CNG INFLUENCE ON COMBUSTION IN AN AUTOMOTIVE DIESEL ENGINE FUELED IN DIESEL-GAS MODE

Silviu ROTARU[1], Constantin PANA[2], Niculae NEGURESCU[3], Alexandru CERNAT[4], Dinu FUIORESCU[5], Cristian Nikolaos NUȚU[6]

The paper presents some results regarding fueling an automotive diesel engine with compressed natural gas (CNG) in dual fuel mode; the purpose of the paper is to analyze the stability of combustion with help of the 5% and 90% of mass fraction burned (5% / 90%MFB) and of the coefficients of variability (COV) for in-cylinder maximum pressure; the use of CNG reduces the angle for 5% MFB with more than 10°CA (crankshaft angle) and extends with 2°CA combustion end of the entire quantity of fuel per cycle; the exhaust gases temperature grows with 7°C more than when engine runs with diesel only.

Keywords: diesel engine, GNC, combustion, diesel-gas, dual-fuel, combustion variability, mass fraction burned.

1. Introduction

Worldwide Automotive Industry is facing a reality that pushes internal combustion engines, especially diesel engine, out of the choices of propulsion utilized in the transport field. More and more countries set greater taxes on diesel engine propulsion whereas numerous cities introduce low emission zone (LEZ) that prohibit means of transportation that use internal combustion engine [1]. Moreover, a greater number of states sign international agreements to reduce greenhouse gases (GHG). For instance, „The Paris Agreement" has been signed by 187 states (12th December 2015) that agreed on implementing ambitious strategies to combat climate change; the developing countries would receive great support to apply more environment friendly industrial processes [2]. There are many experts that posted prediction about the moment when the electric propulsion will represent an important percentage of all means; for instance, Bloomberg New Energy Finance

[1] PhD-Student, Dept. of Thermotechnics, Engines, Thermical and Frigorific Equipment, University POLITEHNICA of Bucharest, Romania, e-mail: Silviu_r.rotaru@yahoo.com
[2] University POLITEHNICA of Bucharest, Romania, e-mail: constantinpana@yahoo.com
[3] University POLITEHNICA of Bucharest, Romania, e-mail: niculae_negurescu@yahoo.com
[4] University POLITEHNICA of Bucharest, Romania, e-mail: cernatalex@yahoo.com
[5] University POLITEHNICA of Bucharest, Romania, e-mail: difuiore@yahoo.com
[6] University POLITEHNICA of Bucharest, Romania, e-mail: cristi_cmt@yahoo.com

(BNEF) annual forecast states that around 50% of passenger cars and 32% of world total fleet will have electric propulsion by 2040 [3]; on the other hand, oil industry magnates have less optimistic opinions that foresee less than 13% worldwide transportation using electric propulsion [4]. By any means automotive constructors need to have a say in this process and come on the market with less expensive alternatives to full electric propulsion: mild hybrid, standard hybrid, plugged-in hybrid [5] [6]. Meanwhile e great amount of research is focused on finding alternative cleaner fuels for internal combustion engines (liquid petroleum gas, CNG, butanol, hydrogen, animal fat etc.). At the fueling a diesel engine with preheated animal fat and butanol there are recorded decreases of nitrous oxides (NO_x) and smoke emissions and of indicated specific energetic consumption (ISEC) [7]. There are benefic aspects when a diesel engine is fueled with liquid petroleum gas (LPG); for instance, in [8] there are recorded lower NO_x and smoke emissions; also, the brake specific energetic consumption (BSEC) is lowered; as far as the combustion is concerned, there are registered higher in-cylinder pressure and maximum pressure rise rate. There are improvements of engine's operate at the fueling with CNG: the NO_x, carbon dioxide (CO_2) and smoke emissions and BSEC drops, and the in-cylinder pressure and pressure rise rate are higher [9]. The diesel engine fueled in dual mode with animal fat as alternative fuel has some advantages like the drop of NO_x and smoke emissions; the brake specific energetic consumption drops [10]. In [11] is presented the effect of CNG through the increase of ignition delay and lower exhaust gases temperature at high engine load and speed. In [12] is also stated the increase of the ignition delay and the increase of COV for 5% MFB and for in-cylinder maximum pressure.

