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Investigation of the Impact of Compressed Natural Gas on Heat Release Rate in Dual-Fuel Operation of a D3900 Diesel Engine at Mid Loads

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Abstract. **This article delves into the influence of Compressed Natural Gas (CNG) on the heat release rate of a D3900 diesel engine operating under a dual-fuel cycle, specifically at mid loads (45%). The study explores the effects of varying proportions of CNG in the total fuel supplied to the engine. Experimental investigations reveal a consistent reduction in heat release rate during dual-fuel operation, proportionate to the quantity of CNG introduced into the combustion process. Notably, the research identifies that the maximum heat release rate occurs at higher crankshaft angles. The article also contains information regarding the in-cylinder pressure and temperature when the engine operates in dual-file mode. These findings contribute valuable insights into the combustion characteristics of dual-fuel systems, shedding light on the interplay between CNG proportions and heat release dynamics in the D3900 diesel engine at mid loads.**

Keywords: heat release rate, diesel engine, dual-fuel diesel cycle, CNG.

I. Introduction

Diesel engines are extensively used in the transportation, agricultural, and industrial sectors worldwide due to their superior energy efficiency, reliability, adaptability and cost-effectiveness than gasoline equivalents [1]. Along with the positives, diesel engines also pose complex problems related to the ecology and their dependence on fossil fuels.

The reduction of petroleum reserves on a global scale necessitates exploration of strategies to decrease fuel consumption or adopt alternative fuels in modern engines. The escalating environmental requirements for contemporary vehicles [2], [3] and engines constitute a crucial precondition for the utilization of alternative fuels or additives to existing ones, aiming to emit fewer harmful emissions during internal combustion engine operation [4].

Compressed Natural Gas (CNG) is one of the most common and easily available gaseous fuels applicable to compression ignition engines for implementing a dual-fuel duty cycle. Natural gas is a hydrocarbon gas formed through the anaerobic decomposition of organic materials. The primary source for its extraction is gas fields, where the predominant component is methane [5]. Natural gas generally consists of a mixture of hydrocarbons with methane (CH_4) as the main constituent. Ethane, propane, butane, nitrogen, and carbon dioxide gases contribute to the remaining composition while traces of water vapor and hydrogen sulphide may be present in some natural gases. The properties of natural gas are vary depending on the location, processing and refining facilities. Usually, the maximum and minimum compositions are specified to enable comparisons to be made – Table 1 [6].

TABLE 1 MAXIMUM AND MINUMUM

Conpound	Typical	Maximum	Minimum
Methane -CH_4	87.3%	92.8%	79%
Ethane – C_2H_6	7.1%	10.3%	3.8%
Propane – C_3H_8	1.8%	3.3%	0.4%
Butane – C_4H_{10}	0.7%	1.2%	0.1%
Nitrogen – N_2	2.2%	8.7%	0.5%
Carbon dioxide-CO_2	0.9%	2.5%	0.2%

The burning process in a compression ignition internal combustion engine of the fuel-air mixture can be conventionally divided into four periods [7], [8]: the ignition delay period, the premixed combustion phase (rapid combustion), the mixing-controlled combustion phase (main combustion period), and the late combustion phase (burning-out period).

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The ignition delay period (induction period), also known as the first period, begins with the start of diesel fuel injection and ends with the autoignition of a portion of the cycle fuel quantity as the pressure curve separates from the compression polytrope (Fig. 1 – Stage I) [9]. During the first period, physical-chemical preparation of the forming fuel mixture takes place, and the first autoignition focal points are created. The portion of the cycle fuel quantity entering the combustion chamber during the first period is approximately 40-50% of total diesel fuel amount and depends on period duration and the integral fuel delivery law. The larger the amount of fuel entering during the first period, the higher the rate of pressure rise in the cylinder after the start of autoignition is. The duration of the first period depends on the cetane number of the fuel, the injection advance angle, and the air-fuel ratio. The angle at which the crankshaft turns during the ignition delay period depends on its duration and the engine speed [6].

