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Research on Fuel Efficiency and Emissions of Converted Diesel Engine with Conventional Fuel Injection System for Operation on Natural Gas

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Abstract: This paper presents a study on the energy efficiency and emissions of a converted high-revolution bore 79.5 mm/stroke 95 mm engine with a conventional fuel injection system for operation with dual fuel feed: diesel (D) and natural gas (NG). The part of NG energy increase in the dual fuel is related to a significant deterioration in energy efficiency (η_i), particularly when engine operation is in low load modes and was determined to be below 40% of maximum continuous rating. The effectiveness of the D injection timing optimisation was established in high engine load modes within the range of a co-combustion ratio of $\text{NG} \leq 0.4$: with an increase in η_i , compared to D, the emissions of $\text{NO}_x + \text{HC}$ decreased by 15% to 25%, while those of CO_2 decreased by 8% to 16%; the six-fold CO emission increase, up to 6 g/kWh, was unregulated. By referencing the indicated process characteristics of the established NG phase elongation in the expansion stroke, the combustion time increase as well as the associated decrease in the cylinder excess air ratio (α) are possible reasons for the increase in the incomplete combustion product emission.

Keywords: compression ignition engine; conventional fuel injection system; natural gas; energy and emission indicators; fuel injection phase

1. Introduction

When comparing autonomous heat engines, modern compression ignition engines (CIEs) are characterised by the highest energy efficiency [1]. This is one of the most important operational indicators, and has a significant influence on the extensive use of these engines in transportation, households, and non-road machines. According to greenhouse gas emission statistics data, CIEs constitute the following parts of the total transport energy balance: 72.1% of the road vehicle sector, 13.6% of marine transportation, and 0.5% of rail transport [2]. Although numerous CIE processes involve heterogeneous fuel mixture combustion across a large temperature inconsistency (at local temperatures up to 2500 to 3000 K and higher, with an air–fuel equivalent ratio up to 2 to 4 units), a relatively high level of harmful component emission is generated in exhaust gas [1,3,4]. Among the harmful exhaust components, the most dangerous to people’s health and fauna are nitrous oxides and particulate matter (PM), which mainly consist of soot [1]. Soot particles of 10 μm and less may cause airway diseases such as mesothelioma [3]. CIEs that burn fuel containing sulphur can release sulphuric oxides that are the main cause of acid rain and can cause damage flora [5].

Therefore, research on the reduction of harmful emissions into the atmosphere as well as CIE standardisation has become prevalent during recent decades (for example, EN590, EPA, 2012). One of the most widespread technologies is the conversion of CIE for dual-fuel operation and the use of petroleum-derived fuel or either compressed natural gas (CNG) or liquefied natural gas [6–11]. The introduction of NG into the cylinder can be achieved with either high-pressure injection or in combination with liquid fuel spray or carburation with air into air inlet manifold. Owing to the inferior NG auto-ignition properties, the fuel mixture may be ignited with electrical discharge, as with Otto engines [12–14], or by using a pilot fuel portion or high reaction fuel (HRF) [15]. The second method for igniting NG is used most frequently, owing to the easy technological realisation of converting CIE to work with NG, without any need for substantial changes in the CIE structure, while the engine can continue to operate on liquid fuel only [16].

These types of engines, also known as dual-fuel engines, can provide significant improvements in efficiency and emissions [16–25]. For example, in research on a dual-fuel cargo truck engine [26] a decrease in NO_x of up to six times and decrease in CO of up to 83% were observed compared to diesel only operation. The MARPOL 73/79 VI annex standard Tier III norms were achieved in a Wärtsillä company average revolution 20DF ship engine when the engine operated with NG fuel feed, without using secondary emission reduction technologies (such as selective catalytic reduction technology) [24]. Compared to petroleum-based fuel, at least on a theoretical basis, the NG chemical composition contains a carbon/hydrogen ratio that is more favourable for decreasing CO_2 emissions responsible for the greenhouse effect by a quarter.

However, serious drawbacks exist when converting a functioning CIE for dual-fuel operation. When using NG greenhouse gas CO_2 emissions are reduced because NG, when compared to diesel, has a smaller proportion of carbon atoms, and also the calorific value of NG is 15–25% higher. Moreover, even when there is an increase in HC emissions from using NG, the overall impact on the greenhouse effect is low, when compared to using diesel only [26]. Also, operation during low load modes results in instability, leading to a reduction in efficiency indicators, and during nominal power load operation and with a large gas fraction of 90% to 95% in the fuel balance, this may result in knocking [24,27]. To solve these problems, motor methods are generally used: adjustment of the compression ratio and combustion chamber optimisation [10,28,29], air vortex movement increase in the cylinder [28], and exhaust gas recirculation (EGR) technology [24,26], among others. Research results attest to the conclusion that depending on the operating conditions, the use of NG can increase [24] as well as reduce instances of engine knocking phenomena [27]. For example, in numerical research [24] combustion pressure high frequency pulsation manifests at early and late (by retarding injection timing more than 40 CAD BTDC) pilot fuel injection timing. While at the same time at injection timing from 25 to 40 CAD BTDC pulsations were observed to be minimal. Authors of this publication used a CIE with a conventional fuel injection system. NG had a positive effect and led to a reduction of high frequency pressure amplitude.

HRF characteristic optimisation is a prevalent method, which includes the pilot fuel portion phase, pressure, injection law change, and multiphase injection [30]. The appeal of this method is related to its relative simplicity, which is relevant for operational CIE conversion, with regulation flexibility across a wide speed and load range, and importantly, enables improvements in engine energy efficiency and emission parameters. The majority of research in this area has been based on complex experimental investigation and internal cylinder mathematical modelling, with respect to the injection phase influence on the operation mixture combustion dynamics. For example, in papers [24,25,30–32], the physical mechanism and factors leading to the diesel and gas–environment air engine cylinder chemical kinetics as well as process cylinder dynamics were investigated.

In [30], a one-cylinder (bore 137.2 mm/stroke 105.1 mm) engine with a common rail (CR) fuel injection accumulative system was investigated, and two separate operational mixture combustion physical mechanisms were established, depending on the HRF. The research was conducted on a 25% partial load and partial part gas phase, with an energy value of co-combustion ratio of NG (CCR NG)

of approximately 75%. During the late diesel injection time (DIT), which does not exceed 30 °CA before top dead centre (BTDC), auto-ignition and combustion dynamics were observed, specific to the CIE cycle: OH radicals were grouped closely with HRF torch, initiating intensive combustion in accordance with the kinetic mechanism, with a further transition to flame diffusion transfer into the peripheral combustion bowl zones. Therefore, a relatively small increase in the induction period can be observed by advancing the DIT by 4 to 5 °CA BTDC, and the combustion process is postponed into earlier crankshaft angles; the maximum pressure P_{max} and maximum combustion temperature T_{max} increase, which leads to superior NO_x emission and efficiency (η_e). The investigated DIT range of 30 to 50 °CA BTDC operational mixture thermodynamics becomes insufficient for rapid HRF combustion. The double increase in the induction period surpasses the HRF increase, bringing the combustion process back to TDC. The active auto-ignition OH centres include a significant part of the combustion chamber (CC) volume, and the fuel gas–air mixture combustion becomes single phase. The local temperature field in the CC is equalised, thereby reducing the NO₂ emissions. Moreover, the reduction not encompassed by the peripheral combustion zones in the CC also leads to a decrease in the emissions of the incomplete combustion products CO and PM.

Analogous results were found in the DIT range between 50 °CA BTDC and 5 °CA after top dead centre (ATDC) in a high-revolution (bore 85 mm/stroke 90 mm) engine with a common rail system [1]. Furthermore, the influence of the DIT on the energy efficiency and emissions was established. The authors of the paper have expanded their research by changing the HRF injection pressure in the range of 500 to 1000 bar, optimising the DIT speed, and separating the HRF into component parts, in combination with pressure optimisation. It was established that HRF injection during the early phases is an effective method for improving dual-fuel engine energy efficiency and emissions. A maritime purpose single-cylinder average rpm CIE, Wärtsillä 20DF (bore 200 mm/stroke 280 mm), with a CR system, was also studied [24].

This research was conducted to establish the HRF distribution and structure, as well as influence of the thermodynamic properties in a compression process on the HRF auto-ignition and fuel mixture combustion process. The experimental research was conducted with medium, nominal engine loads at constant revolutions; the HRF injection pressure was adjusted within the range of 1300 to 2100 bar; the DIT ranged from 15 to 50 °CA BTDC. It was established that, in the DIT range of up to 30 °CA BTDC, the HRF combustion reaction potency, as assessed by the localised HRF equivalence ratio, was high, which is in accordance with the research results. A predominant β value range of 0.2 to 0.8 determines the comparatively short ignition delay of 10 to 17 °CA BTDC and intense combustion. Later, the DIT is accompanied by a noticeable $\beta = 0.2$ to 0.8 part reduction, causing a homogenous operational mixture combustion close to the TDC.

Combustion obtains “soft” characteristics, equalising the localised temperature fields and reducing emissions. Moreover, it was established that the fuel mixture thermodynamic parameter changes in the cylinder (by advancing the inlet valve closing phase) are no less effective than the increased HRF ignition delay period. It is important to note that, during this event, the DIT range is significantly reduced: in the conducted experiment from 45 °CA BTDC up to 32 °CA BTDC, emissions were reduced to or below the MARPOL 73/78 Tier III standard established values. Similar results were obtained in an experiment with a six-cylinder engine (Bore 11.2 mm/Stroke 13.2 mm) [33] with a CR system, by advancing the DIT up to 32 °CA BTDC (CCR NG 90%), and low load mode emissions were reduced. As opposed to the experimental results from [23,27,28,30,34], in which engine parameter deterioration resulted in an increase in hydrocarbon emissions, a reduction in the energy efficiency was observed.

A short overview of the research demonstrates the significance of improving modern engine energy efficiency and emission parameters, changing their operation to NG. However, the separate CIE category conversion to NG operation faces several difficulties. Firstly, problems include models with traditional mechanical fuel injection systems, which exhibit the characteristic of a limited DIT range change and a substantially lower diesel fuel injection pressure compared to the CR system. However, numerous operational CIEs exist with traditional mechanical fuel injection systems. Therefore,

the objective of reducing environmental pollution is inseparable from modernising current generation CIEs to work with NG.

Moreover, it should be noted that most research has concentrated on separate speed and load engine operation mode experiments. However, when a fixed CCR NG gas component exists, there is a lack of data regarding the fuel mixture composition in wide and rational engine work mode ranges, as well as rational dual-fuel distribution in real-life operational range modes. Work that evaluates diesel reading changes under operational conditions has passive properties as a rule, without the realisation of experiments [4,26,35]. This is because of operational health and safety regulations for parameter changes, which limit fast-acting processes.

This paper presents the results from a team of Klaipeda University and Vilnius Gediminas Technical University scientists. The research object was a high-revolution four-stroke engine with a conventional fuel injection system, converted for operation with dual-fuel feed diesel and compressed NG.

The research objectives were as follows:

- Rational estimation of the dual D–NG fuel composition, justifying solutions in accordance with energy efficiency, emissions, and reliability values, while the engine is operating in a wide range of modes.
- The influence of fuel injection timing on the characteristics of the conventional injection fuel system engine.
- The evaluation of rational directions of a converted dual-fuel CIE operation process, with the purpose of establishing a higher level of energy efficiency and environmentally friendly effectiveness.

2. Experimental Methodology

2.1. General Description of Dual-Fuel Engine

Direct injection high revolution four cylinder turbocharged CIE tests were performed at the Internal Combustion Engines Laboratory of the Automobile Transport Department, Faculty of Transport Engineering, Vilnius Gediminas Technical University. A turbocharged 1.9 litre engine with an electronically controlled BOSCH VP37 distribution-type fuel pump and turbocharger was used for the tests. The EGR system was disabled during the tests. Diesel injection timing was controlled using Pulse-Width-Modulation (PWM), by forming electronic control signals for the fuel pump (Figure 1). The main CIE parameters are listed in Table 1.

Table 1. Engine specification.

Displacement (L)	1.896
Bore × stroke (mm)	79.5 × 95.5
Power (kW)/speed (rpm)	66/4000
Torque (Nm)/speed (rpm)	180/2000–2500
Cooling type	Water cooling
Fuel supply system	Direct injection
Cylinders	4 in line
Compression ratio	19.5:1
Aspiration	Turbocharge

2.2. Test Bench

The scheme of the laboratory equipment is illustrated in Figure 1. A KI-5543 engine brake stand was used for the load M_B and crankshaft speed determination. The torque measurement error was ± 1.23 Nm. The hourly fuel consumption B_f was measured by SK-5000 electronic scales and a stopwatch, and the accuracy of the B_f determination was 0.5%.

The NG fuel was measured by a Coriolis-type mass flow meter. The fuel flow meter was a RHEONIK RHM 015 (see Figure 1 pos. 25), connected into the high-pressure fuel supply system before

the gas reducer, which reduced the gas to a pressure of 1.5 bar. The flow meter measuring range was 0.004 to 0.6 kg/min with a high measurement accuracy of $\pm 0.10\%$.

