The expiration date is specified on the label, fig. 4.

Than the Service Life is reached, the cylinders must be removed from service. If cylinders are filled more than [1000 x Service Life in years] before the expiration date is reached the cylinders must also be removed from service [7].

Cylinders require an external re-inspection for defects in the composite wrap at certain intervals after installation or upon reinstallation. Inspection shall always be in accordance with procedures outlined in ISO 19078, and/or also according to the relevant national standard of the country where the cylinder is operated. According to ECE R110 Rev. 2/Amend.4, for natural gas cylinders this inspection shall be performed at least every 48 months after the date the vehicle enters into service [7].

On the bus is also integrated system for measuring the concentration of methane with three sonde (placed in the engine compartment area, in the passengers saloon, and under the CNG tank cover on vehicle roof, (in accordance with the standard ECE R110), and central microprocessor with LCD display, which is placed in the driver working place [3].

ACTUAL TRENDS AND SECURITY OF GAS MARKET SUPPLY

Security of the natural gas market supply represents one of the major strategic and long term goals of a responsible energy policy in the gas field. Concerning gas, Serbia is one of the highly import depending countries of the region, which has suffered problems of a unilateral supply interruption several times in the past. In January 2009, during the gas crisis due to a well-known dispute between the Russian Federation and Ukraine, Serbian natural gas market supply has been totally deferred.

In order to prevent unforeseeable circumstances in terms of gas crisis or other accidental situations affecting the market supply, Public Enterprise Srbijagas has entered investments into construction of an underground storage near Banatski Dvor and intensified activities towards diversification of supply routes as well.

The prospect of the South Stream Project arouses interest in the European energy market. This project would make Serbia a transit country and could considerably contribute to the security of market supply not only in Serbia but also in the countries of South East and West Europe.

In the winter 2009/10 in the crisis time, with short-term measures was conducted follows [10]:

- Increase domestic production to 7 million cubic meters of gas per day,
- Increasing amounts of gas from underground storage Banatski Dvor on 1 million cubic meters gas per day,
- Security Storage 200 million cubic meters of gas in Hungary for Serbia, with delivery at the level of 2 million cubic meters of gas per day, in the winter period,
- Imports from Russia at the level of 10 million cubic meters of gas per day.

In this way, the first time provided the amount of natural gas from 13.7 million cubic meters a day, for the Serbian market and exports in transit for Bosnian Federation [10].

Long-term planning measures to ensure security of supply of natural gas market in Serbia, according to the Energy Development Strategy up to 2015 include the following objectives:

- Completion of construction of the second phase of the Banatski Dvor underground gas storage capacity to 5 million cubic meters of gas per day, and
- Connectivity infrastructure at the regional level, international project South Stream.

The first phase of Banatski Dvor underground gas storage is finalized, so it is with a capacity of 960,000 cubic meters. In the mid of december 2009, began gas production.

Based on the credit of the European Bank for Reconstruction and Development, plans to build underground gas storage in Itebej nearly Žitište, and connect with the regional gas network.

Potential project South Stream is a project of international gas pipeline from Russia that under the Black Sea to Bulgaria and further via two routes transported gas transit through several countries in Western Europe. For now, the supposed route through Serbia would be between Zaječar, Belgrade, and Subotica [10].

In addition to several signed protocols and agreements, 15 may 2009 in Sochi, in the presence of Prime Minister of the Russian Federation and Italy, representatives of gas companies from Russia, Italy, Bulgaria, Serbia, Greece signed an agreement to build an International Gas Pipeline South Stream. According to this decision, 17 November 2009 year, the Gazprom and Public Enterprise Srbijagas start joint venture company South Stream Serbia AD, with direction office in Bern, with previously defined main goals.

In the complete calculation Russian story is just special. Russian Federation produced 651 billion cubic meters of gas in 2007, which represents 41.3% of total produced energy of this state, of which for us interest Gazprom produced 549.6 billion cubic meters of gas in Russia. For 2010 year, the predicted 670, while for the 2030 year forecasts of gas production in Russia reached approximately 997 billion cubic meters [3,10].
The above strategy is in accordance with the assessment that in the period between 2020 and 2025 in the European market require an additional 200 billion cubic meters of gas per year.