The rapid burning period (second period) begins when the pressure curve separates from the compression polytrope and ends when the maximum pressure in the cylinder (Pz_{max}) is reached (Fig. 1 Stage II). During this period, there is a high rate of heat release and a rapid increase in temperature and pressure. By the end of the second period, about 30 - 40% of the total heat input from the fuel is released. The maximum heat release rate – $(dX/d\varphi)_{max}$ is obtained around the top dead center (TDC), just before the pressure maximum. The duration of the second period mainly depends on the rate at which the flame front propagates, i.e., the degree of turbulence of the fuel mixture. The injection advance angle also influences the duration of the second period [7]. The rapid burning period can be in principle divided into two phases: the explosive (uncontrolled) burning phase and the controlled burning phase – corresponding to zones IIa and IIb in Fig. 1 [9]. In the explosive (uncontrolled) burning phase, the fuel injected during the ignition delay period and mixed in a combustible ratio with air in the combustion chamber burns aggressively over a few degrees of crankshaft rotation [9]. It also realizes a very high rate of heat release. Its limits are defined by the separation of the pressure curve from the compression polytrope (point P_c) until reaching the maximum heat release rate point $P_{Z'}$ (Fig. 1 – Phase IIa) [9]. The controlled burning phase begins at point $P_{Z'}$ and ends when the pressure in the cylinder reaches its maximum value. The values and location of Pz_{max} relative to TDC to a large extent determine the efficiency of the fuel process [9].

The main (mixing-controlled) burning period (third period) begins at the maximum pressure in the cylinder and ends when the gas temperature reaches its maximum value (T_{max}) - Fig. 1 – Stage III. The rate of heat release during the third period decreases as the concentration of oxygen in the combustion chamber decreases, and the concentration of exhaust gases increases. During this period, there may be a second (local) maximum in the rate of heat release [8]. The pressure also begins to decrease due to the increase in the volume of the clearance space. The duration of the third period depends on the engine speed and the air-fuel ratio [7].

The late combustion phase (fourth period) begins at the moment of reaching T_{max}, and its end can be determined by the law of heat release – $X = f(\varphi)$ (Fig. 1 – Stage IV). The fuel process is completed when the coefficient of heat release, X, is approximately 0.98 – 0.99. If the analysis of the fuel process is based on the law of active heat release – $Xa = f(\varphi)$, its end is considered the moment when the coefficient of active heat release reaches its maximum value – $Xa_{max} \approx 0.7$ – 0.8. In the case of poor organization of the fuel process, complete combustion of the cycle fuel quantity cannot be ensured, which can be determined by the absence of a maximum in the dependence $Xa = f(\varphi)$ – until the moment the exhaust valve opens [6].

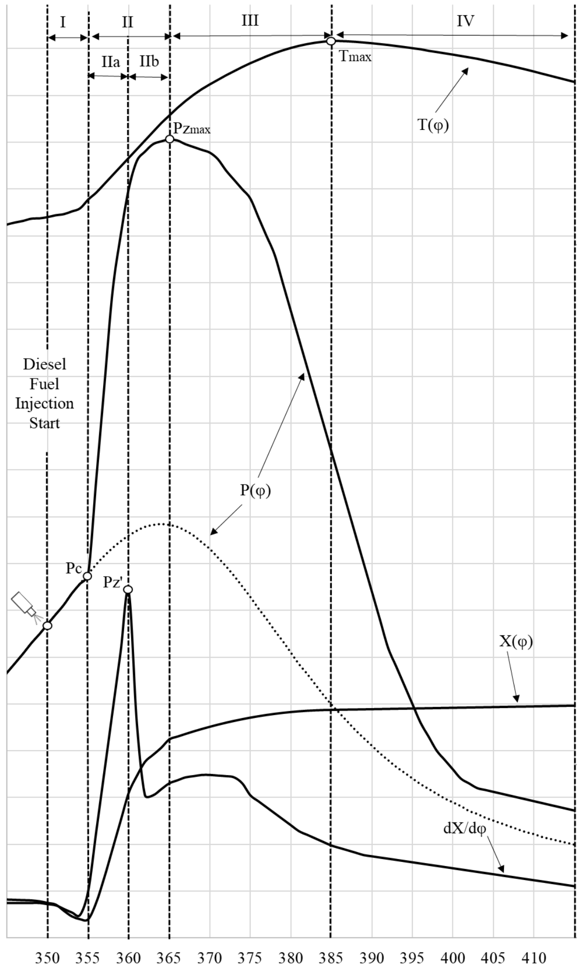

Fig. 1. Combustion parameters of compression ignition engines.