This paper analyzes the 5% and 90% MFB duration, combustion phases, exhaust gases temperature; coefficient of variance (COV) for maximum pressure and for 5% and 90% MFB of a 1.5liter diesel engine fueled in diesel - gas mode (with CNG as alternative fuel) at 2000 rpm and 70% load. The start of combustion (SOC) is calculated using 5% MFB; as the studied literature states, the start of combustion is expected to take place latter in dual-fuel mode. The influence of CNG use over the stability of SOC is analyzed using the coefficient of variance. Using CNG as an alternative fuel is expected to determine a growth of the coefficient of variance for 5% MFB. Using CNG is also expected cu determine a growth of COV for in-cylinder max pressure and for the 90% MFB. In dual fuel operating mode, the combustion is expected to be prolonged and, as an effect, the temperature of exhaust gases will grow.

2. Research methodology

The used test bed is presented in the figure 1. The diesel 1,5liter Renault engine (1) is loaded by an eddy current dyno (2) controlled by electronic control unit (10); this unit also controls the accelerator pedal (9) through electronic actuator (8). The alternative fueling system has the tank (13) that stores compressed natural gas at 225 bar, a manual shutoff valve (14), a two stage pressure reducer (15) that has incorporate an emergency electrical shutoff valve, a manometer (16), a masic flowmeter for gases (17), a flame arrestor (18) and the injector (19); the injector is piloted by a opened type electronic controlled unit (7) connected to computer 20 so that the parameters are always monitored and adjusted. The conventional fueling system is composed from the tank (20), masic liquid flow meter (21), common rail and injectors (27). The in-cylinder pressure is measured with a piezo-electrical transducer (11) mounted in the thermo plunger; the transducer is connected to computer (4) and to oscilloscope (5). The air intake was monitored with a volumic air flow meter (12). To monitor the exhaust gases there were used a gas analyzer (22) and an opacimeter (23).