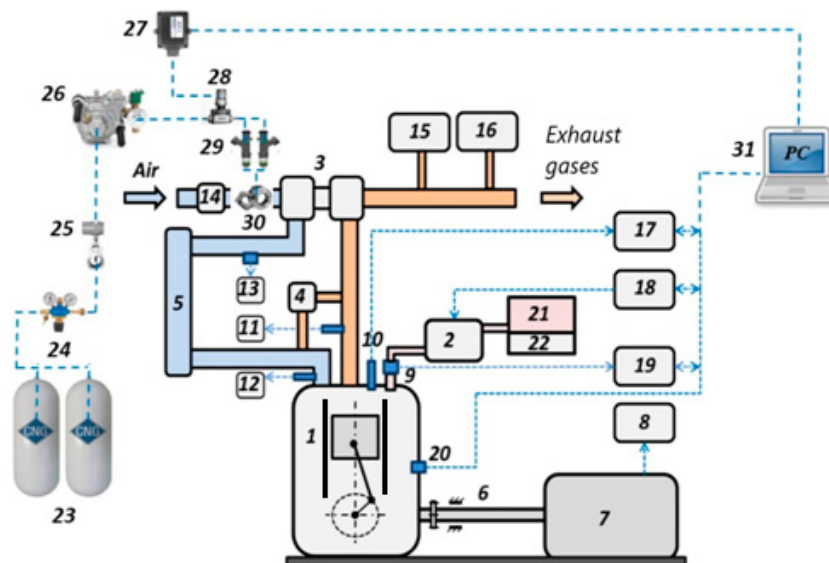


Figure 1. Scheme of the engine testing equipment: 1: engine; 2: high-pressure fuel pump; 3: turbocharger; 4: EGR valve; 5: air cooler; 6: connecting shaft; 7: engine load stand; 8: engine torque and rotational speed recording equipment; 9: fuel injection timing sensor; 10: cylinder pressure sensor; 11: exhaust gas temperature meter; 12: intake air temperature meter; 13: intake air pressure meter; 14: air mass meter; 15: exhaust gas analyser; 16: opacity analyser; 17: cylinder pressure recording equipment; 18: fuel injection timing control equipment (PWM); 19: fuel injection timing recording equipment; 20: crankshaft position sensor; 21: fuel tank; 22: fuel consumption measuring equipment; 23: CNG tank; 24: pressure regulation valve; 25: gas flow meter; 26: pressure reducer; 27: ECU; 28: gas metering valve; 29: gas injectors; 30: air and gas mixer; 31: computer.

2.3. Exhaust Gas Emission Measurement Equipment

The pollutants in the exhaust gas were measured using several gas analysers: AVL DiCom 4000 (AVL, Austria) (Table 2) and HORIBA PG-250 (HORIBA, Japan) (Table 3), TESTO 350 Maritime (TESTO, Indonesia) (for CO, CO₂, HC, and NO_x) (Table 4), and AVL DiCom 4000 and MDO-2 LON (MAHA, Germany) (for absorption coefficient K-value) (Table 5). The HORIBA PG-250, TESTO 350M, and MDO-2LON were used as measurement value control units to ensure the accuracy of the measurement results. The main equipment for the exhaust gas analysis was the AVL DiGas 4000/AVL DiCom 4000 (AVL, Austria).

Table 2. Measurement range and resolution of AVL DiCom 4000 gas analyser.

Parameter	Measurement Range	Measurement Accuracy
Nitrous oxides (NO _x)	0–5000 ppm (vol.)	1 ppm
Hydrocarbons (HC)	0–20,000 ppm (vol.)	1 ppm
Carbon monoxide (CO)	0–10% (vol.)	0.01% (vol.)
Carbon dioxide (CO ₂)	0–20% (vol.)	0.1% (vol.)
Oxygen (O ₂)	0–25% (vol.)	0.01% (vol.)
Absorption (K-value)	0–99.99 m ⁻¹	0.01 m ⁻¹
Lub. oil temperature	0–150 °C	1 °C

HORIBA PG-250 exhaust gas analyser measurement range and resolution.

This equipment contains a gas preparation block, which cools down the gases and thus removes condensate (H₂O) surplus that may accumulate during combustion. During analysis, the quality of the outlet gases is established. The complex exhaust gas analyser TESTO 350 Maritime measurement range and resolution are given in Table 4.

Table 3. Measurement range and resolution of HORIBA PG-250 analyser.

Emission Component	Measurement Range	Measurement Accuracy
CO	0–5000 ppm	± 1% F.S.
CO ₂	0–20 vol. %	± 1% F.S.
SO ₂	0–3000 ppm	± 1 ppm F.S.
NO _x	0–2500 ppm	± 1 ppm F.S.
O ₂	0–25%	± 1 ppm F.S.

Table 4. Measurement range and resolution of TESTO 350M analyser.

Emission Component	Measurement Range	Measurement Accuracy
CO	0–5000 ppm	±5% F.S.
CO ₂	0–50%	±0.3 vol. % + 1% F.S.
NO _x	0–5000 ppm	±5 ppm (0–99 ppm); ± 5% F.S. (+100–+500 ppm)
SO ₂	0–5000 ppm	±5% F.S.
O ₂	0–25%	±0.2%

Exhaust gas opacity analyser MDO-2 LON measurement range and accuracy.

Table 5. Measurement range and resolution of MDO-2 LON analyser.

Measurement Parameter	Measurement Range	Measurement Accuracy
Opacity	0–100%	±2% F.S.
Absorption (K-value)	0–99.99 m ^{−1}	±2% F.S.

The in-cylinder pressure (P_{cyl}) was recorded by an AVL GH13P piezo-sensor (sensitivity 16 pC/bar, linearity of FSO ≤ ±0.3%), which was integrated into the preheating plug and recorded using an AVL DiTEST DPM 800 amplifier (input range 6000 pC, signal ratio 1 mV/pC, overall error complete temperature range 1%) and LabView Real-Time equipment. The intake air mass flow meter was measured by a BOSCH HFM 5 with an accuracy of 2%. The intake manifold pressure was measured with a Delta OHM HD 2304.0 pressure gauge. A TP704-2BAI sensor device with an error of ±0.0002 MPa was mounted ahead of the intake manifold. The exhaust and intake gases temperature meter K-type thermocouple (accuracy ±1.5°C) was used.

2.4. Fuel Specification

Two fuel types were used during the experiment: liquid and gas (Table 6). During the dual-fuel mode, standard diesel fuel (EN 590) and standard compressed NG (ISO 6976:1995) were used.

Table 6. Fuel properties.

Fuel type	Natural gas	Diesel
Density (kg/m ³)	0.74	829.0
Cetane number	-	49
H_U (MJ/kg)	51.7	42.8
Viscosity (cSt 40 °C)	-	1.485
H/C ratio	-	1.907
Component (% vol.)	Methane: 91.97 Ethane: 5.75 Propane: 1.30 Butane: 0.281 Nitrogen: 0.562 Carbon dioxide: 0.0	Carbon: 86.0 Hydrogen: 13.6 Oxygen: 0.4

2.5. Basis for Numerical Research

The CNIDI (Central Diesel Research Institute, St. Petersburg, Russia) mathematical model from the TEPLM software package has been used in order to conduct analysis of experimentally obtained

indicated process diagrams [36], which used a closed thermodynamic cycle energy balance model, evaluating the heat transfer through the cylinder walls. Initial values for the software calculations included the dual-fuel cycle portion (q_c) and lower heating value H_L , while the chemical composition (C, H, O) was established using the following formula:

$$q_c = \frac{q_{cD} \cdot H_{LD} + q_{cNG} \cdot H_{LNG}}{H_L}.$$

where,

q_c : Overall fuel consumption per cycle, g/cycle;

q_{cD} : Diesel fuel consumption per cycle, g/cycle;

q_{cNG} : NG fuel consumption per cycle, g/cycle;

H_{LD} : Lower heating value of diesel fuel, MJ/kg;

H_{LNG} : Lower heating value of NG, MJ/kg.

H_L , the lower heating value of the fuel (MJ/kg) was calculated by the Mendeleev equation [36]:

$$H_L = 337.5 \cdot C + 1025 \cdot H - 108.3 \cdot O,$$

where,

$$C = C_D \cdot (100 - CCR_{NG}) + C_{NG} \cdot CCR_{NG},$$

$$H = H_D \cdot (100 - CCR_{NG}) + H_{NG} \cdot CCR_{NG},$$

$$O = O_D \cdot (100 - CCR_{NG}) + O_{NG} \cdot CCR_{NG},$$

$$CCR_{NG} = \frac{q_{cNG} \cdot H_{LNG}}{q_{cNG} \cdot H_{LNG} + q_{cD} \cdot H_{LD}} \cdot 100\%.$$

where,

CCR_{NG} : Co-combustion ratio of natural gas, %

C_D : Element carbon composition in diesel fuel

C_{NG} : Element carbon composition in NG.

Energy efficiency parameters:

The indicated efficiency (η_i) could be established using the following formula:

$$\eta_i = \frac{3.6 \cdot P_e}{H_{LD} \cdot G_{fD} + H_{LNG} \cdot G_{fNG}}.$$

where, H_{LD} and H_{LNG} : lower heat values of diesel fuel and NG, respectively, MJ/kg; B_{fD} and B_{fNG} : diesel fuel and NG consumption, respectively, kg/h.

The brake thermal efficiency could be established using the classical expression $\eta_e = \eta_i \cdot \eta_m$, where η_m is the mechanical resistance coefficient established using the combined mechanical engine losses P_m , and P_m could be determined from the experimental indicator diagrams.

2.6. Experiment Execution Plan

The experimental engine efficiency and emission research was conducted with a wide range of loads (P_{me}) and rpm (n), as well as various HRF injection timing angles (φ_{inj}) (see Table 7). In every mode, characterised by different combinations ($P_{me(D)}$, n , φ_{inj}), the engine parameters were measured using diesel only (D), and dual D and NG fuel: D60/NG40, D40/NG60, and D20/NG80 (here, the numbers following “D” and “NG” correspond to the diesel and NG percentage parts of the total energy balance). The engine load modes were named in the following manner: $P_{me} = 5.97$ bar, high load mode (HLM); $P_{me} = 3.98$ bar, medium load mode (MLD); and $P_{me} = 1.98$ bar, low load mode (LLM).

Table 7. Experiment execution plan.

	$n = 2500 \text{ min}^{-1} \text{D/GD}$					$n = 2000 \text{ min}^{-1} \text{D/GD}$				$n = 1500 \text{ min}^{-1} \text{D/GD}$			
	φ_{inj}	D	D60/G40	D40/G60	D20/G80	D	D60/G40	D40/G60	D20/G80	D	D60/G40	D40/G60	D20/G80
$P_{me} = 0.597 \text{ MPa}$	−1					X	X	X	X				
	−7					X	X	X	X				
	−13					X	X	X	X				
$P_{me} = 0.398 \text{ MPa}$	−1	X	X	X	X	X	X	X	X				
	−7	X	X	X	X	X	X	X	X				
	−13	X	X	X	X	X	X	X	X				
$P_{me} = 0.198 \text{ MPa}$	−1					X	X	X	X	X	X	X	X
	−7					X	X	X	X	X	X	X	X
	−13					X	X	X	X	X	X	X	X

Note: The engine parameters were investigated in the following ranges: $\varphi_{inj} = 1; 4; 7; 10; 13$ °CA BTDC; the provided results are in the range: $\varphi_{inj} = 1; 7; 13$ °CA BTDC. X: investigated parameters.

For every load mode energy efficiency value, the emission and indicated parameters were registered at least three times, with further average calculations and rough error removal using statistical methods (MATHWORKS—Matlab, Microsoft Excel). The indicator diagram pressure data arrays were established as the averages of 100 registered indicator diagrams.

3. Research Results and Discussion

The conversion of a CIE for operation with NG fuel feed encompasses the evaluation of energy efficiency, emissions, and reliability values (cylinder used, piston group detailed mechanical load ratio criteria, and maximum cylinder pressure P_{max}).

3.1. Changes in Energy Efficiency

The effective η_e and indicated η_i efficiency were used as the energy efficiency parameters. Changes in the parameter η_e when increasing CCR NG in different load and rpm modes (P_{me}), and the injection timing ϕ_{inj} range results are provided as well (see Figure 2).

