

Article

The Use of Hydrogen in the Automotive Diesel Engine—An Efficient Solution to Control Its Operation with Reduced Carbon Emissions

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Abstract

The use of hydrogen in the diesel engine automotive can be a viable and sustainable solution through which to reduce carbon-based emissions for the same operating regimens, without major constructive modifications of the engine. The use of hydrogen in the automotive diesel engine constitutes an efficient solution for controlling its operation with low carbon-based emissions and with reduced energetic specific consumption. This way, the requirements of the EC's Green Deal policy can be met without major costs. Substitution of fossils fuels with hydrogen ensures the reduction of the carbon content of the air–fuel mixture in the engine cylinder, with favourable influences on the processes of the air–fuel mixture formation and combustion, making it possible to reduce carbon-based emissions. The improvement of the combustion process due to hydrogen use leads to the reduction in carbon-based emissions. For the experimental investigations carried on an automotive diesel engine with 1.5 L by displacement, the authors highlighted the following results: a reduction in smoke emissions by over 22%, a reduction in unburned hydrocarbon emissions by over 25%, and a reduction in carbon dioxide by about 20%, even from the use of relatively low cyclic doses of hydrogen at a usual engine load compared with a standard engine. The authors also obtained an 18.5% increase in the level of nitrogen oxides in the partial load engine, even with the use of small amounts of hydrogen added to the intake air; this is a disadvantage, but by applying specific measures, this emission can be reduced. The increase in the homogeneous air–fuel mixture with the use of hydrogen and the combustion processes' duration time were reduced due to the high combustion flame speed of hydrogen comparative to diesel fuel; this ensures a reduction in energetic specific consumption and an increase in thermal efficiency.

Keywords: hydrogen; energetic specific consumption; efficiency; fuel consumption; pollutant emissions; carbon-based emissions; emission reduction



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1. Introduction

Actual Context of the Research

The search for alternative fuels for diesel engines has gained momentum, driven by the need to lower pollutant emissions and the possibility of improving energy efficiency.

Among alternative fuels, hydrogen may be a significant fuel due to its high energy density and clean combustion properties, establishing it as a viable solution for sustainable transportation. Utilizing hydrogen in dual-fuel diesel engines, where hydrogen substitutes the classic diesel fuel, offers a pathway to significantly diminish harmful emissions, particulate matter, nitrogen oxides (NO_x), and greenhouse gases like CO_2 , thereby potentially improving engine performance.

Recent studies have demonstrated the potential benefits of using hydrogen as a fuel in diesel engines. The integration of hydrogen alongside fuels used for diesel engines by using dual-fuelling systems has emerged as a promising strategy through which to reduce greenhouse gas emissions and enhance engine performance. This approach involves the simultaneous combustion of diesel fuel and hydrogen, leveraging hydrogen's high energy content and low carbon footprint to achieve more sustainable operation of the engine.

The retrofitting of existing diesel engines to operate in hydrogen–diesel dual-fuelling systems has also emerged as a key area of focus. This approach offers a cost-effective pathway through which to reduce emission levels without the need for complete engine replacement. Liu et al.'s [1] study showed a 85% reduction in CO_2 emission levels for a diesel engine fuelled with diesel fuel and H_2 using a 2000 rev/min speed regime. Liu et al. [1] showed the feasibility of transitioning to cleaner fuel options, like hydrogen, within existing infrastructure [1]. The engine efficiency increased by 13% with H_2 fuelling [1]. However, the NO_x emission level increased as a result of the increase in the share of the premixture phase of combustion [1].

Wright et al. [2] developed a study to analyse the influence of the air–hydrogen mixture quality and diesel fuel quantity on a heavy-duty diesel engine. Wright et al. [2] showed that the use of 40% H_2 may lead to a substantial reduction in carbon dioxide emission levels, but there was a 24% increase in NO_x emission levels at low loads. The increase in the NO_x emission level may require careful management of combustion. At high loads, the NO_x emission level may increase up to 107%, and exhaust gas recirculation may be a solution for reducing NO_x emission levels [2]. H_2 manifold injection can provide a 40% decrease in CO_2 emission levels for Hydra engines [2].

Akhtar et al. [3] conducted a comprehensive modelling study on a 4.4 L heavy-duty diesel engine, revealing that at engine speeds above 1700 rev/min and a hydrogen mass fraction of 17.5%, there were notable increases in brake power and torque by up to 17% and 16.5%, respectively. However, this enhancement was accompanied by a 43% increase in the NO_x emission level at a speed of 2200 rev/min, indicating a trade-off between performance gains and controlling emission levels [3].

Mukhtar et al.'s [4] experimental investigations carried out on a single-cylinder diesel engine showed the effects of hydrogen inlet injection on operation at 1800 rev/min when diesel fuel and hydrotreated vegetable oil (HVO) were used as pilot fuels. The study showed that hydrogen supplementation led to a reduction in smoke emission levels, with HVO as the pilot fuel resulting in lower smoke levels compared to diesel fuel [4]. Nonetheless, challenges such as engine knocking at lower hydrogen–excess air ratios were observed, underscoring the necessity of precisely controlling operating parameters to optimize performance and mitigate adverse effects [4].