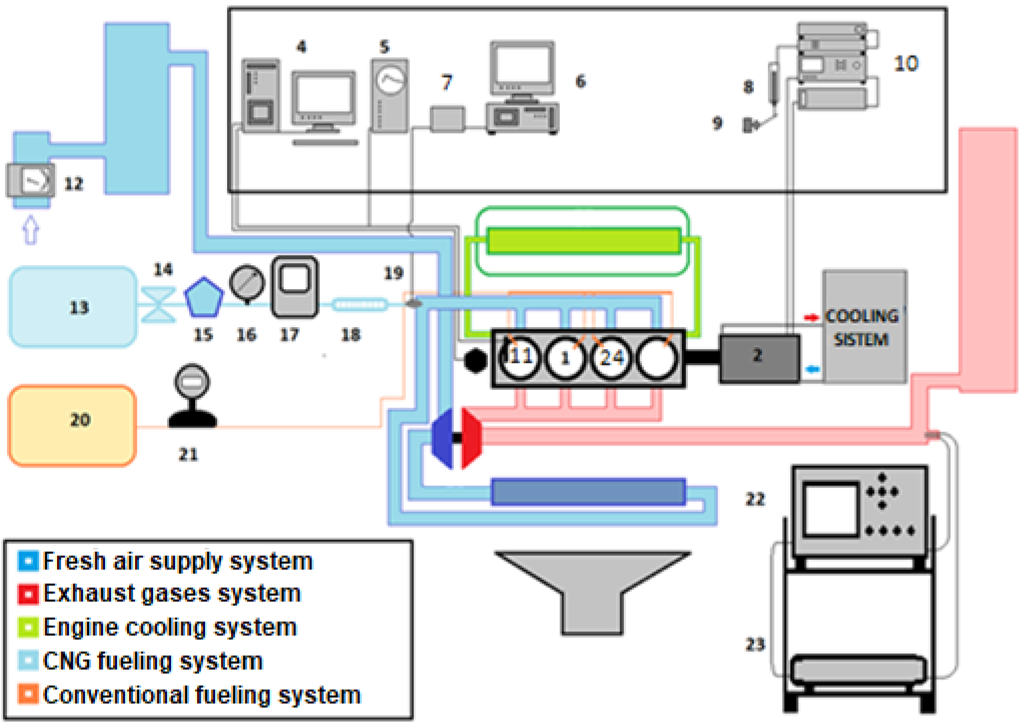

Fig. 1. The experimental test bed

In order to have a very accurate set of data, the entire equipment was calibrated. The benchmark data was recorded when the engine was fueled only with diesel fuel. The operate engine regimen was setup at the speed 2000 rpm and at 70% load. The alternative fuel, CNG (compressed natural gas), is pressured at over 200 bar in a special designed tank (the tank is tested at pressure over 300 bar); the pressure is dropped at fueling pressure of 3 bar by a two stages pressure reducer; the pressure reducer also has an electric shutoff valve that cuts the flow of gas in case of emergencies. Manometer and masic flow meter are assembled in the gas fueling system. CNG is injected into the air manifold, after the turbocharger, at a superior fueling pressure over the supercharging pressure. The testing procedure follows three main steps: 1- the engine is brought to 2000 rpm and the load is set at 70% solely on diesel fuel; 2 - the quantity of diesel fuel injected per cycle is diminished; 3 - CNG is then injected into the intake manifold until the engine power reaches standard value at 70% load and 2000 rpm. The diesel pilot injection timing is kept constant for all energetic substitution ratios.

The energetic substitution ratio (x_c) helps to determine the quantity of CNG which will be injected in the intake manifold and is calculated according to the formula:

$$x_c = \frac{Ch_{CNG}*LHV_{CNG}}{Ch_{CNG}*LHV_{CNG}+Ch_{Diesel}*LHV_{iDiesel}} * 100 \qquad (1)$$

the $Ch_{CNG/diesel}$ represents the fuel consumption in kg/h, and $LHV_{CNG/diesel}$ is lower heating value for CNG/diesel.

3. Results

There have been recorded 200 consecutive combustion cycles. As reference were recorded data of engine running only on diesel fuel. As seen in the articles studied [11], [12], [13], [14], the beginning of the combustion process is longer compared to standard diesel fueling, thus the 5% mass fraction burned was analyzed. In our case, as the substitution ratio grows, the 5% MFB is reached sooner per cycle (figure 2); in some individual cycles (figure 3), 5% MFB is reached 13 °CA sooner than in conventional fueling mode.

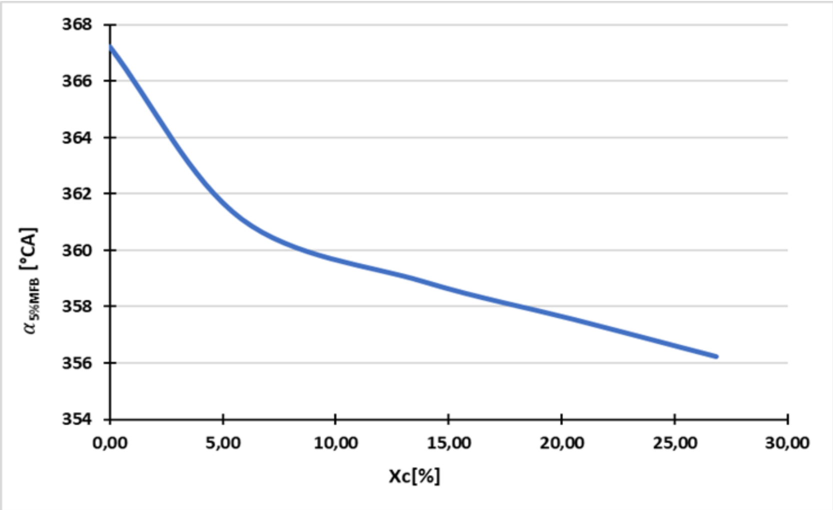

Fig. 2. Crankshaft position correspondent to 5% mass fraction burned as function of substitution ratio.