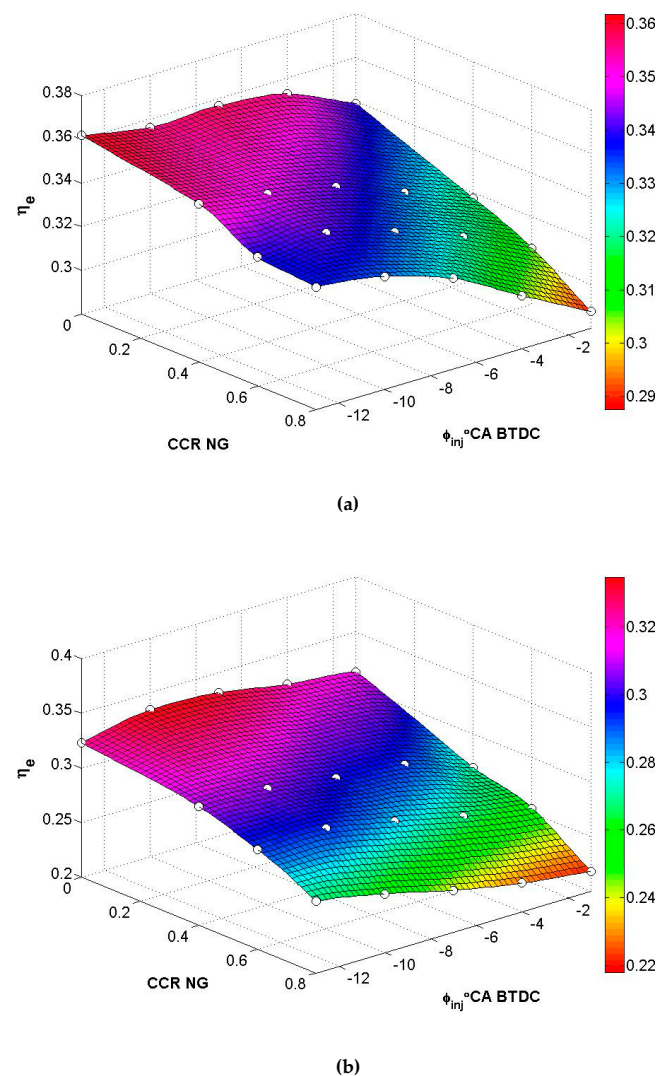
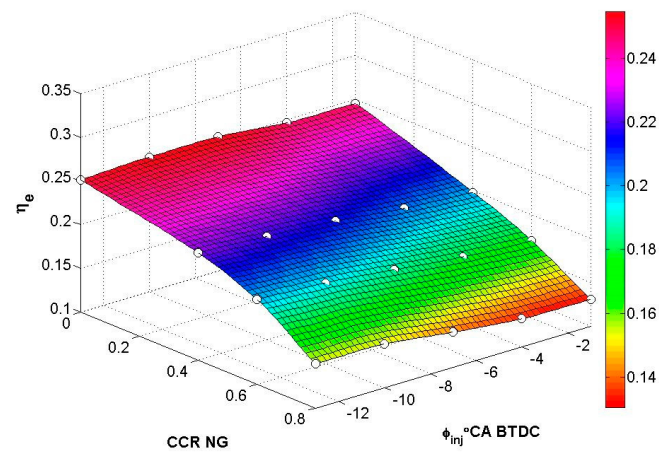
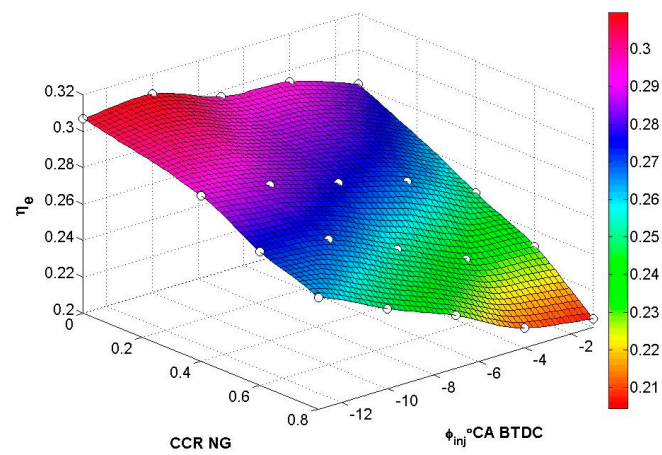


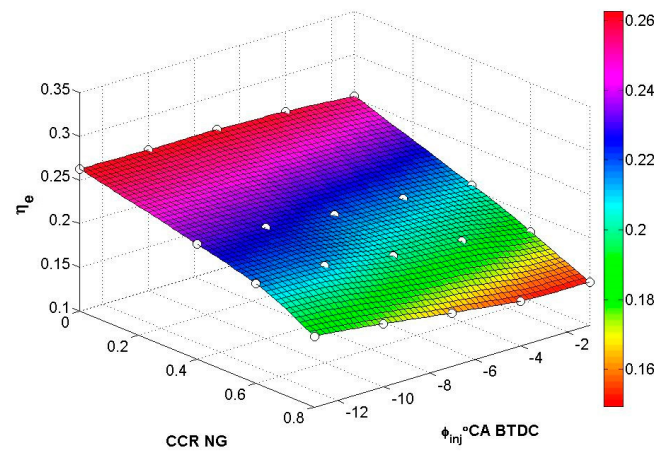
Figure 2. Cont.



(c)



(d)



(e)

Figure 2. Influence of CCR_{NG} and ϕ_{inj} (ϕ_{inj}) on engine η_e at (a) HLM, $n = 2000 \text{ min}^{-1}$; (b) MLM, $n = 2000 \text{ min}^{-1}$; (c) LLM, $n = 2000 \text{ min}^{-1}$; (d) MLM, $n = 2500 \text{ min}^{-1}$; (e) LLM, $n = 1500 \text{ min}^{-1}$. The experimental data are denoted by dots in the graph.

It should be noted that the obtained dependencies $\eta_e = f(P_{me}, \varphi_{inj}, CCR\ NG)$ are qualitatively and quantitatively identical to those of an engine operating in the investigated revolution range of $n = 1500\text{ min}^{-1}$, $n = 2000\text{ min}^{-1}$, $n = 2500\text{ min}^{-1}$ (see Figure 2).

Based on this change in the CCR, the engine parameters were specified as $n = 2000\text{ min}^{-1}$.

The load characteristics at $n = 1500\text{ min}^{-1}$ and $n = 2500\text{ min}^{-1}$ data were limited, only disclosing their specific features. However, the CCR NG quantitative influence did differ strongly for different load modes.

Operating in a mode close to the nominal engine load $P_{me} = 5.98\text{ bar}$ CCR NG, the increase in the influence on the η_e parameter was minimal: a 0.8% to 1.8% η_e decrease for every CCR NG increase of 10%. The range top limit values were obtained with relatively low process dynamics, at $\varphi_{inj} = 1$ to 4°CA BTDC; lower values are specific to processes with high work process dynamics at $\varphi_{inj} = 10$ to 13°CA BTDC (see Figure 3).

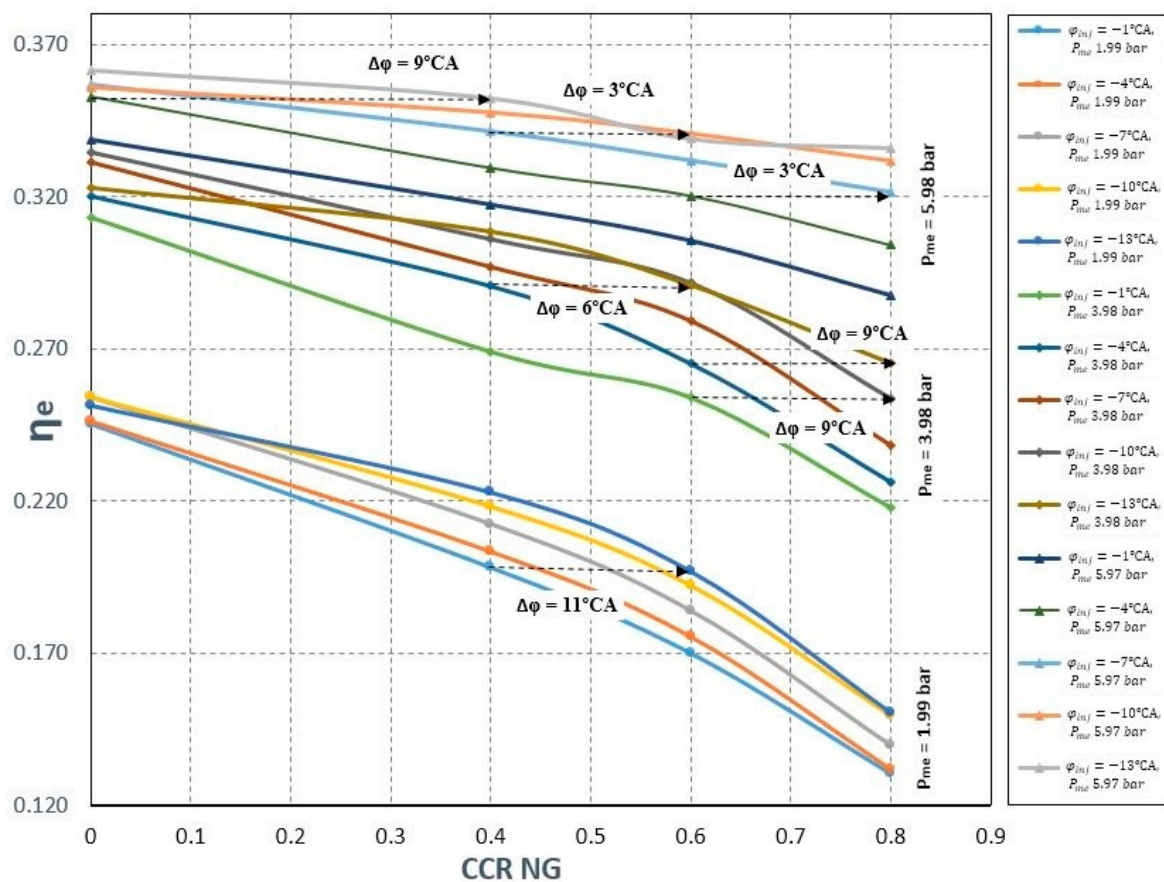


Figure 3. Influence of CCR NG portion increase on diesel engine η_e .

The analysed φ_{inj} change was evaluated as one of the most technologically simple measures, capable of improving the engine parameters for dual-fuel operation. Moreover, an experiment on the changes in φ_{inj} within the range of 1 to 13°CA BTDC was conducted to expand the obtained results for engines with different dynamic characteristics (P_{max} , pressure increase for maximum speed $(dP/d\varphi)_{max}$, average speed $(dP/d\varphi)_{mid}$, and pressure increase for (λ)).

The MLM $P_{me} = 3.98\text{ bar}$ parameter η_e decreased for every CCR NG increase of 10%, making up 2.5% to 3.5%, and for the LLM $P_{me} = 1.99\text{ bar}$ from 4.7% to 6.0%. It is possible to adjust the parameter φ_{inj} to improve η_e for operation with NG in different load modes, and P_{me} differs significantly. When the engine is operating in the HLM φ_{inj} advancement ($\Delta\phi_{inj}$) to 3°CA BTDC, compensating for the negative effect of an increase in the CCR increase on η_e (Figure 3).

Overall, it is rational that for every $CCR\ NG$ increase of 20%, there should be an advancement of φ_{inj} ($\Delta\varphi_{inj}$) by 3 °CA BTDC. The obtained results are specific to the investigated φ_{inj} range, which means that it is applicable for engine models with different operational process dynamics: in the investigated object, the pressure increase P_{max}/P_c was changed (where P_c is the final compression pressure) within the range from 0.9 to 1.8.

In the MDL mode, the influence of the $P_{me} = 3.98$ bar parameter φ_{inj} on η_e decreased. In order to compensate for the η_e decrease with the engine operating on NG, it is necessary to advance φ_{inj} ($\Delta\varphi_{inj}$) to 3 to 6 °CA for every 20% increase in $CCR\ NG$: the lower limit values are consistent with a low dynamics process. For the LLM $P_{me} = 1.99$ bar, the injection timing φ_{inj} must be advanced ($\Delta\varphi_{inj}$) to 9 to 12 °CA or even more.

The adjustment of φ_{inj} for improving η_e is inseparable from the necessity to control the operation process dynamic parameters, including the maximum cycle pressure, in order to avoid mechanical overloading of the engine. The overall tendency of the investigated engine load P_{max} is that a decrease in P_{max} occurs with an increase in $CCR\ NG$ (see Figure 4). It is known that the size of P_{max} is determined by the heat release $Q_{P_{max}}$ at the maximum pressure phase $\varphi_{P_{max}}$ [37–40]. The size is mainly formed by the heat released during the first kinetic phase, which is influenced by the auto-ignition delay period φ_i . A decrease in the amount of fuel injected through φ_i or in the period of injection φ_i ensures a decrease in P_{max} [41–43]. The P_{max} dependence on the $CCR\ NG$ portion variable is nonlinear: at a $CCR\ NG$ increase of up to 0.4, P_{max} changed only slightly; however, when the $CCR\ NG$ increased to a larger portion than 0.4 and up to 0.8, the decrease in P_{max} reached 10 to 15 bar (see Figure 4).

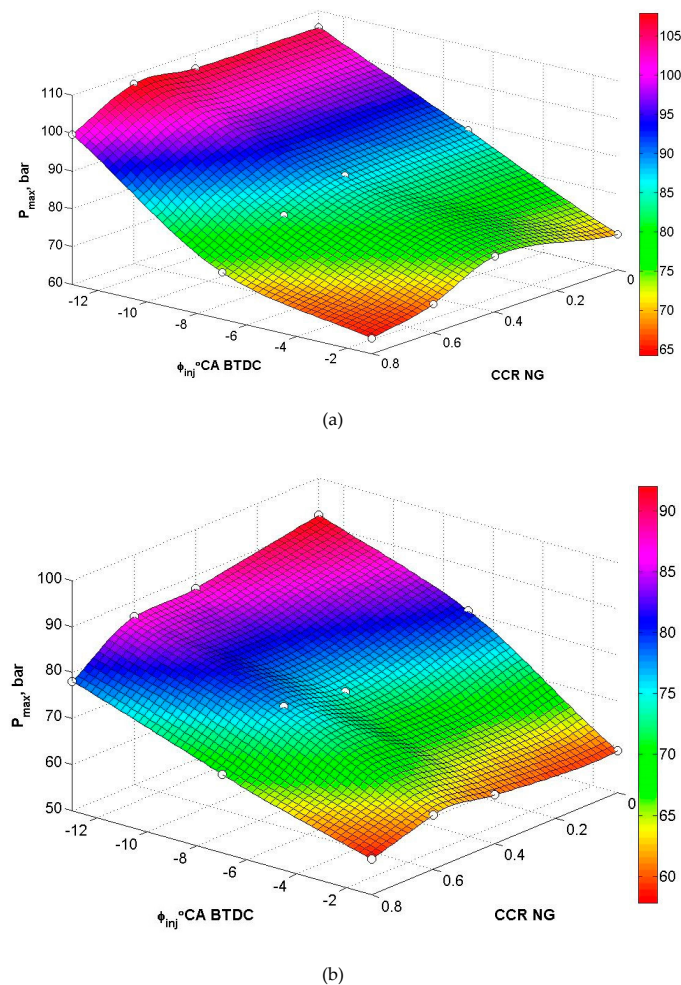


Figure 4. Cont.