REASONS FOR USE OF CNG BUSES IN KRAGUJEVAC

Today the world has about 9.56 million vehicles operated on natural gas (CNG bus total number is about 256,820) and over 14,500 filling stations. According to estimation, it comes up with lower CO_2 emissions by over 15 billion tons [11]. Since 1991 until today, the middle rate of growth of motor vehicles with CNG drive in the total fleet is 18% per year. If continue with this rate by 2020, in the world will be around 65 million gas vehicles, while natural gas as a fuel a day to change to 3.5 million barrels of oil [11].

By using buses with CNG drive, primarily to a large extent contribute to the preservation of health, as compared to conventional diesel vehicles have the following benefits [8,11]:

- For more than 50% lower emissions of Non Methane Hydrocarbons (NMHC),

- From 30% to 60% lower NO_X emissions,
- More than 80% of the less reactivity of the ozone layer,
- Reduced the emission of sulphur compounds in the products of combustion,
- Reduced the emission of fuel vapours during the fuel filling,
- For more than 90% reduction of Particulate Matter emissions,
- Eliminated black smoke from exhaust and carcinogenic particulate emissions, and
- For more than 15 dB below the level of external noise to the diesel vehicle.

Figure 5 shows a diagram of changes in prices of oil and gas. Price of gas is significantly lower than the price of a litre of diesel fuel, especially if the company has its own CNG fuel station [8].

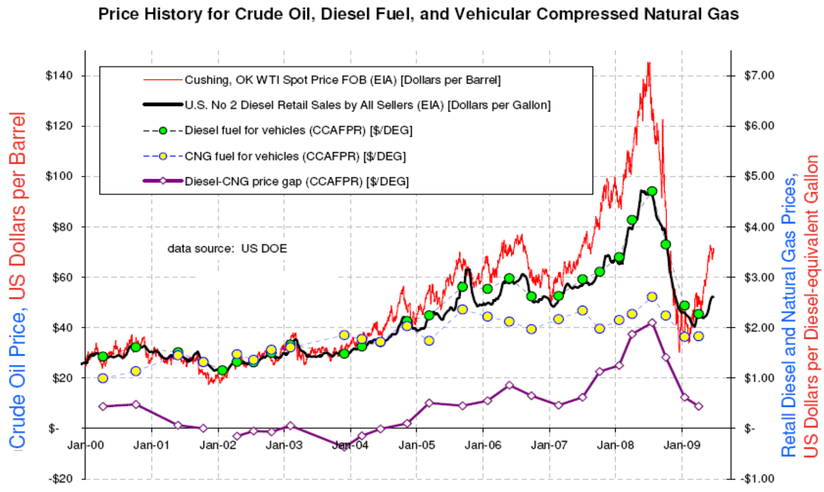

__Figure 5:__ Prices change for Crude Oil, Diesel and CNG

Costs of bus fleet conversions to natural gas technologies are concentrated in the initial capital investment. The purchase cost of a CNG bus can range from $25,000 to $50,000 more than a diesel bus. This price differential is expected to grow narrower as production volumes of CNG buses increase. Also, investment in natural gas fuelling infrastructure usually requires construction of a new fuelling facility. These fuelling facilities can be an investment in future natural gas vehicle fleets that will demand public refuelling stations. The Federal Transit Administration has estimated the cost of a fuelling station for a fleet of 200 buses at approximately $1.7 million. Finally, National Fire Protection Association (NFPA) codes may require modification of bus storage depots to incorporate gaseous fuel detection systems. Despite these added costs, operational and maintenance expenses tend to be lower than those for diesel-powered buses, including newer “clean diesel” models fitted with after-treatment devices. Natural gas-powered buses save money because they do not require frequent oil changes and regular cleaning and eventual replacement of new filters. Since natural gas engines are inherently cleaner burning and experience less engine wear, normal maintenance costs tend to remain low, but these costs have varied among transit.

