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THE IMPACT OF A TWO-PHASE DIESEL FUEL PILOT INJECTION ON THE COMPRESSED NATURAL GAS AIR–FUEL MIXTURE COMBUSTION PROCESS IN A DIESEL ENGINE

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Abstract. Nowadays, there is a global trend towards the use of alternative fuels in order to reduce environmental pollution. For example, Compressed Natural Gas (CNG) has become more widely used around the world. The use of different fuels in engines affects the combustion process and efficiency, with the latter potentially being reduced by such means as, for example, the use of gaseous fuels in conventional diesel engines. Therefore, it is also important to know how CNG combusts in a diesel engine and how the combustion process can be improved. Consequently, the aim of the study is to give an overview of the effect of divided Diesel Fuel (DF) pilot injection on the combustion process of a naturally aspirated diesel engine using dual-fuel mode, with one fuel being DF and the other CNG. The focus of the article is on the commonly used engines on which the diesel injection system works regularly, and CNG fuel is injected into the intake manifold as an additional fuel. The engine DF quantity and injection timing are regulated by the acceleration pedal. The article provides an overview of the diesel and dual-fuel combustion process, and compare the DF and dual-fuel combustion processes. For this purpose, a test was carried out in order to measure the various involved parameters, such as the combustion pressure, torque, and fuel consumption. The results demonstrated that ignition delay does not significantly vary with the use of gas as a fuel source, and the maximum combustion pressure is actually higher with gas. The combustion is more rapid in dual-fuel mode and results indicate that when using dual-fuel mode on regular engines, it would be necessary to regulate the pre- and main-injection timing.

Keywords: heat release, combustion pressure, relative heat release, dual-fuel engine, gaseous fuel.

Notations

ATDC – after top dead center;
BMEP – brake mean effective pressure;
BSFC – brake specific fuel consumption;
BTDC – before top dead center;
CAD – crank angle degree;
CNG – compressed natural gas;
CO – carbon monoxide;
DF – diesel fuel;
DF+CNG – DF and CNG fuel;
EGR – exhaust gas recirculation;
HC – hydrocarbon;
HRR – heat release rate;
NO_x – nitrogren compounds;
RHR – relative heat release;
SOC – start of combustion;
SOI – start of injection.

Introduction

A directive adopted in the European Union stipulates a 10% target for share of energy from renewable sources in the transportation sector in all Member States (Küüt *et al.* 2017). As a result, when it comes to internal combustion engines, engineers have proposed various solutions in terms of the use of different renewable fuels and supporting systems of fuels (engine sub systems, tanking systems, etc.) (Ilves *et al.* 2019; Safronov *et al.* 2020). A potential solution is the use of gaseous fuel with DF in diesel engines where gas is injected into the engine's intake manifold and DF is injected directly into the cylinder (Karim 2003; Namasivayam *et al.* 2010; Azimov *et al.* 2011; Korakianitis *et al.* 2011; Ryu 2013). The main reason for the use of a dual-fuel system (using liquid and gaseous fuel) is to reduce the use of fossil fuels, and, additionally, to reduce the soot particles and nitrogen compounds in the exhaust

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gas. One type of gaseous fuel is methane, which has a high auto-ignition temperature and cannot be used in a diesel engine without the addition of DF. Therefore, in order to use methane in a diesel engine, DF pilot injection must be used, which ignites gaseous fuel in the cylinder.

An advantage of using methane can be associated with the lower amount of soot in the exhaust gases when compared to using DF (Lounici *et al.* 2014; Liu *et al.* 2013). When using DF with methane, the combustion process in the engine should be as effective as possible. This is because the gas is sucked into the cylinder with air, so the amount of air in the cylinder is reduced, and, therefore, the amount of oxygen in the cylinder is reduced. This may result in the DF combustion process becoming ineffective. Therefore, the power of the engine is also somewhat lower than that of a diesel-only engine (Yuvenda *et al.* 2020)

A study that was carried out by Tarabet *et al.* (2014) analysed the combustion of DF and a mixture of DF and CNG (an air–fuel mixture DF+CNG), as well as a mixture of bio-DF and CNG (bio-DF+CNG) in a diesel engine at light (20%) and heavy (90%) loads. Research indicated that at 20% load, the maximum combustion pressure of the DF was somewhat higher than that of the alternative. However, at maximum load, the maximum combustion pressure of DF was significantly lower than the pressure of gaseous fuel. Similarly, the HRR was lower at 90% load with DF, however, at 20% load, it was the same as with DF+CNG. It also became apparent that the ignition delay of DF+CNG was longer than that of DF, but the ignition delay of bio-DF+CNG was shorter because its cetane number is higher (Tarabet *et al.* 2014).

The research by Yuvenda *et al.* (2020) examined the effects of the fuel injection timing of DF+CNG on the combustion process at light load. The results indicated that by delaying the gas injection and increasing the amount of gas in the intake manifold, the engine's power and thermal efficiency increased and fuel consumption decreased. The optimal quantity of replaced gas was noted as comprising 67.69% of the fuel's energy, however, when the amount of gas in the air–fuel mixture was too high, the engine's efficiency parameters and fuel consumption deteriorated. It was also pointed out that the ignition delay increased and the duration of the combustion process for DF+CNG decreased due to lower temperatures in the cylinder (Yuvenda *et al.* 2020).