Computational analyses have provided further insights into the impact of hydrogen enrichment on combustion dynamics. Zhang et al. [5] developed numerical simulations to assess diesel fuel–hydrogen dual-fuel engines, demonstrating that combined injection strategies could simultaneously reduce NO_x emission levels and improve indicated mean effective pressure (IMEP). These findings suggest that strategic optimization of fuel injection parameters is crucial for balancing performance enhancements with emission reductions [5]. The integration of hydrogen use as fuel for diesel engines is not without challenges. Issues

such as backfire, premature ignition, and detonation at high loads have been reported, which can impede the rapid development and deployment of hydrogen-fuelled internal combustion engines. Studies by Zhang et al. [5] indicate that while hydrogen supplementation significantly improves thermal efficiency, it also raises combustion temperatures, leading to increased NO_x emission level. Therefore, ongoing research is essential to address these technical hurdles and to develop robust engine designs capable of safely and efficiently utilizing hydrogen as a supplementary fuel [5]. In the case of H_2 addition, for 10% H_2 and 20% H_2 , the in-cylinder maximum pressure, the peak of HRR, and the combustion duration were increased [5]. When using H_2 as an additional fuel, the rapid pace of combustion was pushed later on in the cycle, from 4.8 CAD to 5.1 CAD [5]. For a 30% H_2 level, which is the level in soot, CO and HC emissions were reduced sharply. In the case of H_2 substitution, for 30% H_2 , 60% H_2 , and 90% H_2 (which replace the diesel fuel), the maximum pressure increased above 280 bar, and the peak pressure angle was closer to TDC [5]. The HRR peak appeared later on in the combustion cycle, with higher values, especially for 90% H_2 . The combustion duration decreased, and the rapid combustion phase appeared before TDC (from 4 CAD to -3.9 CAD) for the case in which H_2 was used as a substitution fuel [5]. In the case of substitution of diesel fuel with H_2 , the CO, CO_2 , and HC were decreased, but the NO_x was increased in a more significant way, reaching 100% for 30% H_2 , 250% for 60% H_2 , and 480% for 90% H_2 [5]. The NO_x emission level was reduced only when the pilot injection timing was tuned, with a 53% decrease being achieved for a 10 CAD delay in injection timing [5].

A fundamental aspect of hydrogen's applicability in internal combustion engines lies in its physico-chemical properties, which distinguish it from conventional hydrocarbon fuels. Hydrogen possesses a notably high energy content per unit mass, at approximately 120 MJ/kg, which is nearly three times that of diesel fuel, making it a highly efficient energy carrier, according to the work of Alzohbi [6]. Additionally, its wide flammability range (4–75% by volume in air) enables lean combustion, promoting fuel efficiency and reducing carbon-based emissions, according to Badea [7]. The high flame speed of air–hydrogen further enhances its combustion characteristics, contributing to improved engine efficiency but necessitating precise ignition control to mitigate issues such as backfire and pre-ignition, according to Han et al. [8]. Hydrogen's high diffusivity ensures rapid mixing with air, facilitating homogeneous combustion and lowering particulate emissions, whereas its low ignition energy (0.02 mJ) increases the likelihood of premature ignition, requiring sophisticated engine management strategies, according to Levikhin [9].

The use of hydrogen as an alternative fuel for diesel engines also necessitates careful consideration of fuelling methods, which significantly influence combustion characteristics and overall engine and fuelling system efficiency. Among the commonly explored techniques, port fuel injection (PFI) provides the injection of hydrogen into the intake valve port, promoting thorough mixing with air before combustion. While this approach is relatively simple, it can reduce volumetric efficiency due to hydrogen's displacement of intake air, according to Mekonnin et al. [10]. High-pressure storage and cost remain issues related to H_2 use on automotive boards, according to Mekonnin et al. [10]. Mohamed et al. [11] studied port injection and direct injection of hydrogen and showed that increased efficiency and decreased carbon emissions are possible. For an SI engine, the direct injection (DI) of hydrogen into the combustion chamber under high pressure offers superior control over combustion timing, mitigating premature ignition risks, though it demands advanced injector designs and high-pressure storage systems [11]. For hydrogen DI, the efficiency increases by 2%, and backfire issues are limited [11]. Another viable method, fumigation, entails injection of the hydrogen into the intake air stream before the intake valve, making adaptation of existing diesel engines to H_2 fuelling easy but causing risk of uncontrolled

