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INVESTIGATION OF COMBUSTION PROCESS IN DUAL FUEL DIESEL ENGINE

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Abstract

In this paper are presented results of dual fuel diesel engine bench testing carried out by author. The engine was fuelled parallel with two fuels. The second, additional fuel was compressed natural gas (CNG). CNG was injected into engine inlet system before inlet to engine turbocharger. There was installed a pressure transducer in one of engine cylinders and cylinder pressure was registered at the time of testing. There is discussed place and way of pressure sensor installation and impact on quality of indicating signals of the channel connecting the transducer with combustion chamber. The carried out investigations included measurement of emissions and engine indicating completed with the heat release characteristics. There was considered influence of CNG on CO, NOx, THC and NMHC emissions, especially how pollutant emissions depend on the share of methane in fuel. The effect of using natural gas as an additional fuel injected to regular CI engine on heat release rate was investigated, as well as its rate and duration. Discussed the differences between combustion of methane and diesel oil and ways of spreading flames into air-fuel mixture for these two mixed fuels The processes of combustion of diesel oil and methane are totally different. The study searches for the limits of methane content due to knocking combustion. Additive of methane to the diesel fuel is a reason of retarded heat release, decrease of engine efficiency, greater fuel consumption and changes in emissions corresponding to the lag of self-ignition.

Keywords: *road transport, combustion engines, air pollution, environmental protection, alternative fuels, combustion*

1. Introduction

In recent years, dual fuel diesel engines has become relatively popular as an effect of preferences made by the governments of several European countries allowing users of these types of engine for reduction of vehicle operating costs, or even to allow to entry so modified vehicles to ‘green’ zones, where traffic was forbidden for vehicles which does not meet the relevant requirements [1-4].

Around this type of adapted engines were accumulated many myths disseminated mainly by the distributors of gaseous installation, promoting dual fuel systems as a way to improve environment and giving the solution to limit pollutant emissions.

The objective of reported work was to gather knowledge on the combustion process and heat release in a modern compression ignition (CI) engine, adapted to dual fuel feeding through applying CNG installation supplying additional gaseous fuel to the engine intake system [5, 6, 9-11].

2. Tested engine

The tested engine (Fig. 1) was a direct injection compression ignition engine designed for light duty truck. Its essential data are shown in table 1. The engine was adapted to dual fuel feeding throughout installing an injector of natural gas in inlet tube and injecting natural gas into engine intake system. The engine was indicated with using AVL indicating system. Pressure transducer was installed only in one cylinder.

Tab. 1. Essential data of the tested engine

Number of cylinder:	4
Capacity [cm^3]:	2637
Power output [kW]:	85
Rated power speed [rpm]:	3600
Maximal torque [Nm]:	250
Maximal torque speed [rpm]	2000-2600

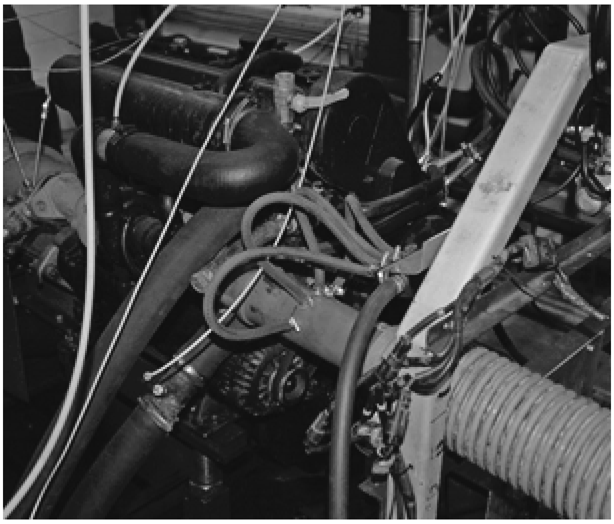

Fig. 1. CNG system installed on tested engine

Preparing of the engine to the testing comprised installing on crankshaft front end the encoder to register crankshaft angel of revolution and mounting in the glow plug completely special adapter containing pressure transducer. Location of the pressure transducer in the combustion chamber is relatively important part of the engine indicating process. The channel connecting pressure sensor with combustion chamber cannot be too narrow because it will be generate acoustic wave interfering measured signal. In such case, pressure sensor will register not only cylinder pressure itself but also stationary acoustic wave interfering with pressure signal giving as in effect false signal.