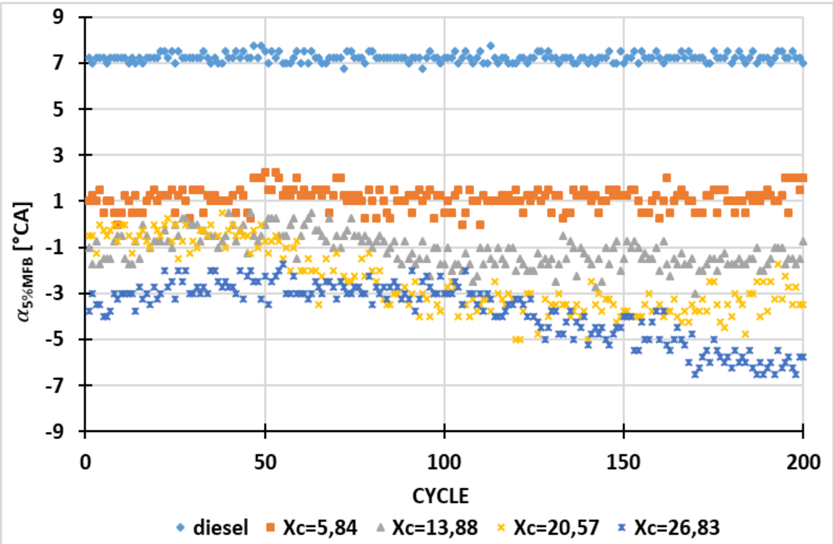

Fig. 3. Crankshaft position correspondent to 5% of masic fraction burned for 200 cycles.

From the figure 3 can be observed that at the engine fueling only diesel fuel the 5% MFB crankshaft angle is about ~7.2 °CA after top dead center (TDC) (less than 1 °CA dispersion). For energetic substitution ratios higher than 13.88%, the 5% MFB crankshaft angle is occurs before TDC; at this x_c the dispersion for $\alpha_{5\% MFB}$ is also the highest (more than 5 °CA;) as the energetic substitution ratio reaches highest, the dispersion of this parameter drops.

A. Jamrozik [15] states that the start of combustion represents the moment (expressed in crankshaft position) until 10% of total heat has been released. A. Yuosefi [16] refers at SOC as being the moment when 10% of mass fraction has burned. As the author states, ignition delay is the duration between start of injection and start of combustion; SOC will be considered the moment after the end ignition delay (the moment when a specific percentage of mass fraction has burned). Befrui et all [17] presents the ignition delay as the period from 0 to 5% MFB and the main combustion phase from 5 to 90% MFB; this indicates that ignition has taken place when 5% of mass fraction has burned. Atkinson [18] refers to SOC as discernable combustion occurrence; he presents two ways to determine this moment: first method states that SOC is the moment when the pressure history deviates from the theoretical motoring curve; the second choice iterates the idea that SOC is when 1% to 10% of mass fraction has burned. He chooses the first method. Lyn W. T. and Valdanis [19] use a percentage of mass fraction burned to determine the start of combustion. In this paper the start of combustion will be identified as being the moment in CA° when 5% of mass fraction has burned.

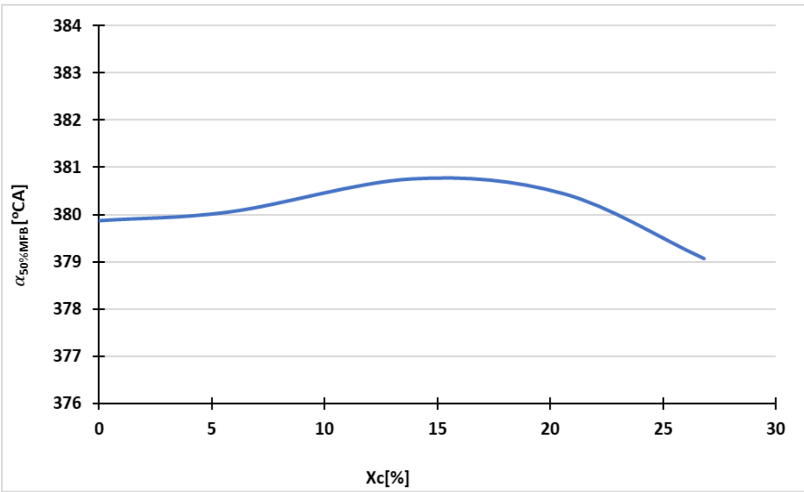

Fig. 4. Crankshaft position correspondent to 50% mass fraction burned as function of substitution ratio.

The premixed combustion phase is very affected when CNG is used; when only diesel is used, this phase covers around 12 °CA. At the highest substitution ratio this phase takes more than 20 °CA. As stated in [20] this phase is several times faster than that of spark ignition engine; when CNG is added, the fuel combustion tends to get slower because of high octane number of CNG; as the energetic substitution ratio gets higher the combustion has similarities with that of spark ignition engines.