combustion events if not properly managed. Barik et al. [12] used a 0.2 lpm H_2 flow to fuel the inlet of a diesel engine fuelled by diesel fuel–diethyl ether pilot in percents of 5–25% [12]. Hydrogen fumigation in a compression ignition engine has been shown to improve combustion, increase thermal efficiency, and reduce emissions like hydrocarbons, carbon monoxide, and smoke, demonstrating its potential for cleaner and more efficient dual-fuel operation compared with classic fuel supply [12]. When using hydrogen, an increase in maximum pressure up to ~67 bar and of the HRR up to 32 J/CAD may appear. Compared with diesel fuelling, when using hydrogen, the thermal efficiency increases by 0.6%, and the brake-specific fuel consumption decreases by 3.7%. The CO emission level decreases by 29%, and the HC emission level is reduced by 35% with hydrogen use at full load, compared to the case in which hydrogen is not used. However, when using hydrogen, the level of the NO emission increases by 29%, and that of CO_2 emission increases by 17% [12]. The hydrogen fractions reaching the exhaust system can influence the state of the exhaust gases. Mohamed showed that possible hydrogen leaks into the exhaust system increase significantly when using lean mixtures and high engine loads [13]. The experiment of Mohamed et al. [14] showed that hydrogen leaks into the engine crankcase can reach 100,000 ppm if the classic crankcase ventilation system is maintained [13]. By adopting forced crankcase ventilation, an 86% reduction in the amount of hydrogen accumulated in the engine crankcase is ensured [13]. Recovering hydrogen from the crankcase through a forced ventilation system ensures a 1.2% increase in the effective efficiency [13].

Kumar et al. [14] showed that the presence of high amounts of hydrogen leads to an increase in exhaust gas temperature compared to diesel fuel, especially when using 30% hydrogen. For 30% H_2 , the levels of CO and CO_2 emissions were reduced, but the level of NO_x emissions increased slightly [14].

Armbruster et al. [15] showed that valve overlap and combustion chamber crevices around the piston head are the main sources of hydrogen slip. In modelling, Armbruster et al. [15] showed that there is no direct correlation between an accidental increase in the flow rate of hydrogen leaks and phenomena such as preignition and detonation [15]. When there are large and continuous leaks of H_2 above 34,000 ppm, preignition or detonation are not present, but exhaust misfires occur [15]. In this case, the effective efficiency of the engine is reduced, and the content of unburned H_2 in the exhaust gases increases. Armbruster et al. [15] showed that H_2 leaks cause exhaust misfires in 93% of cases, preignitions in 8% of cases, and detonation in 12% of cases, depending on the engine's operating mode. These undesirable phenomena occur when H_2 leaks exceed 3000 ppm and the separate phenomena of preignition and detonation do not favour an increase in unburned H_2 in the exhaust gases [15]. During these phenomena, the air– H_2 mixture ignites and burns; it does not remain unburned to be found in the exhaust system. However, high H_2 leaks, from 14,000 ppm to over 30,000 ppm, no longer allow complete combustion of H_2 in the cylinder, with the presence of exhaust misfires being frequent [15]. Reducing the duration of overlapping valve openings can reduce the frequency of exhaust misfires [15].

Mohamed [16] used only hydrogen to fuel an SI engine and obtain an extremely low level (near zero) of CO, HC, and CO_2 emissions. With low loads, H_2 emissions are below 1000 ppm, and with high loads, H_2 emissions are below 500 ppm; this is because the combustion process is more stable when the engine is fuelled with hydrogen compared with classic fuel use when using a stoichiometric mixture [16].

Zambelli [17] shows that use of a lean mixture offers the opportunity to decrease inlet backfire when using hydrogen use, if it is initiated by preignition phenomena. For Euro VI engines fuelled by hydrogen, it may be necessary to use only the catalyst to reduce NO_x emissions, since the CO, HC, and particulate emissions are reduced and within norms [17].

Mohamed et al. [18] used forced ventilation of the engine crankcase, which cuts off the hydrogen supply if the hydrogen concentration exceeds 4%. During the experiment, the maximum exhaust gas temperature exceeds 700 °C, and hydrogen leaks must be limited to 6000 ppm to avoid burning in the exhaust system [18]. Mohamed et al. [18] obtained a slightly increased NO_x level for mixtures with lambda 1.5–2, but when using leaner mixtures with lambda 2.2–3.5, the NO_x emission level was considerably reduced.

2. Experimental Investigations

2.1. Scope of the Experimental Research

The scope of the actual experimental research is a reduction in carbon emissions through the use of small hydrogen doses. If a simple method can be used for hydrogen fuelling and can be easily applied to many diesel engines in order to reduce their exhaust emissions, the future of the diesel engine can be secured. If manifold hydrogen injection can provide a decrease in carbon-based exhaust emissions, in the form of HC, smoke, and CO₂, then we can present advantages to keeping diesel engines in use. The method chosen for hydrogen fuelling must be simple and easy to apply to new- or old-generation engines. The use of small H₂ doses injected into the inlet manifold can be practically obtained through the use of an automotive on-board hydrogen generator. There are many hydrogen generators available on the market, and these can be used in the design of hydrogen fuelling systems. First of all, it is necessary to verify the possibility of operating the diesel engine with initial additions of hydrogen in the intake air, to establish the amount of hydrogen that can be used under conditions in which the level of emissions from the exhaust gases can be reduced. Then, experimental research can be extended to several operating modes and settings of the hydrogen supply system.