The tested engine was adapted to dual fuel supply. For this purpose, into its intake pipe at the inlet to the turbocharger was installed CNG injector taken from four-cylinder engine. Its opening time and injection frequency was controlled by an external controller. Every of four injector outlet pipes were connected to the engine inlet pipe between air filter and turbocharger.

3. Engine testing

The addition of a methane to air entering into cylinder influences on the ignition in the combustion chamber and combustion rate. In the compression ignition, engine combustion process is initiated by a self-ignition of diesel fuel injected into combustion chamber. The higher fuel cetane number is the easier self-ignition. Methane injected to the fresh charge has high octane number and low cetane number; therefore, it needs external source of ignition. This source can be pilot dose of diesel fuel.

In the dual fuel engine with increase of natural gas, content in a cylinder the share of diesel oil decreases. It leads to a limitation of the number of points where self-ignition take place, and from where the flame gradually can spread in air-fuel mixture filling the cylinder. This limitation can slow down the heat release rate, particularly in its initial phase. Greater ignition lag usually leads to harder engine work caused by injection a greater part of fuel during the lag. In effect, after late ignition heat release is more intensive.

The front of flame in a dual fuel engine has a dual character. Injected diesel oil is burnt after evaporation and mixing fuel droplets with air, just as in a regular CI engine. On the other hand, natural gas filling combustion chamber is burning with a visible front of flame that spreads in a similar manner like in spark ignition (SI) engines. We meet here two different models of combustion, which are in superposition. This may influence emissions of pollutants and engine parameters [7].

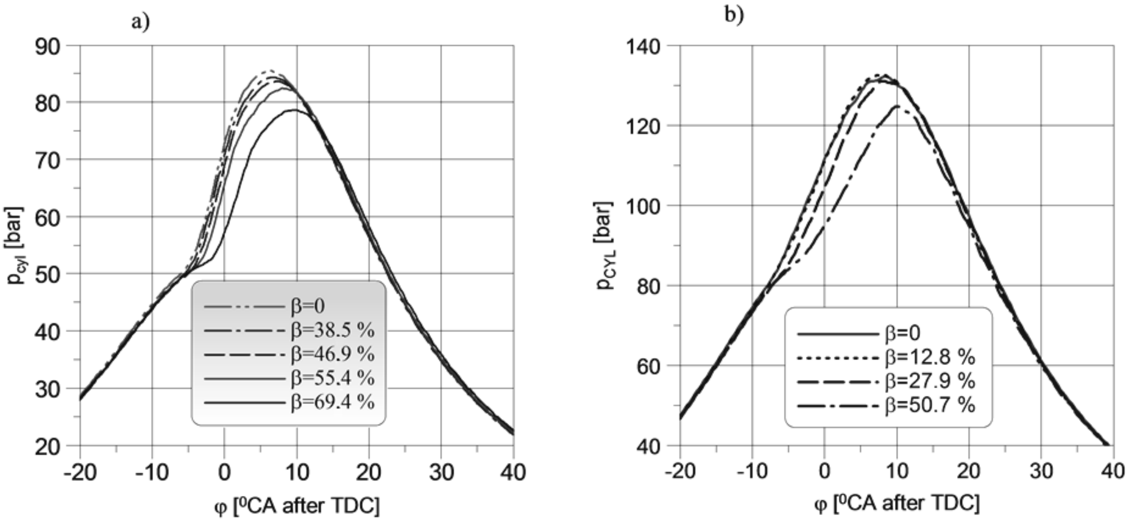

Fig. 2. Cylinder pressure versus crank angle for various mixture of CNG and diesel oil; a) n=1500 rpm, T=100 Nm; b) n=2500 rpm, T=200 Nm

Additionally, picture of combustion inside cylinder is complicated because in CI engine we can distinguish two phases of combustion: premixed (kinetic) combustion and diffusive combustion. In the first phase, the heat release rate depends only on burning speed because in this kind of combustion evaporated fuel is mixed with air and immediately ready to be burnt. In the phase of diffusive combustion heat, release rate depends on diffusion process, which supplies evaporated fuel into the front of flame. In this process combustion rate is equal to the rate expressing how fast fuel is prepared to combustion (evaporating and mixing with air) and how fast is transported to the flame.

