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Effects of Thermal Barrier Coated Piston on Performance and Combustion Characteristics in Dual-Fuel Common-Rail Diesel Engine

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Abstract

In this study, performance and combustion characteristics of biogas and biogas+hydrogen mixtures were experimentally analysed and compared with baseline diesel fuel in a common-rail diesel engine with and without thermal barrier coated piston. Tests were conducted at three different loads (50 Nm, 75 Nm, and 100 Nm) and a constant speed of 1750 min^{-1}. Engine pistons were coated with Yttria Stabilized Zirconia by atmospheric plasma spray method. Results showed that by replacing the standard pistons with the coated pistons, an increase for diesel, biogas, and hydrogen enriched biogas was respectively defined by 8.1%, 6%, and 23% in cylinder pressure, and 19.8%, 12.6%, and 25% in HRR at medium load. Similarly, there was an increase in range of 1.05-12.8% in gas temperature and 20.5-117.2% in knock intensity by the piston coating. CA10-90 was prolonged between 1-15 °CA with gaseous fuel modes and increased with the engine load. Volumetric efficiency was reduced by 0.1-4% with the gaseous fuel operations, while it was increased by using the coated piston for all fuels. Exhaust gas temperature increased with the gaseous fuels whereas showed discrepancies with the coated piston engine. Dual-fuel mode and coated piston application caused brake specific energy consumption to increase significantly.

Keywords: Piston coating; Common-rail diesel engine; Dual-fuel; Combustion.

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1. Introduction

Environmental pollution is increasing at an alarming rate due to pollutants emitted from the internal combustion engines (ICEs) in transportation, power systems, and industrial applications. Moreover, the increasing cost of the conventional liquid fuels makes alternative fuels inevitable in ICEs to solve the problems in view of both environmental and financial. Reducing the pollutants from the petroleum-based fuels used to drive transportation, industry, and propulsion of the power systems is a mandatory requirement. Biogas, alcohols, biodiesel, liquefied petroleum gas, natural gas (NG), and hydrogen (H_2) are the most popular alternative fuels. When running with these fuels, the engine is expected to produce the same power as being with standard fuel without any modification in the mechanical design of the engine. When dual-fuel (DF) mode is applied to the diesel engines, a small amount of diesel fuel (pilot fuel) is used to ignite the air-fuel mixture [1, 2]. Therefore, it enables the consumption of conventional petroleum-based fuel to decrease.

Biogas, mainly composed of methane (CH_4) and carbon dioxide (CO_2), could be used in transportation and stationary power machines for supplying the energy requirement. Biogas is yielded by anaerobic fermentation of the organic resources in absence of oxygen and has similar contents to those of NG, which is mainly composed of CH_4 of 50-70%, CO_2 of 30-50%, and a small amount of other gases such as hydrogen sulphide, nitrogen, carbon monoxide (CO), oxygen, water vapor, ammonia, H_2, and so on [3]. Their proportion depend on resources of raw matter.

Since having high octane number and auto-ignition temperature, the biogas is a proper fuel for ICEs with high compression ratio. Biogas DF mode can be applied to increase brake thermal efficiency (BTE), and to reduce NO_x and particulate emissions, which are the main problems of the diesel engines [1, 4]. However, CO_2 presence in biogas leads to lower flame propagation speed and combustion enthalpy. Reduction of the CO_2 content in biogas can enhance the flame propagation speed and calorific value of the biogas. On the other hand, the enrichment or combination with the gases having higher combustion speed such as H_2 can significantly improve the combustion characteristics of the biogas engine [5]. Compared with other gaseous fuels, H_2 has high flame propagation speed and widespread flammability limit. The use of H_2 in ICEs is a satisfactory way to improve the combustion of DF engines, enhancing engine performance and combustion characteristics, and reducing carbon-related emissions.

By optimizing specific engine parameters and gas combinations, diesel engines fuelled with biogas or biogas/H_2 blends can demonstrate improved performance and reduced emissions. Barik and Murugan [6, 7] performed experimental studies to investigate the performance, emission, and combustion of the biogas DF diesel engine at different loads, biogas flow rates, and ignition timings. For different biogas substitution rates and ignition timings, in-cylinder pressure, heat release rate, combustion duration, and ignition delay in the biogas fuelled diesel engine enhanced, whereas BTE and exhaust gas temperature decreased. Concerning the emissions, NO, smoke, and CO_2 emissions decreased while HC and CO emissions increased with the flow rates of biogas. They achieved to the best results of the performance and emissions with a particular injection timing before top dead center (TDC) [7]. Shan et al. [4] experimentally investigated the effects of exhaust gas recirculation (EGR) and H_2/CO combinations in the mixture on the emissions in DF biogas engine. Biogas with high H_2 content shortened the ignition delay, increased the combustion pressure and temperature, and improved the engine performance. In their study, an increase in EGR ratio boosted HC and CO emissions but diminished NO_x emissions. Abdul Rahman and Ramesh [8] observed no remarkable improvement in BTE for various CH_4/CO_2 fractions of the biogas. Bora et al. [9] investigated the effects of the compression ratio on the performance, combustion, and emissions of a DF biogas engine at different loads. They reported that hydrocarbon (HC), CO and CO_2 emissions increased with the DF biogas engine; however, cylinder pressure and heat release rate decreased. Kalsi and Subramanian [3] used biodiesel as pilot fuel in a biogas working engine and reported lower BTE when the CO_2 fraction in the biogas increased. They claimed that this was because the heat release rate was decreased, and this probably caused by slowing down the chemical reactions. Yoon and Lee [10] documented that the biodiesel pilot-fuelled DF biogas engine produced lesser CO, NO_x, and HC emissions as well as the shortened combustion duration and ignition delay period. Bougessa et al. [11] showed that H_2 enriched biogas decreased CH_4, HC, and CO emissions in comparison with pure biogas in a diesel engine. Talibi et al. [12] reported that H_2 enrichment slightly influenced on ignition delay and peak heat release rate of DF biogas engine. Verma et al. [13] documented that H_2 enrichment advanced peak cylinder pressure and heat release rate whereas unaffected their peak values. Ahmed et al. [14] revealed H_2 enrichment resulted in a substantial increase in peak heat release rate and cylinder pressure because of rapid combustion in comparison with the biogas operation. In addition, locations of the peak pressure and heat release rate for the H_2 enriched biogas were earlier than those of the biogas. As compared to the biogas operation, Khatri and Khatri [15] obtained higher BTE and lower fuel consumption by H_2 enriched biogas operation, which might be attributed to the higher flame propagation speed and wider flammability limit of H_2. Sharma et al. [16] investigated H_2-rich reformed biogas in a common-rail single cylinder diesel engine. The study showed an increase in BTE by 10.5% and a decrease in brake specific energy consumption by 13.6%. Cylinder pressure and heat release rate peaks were advanced and reduced with reformer gas usage because of lower heating value of the gaseous fuel.