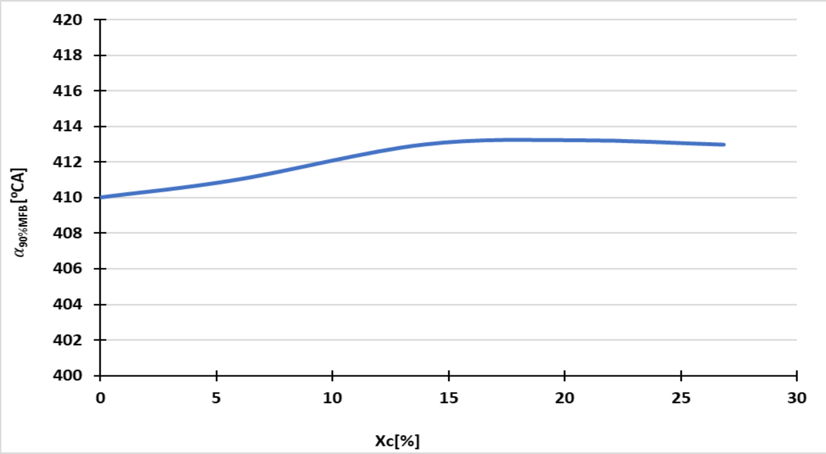

Fig. 5. Crankshaft position correspondent to 90% mass fraction burned as function of substitution ratio.

The combustion of the 90% of mass fraction burned was also recorded and compared with the reference – engine running only on diesel fuel. It can be observed in figure 4 that crankshaft's position corresponding to 50% mass fraction burned ends about same time independently of the substitution ratio (figure 5), whereas crankshaft's position corresponding to 90% MFB has extended from 49 °CA to 51.5 °CA after TDC. Taking these facts into consideration, one can conclude that the moderate combustion phase is very little influenced by using GNC as an alternative fuel.

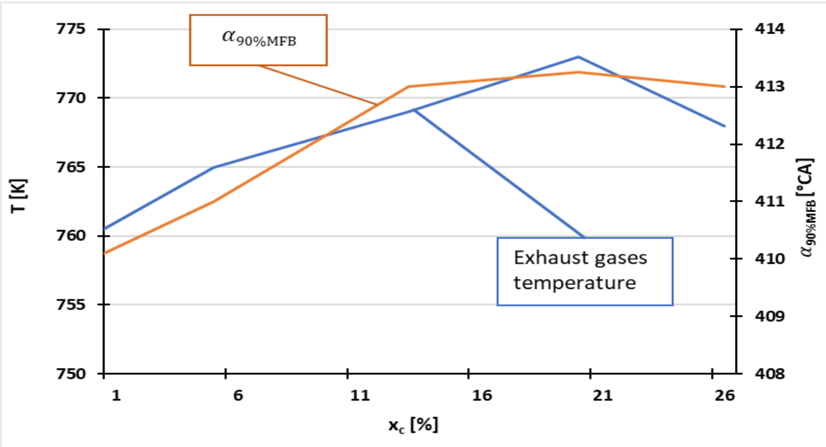

Fig. 6. Crankshaft position correspondent to 90% mass fraction burned and exhaust gases temperature as function of substitution ratio.

To support the idea that replacing a percentage of diesel fuel with CNG has as effect the extending of the moderate combustion phase, the exhaust gases temperature data was recorded. As can be seen in the figure 6 temperature of exhaust gases rises with 13°C when x_c reaches 21%. In this figure it can be observed how the temperature line keeps the same trend as the allure of the line for crankshaft position for 90% of mass fraction burned has.

To determine the stability of combustion the coefficient of variation (COV) were calculated using recordings of in-cylinder pressure data, 5% and 90% MFB. As presented in [21] COV is a measure of relative variability and it is given by formula:

$$x = \frac{\sigma_x}{x_{avg}} * 100 \tag{2}$$

where σ_x is standard deviation and, according to [20], is given by formula (3); x_{avg} is the mean of all analyzed data (in our case all maximum in-cylinder pressure data recorded).

$$\sigma_x = \sqrt{\frac{\sum_{i=1}^{n}(x_i - x_{avg})^2}{n-1}} \tag{3}$$

where n is number of analyzed points, x_i is the value of i^{th} point, and x_{avg} is the mean value of recorded data set [22].