Since methane has almost two times lower laminar burning speed compared to diesel oil, initial stage of the flame development as an effect of the injection of diesel oil pilot dosage, develops much more slowly than it would be for pure diesel oil. When size of flame is similar to the size of turbulent eddies in the combustion chamber, and then the combustion speed is accelerated by the turbulence. This phenomenon can be an explanation of slowing down of the heat release rate due to methane addition to the fuel.

On Fig. 2 is shown cylinder pressure in tested engine as a function of crankshaft angle (CA). These drawings shows that regardless of engine speed and engine loading, with the increase of methane weight content factor (β) in the fuel, maximum cylinder pressure is getting smaller. These differences of pressure occur in the CA range from app. 7° CA before TDC, where combustion begins, up to app. 13-20° CA after TDC (depends on engine loading), where expansion curves practically coincide each other regardless of the methane content in the fuel.

On Fig. 3, the change of specific emissions of selected exhaust gas components in a function of methane share (β) is shown. It can be concluded that with the increase methane share in a fuel, emissions of nitrogen oxides (NO_x), non-methane hydrocarbons (NMHC) and carbon dioxide (CO_2) decrease. However, emission of carbon monoxide (CO) and total unburned hydrocarbons (THC) increased.

Increase of THC emissions is caused by escape of fresh charge containing methane at the time of valve overlapping. However, in tested engine THC emissions consist mainly of nontoxic methane and share of toxic non-methane hydrocarbons (NMHC) decreases with an increase of methane share in the fuel. It is positive effect. Moreover, decrease of NMHC emissions can be asses as 10% for increase of methane share in fuel by every 10%.

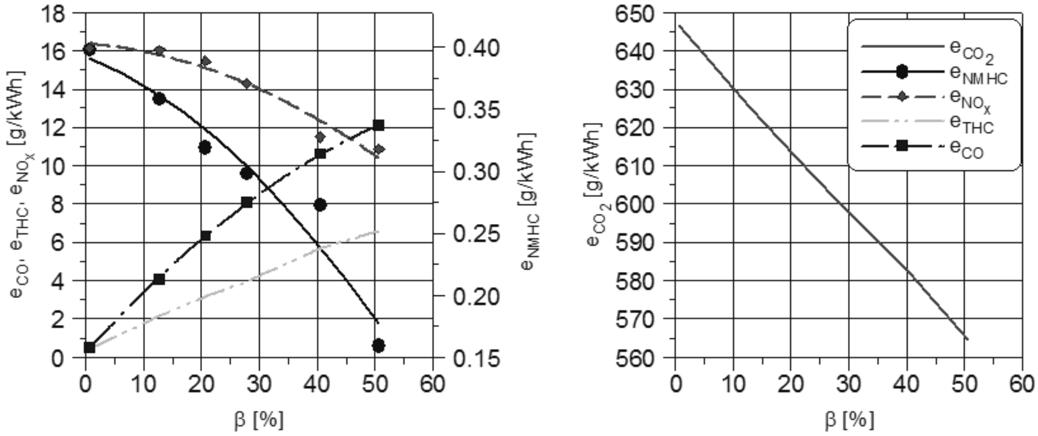

Fig. 3. Specific emissions of pollutants versus metane share in fual (β); n=2500 rpm, T=200 Nm

The change of NO_x emissions is usually affected by two factors: air-fuel ratio (oxygen availability) and the temperature at which the oxidation reaction of nitrogen takes place. In the case of dual fuel engine, methane added to diesel oil enriches air-fuel mixture in the cylinder and should rather cause an increase of NO_x emissions. Observing decrease of NO_x emissions we can assume that the result of applying methane to the engine causes an effect of heat release shifting similar to the effect of injection retarding. Retarded injection reduces the maximum cylinder temperature and causes decrease of NO_x emissions.

One more additional factor influences lower NO_x emission. This is exhaust gas composition. Methane has greater hydrogen weight factor than diesel oil, and exhaust gas of the dual fuel engine contains more water and less CO_2 compared to CI engine. Specific heat of H_2O is more than two times bigger than specific heat of CO_2. It results in a lower temperature in cylinder and gives effect similar to exhaust recirculation reducing NO_x emissions.

On average, the 10% methane additive to the fuel causes decrease of NO_x emission by 6% and CO_2 by 3%. The greatest impact on emissions was observed for carbon monoxide. The 50% methane additive results in increase of CO specific emissions from 0.5 g/kWh to 12 g/kWh, what is over 20 times more. The reason of this can be enrichment of the air-fuel mixture.