Coating the parts of the engine's combustion chamber generally enhances the engine performance because of decreasing the heat transfer to the cooling system, meaning that the heat energy released by fuel can be converted to work [17]. Parts of the combustion chamber can be insulated by materials with lower thermal conductivity coefficient. The temperature of the chamber wall and combustion gases thus increases, leading to increasing the exhaust losses [18, 19]. On the other hand, the thermal barrier concept in combustion chamber was resulted in higher NO_x emissions due to high gas temperature [20]. Volumetric efficiency (VE) can reduce which is due to a decrease in charge intensity, caused by heat transferred to the intake charge from the combustion chamber walls during intake process [21]. In the literature, it was mostly reported the biodiesel could be used as alternative fuel in low heat rejection (LHR) engines. Numerous researchers documented higher NO_x emissions in the LHR engines fuelled with the biodiesel [17, 22, 23]. HC, CO, and smoke emissions could be decreased with LHR engine [17, 23, 24]. Higher gas temperatures of the LHR engines cause higher NO_x emissions and shorter ignition delay, resulting in a decrease in the pre-mixed combustion phase with an increase in amount of the fuel burned during the controlled combustion phase [25]. Consequently, the heat release can shift toward late combustion phase and hence resulted in lower BTE [26]. Paparao et al. [27, 28] investigated effects of oxy-hydrogen DF mode in coated piston engine. Conventional diesel and biodiesel blend were used as pilot fuel to ignite the mixture. They revealed that BTE was enhanced about 5.9% and CO, HC, and smoke emissions were respectively decreased 44.1%, 46.7, and 21.5% with coated piston engine fuelled with biodiesel-diesel blend, compared to conventional diesel engine. Cylinder pressure and heat release rate increased and ignition delay reduced with dual-fuel mode under the coated piston engine.

As can be seen in the published literature, there are fewer studies on usage the gaseous fuels in LHR engines. It was seen in the current literature that the biodiesel studies were mostly addressed in LHR engines. For that reason, in this study dual-fuel mode under the LHR engine was investigated, and the aim of the study is to reveal the combustion characteristics of the biogas and hydrogen enriched biogas under LHR application. When the standard engine (STD) is converted to LHR engine, results of important combustion parameters (cylinder pressure, heat release rate, gas temperature, knock intensity, combustion duration) and performance parameters are discussed for the conventional diesel, biogas and H_2-biogas modes.

2. Experimental Setup and Test Procedure

2.1 Test procedure

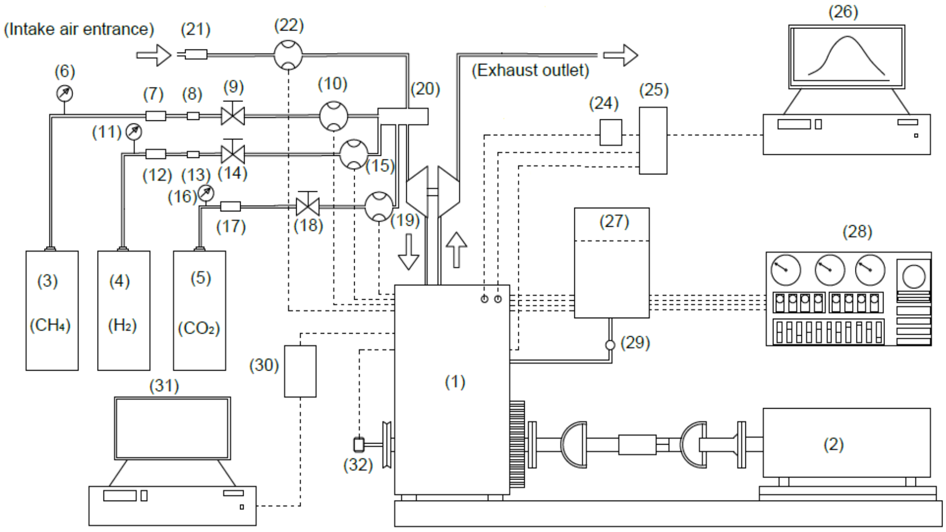
Automotive common-rail diesel engine with a 4-cylinder, 4-stroke, and turbocharger was employed for the experimentation. The engine has two injections stages which are pre-injection and main injection. Technical specifications of the test engine are listed in Table 1. Experimental instrumentation system is illustrated in Fig. 1. An eddy-current dynamometer with water cooling was used to fix the speed and load of the engine. Speed, load, gas flow rates, and temperatures at specific points of the test engine were watched on control panel of the dynamometer. Omega brand K-type thermocouples were used for the temperature measurement of the coolant inlet and outlet, lubrication oil, intake air, exhaust gas, and diesel fuel line. Tests were conducted at three different loads of 50 Nm, 75 Nm and 100 Nm at a fixed speed of 1750 min-1. Diesel fuel consumption was metered by volume with the help of fuel burette of 30 ml and a stopwatch. Air flow rate was measured via a digital mass flow meter. CH4 and H2 gases flow rates were measured by Sierra 5858E model gas flow meters while CO2 gas amount was measured via New Flow TMF model gas flow meter. Compressed CH4, H2, and CO2 gases in high pressured tanks with 50 litre volume were purchased from a commercial gas supplier, which is Hatgrup located in Kocaeli. CH4, H2 and CO2 gases have respectively purities of 99.5%, 99.999% and 99.5%. They were mixed with intake air in a mixing chamber before the engine turbocharger. Flame arresters were used on gas pipeline of CH4 and H2 to prevent backward flame propagation.

Test engine was run about 15 minutes to achieve steady-state conditions with a conventional diesel fuel at turbocharger mode without EGR. Initially, baseline diesel tests were performed at 50 Nm, 75 Nm and 100 Nm loads. Then biogas tests were conducted at the same loads. The load was adjusted by increasing the CH4 and CO2 flow rates. Biogas in this study contains CH4 of 80% and CO2 of 20% by volume. After the biogas tests, the CO2 amount by 50% in the biogas content was replaced with the same amount of the H2 gas. Energy share rate of the pilot diesel fuel was kept constant at about 20% of the total energy rate at a given load. As increasing the engine load, the amount of the pilot fuel supplied was adjusted by taking into consideration the amount of the baseline diesel consumed at the same load.

Table 1. Engine specifications

Engine Type	Four-cylinder, four strokes, in-line, turbocharged
Fuel injection system	Common-rail direct injection
Bore/Stroke (mm/mm)	76 / 80.5
Number of valves	8
Number of injector holes	5
Compression ratio	18.25
Displaced volume (cm^3)	1461
Engine power (kW)	48 (4000 min^{-1})
Firing order & arrangement	1-3-4-2 & ECU controlled

In-cylinder pressure was measured by Optrand pressure transducer fixed on the first cylinder of the engine. It was used a Kubler incremental encoder of 360o to determine the crank shaft angle. Signals collected from the cylinder pressure transducer together with the crank encoder were passed throughout National Instruments data acquisition card and sent to a PC. 200 consecutive pressure data with 1 o CA resolutions were sampled for each test case and averaged to eliminate the effects of cycle-by-cycle variations. Uncertainty of some measured parameters is listed in Table 2.