II. MATERIALS AND METHODS

The investigation of CNG impact on heat release rate (HRR) is based on experimental results obtained by the application of the test setup described on Fig. 1. For the purpose of the study a four cylinder, water cooled, direct injection diesel engine *Perkins D3900* is used. The engine has the following parameters: bore– D = 98.4 mm; stroke – S = 127 mm; compression ration – ε = 16; nominal brake power – N_e = 57.4 kW at engine speed – n = 2500 min^{-1}. The test bench is equipped with the systems listed below which allows detailed monitoring of the performance parameter of the engine operates on CNG-diesel cycle:

- Direct current dynamometer– 11, used to define engine brake power and corresponding load conditions;
- The system used for measuring engine intake air flow includes throttle device (orifice) – 2 and differential water pressure gauge - 1;
- The measurement system of the diesel fuel provided to the engine is consist of fuel tank, two solenoid valves – 14, two photoelectric sensors – 15, weight measuring device (scale) – 16 and control unit 13;
- The CNG injection system is consist of CNG tank – 21; pressure reducer – 22; electronic device – 7 which controls the duration of open condition of the gas valve – 6 while the timing of CNG injection is defined by tandem work of sensor 20 and the pin mounted on the crankshaft pulley;
- Measurement system determining the amount of gas fuel injected in the intake manifold consist of manometer – 3, gas flow meter (G4 type) – 5, thermocouple (K type) – 4 connected to the measurement device 9;
- Engine speed measurement device – 12 receiving signal from induction sensor – 17 which works in combination with tooth gear with 60 teeth;
- In cylinder pressure is measured by AVL system consist of – piezoelectric sensor – 8 mounted in the cylinder head, two speed sensors: one for determining the engine speed – 19 and one for defining the top dead center of the measured cylinder, visualizing and storage system 10.

The amount of CNG injected in the engine is defined as percentage of the total amount of fuel consumption by means of coefficient *K*.

$$K = \left(\frac{B_{CNG}}{B_h}\right).100,\% \qquad (1)$$

where B_{CNG} is he hourly CNG consumption, kg/h; B_h – the total hourly fuel consumption, kg/h.

The total hourly fuel consumption can be calculated by the following equation:

$$B_h = B_{CNG} + B_D \qquad (2)$$

where B_D is the hourly diesel fuel consumption.

The methodology for the experimental study consists in based on comparison of the engine heat release rate as a function of the mass fraction of the gaseous fuel - regulating characteristics with variable gaseous fuel content. The regulating characteristics are obtained at constant engine load (constant mean effective pressure). More details on the on the methodology and obtaining of the relevant characteristics can be found in [10].

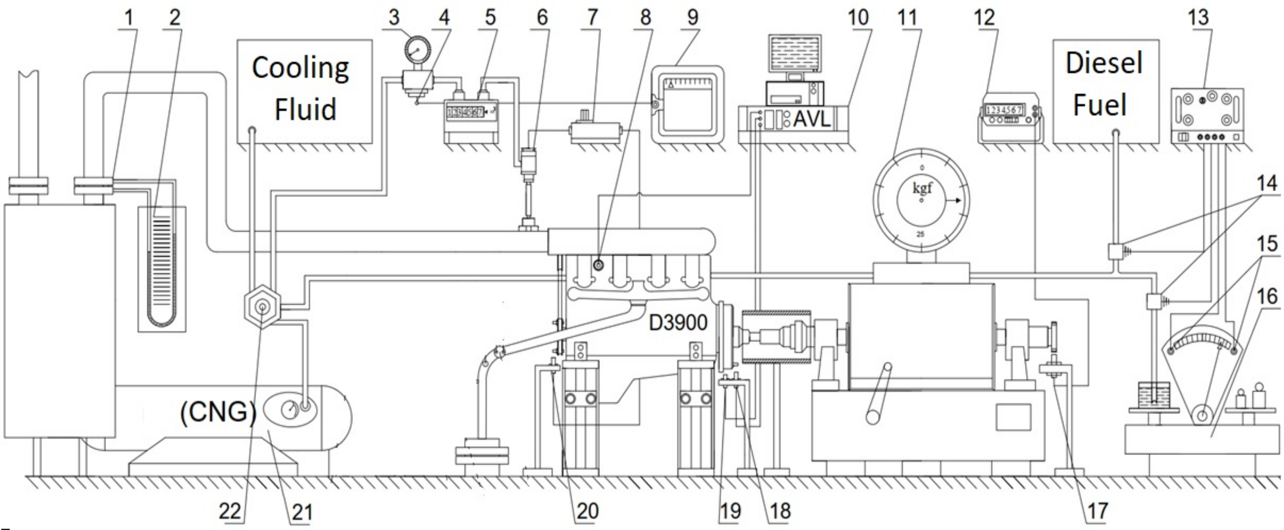

Fig. 2. Test setup for dual-duel mode of *D3900* engine .