As seen in the figure 7, the growth of energetic substitution ration determines a higher COV for in-cylinder maximum pressure. This feature reveals that the more CNG is injected into the cylinder the more instable the combustion is. On the other hand, the combustion evolution depends very much on the percentage of oxygen left into cylinder and on the proximity of diesel droplets to it. Immediately after the pilot ignites, the preformed charges (air-CNG-diesel fuel) will burn. The probability that the oxygen left might be separated of diesel fuel by CO_2 or other exhaust gases is very high and so only a small part of diesel will be oxidized and have a real contribution to combustion. This barrier might have a huge impact on combustion's stability.

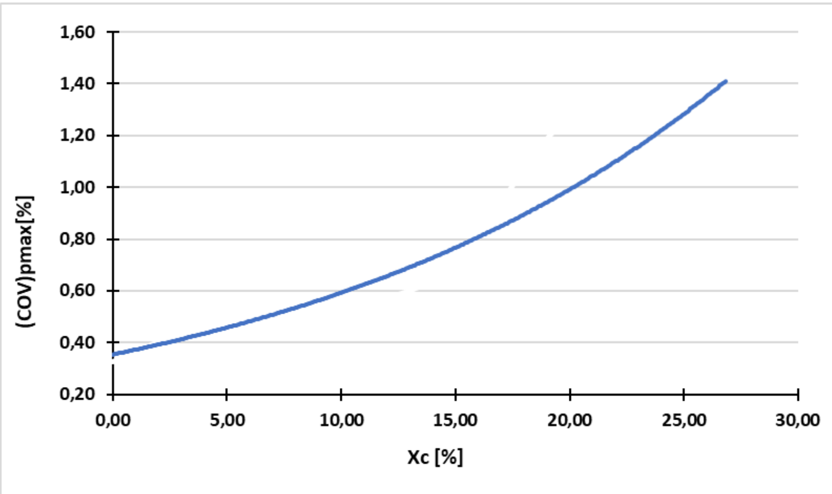

Fig. 7. Coefficient of variability for in-cylinder maximum pressure as function of substitution ratio.

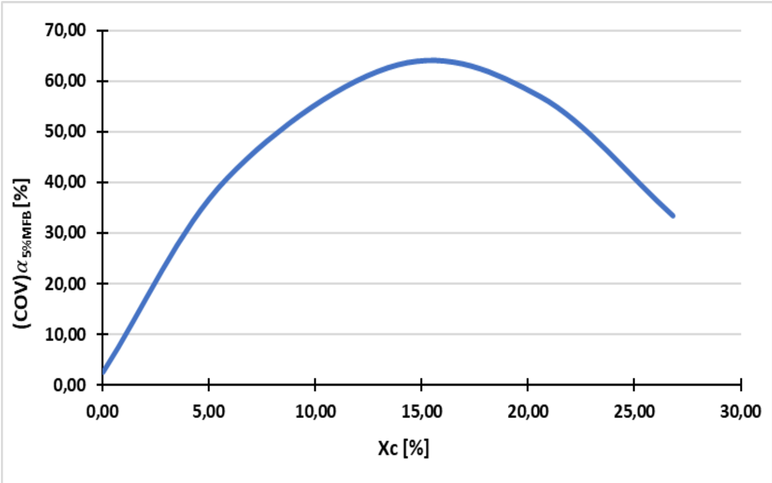

Fig. 8. Coefficient of variability of crankshaft position for 5% MFB as function of energetic substitution ratio.

The coefficient of variability for $\alpha_{5\%MFB}$ as function of energetic substitution ratio is presented in figure 8. As presented in [21], a set of data can be considered stable and the average (mean) value of its elements is representative if the coefficient of variability of the set is less than 10%. In our case the COV for $\alpha_{5\%MFB}$ reaches almost 65%; as seen in figure 3, variation intervals for x_c exceeding 13%, in some cases, are 2 or 3 times higher than the mean value of all data for the 200 consecutive recorded cycles; using CNG as alternative fuel for an automotive diesel engine has a negative effect over stability of the start of combustion; this makes prediction for $\alpha_{5\%MFB}$ when engine runs in dual-fuel very difficult.