1.Test engine, 2.Eddy-current dynamometer, 3.CH4 tube, 4. H2 tube, 5. CO2 tube, 6-11-16.Manometers, 7-12-17. Pressure regulators, 8-13.Flame arresters, 9-14-18.Control valves, 10-15-19. Gas flow meters, 20.Mixing chamber, 21.Air filter, 22. Air flow meter, 23. Exhaust , 24.Charge amplifier, 25. Data card, 26. PC for combustion and emissions data, 27.Diesel tank, 28.Dynamometer control, 29. Burette, 30. On board diagnostics card, 31. PC for OBD interface, 32. Crank shaft encoder.

Fig 1. Schematic of experimental instrumentation

For the coating of the pistons, 8% Yttria Stabilized Zirconia was used as a main coat and nickel-aluminium (Ni-Al) as a bond coat. Thickness of main and bond coats is 400 µm and 100 µm, respectively. Atmospheric plasma spray method was used for the coating application. This application was made by a commercial company.

Table 2. Uncertainty of the equipment

Measured quantity	Measurement range	Uncertainty
Dynamometer load	0-150 kW	±0.25%
Cylinder pressure	0-200 bar	±0.5%
Fuel line pressure	0-3000 bar	±0.8%
Crank angle	0-360 °CA	±0.2%
Air flow rate	0-1200 m^3/h	±1%
CH_4 flow rate	0-200 l/min	±1%
H_2 flow rate	0-200 l/min	±1%
Diesel flow rate	0-30 ml/s	±0.5%
Calculated values		
BSEC	MJ/kWh	1.6%
HRR	J/°CA	0.74%

2.2 Calculation method

Evaluation of brake specific energy consumption (BSEC) will be proper if the different fuels are used in the tests [3]. It is a measure of how much chemical energy of the test fuels is consumed to load the engine and is calculated Eq. (1) below,

$$BSEC = \frac{\dot{m}_D LHV_D + \dot{m}_{CH4} LHV_{CH4} + \dot{m}_{H2} LHV_{H2}}{P_b} 3.6 \quad (1)$$

Here, m and LHV specify brake power (kW) of the engine, mass flow rate (kg/s) and lower heating value (kJ/kg) of the fuels, respectively.

Knock intensity is third derivative of the pressure. It is calculated in this study by Eq. (2) [29];

$$dP(\theta) = \frac{\{86 * [P(\theta-4) - P(\theta+4)] + 142 * [P(\theta+3) - P(\theta-3)]}{(1188 * d\theta)} +$$

$$+ \frac{193 * [P(\theta+2) - P(\theta-2)] + 126 * [P(\theta+1) - P(\theta-1)]\}}{(1188 * d\theta)} \quad (2)$$

where, represents the pressure derivative at crank angle, and is the cylinder pressure at crank angle is the prior pressure data and is the subsequent pressure data. is the pressure data interval.

Cumulative heat release (CHR) is used to achieve the findings about burned mass fraction of the fuel. Its calculation is restricted between inlet valve closing (IVC) and exhaust valve opening (EVO). On this basis, CHR is calculated with consecutive accumulation of in Eq. (3).

$$CHR = \int_{IVC}^{EVO} \frac{dQ_{net}}{d\theta} \theta d\theta \quad (3)$$

Average in-cylinder gas temperature (in K) was calculated via Eq. (4) below,

$$T_g = \frac{PV}{mR} \quad (4)$$

where, P and V are in-cylinder pressure and cylinder volume depending on crank angle. m_a and R denote air mass when the valves were closed and universal gas constant in kJ/kgK, respectively.

Volumetric efficiency was calculated by Eq. (5)

$$\eta_v = \frac{2.m_a}{\rho_A v_S N} \quad (5)$$

where m_a denotes air mass flow rate (kg/s). ρ_a, v_s, and N respectively symbolize density of air (kg/s), swept volume (m^3), and engine speed (min^{-1}) [30].

Energy share rates of CH_4 and H_2 fuels are calculated from Eqs. (6 and 7) respectively

$$ESR = \frac{\dot{m}_{CH4} LHV_{CH4}}{\dot{m}_{CH4} LHV_{CH4} + \dot{m}_{H2} LHV_{H2} + \dot{m}_d LHV_d} \quad (6)$$

$$ESR = \frac{\dot{m}_{H2} LHV_{H2}}{\dot{m}_{CH4} LHV_{CH4} + \dot{m}_{H2} LHV_{H2} + \dot{m}_d LHV_d} \quad (7)$$

Table 3 represents ESR results for H_2 enriched biogas studies at all tests.

Table 3. ESR values for H2-biogas operation under test loads and both engines

	Diesel	Biogas	H_2
	STD engine		
50 Nm	21.1%	76.5%	2.4%
75 Nm	22%	75.5%	2.5%
100 Nm	23%	74.5%	2.5%
	LHR engine		
50 Nm	19.8%	77.5%	2.7%
75 Nm	21%	76.2%	2.8%
100 Nm	20%	77.3%	2.7%

In Fig. 2, it is shown sample pattern of several combustion curves, including cylinder pressure, cumulative heat release, knock intensity, and fuel line pressure. The pre-injection phase of fuel injection system is at about 320 °CA, main injection phase is close to TDC. Cylinder pressure and CHR then rapidly enhance. CHR profile presents mass burnt fraction from 0 (start of combustion) to 1 (end of combustion). Essential combustion duration (CA10-90) is determined as the crank angle interval between 10% and 90% defined on the CHR profile [6, 10]. Knock intensity has two positive peaks and one negative peak. The lowest negative point in knock intensity profile is of critical importance to define the knock indicator during the combustion process.

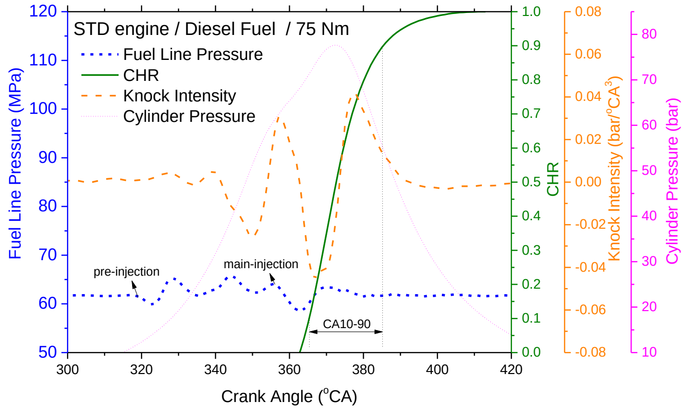


Fig. 2. Sample illustration of fuel line pressure, cylinder pressure, knock intensity, and CHR

3. Results and discussions

3.1. Cylinder pressures and heat release rates

Variations of cylinder pressures and HRRs as a function of crank angle for diesel, biogas and H_2-biogas mixture under STD and LHR engines at 75 Nm load are shown in Fig. 3. Also, maximum cylinder pressure (P_{max}) values and their corresponding crank angle locations under different test cases are seen in Fig. 4. It was firstly observed higher cylinder pressure and HRR peaks with the LHR engine compared to the STD engine. When the standard pistons were replaced with the coated pistons, P_{max} enhanced from 77.6 bar to 83.9 bar with the diesel fuel, and from 101.6 bar to 107.7 bar with the biogas operation, and from 101.8 bar to 125.2 bar with the H_2 enriched biogas operation at medium load of 75 Nm. Corresponding peak HRR values were enhanced from 32.3 J/°CA to 42.5 J/°CA with the conventional diesel, and from 28.3 J/°CA to 31.3 J/°CA with the biogas operation, and from 29.01 J/°CA to 44.5 J/°CA with the H_2 enriched biogas operation, as operating the LHR engine. Higher temperature content with coating pistons in the cylinders promote to enhance both cylinder pressure and heat release rate [31, 32]. Consequently, cylinder pressure and HRR peaks with the LHR engine in a constant load were notably increased compared to the STD engine.