Research by Papagiannakis and Hountalas (2003) examined the effects of gaseous fuel on engine performance and exhaust gases. The tests were carried out at engine speeds of 1500, 2000 and 2500 rpm, and at loads of 40, 60 and 80%. For example, at an engine speed of 2000 rpm, on a load of 80% with the proportion of CNG of the injected fuel being 52%, the test resulted in a significantly lower combustion pressure, longer ignition delay, and higher HRR in the expansion phase, which can imply that the combustion of DF+CNG fuel lasts longer (Figure 1) (Papagiannakis, Hountalas 2003). The decrease in the combustion pressure during the use of dual-fuel is a trend

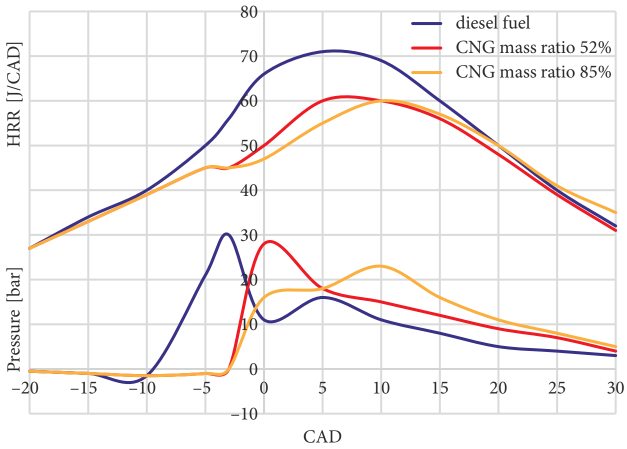

Figure 1. Combustion pressure and HRR at an 80% load and at 2000 rpm – is based on data from research by Papagiannakis and Hountalas (2003) and illustrates the principles of changing pressure and HRR

in other research (Imran *et al.* 2014; Srinivasan, Krishnan 2014; Srinivasan *et al.* 2006).

Lounici *et al.* 2014 give an overview of the combustion of DF and CNG fuel in a diesel engine at different modes. The quantity of DF provided 10% of engine power. The results indicated that in dual-fuel mode, the BSFC was higher at low loads and lower at high engine loads as compared to the use of DF. The HRR in dual-fuel mode at low engine loads was similar to the use of DF, however, at high loads, the HRR in dual-fuel mode was significantly higher. It can be said that due to the more efficient combustion of gaseous fuel, energy was released faster in the process. The proportion of soot in the exhaust gases decreased in dual-fuel mode. NO_x emissions generally decreased in dual-fuel mode at low and medium loads, however, at high loads, NO_x emissions increased in dual-fuel mode. An interesting nuance that can be pointed out concerns the gas temperature in the cylinder, which was higher in dual-fuel mode at all loads. HC emissions increased in dual-fuel mode at all engine loads, and CO was lower at high loads. The proportion of CO was higher in dual-fuel mode at low and medium engine loads. (Lounici *et al.* 2014).

Next, a study by Abdelaal and Hegab (2012), which deals with the impact of the EGR valve on engine efficiency parameters and emissions in dual-fuel mode will be referred to. The experiments were performed with a mixture of diesel, diesel and CNG, and EGR, at positions of 0, 5, 10 and 20%. The pilot injection provided 20% of engine load. The results indicated that in dual-fuel mode, the ignition delay increased synchronically to the increase in EGR valve opening. Furthermore, the ignition delay also increased when CNG and diesel were used together as compared to DF. The maximum pressure increase in the cylinder was highest during the use of DF, however, it decreased in accordance with the opening of the EGR valve. The only variation concerned the rise in pressure at EGR valve opening of 10% during which, at medium engine loads, the rise in pressure was higher than at 5 and 20% of EGR valve opening positions. Engine efficiency increased by 2% at medium and high loads in dual-fuel mode. For HC, CO, and NO_x, the results were similar to the study conducted by Lounici *et al.* (2014). However, in this case, NO_x emissions were also reduced at high engine loads. This may be due to the different proportions of gaseous fuel in the air–fuel mixture (Abdelaal, Hegab 2012).

Nwafor (2007) studied the effect of pilot injection timing on the emissions of a diesel engine running in dual-fuel mode. The fuels used were diesel and CNG, however, the article does not specify their proportions in the air–fuel mixture. The results indicated that the ignition delay increased with the use of CNG fuel. It is crucial to note that by varying the timing of pilot injection, the ignition delay in dual-fuel mode was reduced. Depending on the speed mode, the HC in the exhaust increased in dual-fuel mode, however, the timing of the pilot injection did not change the HC share significantly. The measurement results of the CO coincided with the results of the above-mentioned studies; however, in the speed mode of 2400 rpm, the CO level in exhaust gases was higher during the use of DF at high load than in the dual-fuel mode. It is relevant to point out that varying the ignition timing in an extensive manner (for example, 5 CAD) results in unstable engine work. Therefore, adjusting the engine operation is limited (Nwafor 2007).