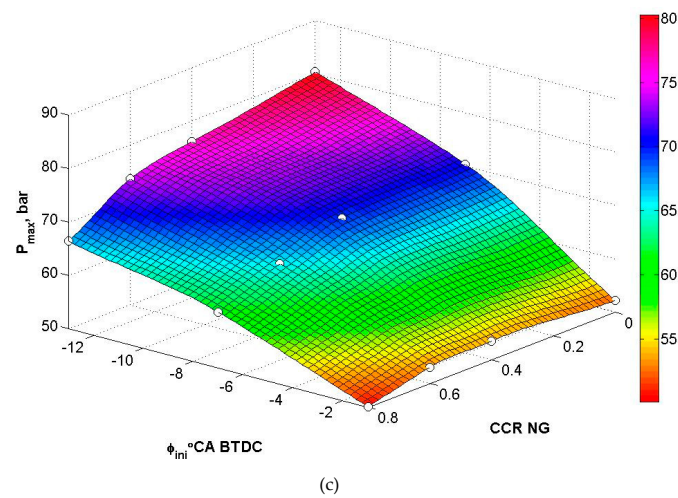


Figure 4. Injection timing φ_{inj} influence on maximum cycle pressure $P_{max}(n = 2000 \text{ min}^{-1})$: (a) HLM; (b) MLM; (c) LLM. The dots in the graph represent the experimental data.

Therefore, it is appropriate to increase the CCR NG and compensate for the η_e losses with an increase in φ_{inj} . Thus, in order to evaluate the changes in P_{max} for CIE conversion for operation with NG fuel feed it is necessary to use different portions of NG, such as CCR NG 0 to 0.2 and 0 to 0.4. Figure 5 provides the engine indicator diagrams with the speed of heat release characteristics $dQ/d\varphi$.

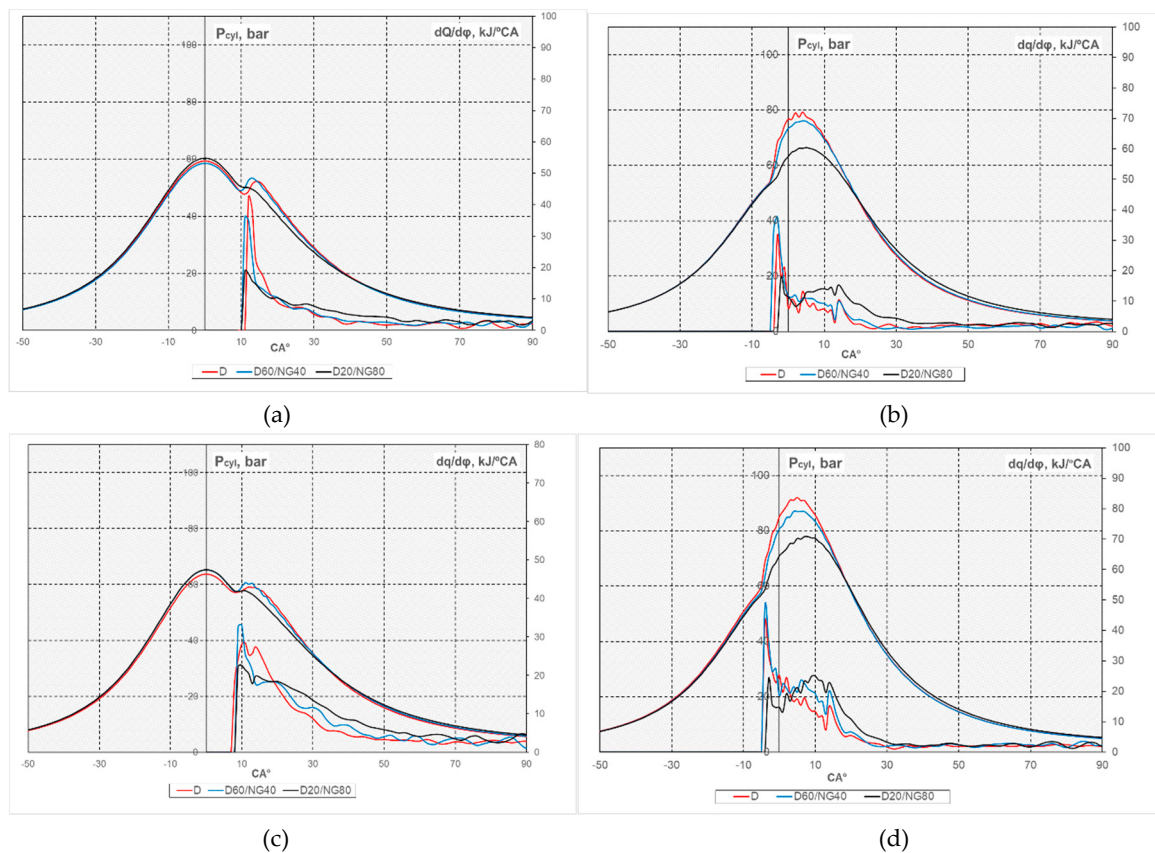


Figure 5. Cont.

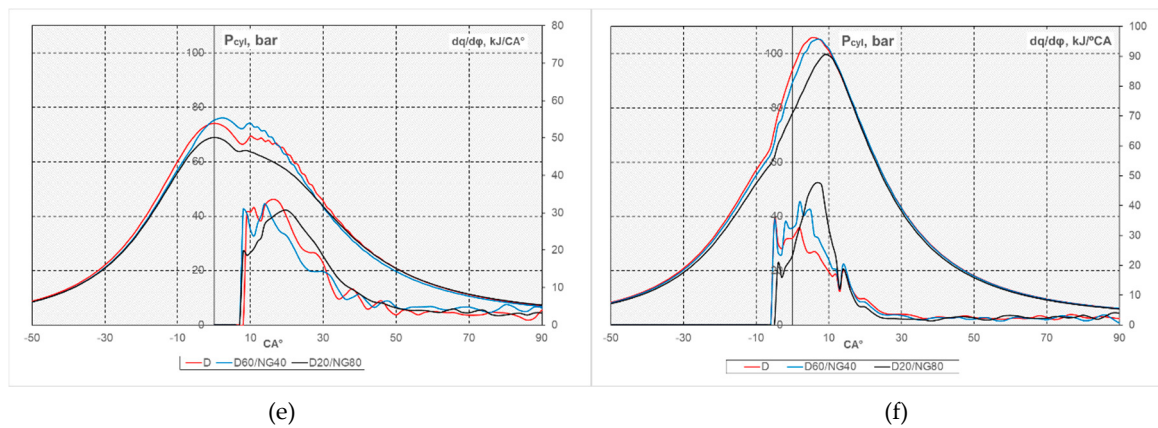


Figure 5. Combustion process indicator diagrams and heat release dynamics $dQ/d\varphi$ for different φ_{inj} ($\varphi_{inj} = -1$ and -13 °CA), ($n = 2000 \text{ min}^{-1}$): (a) and (b) correspond to LLM; (c) and (d) correspond to MLM; (e) and (f) correspond to HLM.

For every engine load mode, the diesel auto-ignition phase for $\varphi_{inj} - idem$ did not change in the CCR NG portion range from 0 to 0.8. The ignition delay period φ_i also did not change for $\varphi_i = 8 \pm 1$ °CA in all investigated φ_{inj} . Overall, the decrease in P_{max} can be linked to the changes in the heat release dynamics when the CCR NG is increased. A decrease in the diesel portion decreases the heat release in the heterogeneous combustion kinetics phase, and increases the total energy balance part of the NG volume specific combustion part [18,23,44]. The combustion process moves towards an expansion stroke, and therefore the decrease in P_{max} occurs in parallel with the deterioration in η_e .

The advancement of the injection timing φ_{inj} moves the combustion process towards TDC, compensating for the changes in η_e relating to the CCR NG increase. However, the φ_{inj} advancement in engines with conventional fuel injection systems exhibits limited capabilities in the aforementioned range. One method for improving the η_e characteristics is accelerating the combustion process in the second (main) phase, which forms the energy efficiency indicators of the cycle. In order to achieve this, the cylinder air load is increased; moreover, the fuel portion vortex movement degree is increased by using different technological methods, such as incorporating fuel additives [42,43].

In the HLM, in which the CCR NG constitutes 0.2, to compensate for the deterioration in η_e , φ_{inj} is advanced to 3°CA BTDC, and the maximum cycle pressure is increased to 7 to 10 bar (see Figure 4). In the MLM, the φ_{inj} advancement to 6°CA BTDC leads to an increase in P_{max} of up to 15 to 25 bar: the result is that P_{max} is at a larger pressure than that obtained by using diesel only (CCR NG = 0).

The LLM P_{max} can reach 20 to 25 bar. Although P_{max} is lower during the medium and low load modes compared to the modes that are closer to the nominal engine load, and engine operation in the HLM only occurs for a limited time, a meaningful increase can significantly decrease the engine reliability. Therefore, it is not considered as rational for practical use. A CCR NG increase for the LLM up to 0.4, together with a φ_{inj} advancement to 6 °CA, can lead to an increase in P_{max} to 15 to 25 bar.

During the MLM, the P_{max} increase can reach up to 30 bar and more. It should be noted that changes in P_{max} are established when the engine is operating with diesel fuel feed in a stationary state of $\varphi_{inj} = 1$ to 4 °CA BTDC, or low process dynamics. When the system is in a higher state of $\varphi_{inj} = 7$ to 13 °CA BTDC or in a more dynamic DE operation state, the advancement of φ_{inj} becomes irrational: for a small η_e restoration effect, a large increase in P_{max} is necessary.

Overall, it should be stated that application of the φ_{inj} advancement to compensate for the decrease in energy efficiency when the engine is operating in the dual-fuel mode is limited during the HLM and portions of the CCR NG from 0 to 0.4.

The exchange of diesel for NG and the use of NG in different load modes are essentially expanded by applying the φ_{inj} advancement method, when there is also a decrease of approximately 3%. The LLM CCR NG is expanded to 40%, while the MLM possible CCR NG application reaches 20% when decreases

of 34% and 6% occur, respectively. An increase occurs when operating with diesel fuel equal to 7 bar, which remains unchanged.

3.2. Emission

The engine emission evaluation is directly related to the engine purpose. Vehicle engines are regulated by international standards throughout the entire range of harmful emissions: NO_x , CO, C_nH_m , and SO_x , as well as PM [45–48]. Moreover, marine-purpose DEs are regulated by MARPOL 73/78 convention Annex VI [49] for NO_x and SO_x emissions. However, as opposed to DEs for other purposes, marine power plants are also regulated by the decision of the International Maritime Organization to limit greenhouse gas CO_2 emissions [50]. Therefore, it is correct for the evaluation of converted dual-fuel engine emissions to be conducted in accordance with their purpose.

3.2.1. Nitrous oxides

The majority of research has pointed out that CIE conversion for operation with NG fuel feed fundamentally decreases NO_x emissions, owing to the equalisation of the combustion temperature field in the cylinder and decrease in high-temperature zones [45–47,50]. The results obtained from this research agree with this tendency. The effects of the NO_x emission decrease with an increase in the CCR are nonlinear – the maximum effect in the investigated options was detected in the CCR NG > 40% range. The NO_x mainly decreased in the LLM $P_{me} = 1.99$ bar.

Moreover, depending on the stationary φ_{inj} value while operating on diesel only, for every CCR NG increase of 10%, the NO_x emissions decreased by 7% to 3% in the HLM, and 9% to 10% in the MLM and LLM (earlier values of φ_{inj} exhibited lower changes in NO_x emissions). With this manner of exchanging diesel for NG, the NO_x decrease constitutes approximately 65% in the HLM, and a decrease in NO_x of 90% to 95% can be reached in the MLM and LLM (see Figure 6).

However, the CCR NG increase is followed by a decrease in η_e of up to 20% in the MLM and 45% in the LLM. It is obvious that the changes in η_e and NO_x must be coordinated, for example, by applying the φ_{inj} advancement method (see Figure 6).

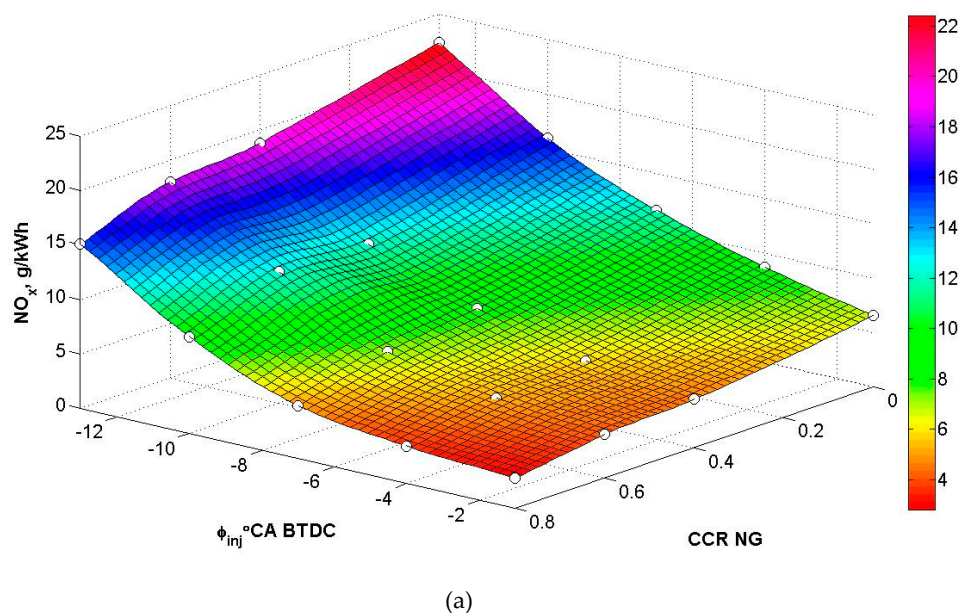


Figure 6. Cont.