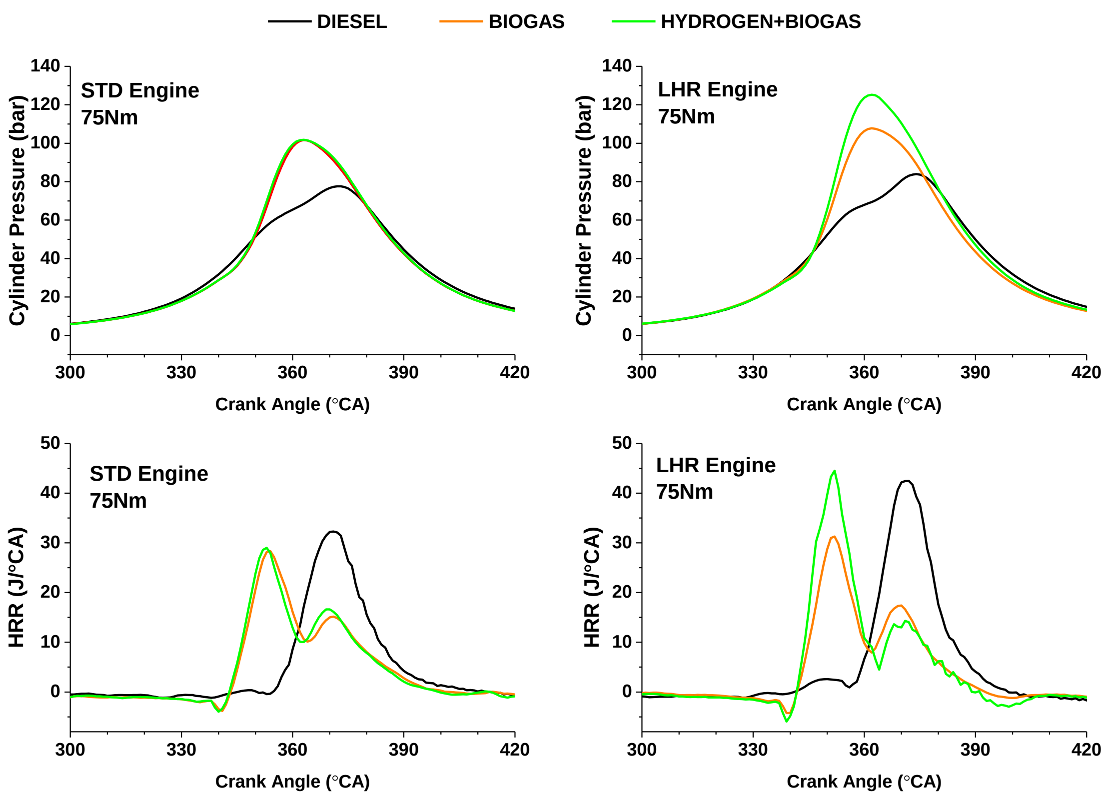


Fig. 3. Cylinder pressure and heat release rates of the fuels for both engines

Combustion was quite different for DF modes and was generated both higher and earlier P_{max} with DF modes, as seen in Figs. 3. and 4. In both engine types, P_{max} values were obtained between 360-366 °CA with the DF combustion, and between 371-374 °CA with the conventional diesel combustion. In operation with the gaseous fuels, the ignition of the homogeneous air-fuel mixture is triggered by the pre-ignition stage of the common-rail injection system. The pre-ignition stage was further earlier than the main injection stage in this test engine [33], as observed in fuel line pressure in Fig.2 and HRR profile in Fig. 3. This causes earlier starting of the combustion with DF modes compared to conventional diesel. Combustion process with DF operations thus takes place shifted and advanced towards the compression stroke. During the ignition delay period, it was appeared a sharp descent in HRR profile and lower cylinder pressure during compression stroke with the DF modes due to having higher specific heat capacities of the gaseous fuels and absorbing the heat of the charge with the injected diesel [34]. In contrast to diesel combustion, pre-mixed combustion phase of the DF combustion is rapidly completed with a higher first peak of HRR and then observed a lower second peak of HRR, which is a consequence of pilot fuel addition with the main injection stage. In diesel mode, less effective pre-mixed combustion phase shows relatively a lower peak of HRR and then a stronger diffusion combustion phase depicts a higher peak of HRR. H_2 introduced mixture showed higher P_{max} and HRR peak especially under LHR engine; however, their peak locations remained nearly constant. H_2 usually causes to faster flame velocity and immense flammability, promoting the pre-mixed burning phase of the mixture and thus increasing the HRR and cylinder pressure of the H_2 enriched biogas mixture [34].

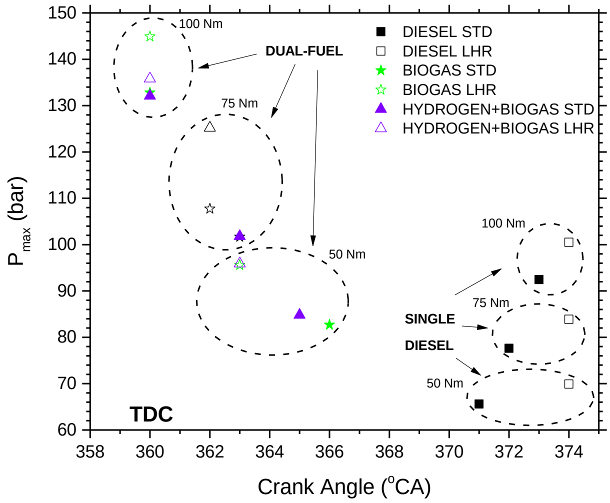


Fig. 4. Maximum pressure values and locations of the fuels at different conditions

3.2. *Gas Temperature*

Fig. 5. depicts gas temperature variations with crank angle for all fuel types, load ranges, and engine coating cases. In conventional diesel mode, gas temperature values at 50 Nm, 75 Nm, and 100 Nm loads were respectively detected as almost 1763 K, 1997 K, and 2264 K under STD engine operation whereas 1978 K, 2252 K, and 2463 K under LHR engine. As changing the STD engine to LHR engine, gas temperature values for conventional diesel, biogas, and H_2-biogas operations respectively increased by 12.2%, 4.8%, and 12.8% at 50 Nm, 12.7%, 2.9%, and 10.5% at 75 Nm, 8.8%, 8.2%, and 1.05% at 100 Nm. Gas temperature for a given fuel significantly enhanced with the elevated load, and the LHR engine showed the highest gas temperatures for all fuels. This is clearly consequence of a decrease in the heat loss to the piston surface [19]. Furthermore, under a constant load and engine type, DF operations showed relatively earlier and lower gas temperature peaks compared to corresponding conventional diesel operation, which are in good agreement with the early discussed cylinder pressure and heat release results. CO_2 presence in the DF mixtures shows inert gas properties, which partly absorbs the heat energy of the mixture and thus promoting lower burning gas temperatures with the DF modes [12].