The aim of the study by Liu *et al.* (2013), was to investigate the effect of different amounts of DF injection on the exhaust gases of a dual-fuel engine. The quantity of CNG fuel was fixed and the amount of pilot injection was gradually increased. Varied injection timings were used for different injection quantities. The results implied that as the amount of pilot spray increased, the share of CO also increased to an extent. At the same time, the proportion of CO decreased when the amount of pilot injection in the air–fuel mixture provided sufficient to increase the combustion temperature in the cylinder. Concerning this particular study, it can be pointed out that CO emissions depended on the amount of DF injections, since in case of small and large pilot injections, it decreased, and in case of medium injection, it increased. The proportion of NO_x compounds in exhaust gases decreased in the dual-fuel mode. In addition, NO_x emissions increased as the amount of pilot injection decreased. The proportion of HC decreased significantly as the amount of pilot injection increases. Compared to the use of DF, the proportion of HC in the exhaust was significantly higher in dual-fuel mode. The soot level in the exhaust gases increased simultaneously with the amount of pilot spray (Liu *et al.* 2013).

The effect of CNG port-injection timing on the combustion process and exhaust emissions of a dual-fuel engine was investigated by Yang *et al.* (2014). The results indicated that the smaller the CNG injection timing, the higher the combustion pressure in the cylinder and the higher the HRR release in the combustion-controlled phase. CNG injection timing varied from −500 °CAD to −240 °CAD ATDC. The flame propagation time decreased at low engine loads as the CNG injection timing decreased. At high loads, the flame propagation time did not change significantly. The RHR50 varied from 45 to 32 degrees ATDC at low engine loads in dual-fuel mode, which is characterized by delayed combustion. At high loads, it remained at 15…25 degrees at all CNG injection timings. The efficiency of the engine decreased somewhat as the CNG injection timing decreased. At medium and high loads, the efficiency of the engine increased by a maximum of 2% as the injection timing decreased (Yang *et al.* 2014).

The article by Wei and Geng (2016), deals with the use of natural gas in a diesel engine. The current article covers exclusively the information on combustion process as presented by Wei and Geng (2016). In their article, it is pointed out that in most of the research articles discussed, in dual-fuel mode, the combustion pressure in the cylinder decreases to an extent and the ignition delay increases. Peak in-cylinder pressure and pressure rise rate at dual-fuel mode are lower compared to DF combustion. The air–fuel mixture is lean, which inhibits the spread of flame. The slow spread of the flame leads to slow combustion as well as the slow rise in the combustion pressure in the cylinder. However, Lounici *et al.* (2014), present somewhat different results. In addition, pilot injection quantity and timing are closely related to cylinder pressure and rise in pressure. Cylinder pressure increases as the amount of pilot injection increases and the timing is advanced. Moreover, ignition delay increases in dual-fuel mode. Lounici *et al.* (2014), point out the cooling process of the gases in the cylinder when gaseous fuel and chemical factors are used. The HRR in the premixed controlled combustion phase is generally lower than with conventional DF. In the combustion-controlled combustion phase, the increase in HRR is not very high, and the energy release time is prolonged. As a result, the exhaust gas temperature may also increase. It is important to note that all combustion phases are affected by the timing, quantity, proportion of gaseous fuel in the heating mixture, etc. (Wei, Geng 2016).

In all these studies, the common characteristic is the pilot injection, which is necessary for igniting the gaseous fuel. However, none of the described studies use or describe the two-phase DF pilot injection for igniting gaseous fuel. This may be due to the assumption that the smallest injection of DF into the dual-fuel engine ignites the air–fuel mixture, which may cause premature combustion in the cylinder. In addition, several CNG systems are being used on diesel engines, in which the amount of fuel to be directed into the engine is controlled by injection pressure (Ismail *et al.* 2018). With this approach, the injection phases are not controlled. Therefore, it would be important to determine how dividing the DF pilot injection into phases would affect the diesel engine combustion process. The adjustment of the engine and pilot injection (in terms of injection quantity, quality by the fuel drops size (Kägo *et al.* 2019) and timing) have a lot of impact, nevertheless, it is important to divide the pilot injection into phases. Since pre-injected DF does not always ignite in the normal operational process of a diesel engine, instead, it ensures a steady combustion of the air–fuel mixture, thus, the pre-injected DF may also not ignite in the gas environment. However, it is likely that pre-injection ensures a more smooth combustion of the gaseous air–fuel mixture in the cylinder.

Based on the above, the aim of the study is to give an overview of the effect of divided DF pilot injection on the combustion process of a naturally aspirated diesel engine using dual-fuel mode, with one fuel being DF and the other CNG. The focus of the article is on commonly used engines, where the diesel injection system works regularly, and CNG fuel is injected into the intake manifold as an additional fuel. The quantity of DF and injection timing are regulated by the acceleration pedal. The article provides an overview of the diesel and dual-fuel combustion process, and compare the DF and dual-fuel combustion processes. In the tests, it was ensured that the DF system's control unit was operating normally, and the gaseous fuel supply system's control unit regulated the dosing of gas into the engine. The proportion of injected gaseous fuel in the total amount of fuel was 50%.