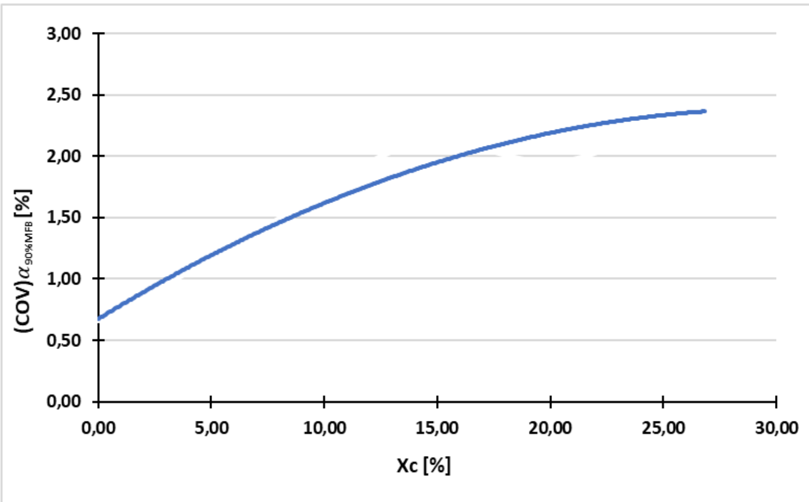

Fig. 9. Coefficient of variability for crankshaft position for 90% MFB as function of substitution ratio.

The diagram of COV for 90% MFB as a function of x_c is represented in the figure 9; it is increasing continuously with the growth of the energetic substitution ratio. This coefficient does not exceed 3%, meaning that using CNG as an alternative fuel for diesel engine has little impact on the stability of moderate combustion phase.

4. Conclusions

The analysis of the recorded and processed data reveals several facts that comply with existing related scientific literature:

- 5% MFB is reached with 10 °CA earlier comparative with the fueled engine with only diesel
- 50% MFB ends around 380 °CA for 0-7% energetic substitution ratios and increases until 380.7 °CA for 7-22% energetic substitution ratios and then decreases for others energetic substitution ratios, but the premixed combustion phase takes up to 14 °CA longer
- 90% MFB ends with 1–2 °CA later for high energetic substitution ratios
- Combustion of the entire fuel quantity introduced in cylinder per cycle is with 12 °CA longer comparative with diesel fuel, exhaust gases temperature being with 7°C higher at the greatest energetic substitution ratio
- COV for in-cylinder maximum pressure gets higher as the energetic substitution ratios grows, it never exceeds 1.5%, so it can be said that using CNG as alternative fuel has little effect on variation of in-cylinder maximum pressure.
- COV for 5% MFB shows that adding CNG to the cyclic dose of fuel has a great influence on stability of combustion beginning: if, for conventional fueling mode, SOC variation interval is only one 1°CA, in dual fuel mode it reaches 13 °CA; $COV_{5\%\ MFB}$ reaches highest level for energetic substitution coefficient at 13.8% (60%); as x_c grows to 26% the coefficient of variation gets lower (33%) but still much higher than the limit stated in [21].
- COV for 90% MFB increases continuous with the growth of the energetic substitution ratio, being smaller than 3%. The limit set in [21] makes it safe to say that using CNG as alternative fuel has little impact on stability of diffusive combustion.

All initially set objectives were addressed throughout this paper: as expected, using CNG as alternative fuel has a negative influence over the stability of 5% and 90% MFB and over in-cylinder maximum pressure. While stability of 5% MFB is severely affected (COV reaches 65%), COV for 90% MFB and in-cylinder maximum pressure are less than 10%. As expected, the combustion in dual fuel mode had been with around 2 °CA longer than in conventional operating mode.

As an effect the temperature of exhaust gases had grown with 13 K. Opposite to the expected result the 5% MFB takes place sooner in dual fuel mode.

Acknowledgements:

This work has been funded by the European Social Fund from the Sectoral Operational Programme Human Capital 2014-2020, through the Financial Agreement with the title "Scholarships for entrepreneurial education among doctoral students and postdoctoral researchers (Be Antreprenor!)", Contract no. 51680/09.07.2019 - SMIS code: 124539.

The authors address special thanks to AVL GmbH Graz Austria for providing the necessary equipment. Gratitude is also presented to Mr. Ovidiu Pavelescu for providing the CNG fueling system.

This work was supported by a grant of the Romanian Ministry of Research and Innovation, CCCDI – UEFISCDI, project number PN-III-P1-1.2-PCCDI-2017-0404 / 31PCCDI/2018, within PNCDI III.

R E F E R E N C E S

[1]. https://dieselinformation.aecc.eu/ ”WHERE IN EUROPE CAN I DRIVE MY DIESEL CAR?”