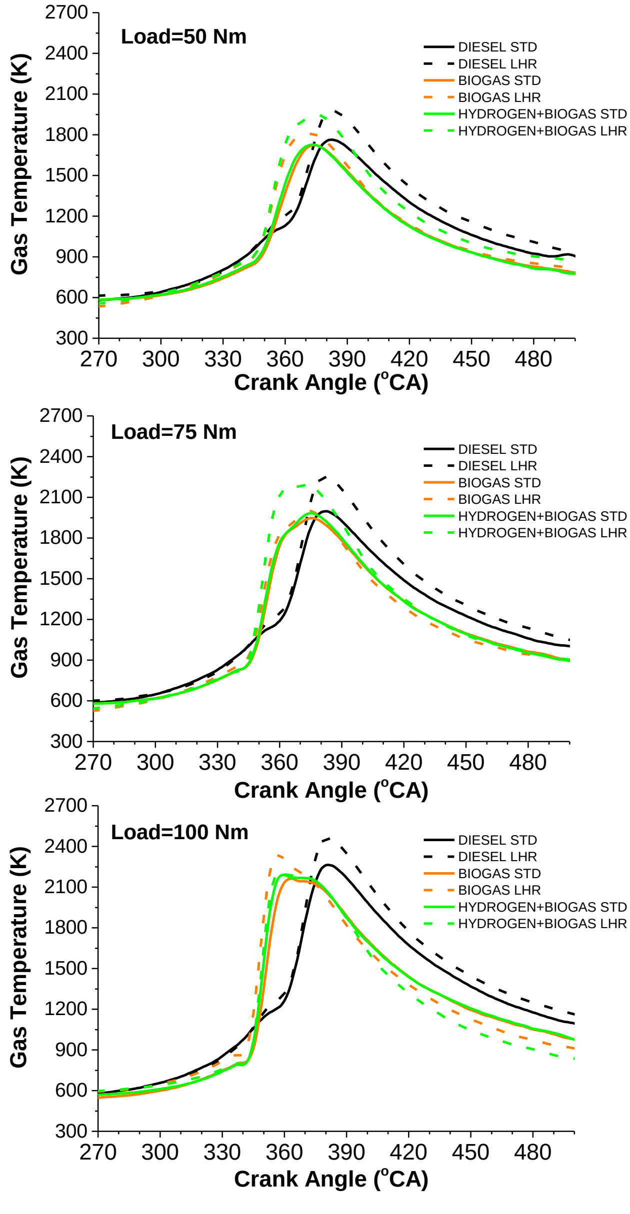


Fig. 5. Temperature variations for all fuels at different conditions

3.3. *Cumulative heat release*

CHR variations for the test fuels under coated and uncoated engines at different loads are depicted in Fig. 6. It is clear from the figure that CHRs of the DF operations show faster initial stage of the combustion due to the homogeneous gaseous fuel-air mixture ignited by earlier injection of the pilot diesel. In diesel fuel mode, the energy need of the engine is met by main injection while pre-injection is just made to shorten the ignition delay thus resulting in lower NO_x emissions and diesel noise. LHR engine clearly shows earlier combustion phase for all fuels compared to STD engine. With LHR engine, burnt fuel amount at a given ºCA is more than that with STD engine. This is because LHR engine operates with higher temperatures thus advancing the combustion phase. As increasing the load, faster pre-mixed combustion phase was seen with DF modes. It can be concluded from the figure that approximately 65-80% of the air-fuel mixture is burned around TDC at 100 Nm with DF modes. Moreover, the higher the engine load, the earlier start of combustion for all fuels. This is due to the fact that the flame speed and vigorous burning of mass fraction of the fuels are boosted [2].

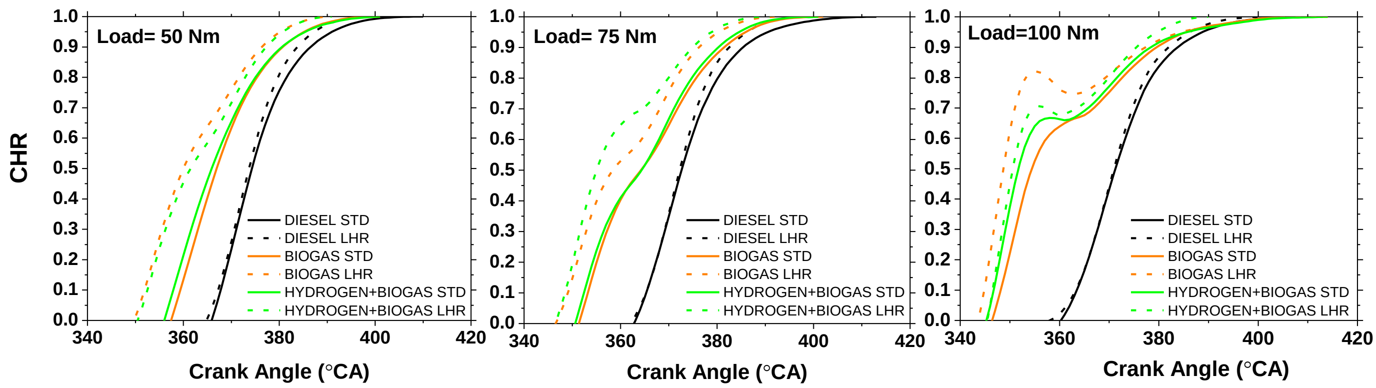


Fig. 6. CHR for all fuels at different conditions

3.4. Knock intensity

In Fig. 7, variations of the knock intensity with respect to the crank angle are shown for all tests. Knock intensity is the third derivation of in-cylinder pressure and has two positive peaks and one negative peak, as shown in the graphics below. It is noteworthy that the knock detection is based on the negative side in the slope of knock intensity [29]. A large negative value in the slope of knock intensity is an indicator defining the degree of abrupt pressure rise [35]. The narrow pressure region is generally related to end gas auto-ignition. Values presented in each graphic indicate degree of abrupt pressure rise. It was apparent that the knock indicator was always larger in LHR engine for both diesel fuel and dual-fuel modes at all test loads. When the STD engine was converted to LHR engine, the knock intensity for the conventional diesel, biogas, and H_2 enriched biogas were respectively increased by 74.4%, 58.7%, and 117.2% at 50 Nm, 75.5%, 20.5%, and 22.7% at 75 Nm, 24.3%, 81.8%, and 33.3% at 100 Nm. High pressure oscillations lead to increase the heat losses in the LHR engine [36, 37] and this would undoubtedly cause the engine performance to deteriorate. As the load was increased, it was also appeared that the knock intensity was enhanced since more fuels were burned. Hence heat release increased, providing that the pressure oscillations were improved [38]. With the DF modes, it was indicated that the knock intensity significantly increased, it will be relatively resulted in worse combustion efficiency and higher heat losses. Erdogan et al. [39] reported that the knock intensity of high viscosity biodiesel started earlier and had higher amplitude. Similarly, Okçu et al. [40] documented that higher knock intensity occurred with biodiesel operation in a reactivity-controlled compression ignition engine. The authors also revealed that pressure rise rate trend were well consistent with the knock intensity trend. It was clear that DF modes increased knock intensity, and H_2 addition mostly resulted in higher knock intensity. Earlier start of combustion and faster pre-mixed combustion phase with DF modes promoted higher knock intensity therefore the combustion phases became were advanced and the pressure oscillations were enhanced.