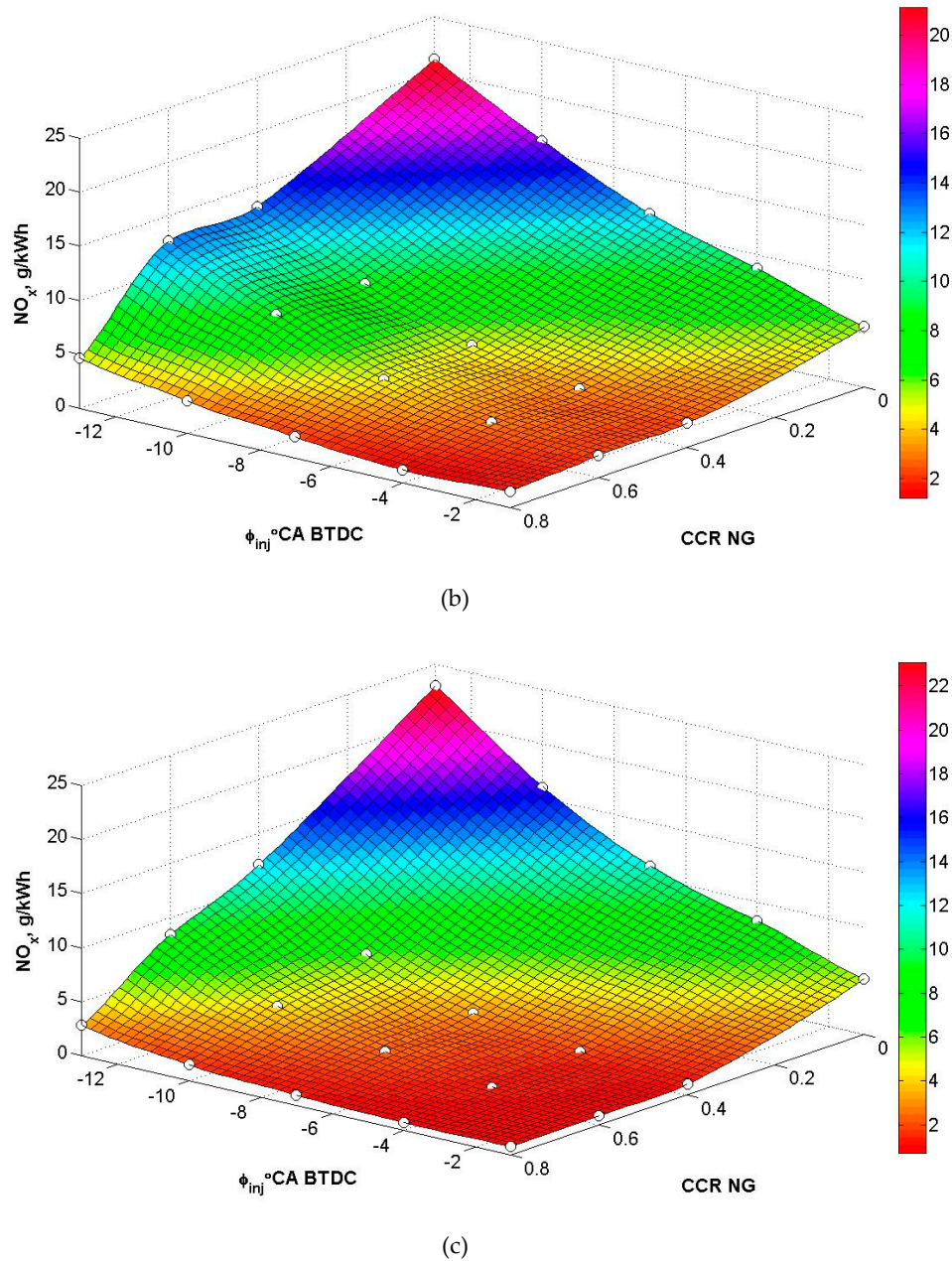


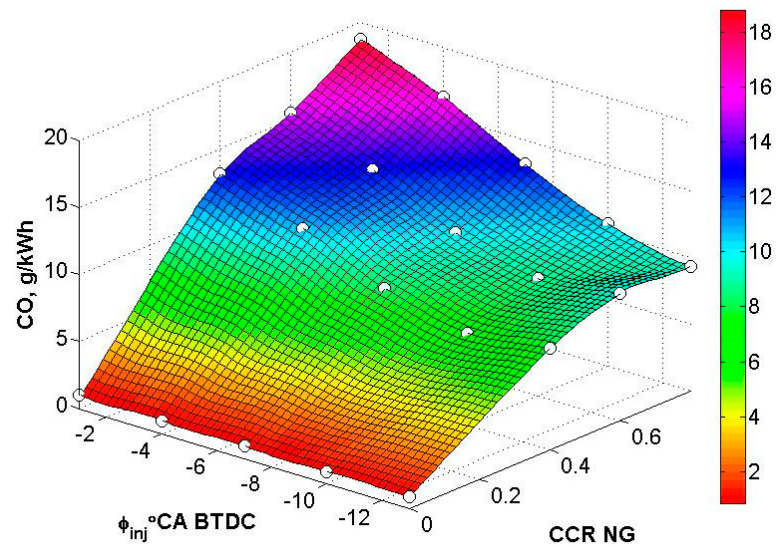
Figure 6. Influence of dual-fuel CCR NG portion and φ_{inj} on NO_x emissions ($n = 2000 \text{ min}^{-1}$) : (a) HLM; (b) MLM; (c) LLM. The dots in the graph represent the experimental data.

A CCR NG increase to 0.2 in the HLM was implemented practically without a significant decrease in the energy efficiency; therefore, even without the use of φ_{inj} advancement, the NO_x emissions decreased. When increasing the CCR NG portion up to 0.4, it is rational to use a limited φ_{inj} advancement solution, which will ensure a certain amount of η_e and NO_x reduction. For example, in the HLM, a φ_{inj} advancement to 3 °CA BTDC instead of 6°CA BTDC ensured a reduction in η_e and NO_x of 3% and 0.9 g/kWh, respectively. Without changing φ_{inj} , η_e was reduced by 6.5%, and the NO_x emissions were reduced by 1.2 to 1.7 g/kWh or 35% compared to the engine operating on diesel only.

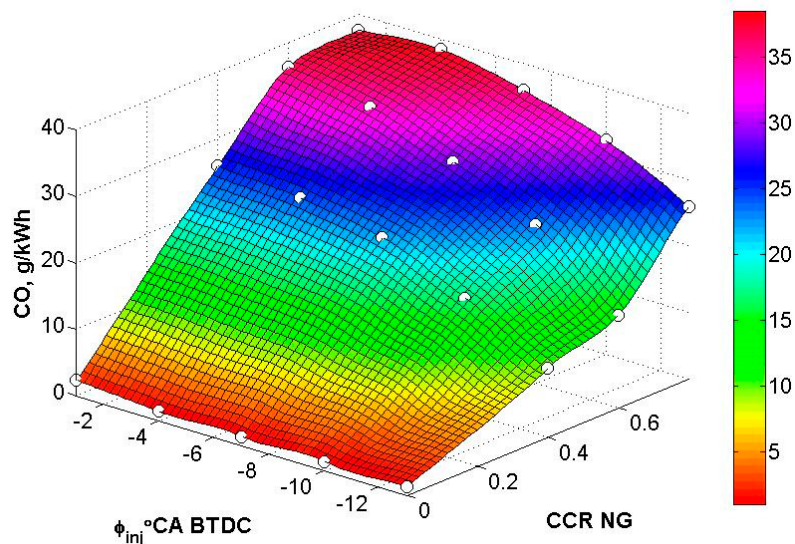
In the MLM, a partial deterioration of the η_e compensation with a change in φ_{inj} ensured a NO_x decrease of 0.7 to 1.1 g/kWh or approximately 15%. In the LLM, exchanging 0.2 of diesel with NG, without φ_{inj} adjustment, a reduction in NO_x emissions of 2 to 3.3 g/kWh or 50% was achieved.

3.2.2. Carbon monoxide

According to various sources, the conversion of a CIE for dual-fuel operation is related to a significant increase in harmful partial combustion components, such as CO and HC emissions [45–48]. During the experiment, it was determined that when a portion of CCR NG of 0.8 was reached, the CO emissions increased 8 to 30 times in the HLM from the base level of 0.5 g/kWh, 20 to 30 times in the MLM from approximately 1 g/kWh, and 10 to 20 times in the LLM from approximately 2 to 6 g/kWh, during different operation dynamics (see Figure 7).



(a)



(b)

Figure 7. Cont.

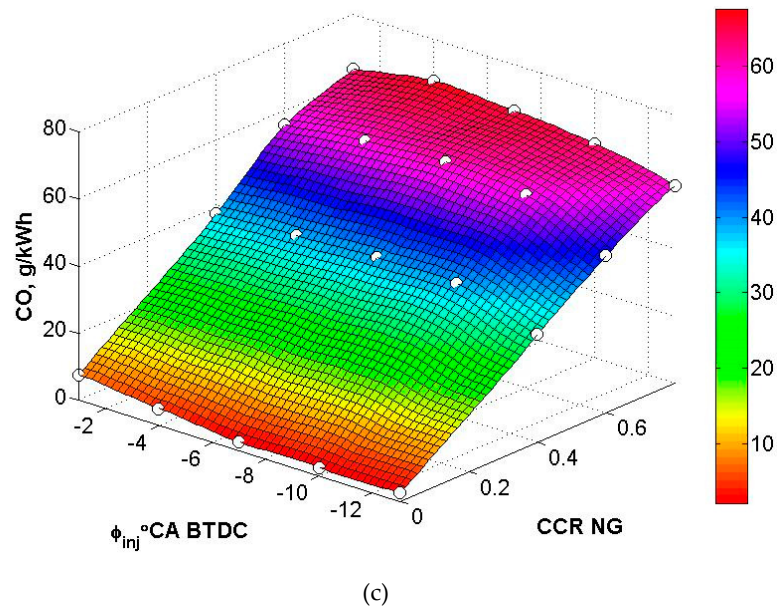


Figure 7. Influence of dual-fuel CCR NG portion and ϕ_{inj} on CO emissions ($n = 2000 \text{ min}^{-1}$): (a) HLM; (b) MLM; (c) LLM. The dots in the graph represent the experimental data.

In the investigated range, ϕ_{inj} advancement becomes an ineffective measure for CO emission control. Overall, for CCR NG 0.2 and 0.4 portions, an increase in CO emissions could be detected in the HLM of 3 to 5 g/kWh and 5.5 to 10 g/kWh, and in the MLM, with CCR NG 0.2, an increase occurred for 7 to 10 g/kWh.

3.2.3. Hydrocarbons

No less intensive is the HC increase when the engine is operating on NG (see Figure 8).

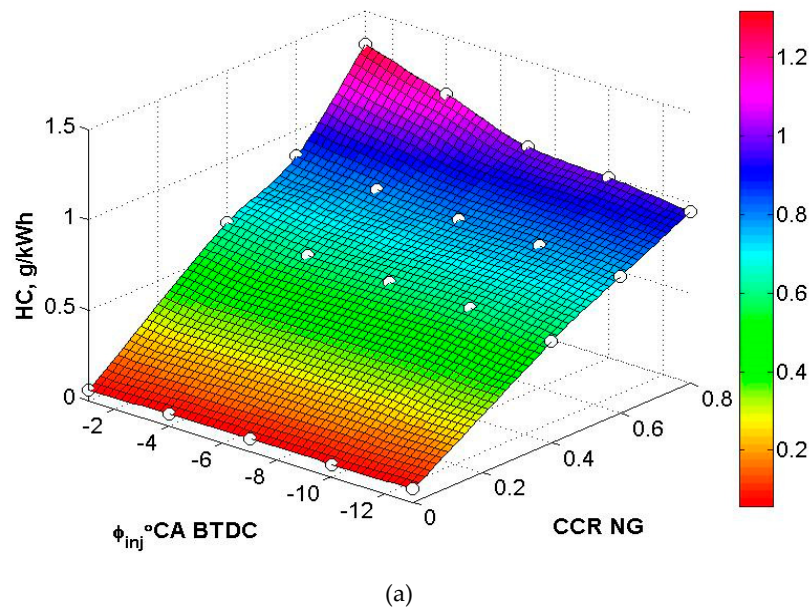


Figure 8. Cont.