III. Results and discussion

The results of the heat release (Fig. 4), heat release rate (Fig. 5) and in cylinder temperature (Fig 3) are derived by numerical processing of the indicator diagrams (Fig. 3) by usage of dedicated computer program described in [11]. It is notice that the usage of CNG as a gas fuel for dual-fuel duty cycle of *D3900* engine at mid loads lead to decrees of the heat release and heat release rate. It is also observed that the total heat release decrease in dual-fuel mode which is an indication of less effective combustion process. The effect is proportional to the amount of gas fuel provided to the engine. The reason for the change with an increase in the amount of natural gas supplied to the engine is the lower value of maximum pressure in the cylinder and an increased duration of ignition delay. Consequently, there is a displacement (in the direction after top dead center) of the angle at which the maximum pressure in the cylinder is reached. In addition to the reasons mentioned so far, it should be noted that the results were obtained at 45% load, which makes oxidation of part of the air-fuel mixture difficult. This is due to the relatively cold walls of the combustion chamber. Also the small quantity of diesel fuel supplied, the autoignition of which does not generate enough heat for the oxidation of the entire quantity of the gas-air mixture in the cylinder leads to decrease of the heat release rate proportional to the volume of gas fuel (increase of gas fuel portion leads to decrease the diesel quantity).

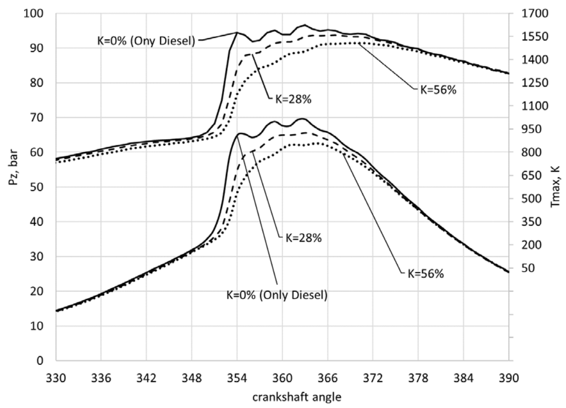

Fig. 3. Cylinder pressure and temperature of *D3900* at 45% load and 1400 min^{-1}.

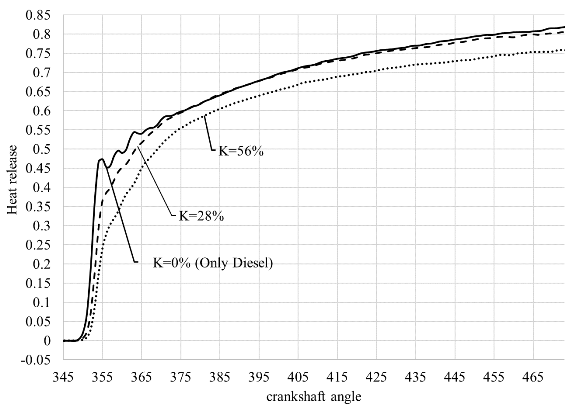

Fig. 4. Heat release of *D3900* at 45% load and 1400 min^{-1}.

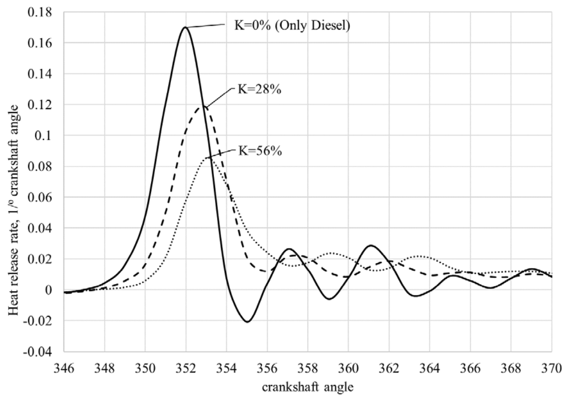

Fig. 5. Heat release rate of *D3900* at 45% load and 1400 min^{-1}.

IV. Conclusions

Based on the performed experimental study of the CNG effect on the heat release rate of *D3900* diesel engine operating on dual-fuel mode at mid loads (45%load at 1400 min^{-1}) the following conclusions can derived:

- In cylinder pressure decrease proportional to the amount of gas fuel provided to the engine;
- Peak of cylinder pressure is observed at higher crankshaft angle when the engine operates on dual-fuel mode;
- In cylinder temperature decrease when the engine operates in dual-duel mode;
- The heat release and the rate of heat release decrease with increase of the amount of CNG provided to the engine;
- The maximum of the heat release rate is achieved at higher crankshaft angle.

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