Public transport in Kragujevac city operates with 50 buses with average 280 km/per day, every 315 working days per year. In this moment, actual situations with diesel and CNG buses are in the next [3,12]:

- Average diesel fuel consumption is about 40 L/100 km. With range of 280 km per day, this is about 112 L of Diesel fuel daily, for one bus. With 50 buses for Public Transport in Kragujevac final average diesel fuel consumption is 5,600 L of diesel fuel daily
- Average CNG consumption is about 33 kg/100 km, and this is about 4,620 kg of CNG daily
- Price for Euro Diesel fuel in this moment is about 1 €/L, whereat for CNG is about 0.41 €/kg on proper CNG filling stations, or 0.65 €/kg on liberal sale

Taking into account these parameters, we obtain the following conclusions about a **year operation** with a 50 buses operated on diesel fuel and CNG especially in Kragujevac city in 315 working days [3,12]:

- Total consumption in case of diesel drive is 1,764,000 L of diesel fuel or 1,764,000 €
- Total CNG consumptions is 1,455,300 kg of CNG or 596,673 € *(proper fuel station)*
- If transporter uses the public station, then the annual CNG cost is about 946,000 €

If we replace complete diesel buses fleet in Kragujevac city with new CNG buses, (our proposal project "KRAGUJ") total money savings is about 1,167,000 € yearly.

Costs for additional infrastructure and new CNG filling stations are about max 450,000 € for two stations with filling capacity of 1,800 m^3 of CNG an hour. Also should take into account the depreciation for buses.

Looking the complete calculation for the period from 5 to 10 years, according to contract validity for Passengers Public Transport, it is clear what the level of fuel savings is much major.

All this should add the second benefits, taking into account the reduction of vehicle noise at work on CNG, as well as reducing emissions of toxic and harmful combustion products in exhaust gases.

Another main purpose was to produce biogas for the city buses to reduce the local, regional and global emissions from the urban transport (some projects started in Kragujevac on this topic).

The city is located in the middle of an agricultural district on the plains central Serbia. The prerequisites for building a biogas plants were thus obvious. The manure from cattle and pigs in the area could be co-digested with abattoir waste and organic waste from other food industries.

The biogas plant has made it possible for the city of Kragujevac like example to decrease the CO_2 emissions from transport and also to decrease the local emissions of dust, sulphur and NO_X.

During the prototype bus exploitation was confirmed a better fuel economy with CNG, compared to diesel drive. Fuel cost per kilometre is lower for about two or three times with CNG, especially in situations than the transportation company has its properly CNG fuel station.

In the period from 26/05/2009 to 17/06/2009, during the exploitation in City Transportations Company Belgrade, CNG bus achieved an average gas consumption of 40.7 kg/100 km (on the city routes: 74, 94, 55 and 58). During the work on private line number 707 in Belgrade city, average gas consumption was 32.5 kg/100 km, while in the city of Kragujevac, during the exploitations on mixed regimes in all routes achieved an average gas consumptions of 33 kg/100 km. On the open road, while a drive at intercity conditions, average gas consumption measured was 22.4 kg/100 km. Also, it is important to note that the vehicle radius with one filling is around 560 km in the urban conditions [3,12].

According to possibilities of gearbox regime settings to work on economy mode, referred to the experience during the prototypes exploitations, was realised the vehicle adjustment-settings for further serial production. Also it`s very important that the CNG engine working temperature moving in the projected range, without deviations. Visual control of CNG fuel line is not significant changes, and all detectors are not registered methane presence in the measuring areas.

CONCLUSIONS

Use of Compressed Natural Gas – CNG as an alternative fuel is an effective, currently available way to help solve pressing environmental and fuel – resource problems.

The goal of the European Union is that alternative sources of energy will represent 20% of total consumption by 2020, confirming the strong political will amongst European countries to reduce global pollution and improve living standards.

Using the gas tanks composed of light material type 3 and projecting the CNG vehicle installations in accordance with regulations ECE R 110, on the buses achieved a remarkable progress in terms of safety, and also regarding to fuel economy and emissions lowering.

Mounting of original CNG engine reached the maximum effectiveness of the used fuel, the vehicle meets the environmental norms defunded with Enchanced Enviroment friendly Vehicle norm. Combination with automatic gearbox with six speeds have a positive effect on passengers and drive comfort, as well as on cost, because of effective using of engine power output.

By using buses with CNG drive, primarily to a large extent contribute to the preservation of health, as compared to conventional diesel vehicles have environmental benefits.

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