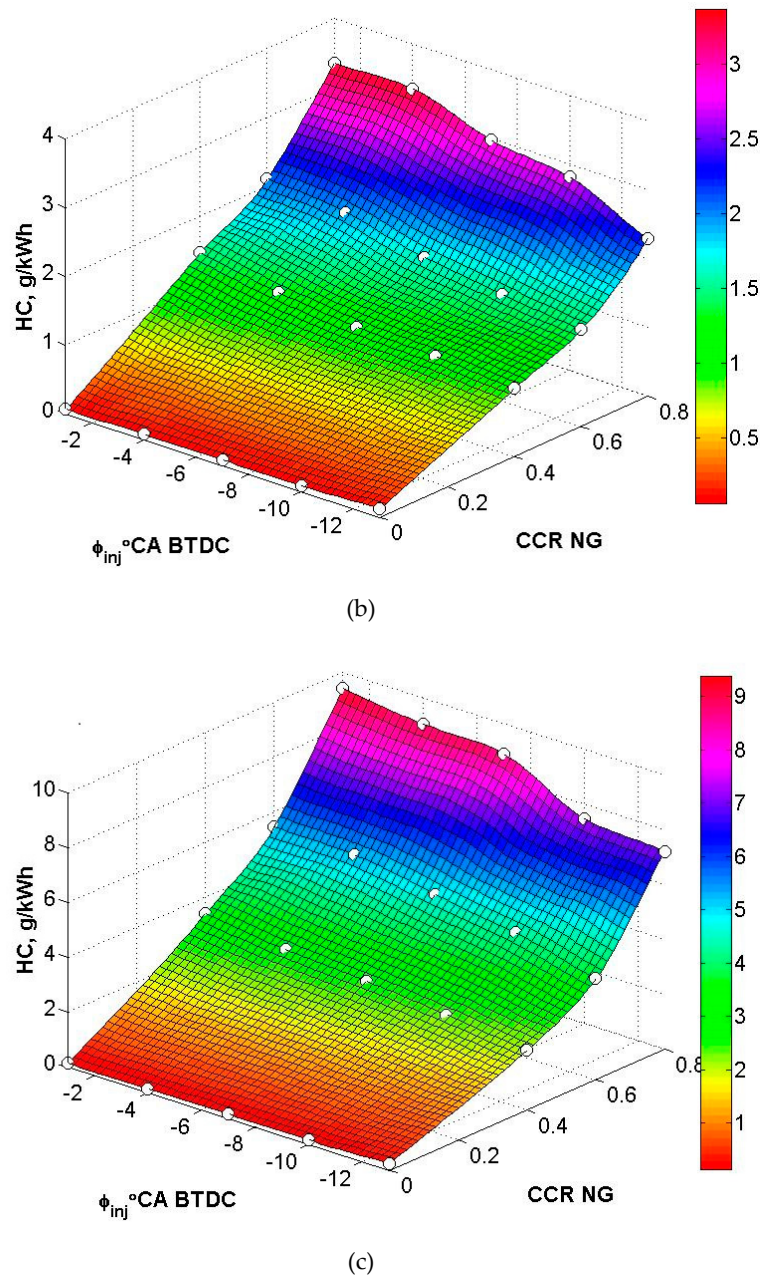


Figure 8. Influence of dual-fuel CCR NG portion and ϕ_{inj} on HC emissions ($n = 2000 \text{ min}^{-1}$): (a) HLM; (b) MLM; (c) LLM. The dots in the graph represent the experimental data.

The HC emissions increased when the engine load was increased, and the influence of the ϕ_{inj} advancement was not significant. In the HLM, for every CCR NG portion increase of 0.1, the HC emission increase constituted approximately 0.1 g/kWh; when the CCR NG was in the range of 0.6 to 0.8, the HC emissions increased by 0.15 to 0.4 g/kWh. In the MLM, for an analogous CCR NG portion, the HC emissions increase constituted 0.4 to 0.6 g/kWh and 0.7 to 1.5 g/kWh, respectively; for the LLM, the values were 1 to 1.5 g/kWh and 2.5 to 3.0 g/kWh.

3.2.4. Carbon Dioxide

Greenhouse gas CO_2 emissions from internal combustion engines are dependent on two factors: the carbon C in the fuel chemical composition and fuel consumption [45,46,51]. Compared to diesel, the NG chemical composition has a C part of 75% versus 85% to 86%. Thus, engine conversion for

operating on NG at the same mass portion results in a decrease in fuel consumption and therefore reduced CO₂ emissions.

During the experiment, a decrease in CO₂ was only achieved for the HLM $P_{me} = 5.98$ bar in the entire CCR NG portion range from 0 to 0.8 (see Figure 9).

Moreover, the ϕ_{inj} advancement reduced the CO₂ emissions in the CCR NG portion range from 0 to 0.8: when $\phi_{inj} = -1^\circ$ CA BTDC, the CO₂ emissions decreased by 9%, for $\phi_{inj} = -13^\circ$ CA BTDC, the decrease could reach 16%. For the MLM and LLM, the increased CO₂ emissions led to a deterioration in energy efficiency. Overall, in the mode of $P_{me} = 3.98$ bar for a range up to a CCR NG portion of 0.6, the increase in CO₂ emissions practically remained unchanged, and with a further increase in the CCR NG up to 0.6 to 0.8 in the LLM, the CO₂ increase totalled 30% to 45%.

The overall effects of the CIE conversion for dual-fuel operation on the energy efficiency and emissions are provided in Table 8.

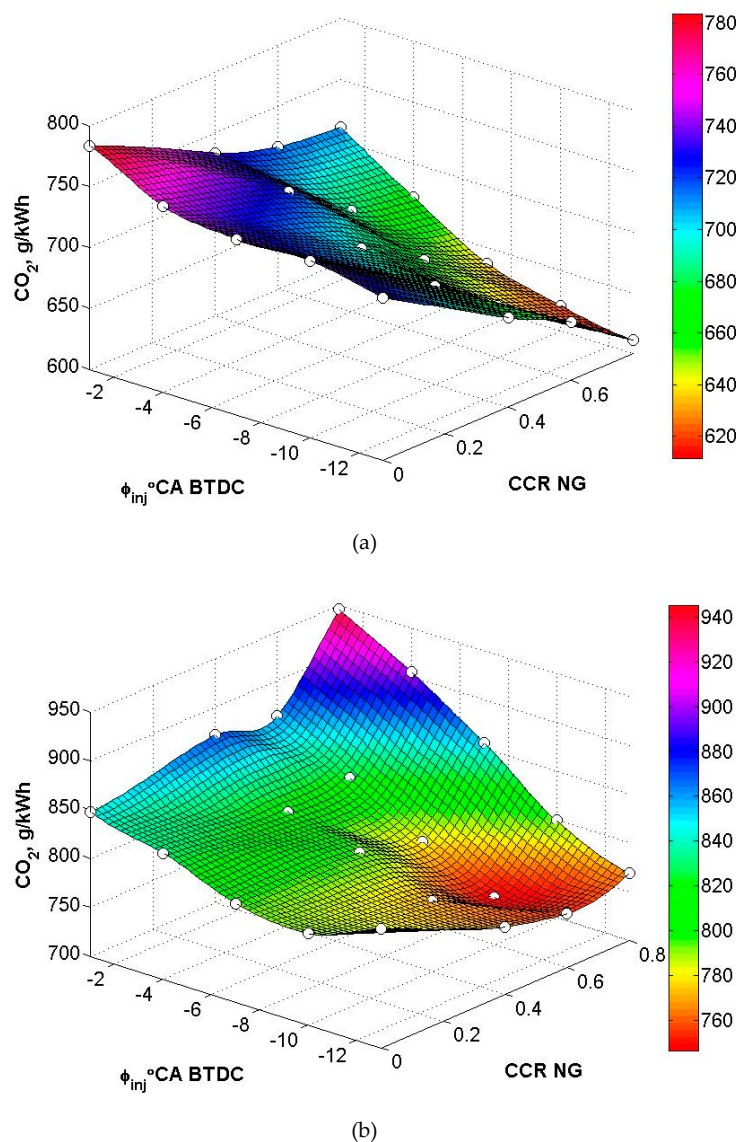


Figure 9. Cont.

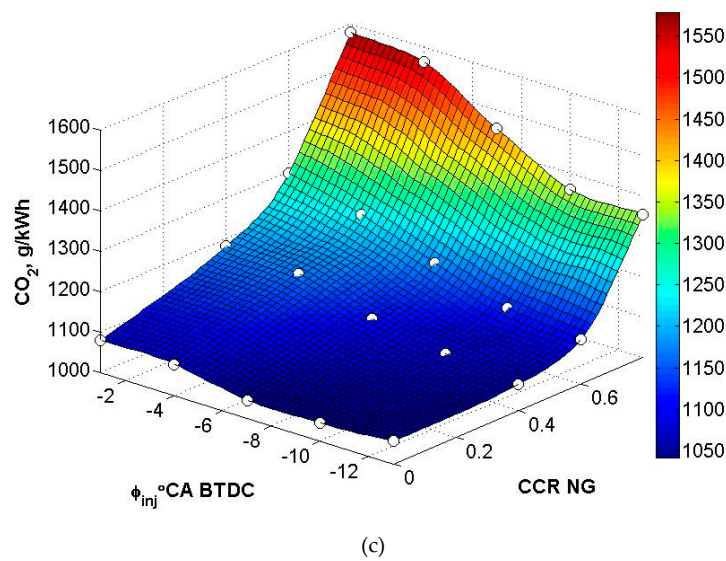


Figure 9. Influence of dual-fuel CCR NG portion and φ_{inj} on CO_2 emissions ($n = 2000 \text{ min}^{-1}$): (a) HLM; (b) MLM; (c) LLM. The dots in the graph represent the experimental data.

Table 8. Influence of investigated engine (79.5/95.5) conversion to operation with NG fuel feed on energy efficiency and emission parameters.

Operating mode: HLM $P_{me} = 5.98 \text{ bar}$								
Changes in engine parameters when CCR NG increases from 0 to 0.2								
Stationary $\varphi_{inj}, ^\circ\text{CA}$	Advanced to $\varphi_{inj}, ^\circ\text{CA}$	$\Delta\eta_e$	$\Delta P_{zr}, \text{bar}$	$\Delta\text{NO}_x, \text{g/kWh}$	$\Delta\text{CO}, \text{g/kWh}$	$\Delta\text{HC}, \text{g/kWh}$	$\Delta\text{CO}_2, \%$	$\Delta\text{CO}_2, \text{g/kWh}$
−1	−4	0	+10	−0.2	+4	+0.22	−9	−47
−1	−1	−3%	+5	−1.2	+5	+0.25	−5.5	−25
−4	−7	~0.7%	+7	+0.2	+3	+0.21	−9	−34
−4	−4	−3.5%	0	−1.7	+4	+0.22	−6.5	−16
Operating mode: HLM $P_{me} = 5.98 \text{ bar}$								
Changes in engine parameters when CCR NG increases from 0 to 0.4								
Stationary $\varphi_{inj}, ^\circ\text{CA}$	Advanced to $\varphi_{inj}, ^\circ\text{CA}$	$\Delta\eta_e$	$\Delta P_{zr}, \text{bar}$	$\Delta\text{NO}_x, \text{g/kWh}$	$\Delta\text{CO}, \text{g/kWh}$	$\Delta\text{HC}, \text{g/kWh}$	$\Delta\text{CO}_2, \%$	$\Delta\text{CO}_2, \text{g/kWh}$
−1	−4	−2.9%	+9	−0.9	+6	+0.45	−15	−63
−1	−1	−6.5%	+4	−1.7	+10	+0.5	−9	−49
−4	−10	−1.3%	+23	+2.6	+5.5	+0.42	−16	−72
−4	−7	−3.5%	+7	−0.6	+6	+0.45	−15	−60
−4	−4	−6.4%	0	−1.2	+8	+0.45	−12	−32
Operating mode: MLM $P_{me} = 3.98 \text{ bar}$								
Changes in engine parameters when CCR NG increases from 0 to 0.4								
Stationary $\varphi_{inj}, ^\circ\text{CA}$	Advanced to $\varphi_{inj}, ^\circ\text{CA}$	$\Delta\eta_e$	$\Delta P_{zr}, \text{bar}$	$\Delta\text{NO}_x, \text{g/kWh}$	$\Delta\text{CO}, \text{g/kWh}$	$\Delta\text{HC}, \text{g/kWh}$	$\Delta\text{CO}_2, \%$	$\Delta\text{CO}_2, \text{g/kWh}$
−1	−7	+1.5%	+24	+1.1	+9.0	+0.36	−6.2	−50
−1	−4	~0%	+9	−0.7	+9.7	+0.43	−3.9	−32
−1	−1	−6.5%	+1	−1.7	+10	+0.50	2.7	+24
−4	−10	0	+13	+2.1	+7.0	+0.30	−7.3	−56
−4	−7	−1.5%	+10	−1.1	+9.0	+0.35	−3.8	−30
−4	−4	−4%	0	−3.0	+9.9	+0.43	−1.5	−12

The evaluation encompassed different η_e compensation possibilities, from φ_{inj} advancement from full η_e establishment when the engine was operating on diesel to $\varphi_{inj} = idem$.

When diesel is exchanged for NG (CCR NG 0.2), it is more rational to use partial φ_{inj} advancement (with the purpose of re-establishing a proper η_e). In the $P_{me} = 1.99$ bar mode, where a decrease in η_e of 3% to 3.5% was observed, the increase in P_{max} did not exceed 5 bar. A significant decrease in NO_x was achieved at 1.2 g/kWh, the HC increase did not exceed 0.2 g/kWh, and the CO_2 emissions decreased. For the MLM with $P_{me} = 3.98$ bar, the η_e deterioration also did not exceed 3% to 3.5%, P_{max} increased by 7 to 9 bar, and CO_2 decreased by 15%.

A CCR NG portion increase to 0.4 it is more appropriate for use in partial φ_{inj} advancement. The engine parameter changes are not significantly different from those of the CCR NG 0.2 values. An effect of a decrease of approximately 15% in CO_2 emissions detected compared to operation on diesel only.