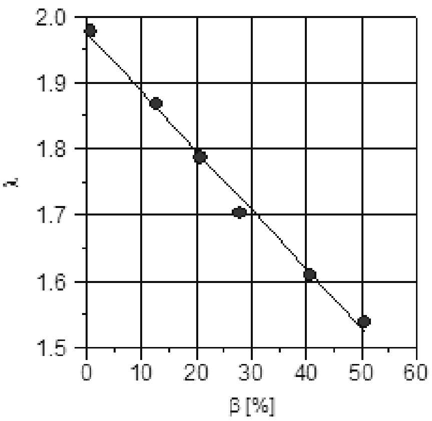

Fig. 4. Equivalent air-fuel ratio versus content of methane in fuel; n=2500 rpm, T=200 Nm

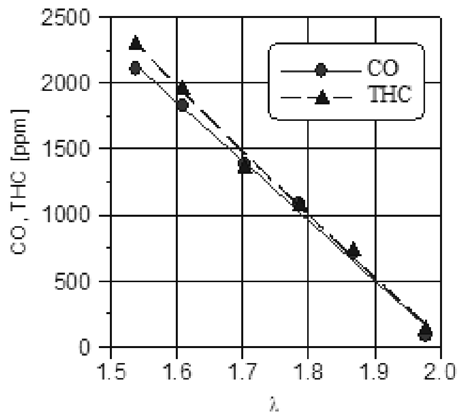

Fig. 5. Concentration of Co and THC versus air-fuel ratio; n=2500 rpm, T=200 Nm

To prevent the engine against increase of emissions the engine exhaust system should be equipped with suitable diesel oxidation catalytic converter (DOC), which will reduce CO and CH_4 emissions. If adopted engine is originally equipped with a DOC, then probably its efficiency of purging methane will be close to zero. This is typical achievement for these reactors.

Methane injected into the engine intake system is nearly under ambient pressure. It makes that in the cylinder is less fresh air because the volume occupied by the portion of the air is replaced with methane. As a result of this, for the same mass of a fuel injected to diesel engine and to dual fuel engine we will have less fresh air inside cylinder and richer air-fuel mixture in a dual fuel engine. Moreover, stoichiometric ratio for methane is higher than for oil due to methane molecule contains more hydrogen than diesel oil. Replacing, a dose of oil with methane, causes enrichment of the air-fuel mixture because of differences between stoichiometric ratios. It is shown on Fig. 4 where for the engine fuelled only with diesel oil (β=0) air-fuel ratio is 1.98, however for 50% share of methane, air-fuel ratio decreased up to 1.53. There is a linear relationship between CO and THC concentration versus air-fuel ratio (Fig. 5), what explains changings of emissions versus methane content. Additionally, the reported changes of THC concentrations can be explained by the valve overlapping.

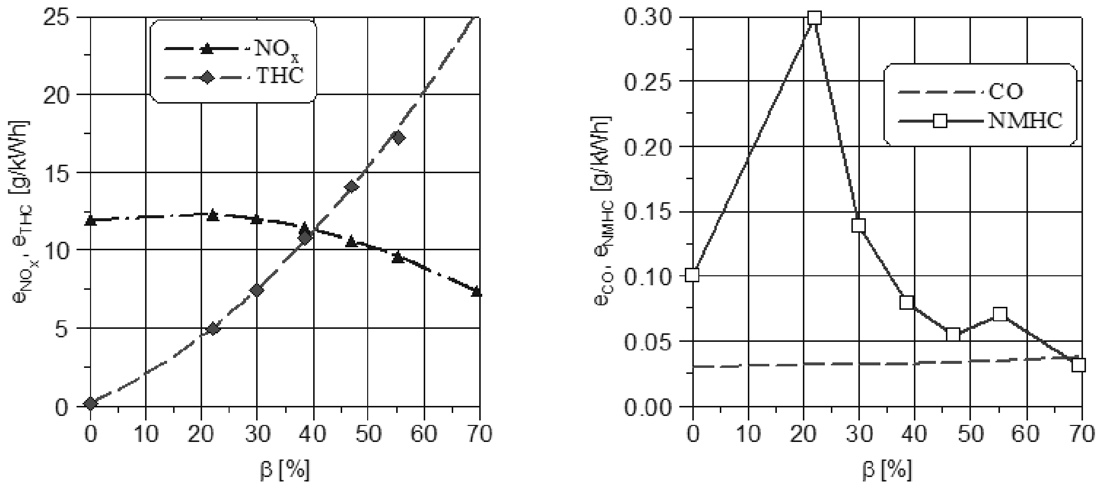

Fig. 6. Specific emissions of various pollutants versus content of methane in fuel; engine equipped with DOC; n=1500 rpm; T=100 Nm