For the first stage of the experimental investigation, the hydrogen used for fuelling was stored in a pressurized tank at 150 bar. The paper presents an analysis of the hydrogen–diesel dual-fuel system, focusing on the effects of hydrogen addition on engine performance and emissions. The insights derived from this research are intended to inform the development of optimized dual-fuel strategies that effectively balance performance improvements with environmental considerations. This paper presents the experimental results of the influence of the hydrogen substitution on carbon-based emissions levels and the energetic performance of a diesel engine.

2.2. Methodology of the Experimental Investigations

The experimental investigations were carried on a Renault automotive diesel engine type K9K with 1.497 L displacement and supercharged at a pressure of 1.8 bar, which develops 50 kW at 4000 rev/min. The principal diagram of the experimental setup is presented in Figure 1. The engine is equipped with a secondary fuelling line which is used for hydrogen inlet manifold injection. The hydrogen tank is pressurized at 150 bar and is equipped with two pressure regulators. The high-pressure regulator controls the hydrogen fuelling pressure in the domain of 150–10 bar, and the low-pressure regulator controls the hydrogen fuelling pressure in the domain of 10–0.5 bar. A flame trap is mounted in front of the hydrogen tank before hydrogen injectors in order to ensure operational safety during experiments. As a technical modification, the inlet manifold is equipped with electromagnetic injectors for hydrogen fuelling. The electronic control unit of the engine (ECU) is connected to an open electronic control unit manufactured by Dastek Unichip. Through this Dastek Unichip unit, the engine diesel fuel injectors and the external injectors used for hydrogen can be controlled in terms of opening duration and timing. With dual fuelling, when diesel fuel and hydrogen are used, the substitution of classic fuel with hydrogen is possible. Thus, with the same operating regime and with a decrease in the diesel fuel cyclic

dose, the cyclic dose of hydrogen is increased until the engine power is restored to the reference value. The reference value is established upon diesel fuel fuelling. Changing the cyclic doses of the fuels is possible by changing the opening times of the respective injectors. The diesel fuel injection strategy remains unchanged during experimental investigation, and only the diesel fuel quantity is reduced with H₂ use, in order to observe if the hydrogen can be used with minimal modifications in injection management.

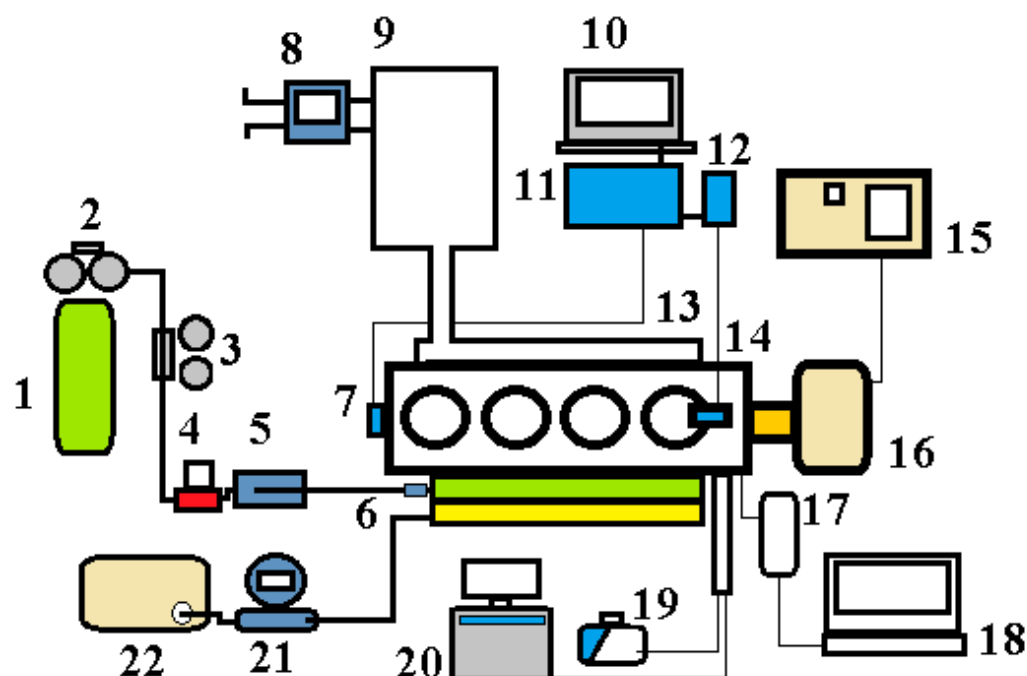


Figure 1. Hydrogen engine test bench diagram.

The share of hydrogen that substitutes the diesel fuel is determined by the energetic substitution ratio, x_E , according to the following relation:

$$x_E = \frac{C_{hH_2} \cdot H_{iH_2}}{C_{hH_2} \cdot H_{iH_2} + C_{hDF} \cdot H_{iDF}} \cdot 100 [\%] \quad (1)$$

where

C_{hH_2} —hydrogen consumption in kg/h;

H_{iH_2} —lower heating value of hydrogen in kJ/kWh;

C_{hDF} —diesel fuel consumption in kg/h;

H_{iDF} —lower heating value of diesel fuel in kJ/kWh.