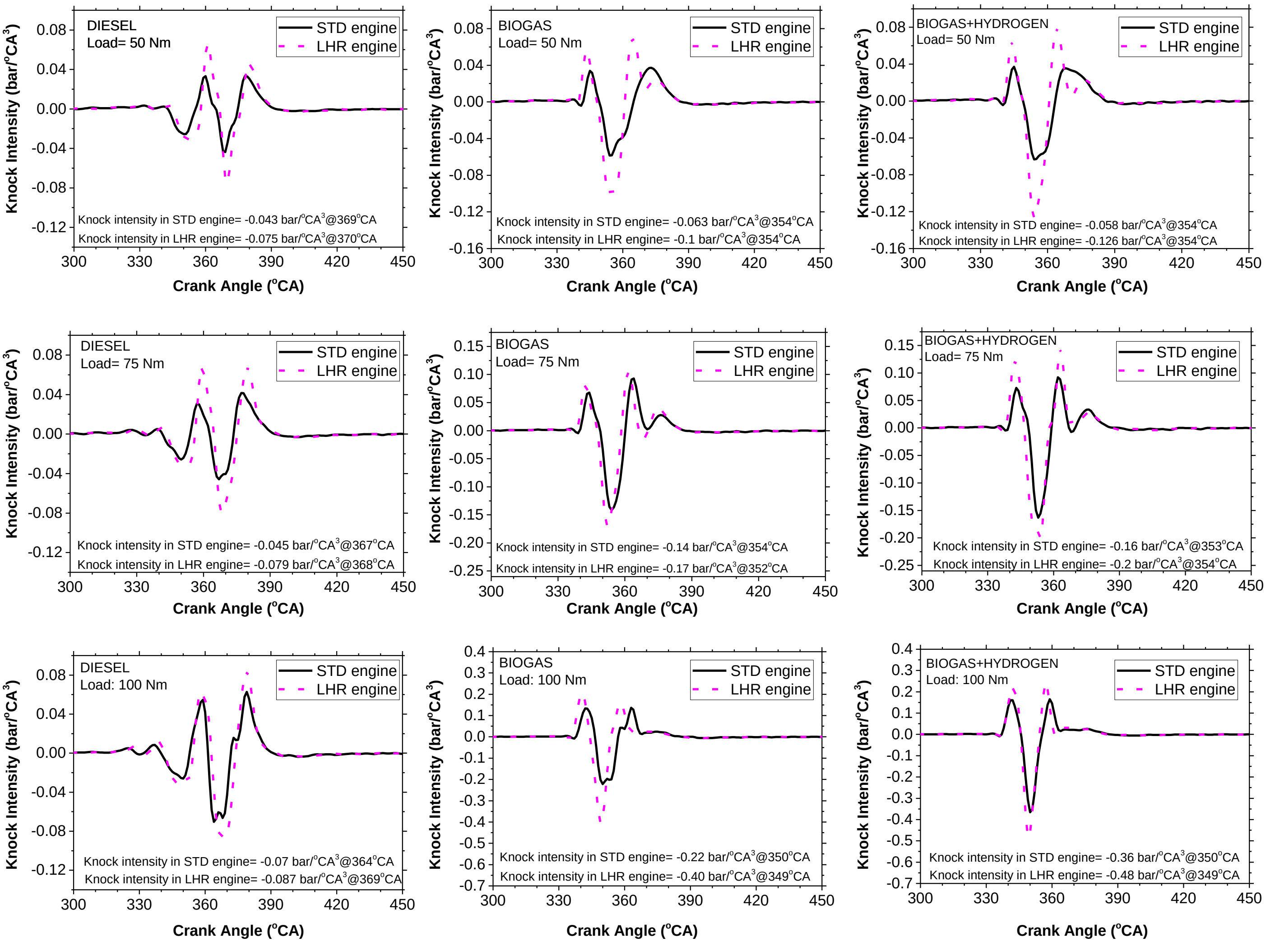


Fig. 7. Knock intensity versus oCA of the test fuels at various loads for STD and LHR engines

3.5. *Duration of combustion*

CA10-90 interval in CHR profile presents combustion duration in this study, as previously depicted in Fig. 2. The combustion duration is important because it impacts on heat transfer to the chamber walls, indicated engine efficiency, fuel economy, tendency to knock and exhaust emissions [41]. Fig. 8 illustrates combustion duration for diesel and gaseous fuels under both engines at different loads. It was seen that CA10-90 duration increased as the engine load was increased. This is due to a further increase in amount of the supplied fuels. It was also observed from the figure that CA10-90 of the DF modes was more (1-15 °CA) than that of the baseline diesel. This difference is more obvious at high load for both STD and LHR engines. CA10-90 duration was respectively increased from 20.3 °CA with diesel fuel to 31.5 °CA and 31.7 °CA with biogas and H_2 enriched biogas under STD engine, and from 18.7 °CA with diesel fuel to 32.5 °CA and 28.9 °CA with biogas and H_2 enriched biogas under LHR engine. The decrease in intake oxygen because of substituting the gaseous fuels with the intake air leads to a prolonging in combustion duration of DF modes, especially observing a remarkable increase in the end-burning phase [7]. Slower rate of burning with the biogas and weak combustion of the main-injected pilot diesel during the diffusion phase due to lack of oxygen additionally contribute to increase the CA10-90 duration with DF modes [10]. The maximum CA10-90 was found 32.5 °CA under LHR engine with biogas mode at 100 Nm load. CO_2 presence in the biogas promotes the CA10-90 duration to extend because of slowing down the burning process [42].