Specific emissions are calculated as the product of the concentrations, exhaust mass flow rates and gas densities. It turned out that for the same power output the engine fuelled with methane produces smaller quantity of exhaust gas. This is mainly due to a smaller mass of air sucked by the engine.

Efficiency of engine thermodynamic cycle increases with the increase of the charge mass. This is associated with the permanent heat losses to the walls of combustion chamber. The smaller mass of the fresh charge inside cylinder is the greater share of heat losses to the walls. This can be seen watching the impact of the engine loading on the engine specific fuel consumption.

On Fig. 6 specific emission of pollutants in a tested engine equipped with DOC is shown. It can be seen that using of catalytic converter was really effective way to reduce the emissions. However, installed catalytic converter did not influence the NO_x, THC and CH_4 emissions.

On a Fig. 7 showed some popular factors usually describing heat release in the cylinder. The angles $\varphi_{0\%}$, $\varphi_{5\%}$, $\varphi_{50\%}$ and $\varphi_{90\%}$ are basic parameters used in the literature. The angle $\varphi_{0\%}$ defines ignition lag having impact on noise of an engine and maximum cylinder pressure. The angle $\varphi_{5\%}$ defines heat release after ignition. This angle is important for SI engines, during this phase speed of laminar burning determines combustion rate. In this phase, laminar flame turns into turbulent flame, which results in sharp acceleration of the combustion. The angle $\varphi_{50\%}$ is position close to the centre of gravity of the heat release rate and is a good representative parameter for determining the location of whole heat release process with respect to TDC.

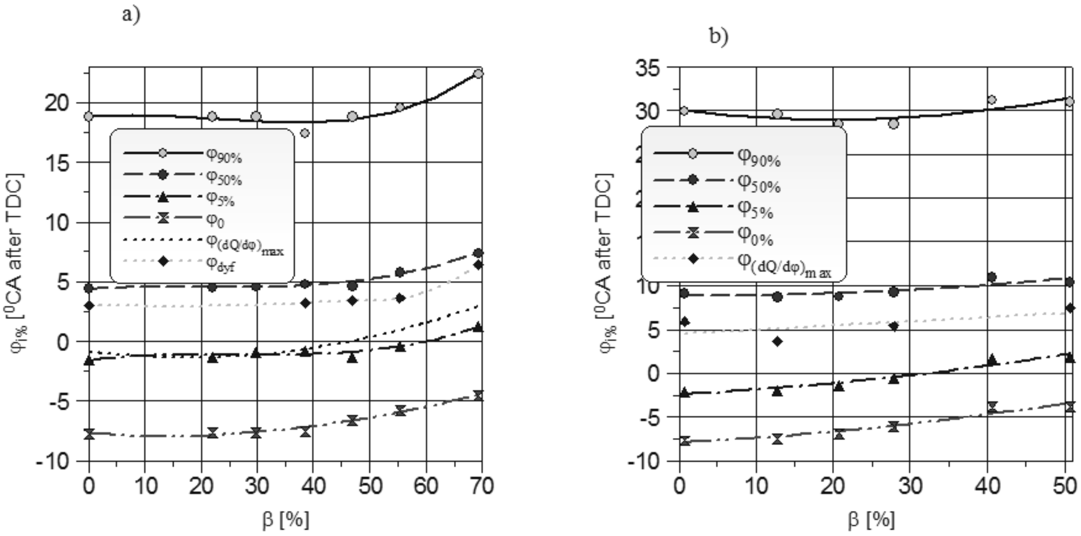

Fig. 7. Characteristic angels describing heat release vs. content of methane in fuel; a) n=1500 rpm, IMEP=6.5 bar, b) n=2500 rpm, IMEP=9.5 bar

The $\varphi_{90\%}$ angle, is angular position of crankshaft, where 90% of fuel contained in the cylinder has been burnt. This position is conventionally regarded to be the end of the regular combustion with clearly visible front of flame. Burning up of remaining 10% of the fuel is relatively extended in time and has little impact on the overall efficiency of the engine. Fig. 7 shows also the angle corresponding to the position of the maximum of heat release rate and the starting point of diffusive combustion on this graph.