The experimental investigations were carried out at a regime of 2500 rev/min engine speed and a 37% load for different energetic substitution ratios in the area of $x_E = 0$ –41.3%, among which the value $x_E = 0\%$ represents diesel fuelling.

The main components of the experimental test bench presented in Figure 1 are as follows: 1—hydrogen tank, 2—high-pressure reducers, 3—low-pressure reducers, 4—Alicat Scientific MC50 flowmeter for hydrogen (Tucson, AZ, USA), 5—flame arrestor, 6—hydrogen injector, 7—AVL angle encoder 650 CC (Graz, Austria), 8—Krohne inlet air flowmeter (Duisburg, Germany), 9—engine inlet air tank, 10—computer of the data acquisition system, 11—AVL Indimodul 621 data acquisition system (Graz, Austria), 12—AVL piezo-transducer amplifier (Graz, Austria), 13—AVL GU 13P piezoelectric pressure transducer (Graz, Austria), 14—K9K diesel engine, 15—Horiba Schenck control cabinet (Oberursel, Germany), 16—Horiba Schenck E90 dynamometer (Oberursel, Germany), 17—Dastek Unichip electronic control unit (Pretoria, Republic of South Africa),

18—computer of the Dastek Unichip electronic control unit (Pretoria, Republic of South Africa), 19—wireless AVL opacimeter (Graz, Austria), 20—AVL MDS 450 gas analyser (Graz, Austria), 21—Optimass flowmeter for diesel fuel (Duisburg, Germany), and 22—diesel fuel tank.

The experimental instruments were calibrated before starting the experimental session. The accuracy of the measuring devices is presented in Table 1.

Table 1. The errors of the experimental devices.

Test Bed System	Operational Parameter	Errors
Power system		
Horiba E90	Torque [Nm]	±0.2%
	Speed [rpm]	±1 rpm
Data acquisition system		
AVL GU 12P	Cylinder pressure [bar]	±0.05%
AVL 365 CC	Crankshaft angular position [CAD]	±0.1%
Fluid flowrates		
Alicat Scientific MC50	H ₂ flowrate [slpm]	±0.2%
Krohne Optimass 3300	Diesel fuel flowrate [kg/h]	±0.1%
Krohne H 250	Inlet air flow [m ³ /h]	±0.3%
Exhaust Gases	Exhaust gas emission	±0.65%vol for CO ₂
AVL MDS 450		±0.10 ppm vol for HC
		±50 ppm vol for NO
		±0.1% for opacity
		0.001 for lambda
Temperatures of fluids		
Thermocouple	Inlet air/cooling water	±1 °C degree
Cromel–Alumel TTC		
Thermoresistance PT100 TTR	Exhaust gases	±2 °C degree
Shimaden SR93	Digital display	±0.3%

For each operating point defined by an x_E value, a set of 200 individual data were recorded, presenting interest parameters such as cylinder pressure, angle of peak pressure, pressure rise rate, and heat release. For parameters such as fuel consumption, level of pollutant emissions, greenhouse gases, and temperatures, five datasets were recorded. Based on these, the study of combustion involves the analysis of the heat release rate, the reaction heat release law, the total duration of the combustion process, and also the influence of the energetic substitute ratio value on the level of carbon-based emissions.

The evaluation of the errors of the results is carried out using the following Formula (2) [13]:

$$\Delta_i = \frac{\delta_i}{v_i} \quad (2)$$

$$\delta_i = \frac{1}{n} \cdot \sum_{i=1}^n \left(r_i - \frac{\sum_{i=1}^n r_i}{n} \right)^2 \quad (3)$$

$$v_i = \frac{\sum_{i=1}^n r_i}{n} \quad (4)$$

The Δ_i is used to determinate the accuracy of the measurement results for a set of five measurements taken for each operating regime, and its values did not exceed 1%.

The diesel fuel consumption was 4.96 kg/h at $x_E = 0\%$, 4.46 kg/h at $x_E = 8.4\%$, 3.65 kg/h at $x_E = 23.5\%$, and 2.79 kg/h at $x_E = 41.3\%$. The hydrogen flow was injected into the inlet manifold by using an electronic injector unit. The electronic injector unit was mounted in the last section of the inlet line, closer to the inlet valves. Thus, the air–hydrogen mixture was easy to create and was admitted into the cylinder. The diesel fuel was energetically substituted for hydrogen with an energetic percentage of 8.4%, 23.5%, and 41.3%. These values correspond to hydrogen consumption flows of 0.1438 slpm at $x_E = 8.4\%$, 0.396 slpm at $x_E = 23.5\%$, and 0.689 slpm at $x_E = 41.3\%$. The hydrogen concentration in the in-cylinder inlet air is 0.121% for $x_E = 8.4\%$, 0.347% for $x_E = 23.5\%$, and 0.608% for $x_E = 41.3\%$.

For the experimental K9K hydrogen-fuelled engine, there was no overlapping valve opening. The intake valve opening timing was 9 CAD before TDC, and the exhaust valve closing timing was 27 CAD before TDC; these valves did not have a simultaneous opening period.