1. Material and methods

This section provides an overview of engine test methodology and test equipment to evaluate the effect of pre-injection of diesel on engine combustion process in dual-fuel mode. The fuels used were regular DF (for direct injection) and CNG (for port-injection). DF quantity and injection timing were regulated by the acceleration pedal to simulate the engine control methodologies of a dual-fuel vehicle in practice. The research was carried out using the engine test bench *Schenck Dynas3 LI250*, and the test engine was an *AVL 5402* compression ignition engine with a common rail injection system. An *AVL 621* unit with an *AVL 2P2E* indicating amplifier was used to measure the combustion pressure, and an *AVL 7351* unit was used to measure fuel consumption. The test equipment is shown on Figure 2 and the technical data for the engine and test equipment is given in Table 1.

The fuels used for the engine tests were DF and CNG (methane with a purity level of >98%). The tests were performed in two stages. In the 1st stage, the tests were carried out on DF only. In the 2nd stage of the experiment, DF and CNG were used jointly. In dual-fuel mode, DF made up 50% (by mass) of the fuel that was used in the air–fuel mixture of stage one. The test measurements were carried out at the crankshaft rotational speed of 2300 rpm and at engine loads of 10, 50, 75 and 100%. CNG was added to the cylinder until the torque value of the engine reached the set torque value of the chosen load with the use of DF. The parameters measured in this research were combustion pressure, fuel consumption, and torque.

Gaseous fuel was injected into the engine intake manifold. Gaseous fuel consumption was measured using the CAS CI-2001A scales. The measuring time was 60 s.

The HRR was calculated for the analysis of the combustion process, which is expressed as follows (Heywood 2018):

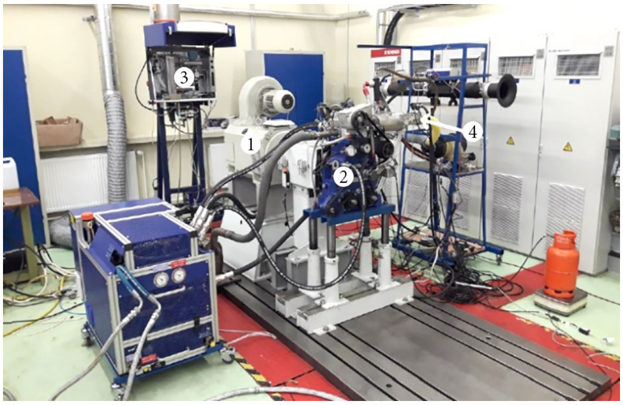

Figure 2. Test equipment: 1 – the engine test bench *Schenck Dynas3 LI 250*; 2 – the compression ignition engine *AVL 5402*; 3 – the *AVL 7351* fuel consumption measuring unit; 4 – the gas injection system

Table 1. Technical data for the test engine and test equipment

Engine test stand: *Schenck Dynas3 LI250*: »» torque range 0–650 N×m; »» max braking power 250 kW; »» max rotational speed 12000 rpm; »» accuracy 0.1%
AVL 5402 CR DI Single Cylinder Research Engine Engine Specifications: »» bore 85 mm; »» stroke 90 mm; »» displacement 510 cm; »» max speed 4200 rpm; »» max firing pressure 170 bar; »» max BMEP ~14 bar at 2300 rpm and supercharged operation; »» max output ~19 kW at 4200 rpm and supercharged operation; »» compression ratio 17:1 (approximately); »» fuel supply system: common rail
Control hardware and software: *AVL RPEMS* control unit and *INCA 7.1* software
Conditioning unit: *AVL 557*
Fuel consumption measurement *AVL 7351*: »» measurement range 0...125 kg/h (at density 0.75 g/cm3); »» accuracy ≤0.12%
Combustion pressure measurement: module *AVL 621* with amplifier *AVL 2P2E*
Pressure sensor *GH13P* with accuracy ≤1.5%

$$\frac{dQ_n}{d\varphi} = \frac{\gamma_{hr}}{\gamma_{hr}-1} \cdot p \cdot \frac{dV_c}{d\varphi} + \frac{1}{\gamma_{hr}-1} \cdot V_c \cdot \frac{dp}{d\varphi}, \quad (1)$$

where: $\frac{dQ_n}{d\varphi}$ – is HRR; φ represents crank angle; γ_{hr} is the ratio of specific heats; p is the combustion pressure; V_c is the cylinder volume.

The heat release is expressed as follows:

$$Q_n = \frac{dQ_n}{d\varphi} + Q_{n.\varphi-1}, \quad (2)$$

where: Q_n is the net heat release; $Q_{n.\varphi-1}$ is the energy released in the combustion process per crank angle.