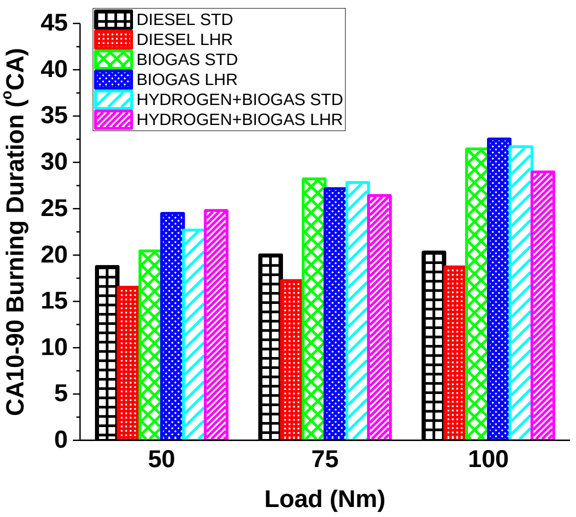


Fig. 8. CA10-90 for all fuels at different loads in STD and LHR engines

H_2 presence in the mixture influenced slightly on combustion duration and occurred unstable variations in combustion duration. In general, H_2 addition accelerates the combustion process and causing shorter CA10-90 duration. Accelerating effect of H_2 addition on combustion phase of the biogas was noticeable at medium and high loads. At low load of 50 Nm, leaner mixture played an important role in longer CA10-90 duration and consequently CA10-90 increased from 20.4 °CA with biogas to 22.7 °CA with H_2 addition in biogas under STD engine, and from 24.5 °CA with biogas mode to 24.8 °CA with H_2 addition mode under LHR engine. Concerning the coating effect, it was seen CA10-90 duration steadily decreased with single diesel operation at all loads and with dual-fuel modes at medium load of 75 Nm in LHR engine. Consistent heat release behaviour at medium load and dependable burning behaviour of the diesel fuel at all loads provided reasonable results of CA10-90. These results are also consistent with the literature studies [7, 41].

3.6. Performance characteristics (brake specific energy consumption, volumetric efficiency, and exhaust gas temperature)

Performance of the ICEs depends on compression ratio, heat transfer to combustion chamber, combustion efficiency, VE, duration and phase of the combustion. Fig. 9a-c shows BSEC, VE, and exhaust gas temperature variations of diesel and DF operations at different loads under STD and LHR engines. When the engine was run on the DF modes at both types of the engines, it was noticed a significant increase in BSEC, suggesting lower conversion of the energy to useful work with dual-fuel combustion strategy. When the engine operation was converted from conventional diesel to the biogas DF mode, BSEC increased from 10.6 MJ/kWh to 15.05 MJ/kWh at 50 Nm, from 9.5 MJ/kWh to 12.7 MJ/kWh at 75 Nm, and from 10.06 MJ/kWh to 12.9 MJ/kWh at 100 Nm under the STD engine, whereas under the LHR engine BSEC increased from 10.6 MJ/kWh to 16.36 MJ/kWh at 50 Nm, from 9.3 MJ/kWh to 13.1 MJ/kWh at 75 Nm, and from 9.8 MJ/kWh to 14.7 MJ/kWh at 100 Nm. Lower combustion efficiency, higher knock intensity, prolonged combustion duration, lower VE, and worse ignition characteristics with the DF modes are other responsible factors for higher BSECs [7]. Gatts et al. [43] reported that BTE for the H_2-diesel DF mode had a similar trend with the combustion efficiency of H_2 in a multi-cylinder diesel engine. They documented that combustion efficiency of conventional diesel is over 99.5%. Clearly, in case of the diesel fuel operation, spray plume does not reach the cylinder walls, and the combustion is restricted within the piston bowl. Surrounding air acts as semi-insulator between burning gases and the piston bowl, meaning lesser heat loss to the combustion chamber walls and thus the best engine performance with single diesel mode [444]. Besides, DF modes showed longer combustion duration (see Fig. 8) leading to more heat losses to the combustion chamber walls, suggesting negative contribution to the engine performance [45]. Lower flame propagation speed and burning gas temperature, and enhanced negative compression work, caused by the induction of a large quantity of air-gas fuel mixture, are associated with the higher BSEC results with DF modes [10]. Moreover, it is well known that DF mode suffers from larger unburned HC emissions at especially partial loads [3, 10, 46], leading to lower fuel conversion efficiency. Totally, the engine performance of DF modes was affected toward worsening side.

As seen in Fig. 9a, LHR engine with the gaseous fuels gives clearly higher BSEC in comparison with STD engine cases. It is thought that this was caused by excessive knock intensity (see Fig. 7), resulted in higher heat transfer and thus less combustion heat. However, it was documented that LHR concept was beneficial to improve thermal efficiency in conventional diesel fuel mode [27]. This can be possible by higher combustion temperature and thus more efficiently combustion of introduced fuels.

With respect to the engine load, the best BSEC for all the fuels is achieved at medium load of 75 Nm. As the engine load is increased, the pressure of the diesel spray enhances and diesel droplet size decreases, and mixture strength enhances [46]. All these contribute to some improvement in the combustion efficiency and fuel conversion efficiency. However, more increasing in the engine torque causes to further increase in the knock intensity. It results in higher heat loss to the combustion chamber walls [37].

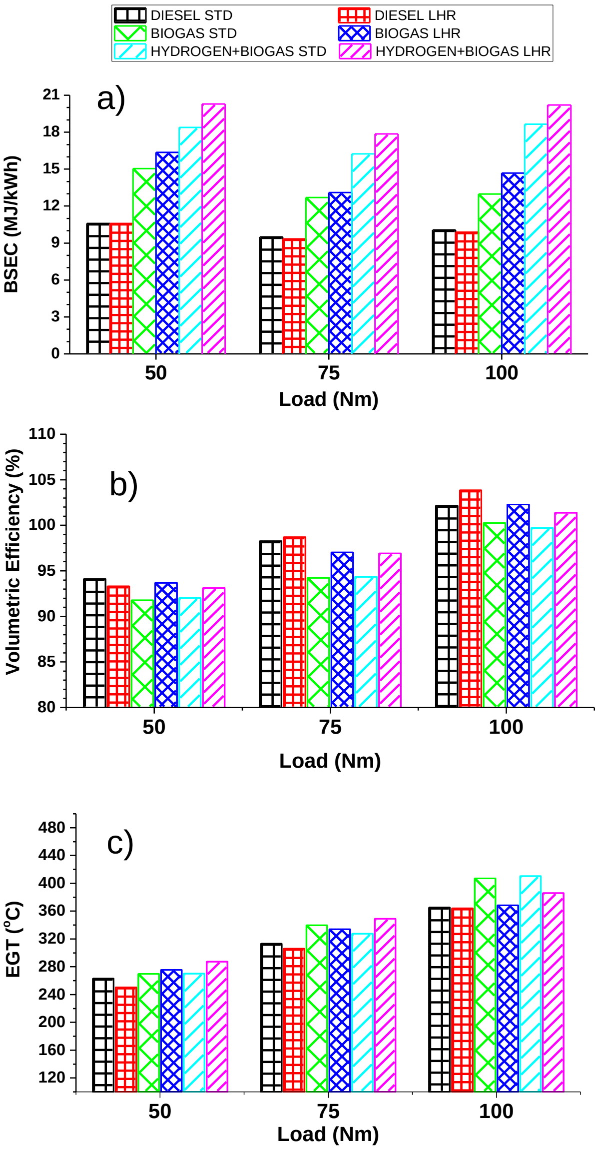


Fig. 9. Performance parameters for all fuels at different loads for both engines, a) BSEC, b) VE, c) EGT

H_2 presence in biogas resulted in higher BSEC values, as shown in Fig.9a. When the biogas DF mode was enriched with H_2 gas, BSEC increased by 22.2% at 50 Nm, 27.9% at 75 Nm, and 43.7% at 50 Nm under STD engine, and by 23.9% at 50 Nm, 36.3%, and 22.7% at 75 Nm, 37.7% at 100 Nm under LHR engine. It is considered that the increase in the knock intensity (see Fig.7) with the H_2 addition is main cause to higher BSEC, namely it leads to enhance the heat transfer and thus increasing the energy consumption needed to obtain same torque [36, 37]. BSEC increasing with H_2 presence can be also associated with several reasons, which are decrease in the volumetric calorific value of the mixture and shorter quenching distance on the walls with the H_2 presence [44, 48]. Another possible reason for higher BSEC is that H_2 rapidly consumes the oxygen during the combustion process. It leads to be deteriorated the combustion process together with the CO_2 presence and thus the overall performance of the H_2 enriched biogas engine to lower [48].