This aspect represents a novelty of this work and offers the advantage of eliminating hydrogen losses during the overlapping opening of the intake and exhaust valves. Given that the intake and exhaust valves were not opened simultaneously, and there was no cylinder sweep and hydrogen leaks, one can refer to internal exhaust gas recirculation. The hydrogen flow rate used was established to restore engine power when the engine was fuelled with diesel, and it thus compensated for any possible losses of hydrogen to the engine crankcase. The hydrogen leaks were imperceptible, considering that the engine operation was normal, without exhaust misfires. Any hydrogen leaks from the crankcase were taken over by the crankcase ventilation system, which remained unchanged, and the gases were returned to the intake before then returning to the cylinder. When analyzing the exhaust gases, no increases in HC emissions were recorded when hydrogen was also supplied.

The optimal correlation among engine load–engine speed–hydrogen cyclic dose–diesel fuel dose–supercharging pressure–exhaust gas temperature–excess air coefficient while using the investigated regime can be considered a novelty aspect of this research. Another novelty is the use of electronically assisted injection for hydrogen supply compared to the inlet–carburettor supply solution, which is specific to the diesel gas method. Another novelty aspect is that for the experimental diesel engine, there was no overlapping valve opening, and this introduced the advantage of eliminating hydrogen losses during the overlapping opening of the intake and exhaust valves.

3. Study Results and Discussion

Results of the Experimental Study

The in-cylinder pressure diagrams, which were obtained by averaging 200 individual smoothed diagrams, are presented in Figure 2 for different energetic substitute ratios.

When fuelled with hydrogen and diesel fuel, the pressure in the cylinder changes significantly, the growth slope becoming steeper, and the maximum values increasing. The maximum value tends to approach the Top Dead Centre (TDC), which may correspond to a faster, more brutal combustion in the presence of air–hydrogen mixtures established in the cylinder at the moment of autoignition of the diesel dose. This trend is amplified by the increase in the cyclic hydrogen dose when x_E increases. Moreover, it can be observed in an individual cycle that at maximum x_E , the pressure oscillations produced by the brutal combustion are amplified, as shown in Figure 2. Similar results were obtained by other researchers [4,6].

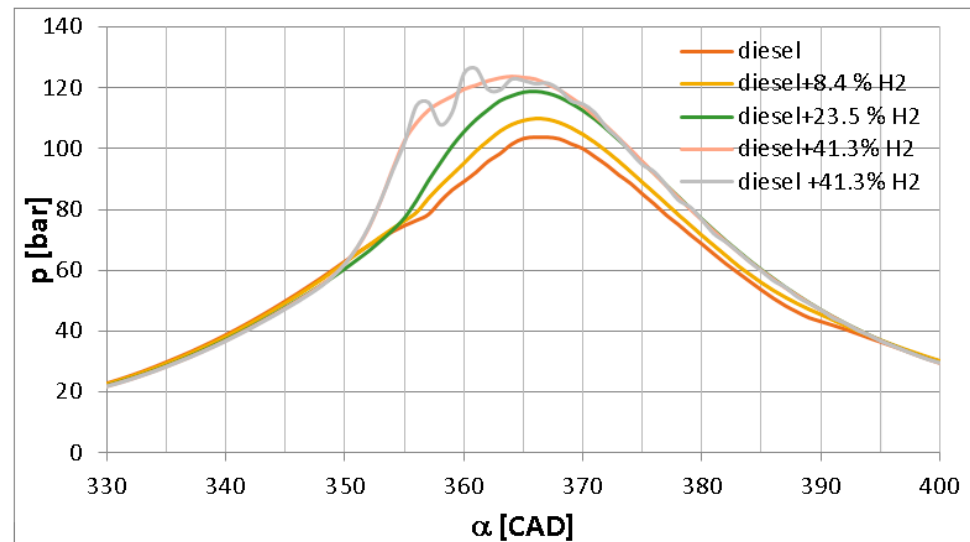


Figure 2. Indicated diagrams of in-cylinder pressure for different energetic substitution ratios of the diesel fuel with hydrogen.

At a higher hydrogen fuelling angle, the maximum pressure appears sooner, with a per-cycle rate of 1 CAD at $x_E = 8.4\%$, 2 CAD at $x_E = 23.5\%$, and 4 CAD at $x_E = 41.3\%$. The approach of the maximum pressure angle to the TDC can be correlated with the increase in the share of the premixed phase of combustion for the preformed mixtures produced by hydrogen fuelling.

The value of the maximum pressure increases by 5.98% at $x_E = 8.4\%$, by 14.77% at $x_E = 23.5\%$, and by 19.11% for $x_E = 41.3\%$ comparative to $x_E = 0\%$, as Figure 3 shows. The higher combustion speed of the air–hydrogen mixtures compared to that of the air–diesel vapor mixtures but with the same excess air coefficient can lead to a reduction in combustion duration, meaning the use of hydrogen increases efficiency.