Table 2. Injection parameters for the diesel supply system

	DF		DF+CNG	
	Pre-injection	Main-injection	Pre-injection	Main-injection
Load 10%				
Duration [ms]	0.27	0.34	0.25	0.28
SOI [degrees BTDC]	18.38	6.30	19.13	8.05
Pressure [bar]	589		597	
Load 50%				
Duration [ms]	0.26	0.47	0.27	0.30
SOI [degrees BTDC]	17.63	5.25	18.30	6.30
Pressure [bar]	736		600	
Load 75%				
Duration [ms]	0.25	0.54	0.26	0.35
SOI [degrees BTDC]	14.97	5.25	18.38	6.00
Pressure [bar]	784		619	
Load 100%				
Duration [ms]	0.25	0.58	0.26	0.44
SOI [degrees BTDC]	13.39	5.20	17.63	5.25
Pressure [bar]	819		673	

The isotropic coefficient has been calculated on the basis on the logarithmic scales of combustion pressure and cylinder volume. The amount of DF was reduced according to the accelerator lever's position when dual-fuel mode was used. The amount of DF injected into the cylinder was equal to half of the amount of DF being used (in terms of mass) in the particular mode. The proportion of gaseous fuel depended on engine torque in stage one that could be achieved when DF was used. The timing for DF injection was divided into two-phases. The timing and duration of the injection process is given in Table 2.

As the position of the acceleration lever is different between the DF test and the DF+CNG test, the timing is different for the pre-injection and main-injection processes. This is due to the fact that the position of the accelerator lever affects the injection timing. This methodology simulates the operation of a normally-used gaseous fuel system on an engine for which the control parameters of the diesel injection system's control unit are changed. This results in differences in the injection timing, duration, and pressure. Compared to other research mentioned in the Introduction, the novelty of the article lies in the two-phase DF injection that can significantly affect the combustion process of a diesel engine.

The test engine torque values at different loads are given in Table 3.

Table 3. The engine torque values at different loads

Load [%]	10	50	75	100
Torque [N·m]	2.50	12.50	18.75	25.00

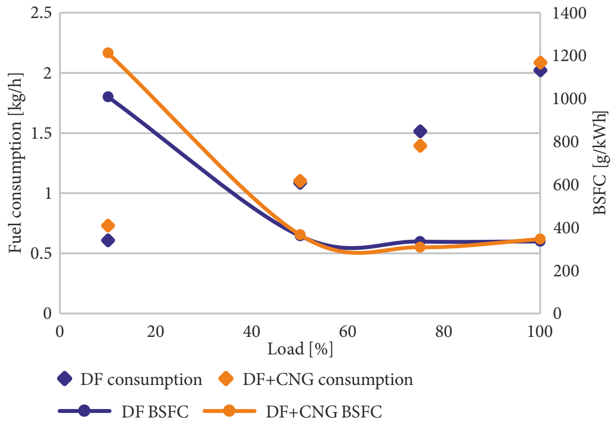

Figure 3. Fuel consumption at workloads of 10, 50, 75 and 100%

2. Results and discussions

The fuel consumption data is presented on Figure 3. In case of DF+CNG, the quantities of DF and gaseous fuel have been summed up. At loads of 10 and 100%, the DF consumption is less than that of DF+CNG consumption, being 10 and 3%, respectively. At a load of 50%, fuel consumption is similar in both tests, and at a load of 75%, the DF consumption is higher than the DF+CNG consumption by approximately 8%. It can be seen on the specific fuel consumption graph how the specific fuel consumption is similar at loads of 50, 75, and 100%. At a load of 10%, the specific fuel consumption when using gas are significantly higher than when DF is being used. Based on data from the studies by Papagiannakis and Hountalas (2003), the specific fuel consumption when using gaseous fuel and DF are higher than when using only DF. According to data in earlier studies (Yuvenda *et al.* 2020; Papagiannakis, Hountalas 2003), the specific fuel consumption is significantly affected by engine adjustment, the timing and duration of the pilot injection, and the quantity of the fuel injected.

In Figures 4–7, the combustion pressure graphs according to the degree of the rotation of the crankshaft are presented. Figure 4 demonstrates the combustion pressure at a 10% load, which shows that the combustion pressure of DF+CNG is higher than with DF. An important difference when compared to the data provided in other research is that when using DF+CNG, the ignition delay is significantly shorter than with DF (around 2 degrees shorter). It is also important to note that DF is injected into the cylinder 2 degrees later than when using gaseous fuel. The ignition delay of DF+CNG may result from the formation of a mixture of DF and gaseous fuel due to pre-injection, and this mixture ignites earlier than when using a single injection phase. Furthermore, the graph in Figure 4 also demonstrates that during the use of DF, the premixed combustion phase in the combustion process is very rapid in this operational mode, and the transition to the mixing-controlled combustion phase is clearly distinguishable. For DF+CNG, the transition from the premixed combustion phase to the mixing-controlled combustion phase is quite smooth. Such smooth transition can be explained by the mixing of DF and CNG due to pre-injection, which also induces the multipoint ignition of the air–fuel mixture and an even spread of the flame. This also leads to a slightly increased combustion pressure in the engine cylinder. The difference with the findings in other research is that SOI begins in dual-fuel mode with the combustion of DF (Papagiannakis, Hountalas 2003; Wei, Geng 2016). This also accounts for a certain increase in the combustion pressure. However, DF HRR increase is more rapid when compared to the DF+CNG. It can be explained by the slow spread of the flame, which causes slow combustion and slow rise of combustion pressure in the cylinder (Wei, Geng 2016). For this reason, it is important that the pre- and main-injection take place earlier than when DF is used, in order to ensure a similar combustion in dual-fuel mode.