Watching at Fig. 7, we can see that adding methane to diesel oil shifts heat release process beyond TDC. The shifting is the greater the greater content of methane in the fuel. Despite this general trend it can be seen that methane share in quantity of $\beta < 15\%$ improves some parameters (e.g. $\varphi_{0\%}$, $\varphi_{90\%}$). Retarded heat release is a reason of lower engine efficiency and knocking combustion.

On Fig. 8 and 9 heat release rate and cumulative heat release, for different content of methane in fuel (β) is shown. With increase of methane share, the heat release rate also is changing. The ignition lag is getting greater, *dQ/dφ* curve becomes lower and wider.

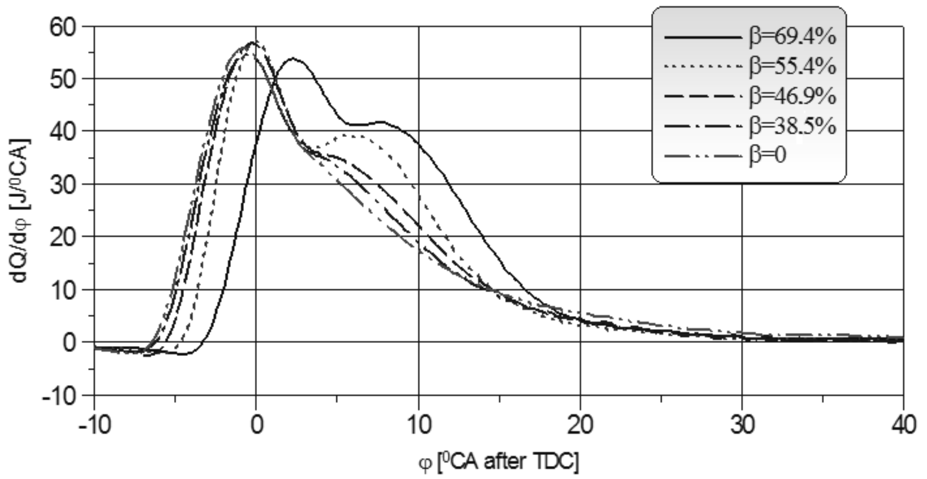

Fig. 8. Heat release rate versus crankshaft angle and fuel composition n=1500 rpm, IMEP=6.5 bar

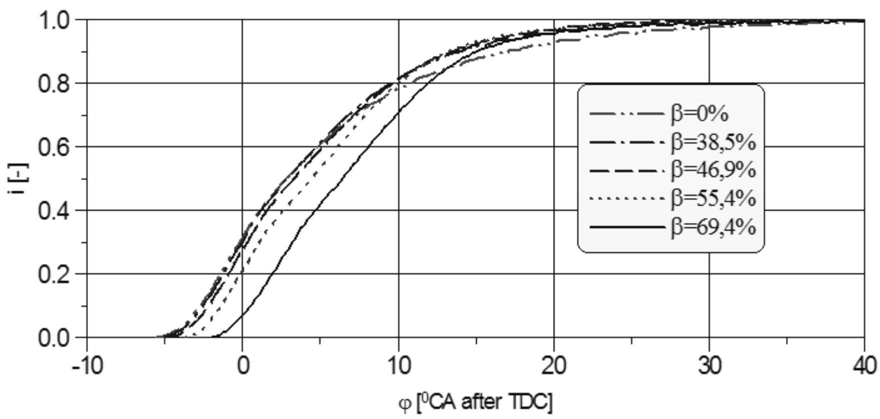

Fig. 9. Normalized cumulative heat release versus crankshaft angle and content of methane n=1500 rpm, IMEP=6.5 bar

4. Conclusions

1. Adding of CNG to diesel oil in tested engine caused increased of THC, CO and CO_2 emissions, whereas emission of NO_x and NMHC decreased. Efficiency of dual fuel engine decrease with increase of methane content in fuel.
2. Increase of methane share in fuel causes proportional increase of ignition lag and delay in heat release process.
3. To compensate influence the combustion delay the injection advance should be adjust.
4. Knock combustion is a limiting factor for the maximum possible dose of methane; for tested engine, this dose was approximately 50%.

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