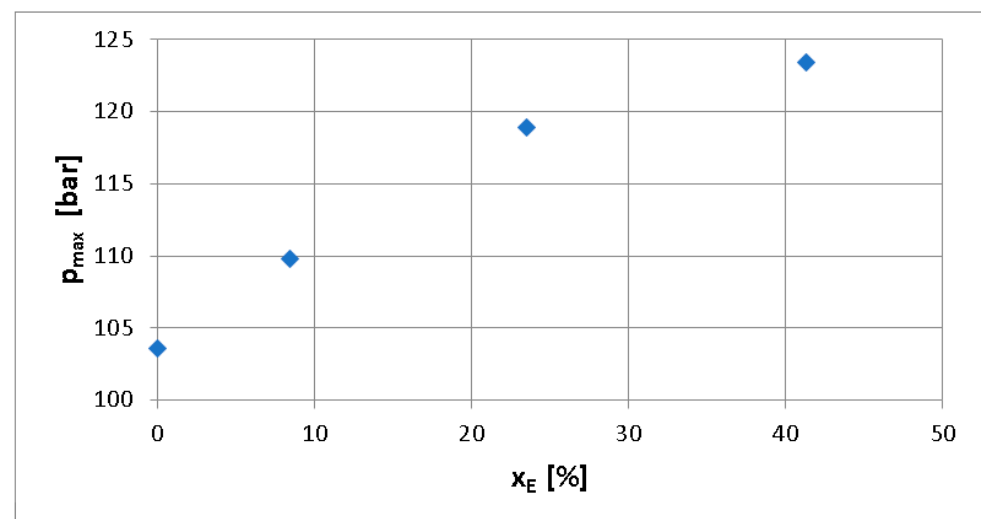


Figure 3. Maximum pressure for different energetic substitution ratios of the diesel fuel with hydrogen.

Higher hydrogen combustion rates influence the variation in the maximum pressure rise, as shown in Figure 4. When using hydrogen fuelling, the maximum pressure rise rate increases by 6.51% at $x_E = 8.4\%$, by 50.93% at $x_E = 23.5\%$, and by 23.94% at $x_E = 41.3\%$, compared to $x_E = 0\%$, as shown in Figure 4. It is noted that there is a correlation between the trends of decreasing maximum pressure angle, increasing maximum pressure, and increasing maximum pressure rise rate. Although the recorded values do not exceed

the range of usual values specific to diesel engines, the increasing trend recorded in the maximum pressure values and the pressure increase rate may be a criterion for limiting the cyclic hydrogen dose—at $x_E = 41.3\%$ —in order to maintain engine reliability. Similar results were obtained by other researchers [5,12].

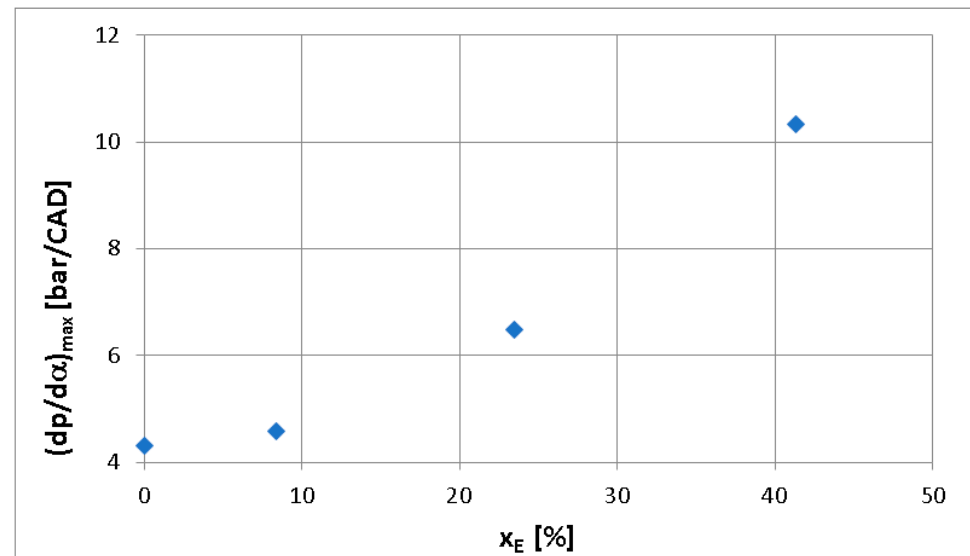


Figure 4. Maximum pressure rise rate for different energetic substitution ratios of the diesel fuel with hydrogen.

The higher value of the LHV of hydrogen, large inflammability limits, and higher flame speed comparative to diesel fuel lead to an increase in the HRR peak when using hydrogen, as shown Figure 5. For hydrogen fuelling, the peak of the increase in heat release rate was 45.77% at $x_E = 41.3\%$. For the energetic substitution ratios $x_E = 8.4\%$ and $x_E = 23.5\%$, similar values were registered, but a tendency to accelerate the combustion process in the presence of hydrogen was observed for all energetic substitution ratios, with the peak of HRR occurring earlier in the cycle, as shown in Figure 5.