A similar trend is also visible at loads of 50 (Figure 5) and 75% (Figure 6), where the transition to the mixing-controlled combustion phase is smoother when compared to the process where DF is used. In addition, the maximum combustion pressure is higher when using DF+CNG. At a load of 50%, the DF ignites 1 degree earlier than with DF+CNG. In this respect, it is important to mention that the pre-injection and main-injection phases of DF begin about one degree later than when using DF+CNG. However, when comparing the results with the results of a single DF injection phase, it can be seen that the ignition delay decreases significantly. The figure illustrates the significance of pre-injection in reducing the ignition delay in the use of a gaseous fuel mix. Based on the analysis of the results, it can be said that in conventional engines, the combustion process using CNG gas is similar to that of DF. The main difference is the maximum combustion pressure that can increase by ~10%. It is important that the pre- and main-injection occurs at least 1 degree earlier than when DF is used, in order to ensure a similar combustion in dual-fuel mode.

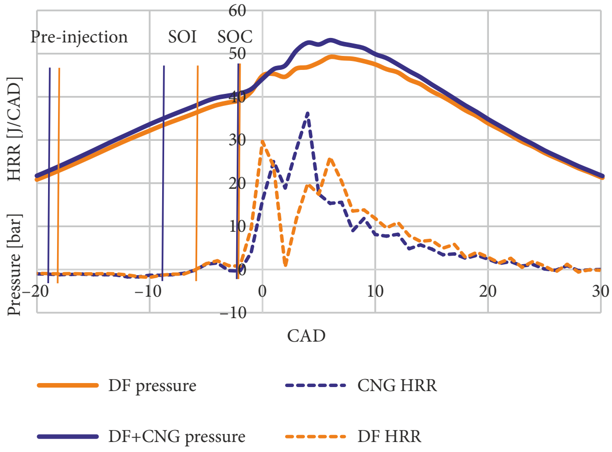

Figure 4. Combustion pressure and HRR at a 10% load

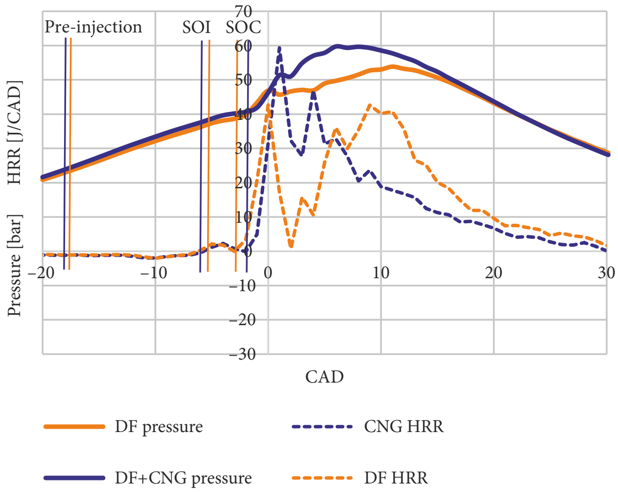

Figure 5. Combustion pressure and HRR at a 50% load

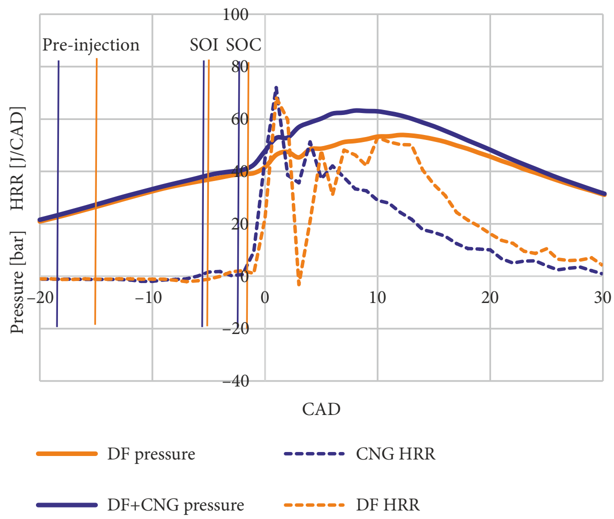

Figure 6. Combustion pressure and HRR at a 75% load

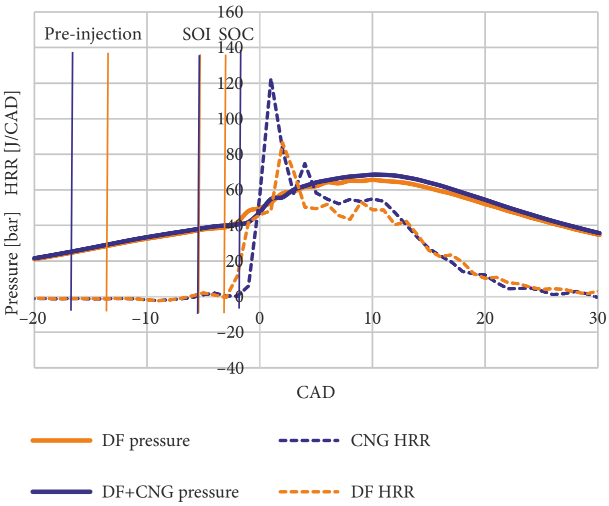

Figure 7. Combustion pressure and HRR at a 100% load

However, at a load of 75% (Figure 6), the DF+CNG fuel mixture ignites approximately 1 degree earlier than when using DF. During the use of DF+CNG, the pre-injection phase of DF starts approximately 3 degrees earlier than when using DF. The difference between the main-injection timing is approximately 0.75 degrees. The process of the combustion is similar to the one at lower loads, however, at a load of 75%, pre-injection should be earlier in dual-fuel mode as compared to the use of DF.