Fig 9b depicts variations in VE at all test cases. DF modes reduced the VE due to replacing the intake air with the gaseous fuels. Reduction in VE with DF modes was about 2.3% at 50 Nm, 4% at 75 Nm, and 1.9% at 100 Nm for the STD engine whereas 0.1% at 50 Nm, 1.7% at 75 Nm, and 1.6% at 100 Nm for the LHR engine. It was recognized that the VE noticeably increased with the LHR engine for all fuels under all loads. This increase can be attributed to the turbocharger existence on the engine. In the natural aspirated diesel engines, the coating helps increase the wall temperatures, which reduces the amount of the fresh air and thus resulted in lower VE together with BTE [20]. Besides, VE of all fuels followed an increase trend with the engine load, which can be ascribed to the increasing exhaust gas energy, providing an improvement in the turbocharger performance and the amount of the intake charge [26].

Fig. 9c indicates exhaust gas temperature (EGT) variations with different fuel combinations and engine loads in the coated and uncoated engines. EGT was detected in range of 250-287 °C at 50 Nm, 305-348 °C at 75 Nm, and 364-410 °C at 100 Nm for all fuel combinations and piston coating cases. From these values, it was clear to increase EGT with increasing the engine load for all fuels under both engines due to the increase of total energy input [10]. For the conventional diesel fuel, as the engine load was enhanced from the lowest to the highest one, EGT values enhanced from 263 to 365 °C for the STD engine whereas from 250 to 364 °C for the LHR engine. Similarly, at high load for all fuel types, less EGT values were recorded with LHR engine. This is an inconsistent result with the gas temperature values in LHR engine. Probably it is caused from combined effects of burning behaviour of the gaseous fuels and pilot fuel at high load condition. Also, with enhanced exhaust energy within LHR engine at higher loads, the turbocharger existence may convey more fresh air together with the gaseous fuels into the engine, leading the variance in intake conditions and burning gas temperature. In the literature, Civiniz et al. [18] and Özer et al. [49] reported higher EGT values by using LHR application to the test engines due to increase in burning gas temperature. Coating concept on the chamber walls generally leads to lower heat transfer to the coolant system, and thereby increasing the accumulated heat in the cylinders due to altering the system towards the adiabatic process. In operation with the gaseous fuels, higher EGT values were detected compared to the diesel fuel for both engines. This is closely consistent with the higher BSEC findings with DF modes. It was apparent that the fuel energy unconverted to the useful work with DF operations was resulted in higher EGT values. It was appeared that the EGT variations for the gaseous fuel mixtures were inconsistent with each other at various load cases. For instance, with the biogas mode, EGT under the LHR engine was slightly higher about 6 °C at 50 Nm load, but lower 6 °C and 39 °C at 75 Nm and 100 Nm loads respectively. Regarding the effect of H_2 addition, EGT values enhanced by 4.3%, 4.4%, and 4.8 % with the H_2 enriched biogas modes at respective loads of 50 Nm, 75 Nm, and 100 Nm, compared to the biogas DF modes under the LHR engine. In the STD engine, EGT increased by 0.1 % at low load and 0.8% at high load but reduced by 3.7% at medium load.

4. Conclusions

As alternative gaseous fuels, biogas and H_2-introduced biogas were run in LHR and STD diesel engines. Combustion and performance characteristics of the dual-fuel diesel engine were investigated in this study. Results were evaluated at different loads, fuel types, and coating cases. Main conclusions obtained from this study are summarized below.

- In-cylinder pressures and their locations under both DF operations and LHR application were always higher and earlier than those of the diesel mode. When the STD engine was changed to LHR engine, P_{max} was respectively increased by 8.1%, 6%, and 23% with diesel, biogas, and the H_2 enriched biogas modes, while peak HRR was respectively enhanced by 31.6%, 10.6%, and 53,4% with diesel, biogas, and H_2 enriched biogas modes at medium load.
- Peak gas temperature noticeably increased with the LHR engine. By changing the STD engine to LHR engine, peak gas temperatures for the conventional diesel, biogas, and H_2-biogas operations were respectively increased by 12.2%, 4.8%, and 12.8% at 50 Nm, 12.7%, 2.9%, and 10.5 at 75 Nm, 8.8%, 8.2%, and 1.05 at 100 Nm. The peak gas temperature with DF modes was reduced and shifted to earlier locations.
- Knock intensity for a fuel type was higher in LHR engine. When the STD engine was used instead of LHR engine, the knock intensity for the conventional diesel, biogas, and H_2 enriched biogas were respectively increased by 74.4%, 58.7%, and 117.2% at 50 Nm, 75.5%, 20.5%, and 22.7% at 75 Nm, 24.3%, 81.8%, and 33.3% at 100 Nm.
- HRR peak with DF modes was earlier than conventional diesel. Earlier CHR with the gaseous fuels compared to the conventional diesel fuel was main reason for higher cylinder pressure, gas temperature, and knock intensity. Pre-mixed combustion with DF mode enhanced, and the combustion center became closer to TDC.
- CA10-90 duration with DF modes was higher than that of the conventional diesel. It prolonged with increasing the engine load for all fuels under both engines, and partly responsible to worse engine performance.
- Compared to those of the diesel operation, BSEC with the DF modes increased in both engines. For both LHR and STD engines, H_2 introduction led to additional increase in BSEC. Coating the pistons showed a negative effect on BSEC. For all test fuels, the best BSEC results were obtained at the intermediate load of 75 Nm.
- With DF modes, it was founded higher EGT values whereas lower VE results compared to diesel fuel. Under LHR engine, while EGT decreased for diesel fuel at all loads, it increased at low load for DF modes and decreased at high load. It was clear that the fuel energy transformed to the useful work in operation with DF operations was resulted in higher EGT values. VE was generally lower with the DF modes and improved in LHR engine with the help of the turbocharger.

The experimental study demonstrated that the common-rail diesel engine can be operated with the coated pistons. However, the dual-fuel operations in the engine need to be further investigated in order to improve the overall performance.

Conflict of Interest Statement

The authors declare that there is no conflict of interest in the study.

CRediT Author Statement

Ali Şanlı: Conceptualization, Writing, Editing, Data curation.
Ilker T. Yılmaz: Conceptualization, Experimentation, Editing.
Metin Gümüş: Conceptualization, Supervision, Editing, Validation.

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