However, the HC emissions increase is not critical, because a valid standard regulates the total $NO_x + HC$ emissions [52]. The sum of the NO_x and HC emissions in the investigated engine conversion to NG was decreased, because the decrease in the absolute NO_x exceeded the increase in HC emissions (see Table 8). As a result, the only increase was in the CO emissions. Very few road transport engine CO emissions are regulated by secondary harmful emission control technologies (oxidation-type neutralisations) [45–47,51,52]. Moreover, marine-purpose power plant CO emissions are not regulated. However, DE conversion for dual-fuel operation purposes exhibits complex improvements in energy efficiency and emission values for conducting research and identifying rational solutions.

3.3. Engine Parameter Improvement Field Evaluation

With reference to previously conducted research on HRF, an injection timing advancement in the range of -1 to -13 °CA BTDC to improve converted CIE operation into dual-fuel is not very effective, because it is strongly related to an intensive increase in P_{max} and NO_x .

The search for rational engine parameters has mainly focused on improving the energy efficiency—particularly the operation indicated process efficiency η_i . Based on internal combustion engine classical theory rules [39,40,52,53], changes in η_i during experiments and improvement reserves are analysed in coordination with heat release dynamics in the engine cylinder, as well as the main operation process parameters (air excess ratio α , dynamic parameters such as P_{max} , and auto-ignition delay period φ_i , among others).

As established in Section 3.2, the increase in the CCR NG under $\varphi_{inj} = idem$ conditions influences the combustion process movement towards expansion stroke, and increases the combustion process period—one of the main η_i influencing factors (see Figure 10). The graph in Figure 10 contains integral heat release characteristic data in the relative form of $X = f(\varphi)$, and indicates a partially meaningful decrease about X during the expansion stroke, which is even more evident when there a larger portion of CCR NG and lower engine load exist.

Larger changes in $X = f(\varphi)$ are characteristic for low dynamic engine cycles ($\varphi_{inj} = 1$ to 4 °CA BTDC). For an earlier φ_{inj} , the differences among the $X = f(\varphi)$ characteristics decrease.

For change evaluation of the quantitative heat release characteristics, the relationship with the η_i uses 50% of the released heat phase CA_{50} in practice is commonly applied to engine cycle process energy efficiency evaluation [30,33]. A direct relationship exists between the CCR NG and CA_{50} parameter interaction: in the HLM, when CA_{50} increased from 0 to 0.8, an increase of 4 °CA occurred; in the MLM, an increase of 12 °CA occurred; and in the LLM, an increase of 42 °CA occurred. When increasing the engine cycle dynamics ($\varphi_{inj} = 13$ °CA BTDC), the CA_{50} exchange of CCR NG in equal parts decreased intensely; for example, up to 6 °CA and 14 °CA for the MLM and LLM, respectively.

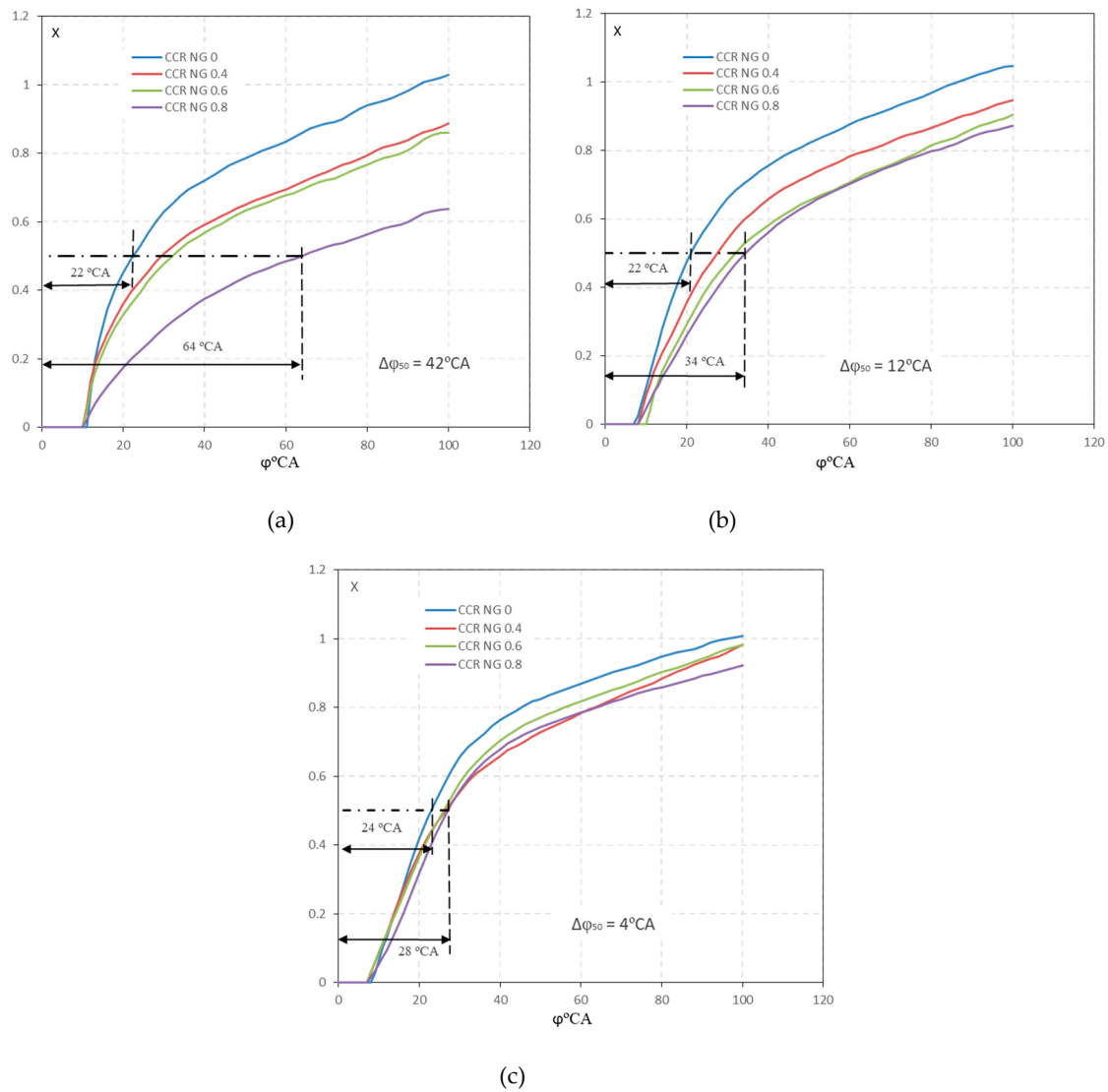


Figure 10. Dual-fuel engine CCR NG fuel portion influence on heat release characteristics: (a) HLM; (b) MLM; (c) LLM.

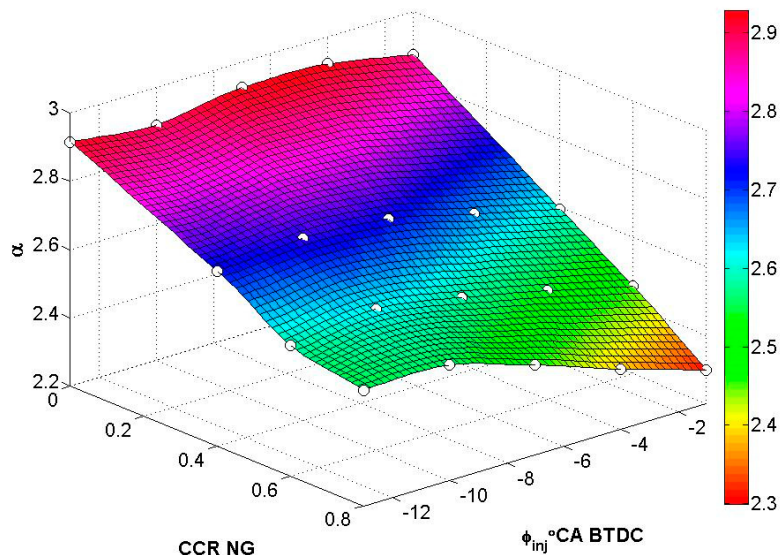
Based on the postulate that the ignition delay has a significant influence (φ_z , based on evaluation of the I.I. Vibe heat release model) on η_i [54–56], it is appropriate to analyse the relationship between φ_z and the engine cycle parameters. For this purpose, Woschi's mathematical modelling I.I. Vibe combustion period φ_z analytical dependency on the engine cycle implementation parameters was used [57,58]:

$$\varphi_z = \varphi_{z0} \left(\frac{n}{n_0} \right)^m \left(\frac{\alpha_0}{\alpha} \right)^k.$$

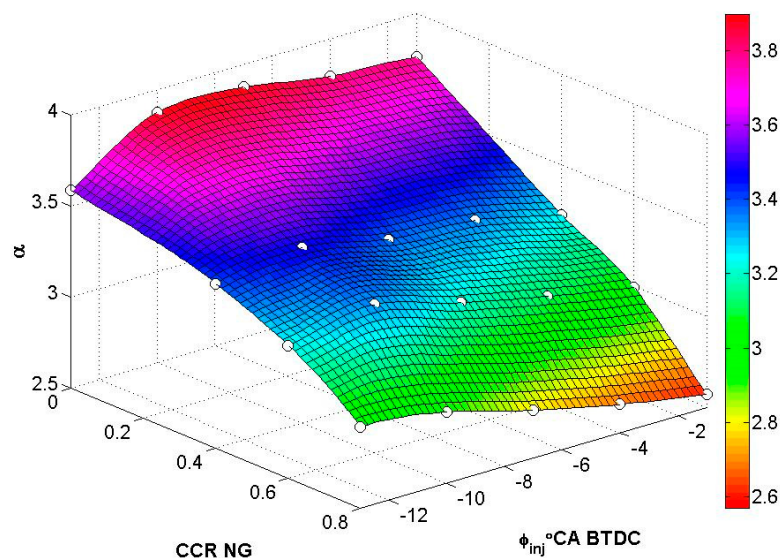
Here, the “0” index is assigned to the respective parameter values, and is also known as the “basis” engine cycle mode, for which it is usual to accept the nominal power mode; n indicates the engine revolutions; α is the excess air ratio; and m and k are engine constants over a wide range of fuel types [56,57]. It should be noted that the φ_z dependency was established based on a large scope of diesel engine experimental data.

In accordance with the φ_z expression during $n = idem$, the variable (α) remains the main influencing factor on the φ_z period, as well as when evaluating the relationship between η_i and φ_z , and the energy efficiency in the engine cycle parameter η_i [58,59]. A change in the CCR NG fuel portion in different load modes attests to the following (see Figure 11):

- An increase in the amount of CCR NG has an inverse effect on the α value, which can be explained by a partial exchange of air with NG, as well as NG having a larger stoichiometric coefficient compared to diesel—17 kg air/kg NG versus 14.6 kg air/kg diesel when the engine is operating in different modes (when the engine is operating in different modes, the CCR NG/air inlet pressure P_k remains constant).
- The largest effect from the CCR NG fuel portion on α was detected in the LLM, and decreased significantly when ϕ_{inj} was advanced.



(a)



(b)

Figure 11. Cont.

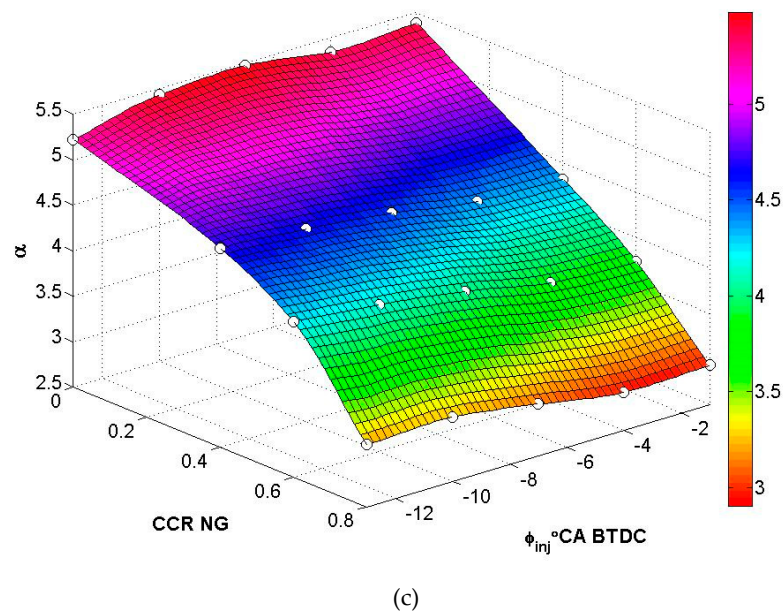


Figure 11. Influence of dual-fuel CCR NG portion and ϕ_{inj} on α ($n = 2000$ rpm) : (a) HLM; (b) MLM; (c) LLM. The dots in the graph represent the experimental data.

The changes in the parameter (α) are attuned with the changes in the parameter (η_i) arising from the same factors of CCR NG and ϕ_{inj} (see Figure 12 compared to Figure 11).

On this basis, the composed graphical η_i and α dependencies (see Figure 13a) confirm that α has a meaningful influence on the engine cycle η_i . The investigated engine load modes of $P_{me} - idem$ exhibited similar dependencies of $\eta_i = f(\alpha)$, independent of the CCR NG and ϕ_{inj} values. The determinant coefficient $R^2 = 0.8$ to 0.994 attests to the strong correlations of the η_i and α parameters.