Figure 7 illustrates the combustion pressure at a load of 100%. At 100% engine load, both graphs are rather similar, nevertheless, the maximum combustion pressure of DF+CNG is somewhat higher (by approximately 5%), and DF ignites around 1.5 degrees earlier than DF+CNG. When comparing the results of other researchers, the combustion pressure maximum value and the course of the pressure are similar to a process when DF is used. At a high load, the pre-injection must be more advanced in dual-fuel mode, and the main-injection timing can remain similar to the one used in DF settings.

The pressure and HRR graphs above indicate that the maximum value of combustion pressure when using gaseous fuel is equivalent to or higher than when using DF. Based on these tests, it cannot be argued that the ignition delay increases with the use of gaseous fuel. On the basis of the results that have been obtained, it, however, can be argued that when using a two-phase DF injection, the ignition delay in using gaseous fuel does not significantly change. During the use of gaseous fuel, the ignition delay increased by a maximum of 2 degrees. An important aspect in this case relates to the ignition delay being significantly affected by the timing of the pre-injection. At loads between 10 and 75%, during the use of gaseous fuel, the transition from the premixing combustion phase to the mixing-controlled phase is smoother, and the combustion of the air–fuel mixture is more efficient. At a 100% load, the transition from the premixed combustion phase to the mixing-controlled phase is similar for both fuels. In the case of gaseous fuel, the HRR is significantly faster in the premixed phase. These combustion phases are generally influenced by engine design and injection phases. The results differ from generally accepted published knowledge where the ignition delay is longer and the combustion pressure drops during the use of gas (Papagiannakis, Hountalas 2003; Wei, Geng 2016).

The following Figures 8–11 present graphs that are related to RHR. Figure 8 demonstrates the RHR at a load of 10%. At a low load, the RHR for DF is slightly lower than that of DF+CNG. The maximum heat release is achieved by DF and DF+CNG at CAD 24. At loads of 50 (Figure 9) and 75% (Figure 10), the heat is released significantly slower with DF. At a 50% load, 100% of the air–fuel mixture was combusted at CAD 35 when using DF, and at 28 degrees, when using DF+CNG. At a 75% load, 100% of the air–fuel mixture was combusted at 33 degrees when using DF, and at 30 degrees, when using DF+CNG. At loads of 50 and 75%, it could be seen that 50% of the heat mixture (RHR50) was combusted at about 10 CAD for DF; the RHR50 was achieved at 6 CAD for DF + CNG. This indicates that the combustion of the DF+CNG fuel mix is significantly faster in the initial phase, which is the premixing combustion phase. This confirms the formation of multipoint combustion zones in the cylinder due to ignition. This differs from the results of the article by Yang *et al.* (2014), where the HRH50 was rather delayed. Based on the results that have been obtained, it appears that the timing of the pre-injection and main-injection phases of DF need to be adjusted, as the combustion is faster when using gaseous fuel when compared to using DF. Adjusting of the injection timing can improve the engine efficiency and ensure durability.

Figure 11 presents the RHR at 100% engine load. The figure indicates that the graphs for DF and DF+CNG are quite similar, however, the heat release is somewhat slower in the case of DF. The maximum heat release is achieved at an angle of 29 degrees for DF, and at 27 degrees for DF+CNG. RHR50 is at an angle of approximately 8 degrees in case of both fuels. At this load, the combustion of both fuel mixtures is stable and has a similar combustion velocity.

Figures 8–10 demonstrate the RHR50 values of combustion process on dual-fuel mode. The combustion process is much faster on dual-fuel mode. Based on the data from the research by Yang *et al.* (2014), the results of the data in this article are different and distinctly demonstrate the importance of the injection phases of DF. Widely used dual-fuel vehicles are likely to have a faster combustion process, and, thus, the use of fuel in engines is not the most efficient. According to this, it is crucial to use injection phases when injecting DF into dual-fuel diesel engines. A small amount of fuel does not ignite the homogeneously-formed air–fuel mixture in the cylinder, however, it does allow for a more homogeneous and rapid combustion of the fuel mixture. What is more, the pre-injection of DF reduces the ignition delay. When using a two-phase pilot injection, the combustion of the DF+CNG fuel mixture is faster than that of DF, so it is important to adjust the timing of the injection phases.