[2]. https://unfccc.int/ ”The Paris Agreement”

[3]. https://about.bnef.com/ ” BNEF’s annual forecast of electric vehicles, shared mobility and road transport to 2040.”

[4]. https://qz.com/ ” Researchers have no idea when electric cars are going to take over”

[5]. https://www.delphi.com/ ” A shrewd solution to hybrid power shortfalls.”

[6]. https://audi-encounter.com/ ” Mild-hybrid technology”

[7]. *Cernat Alexandru, Pană Constantin, Negurescu Nicolae, Nuțu Cristian, Nicolici Adrian,* Butanol Effects on the Fueled Diesel Engine Operation with Preheated Diesel Fuel-Animal Fat Blend AMMA 2018, PAE, pp. 609-616, 2019

[8]. *Nemoianu Liviu, Pană Constantin, Negurescu Nicolae, Cernat Alexandru, Fuiorescu Dinu, Nutu Cristian,* LPG as Alternative Fuel for Clean Automotive Diesel Engine, AMMA 2018, PAE, pp.617-6124,2019

[9]. *Silviu Rotaru, Constantin Pană, Nicolae Negurescu, Gheorghe Lazaroiu, Alexandru Cernat, Dinu Fuiorescu, Cristian Nikolaos Nuțu,*” Researches regarding the CNG use at an automotive diesel engine”, SMAT 2019 Book of Abstracts, ISBN 978-606-14-1548-9

[10]. *A. Nicolici, C. Pană, N. Negurescu, A.Cernat, C. Nuț*u, The use of animal fat in diesel fueled engine, IOP Conf. Series: Materials Science and Engineering 444 (2018) 072003 doi:10.1088/1757-899X/444/7/072003

[11]. *M. Mbarawa, B.E. Milton,* An Examination of the Maximum Possible Natural Gas Substitution for Diesel Fuel in a Direct Injected Diesel Engine, R & D Journal, 2005, 21 (1) incorporated into "The SA Mechanical Engineer"

[12]. *Wan Nurdiyana Wan Mansor*, Dual Fuel Engine Combustion and Emissions – An Experimental Investigation Coupled with Computer Simulation, work submitted for Degree of Doctor of Philosophy Colorado State University Fort Collins, Colorado Fall 2014

[13]. *Lijiang Wei, Peng Geng,* A review on natural gas/diesel dual fuel combustion, emissions and performance, Fuel Processing Technology **142** (2016) 264–278

[14]. *Liu Shenghua*, Zhou Longbao,Wang Ziyan, Ren Jiang,* - Combustion characteristics of compressed natural gas/diesel dual-fuel turbocharged compressed ignition engine, Proc. InstnMech. Engrs, **217,** Part D: J. Automobile Engineering

[15] *A. Jamrozik, W. Tutak and K. Grab-Rogalinski* An Experimental Study on the Performance and Emission of the diesel/CNG Dual-Fuel Combustion Mode in a Stationary CI Engine, Energies 2019, 12, 3857

[16] *A. Yousefi, M. Birouk, B. Lawler*, Performance and emissions of a dual-fuel pilot diesel ignition engine operating on various premixed fuels, ECM, 106, 2015, 322-336

[17] *B. A. Befrui, W. Brandsttter, and H. Kratochwill,* Calculations of Inhomogenous-Charge Combustion in Swirl Assisted Lean Burn Engine SAE Paper 910266 (1991)

[18] *C. M. Atkinson, G. J. Thompson, M. L. Traver, N. Clark*, In-Cylinder Combustion Pressure Characteristics of Fischer-Tropsch and Conventional Diesel Fuels in a Heavy Duty CI Engine." SAE Transactions, vol. 108, 1999, pp. 813–836. JSTOR.

[19] *W.T. Lyn, E. Valdmanis,* The effect of physical factors on ignition delay, SAE No. 680102, 1968.

[20]. *Gunwald Berthlod* Teoria, calculul si constructia motoarelor pentru autovehicule rutiere, Bucharest 1980 (Theory, calculation and construction of engines for road vehicles).

[21]. https://www.statisticshowto.datasciencecentral.com/probability-and-statistics/how-to-find-a-coefficient-of-variation/

[22]. https://www.investopedia.com/terms/s/standarddeviation.asp