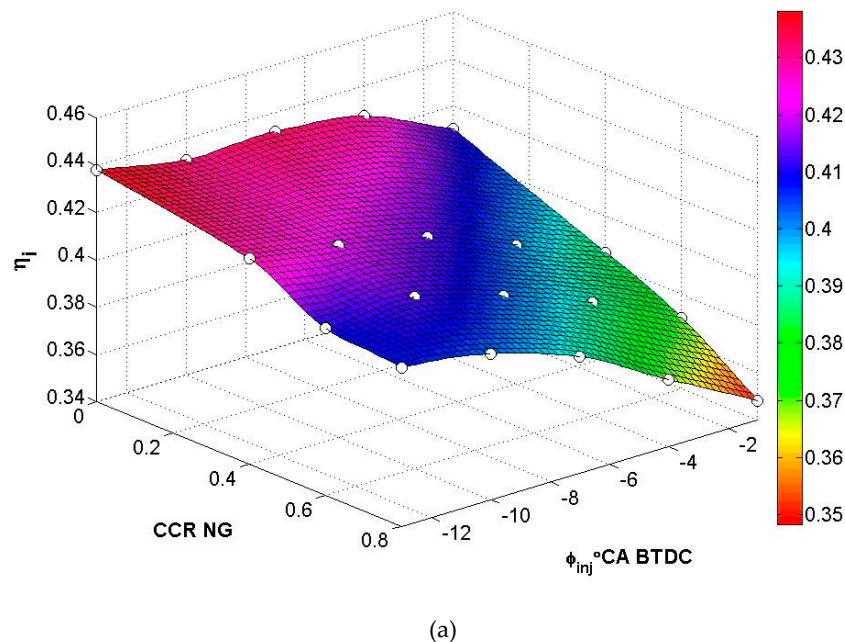
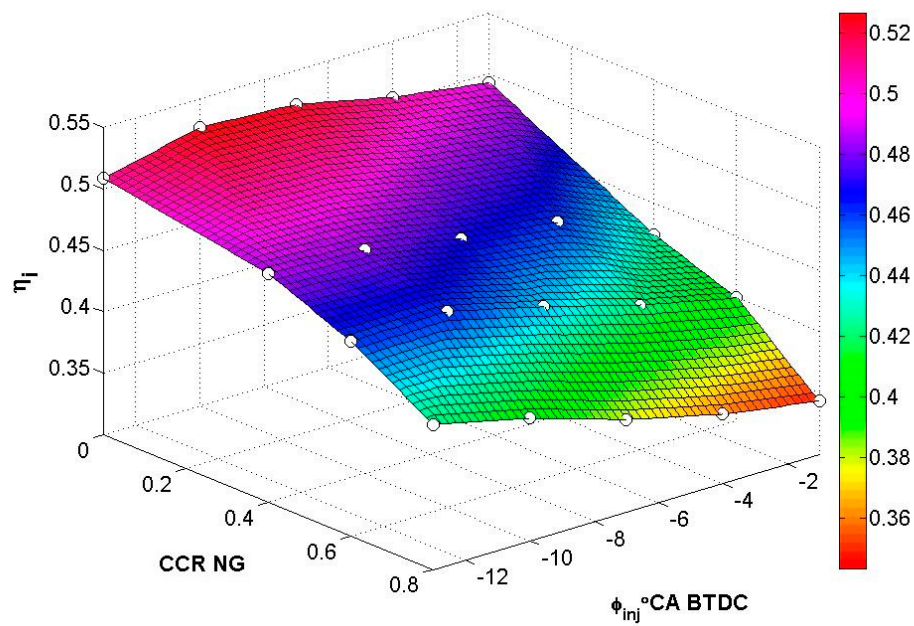
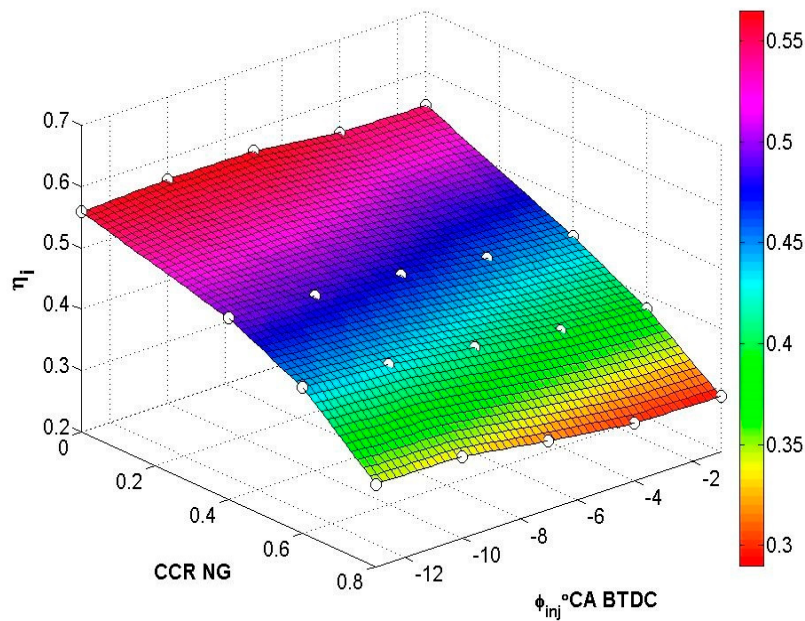


Figure 12. Cont.



(b)



(c)

Figure 12. Influence of dual-fuel CCR NG portion and ϕ_{inj} on α ($n = 2000 \text{ min}^{-1}$) : (a) HLM; (b) MLM; (c) LLM. The dots in the graph represent the experimental data.

The ratio change of the dependency of parameters $\bar{\eta}_i$ and α comes close to a functional with $R^2 \approx 1.0$ (see Figure 13b).

In a practical sense, the increase in the parameter (α) for a converted DE does not result in many technological difficulties. The increase in the supply of air to the ICE engine cycle can be ensured by changing the air compression equipment to that with a higher capacity or modifying the current compressor [52].

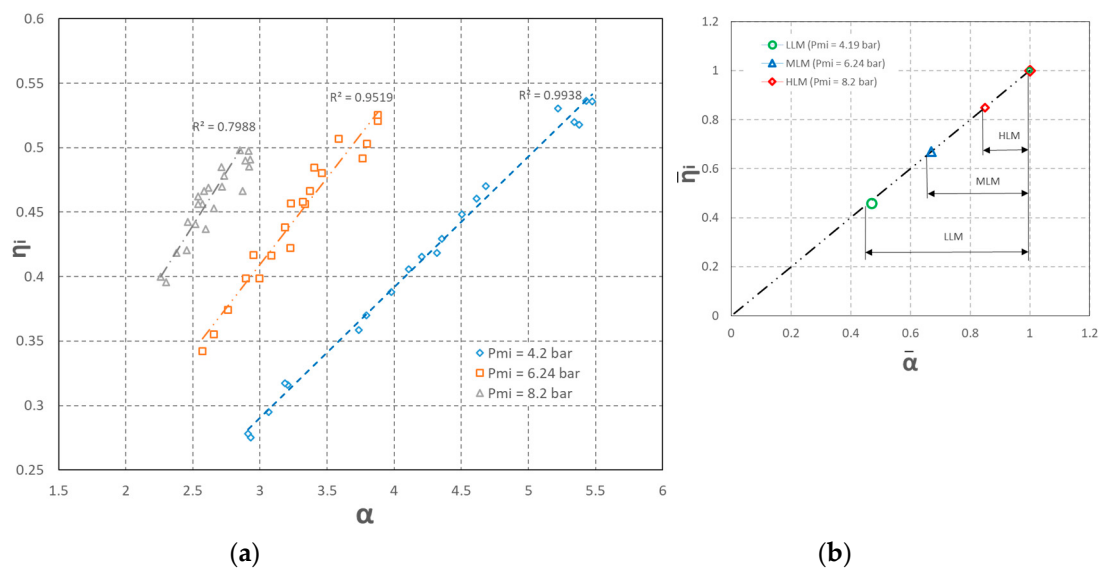


Figure 13. (a) Relationship between η_i and α during dual-fuel engine operation with different loads of CCR NG and φ_{inj} (CCR NG = 0 to 0.8; φ_{inj} = 1 to 13°CA BTDC). (b) Relationship between η_i and $\bar{\alpha}$ with dual-fuel engine operating on different loads of CCR NG and φ_{inj} .

The established relationship between η_i and α attests to the fact that α is not the only variable that forms the η_i value. The engine cycle energy efficiency is also determined by other factors, because it is demonstrated that an increase in the engine load, at the same values of α , leads to an increase in η_i (see Figure 13a). Additional factors influencing η_i may be other engine cycle dynamics parameters: P_{max} and the change in pressure $\lambda = P_{max}/P_c$ (here, P_c is the pressure at TDC) [58]. An increase in the load leads to an increase in the engine cycle dynamic parameters. It is also likely that an increase in the fuel macro- and micro-turbulence influences the combustion intensity positively [40,58]. During large load modes, an increase in the air inlet pressure and the respective air–NG mixture movement speed through the air inlet valve accelerates the fuel combustion main diffusion phase, resulting in an increase in η_i [59].

Micro- and macro-turbulence accelerate the active radical OH and local combustion centre diffusion into the cylinder periphery, ensuring an entire volume combustion process [40,60].

An analogous energy efficiency increase result was obtained in the works of [28,33], where a non-traditional CIE cycle HRF injection timing angle advancement of up to 50 °CA BTDC was used. As noted previously, the energy efficiency increase of a converted CIE for dual-fuel operation also manifests itself as a method for emission control when operating on NG. An increase in the air mass during the engine cycle leads to a decrease in the PM, CO, and HC emissions [19,61–63]. As well as the temperature field being equalised in the cylinder during combustion, there is also a decrease in the number of combustion centre concentrations in the cylinder, and as a result, a decrease in the NO_x emissions [27,33,63]. An important improvement effect on η_i is the greenhouse gas CO₂ emission decrease, as observed in the LLM and MLM, owing to the smaller portion of carbon in the chemical composition of NG of 0.75 compared to diesel with 0.86.

4. Conclusions

The energy efficiency and emissions of a converted (Bore 79.5 mm/Stroke 95 mm) CIE for operation on dual D and NG fuel were established through experimental research:

- A deterioration in the energy efficiency was established when increasing the NG portion in the energy balance up to 80% or the CCR NG up to 0.8: in the HLM, 7 to 15%; in the MLM, 15 to 38%, and in the LLM, 40 to 45%.

- An increase occurred in the incomplete combustion products in exhaust gases: the HC could exceed 30 times in the HLM and MLM and up to 80 times in LLM with 0.1 g/kWh while operating on D; the CO increased up to 30 times from 0.5 g/kWh; although the emissions of the most harmful pollutant, NO_x , decreased approximately 4 to 5 times.
- The pilot D portion HRF injection timing DIT optimisation with respect to energy efficiency could be practically effective only when it did not exceed a CCR NG of 0.4 when the engine was operating in the HLM ($P_{me} \sim 6$ bar): by advancing DIT by 3 to 6°CA, the $\text{NO}_x + \text{HC}$ emissions decreased by 10 to 15%; in the MLM and LLM, DIT optimisation did not have a positive effect.

The indicated process parameter analysis in the investigated CCR NG = 0 to 0,8 and DIT = −1 to −13°CA BTDC range attests to the fact that conversion of a CIE for NG operation does not undermine the CIE characteristic interdependent correlated cycle indicated efficiency and characteristic parameters (such as α and P_{max}). On this basis, one of the main reasons for the deterioration in η_i when the engine is converted for NG operation was the NG combustion phase increase in the expansion stroke, and therewith, the associated cylinder air excess deterioration (evaluated by the air excess ratio α).

The research results can be used for the evaluation of similar experiments, in which converted CIEs with conventional fuel injection systems for operation with dual-fuel feed without significant changes to the engine design were used.

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Nomenclature

k	Optical absorption coefficient (m^{-1})
n	Rotational speed of the crankshaft (min^{-1})
T_g	Exhaust gas temperature (K)
T_K	Air temperature after compressor (K)
P_e	Brake power (kW)
α	Excess air coefficient
β	Fuel excess coefficient
ε	Compression ratio
$\varphi_{inj}(\phi_{inj})$	High reaction fuel injection time (°CA)
H_U	Lower heating value (kJ/kg)
η_m	Mechanical efficiency coefficient
η_i	Indicated thermal efficiency
η_e	Brake thermal efficiency
P_{max}	Maximal cylindrical pressure (bar)
P_k	Air pressure after compressor (bar)
$\lambda = \frac{P_z}{P_c}$	Cylinder pressure increase rate
$X = f(\varphi)$	Relative heat release ratio (°CA)
$\frac{dQ}{d\varphi}$	Heat release rate (kJ/°CA)
P_{me}	Brake mean effective pressure (bar)
P_{mi}	Indicated mean effective pressure (bar)
P_{cyl}	Pressure in cylinder (bar)
CA_{50}	Half of heat released during cycle (°CA)
$(dP/d\varphi)_{max}$	Maximum pressure increase rate in cylinder (bar/°CA)
$(dP/d\varphi)_{mid.}$	Average pressure increase rate in cylinder (bar/°CA)

Abbreviations

CIE	Compression ignition engine
CR	Common rail fuel injection
CCR NG	Co-combustion ratio of natural gas
CC	Combustion chamber
DIT	Diesel fuel injection timing
HRR	Heat release rate
HRF	High reaction fuel
HLM	High load mode
MLM	Medium load mode
LLM	Low load mode
ULLM	Ultra-low load mode
BTDC	Before top dead centre
ATDC	After top dead centre
CA	Crank angle
CNG	Compressed natural gas
LNG	Liquefied natural gas
CO	Carbon monoxide
HC	Hydrocarbon
NG	Natural gas
NO _x	Nitrous oxide
CO ₂	Carbon dioxide

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