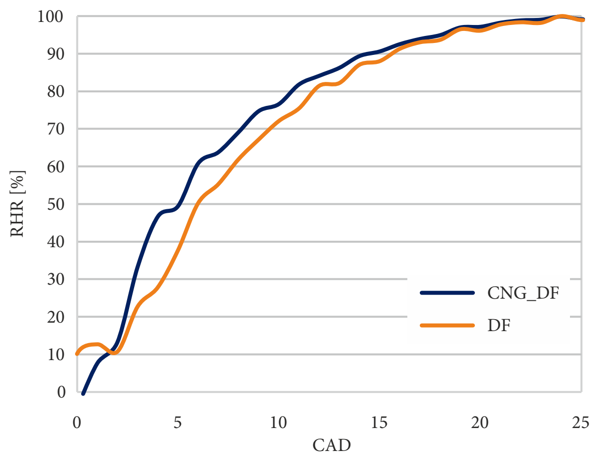

Figure 8. RHR at a 10% engine load

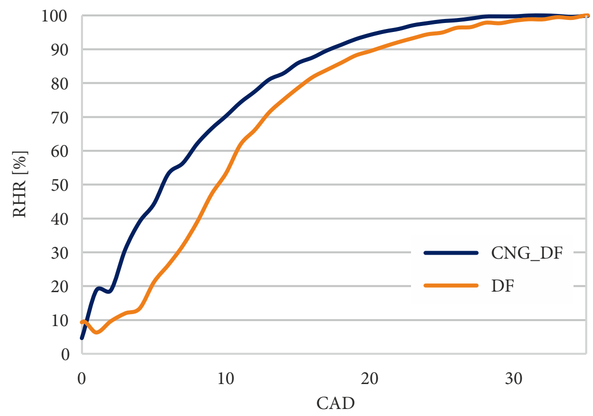

Figure 9. RHR at a 50% engine load

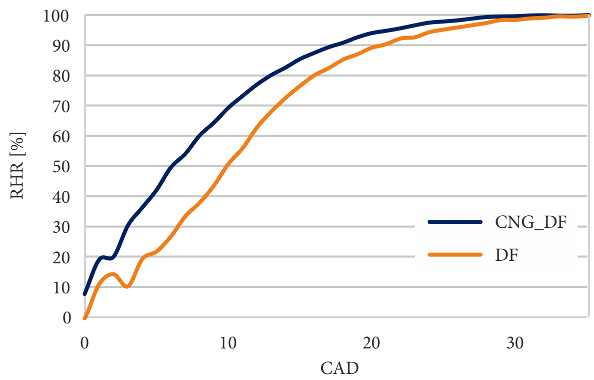

Figure 10. RHR at a 75% engine load

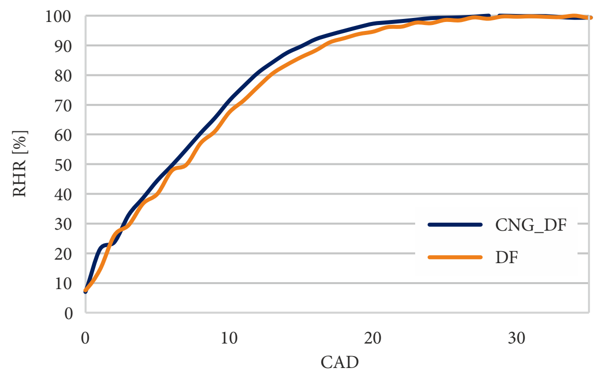

Figure 11. RHR at a 100% engine load

Conclusions

The tests for this particular research were carried out on a dual-fuel diesel engine in order to study the impact of the two-phase injection of DF on the combustion process of gaseous fuel. More specifically, the combustion process of DF and of a DF+CNG fuel mix in an engine were compared to a process in which 50% of the DF had been replaced by gaseous fuel. The fuel mixture was formed by means of direct injection when using DF and by in-direct injection (homogeneously) when using gaseous fuel. The following conclusions can be identified from the results:

- »» injecting DF in two-phases reduces the ignition delay when using gaseous fuel, and the ignition delay becomes comparable to the combustion process where DF is used;
- »» when using pre-injection, the maximum combustion pressure increases slightly (~10%); this is different from the results that have been obtained when using single-phase pilot injection of DF;
- »» the use of pre-injection for DF results in a multi-point ignition of the fuel mix with gaseous fuel, making the transition from the premixing combustion phase to the mixing-controlled combustion phase smoother when compared to using only DF in the engine;
- »» when using pre-injection at light and medium loads, the RHR in the premixing combustion phase is significantly faster when using gaseous fuel; the difference of RHR50 from using DF is approximately 4 degrees. RHR100 is also achieved faster with gaseous fuel;
- »» when using a two-phase pilot injection, the combustion of DF+CNG fuel mix is faster than that of DF, so it is important to adjust the timing of injection phases in dual-fuel diesel engines. As the combustion is faster in dual-fuel mode, results indicate that when using dual-fuel mode on regular engines, it would be required to delay the pre- and main-injection timing.

In this research, the tests were carried out at one engine speed mode, therefore, it is recommended to further study the impact of a two-phase injection of DF at different speed modes in a dual-fuel engine. Additionally, it is important to identify the optimal pre-injection quantity of DF in order to ensure the efficient combustion of gaseous fuel in the cylinder.

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Author contributions

Risto Ilves and *Rauno Põldaru* conceived the study and were responsible for the design and development of the data analysis.

Risto Ilves, *Rauno Põldaru*, *Andres Annuk* and *Jüri Olt* were responsible for data collection and analysis.

Risto Ilves and *Rauno Põldaru* were responsible for data interpretation.

Rauno Põldaru wrote the 1st draft of the article.

Risto Ilves wrote a final version of the article.

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