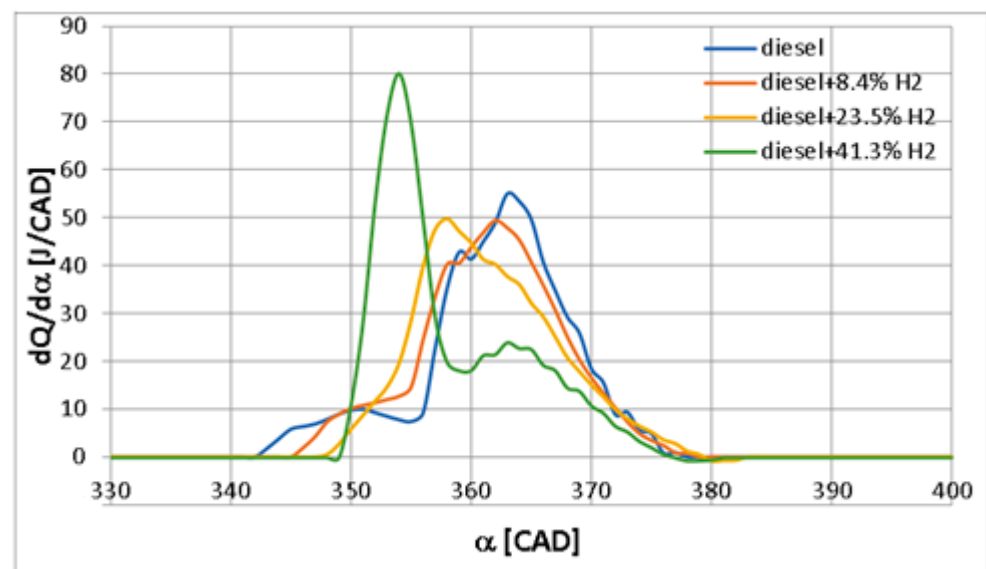


Figure 5. Heat release rate diagrams for different energetic substitution ratios of the diesel fuel with hydrogen.

The angle at which the maximum value of the heat release rate occurs was reduced to 1 CAD at $x_E = 8.4\%$, to 5 CAD at $x_E = 23.5\%$, and to 9 CAD at $x_E = 41.3\%$. The reduction

in the angle of maximum HRR correlates with the reduction in the angle at which the maximum pressure occurs, as shown in Figures 2 and 3. Similar results were obtained by other researchers [5,12].

The heat release law became more aggressive when the hydrogen cyclic dose was at its maximum of $x_E = 41.3\%$, and the rapid phase of combustion process increased its share at $x_E = 41.3\%$, as shown in Figure 6.

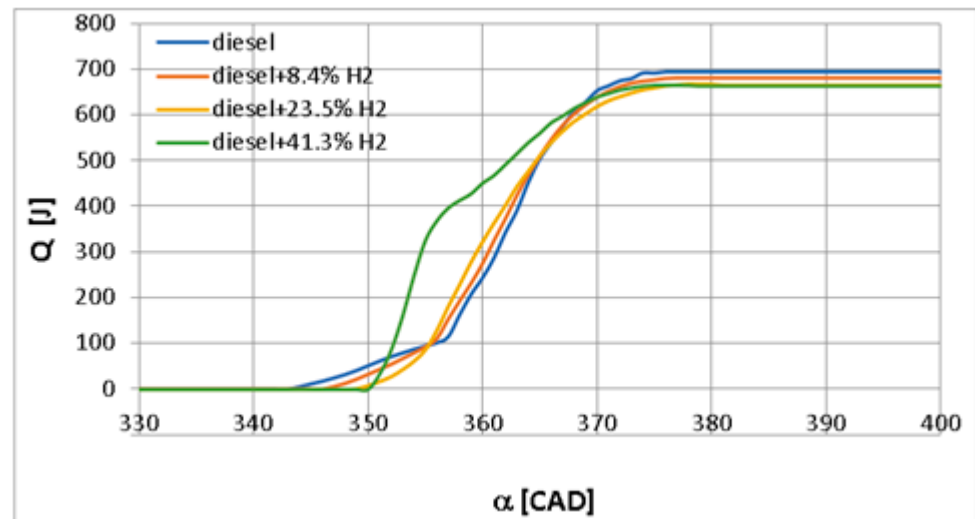


Figure 6. Heat release law diagrams for different energetic substitution ratios of the diesel fuel with hydrogen.

Due to H_2 's higher flame speed and wider inflammability limits, the total duration of combustion was reduced when using hydrogen, as shown in Figure 7. The combustion duration decreased to 2 CAD at $x_E = 8.4\%$, to 3 CAD at $x_E = 23.5\%$, and to 8 CAD for $x_E = 41.3\%$ versus classic fuelling, $x_E = 0\%$. The duration of the combustion process reduced by 5.7% at $x_E = 8.4\%$, by 8.5% at $x_E = 23.5\%$, and by 22.8% for $x_E = 41.3\%$ comparative to diesel fuelling. Reducing the combustion duration led to an increase in maximum pressure and maximum pressure rise rate when using H_2 , as shown in Figures 3 and 4. Similar results were obtained by other researchers [5].

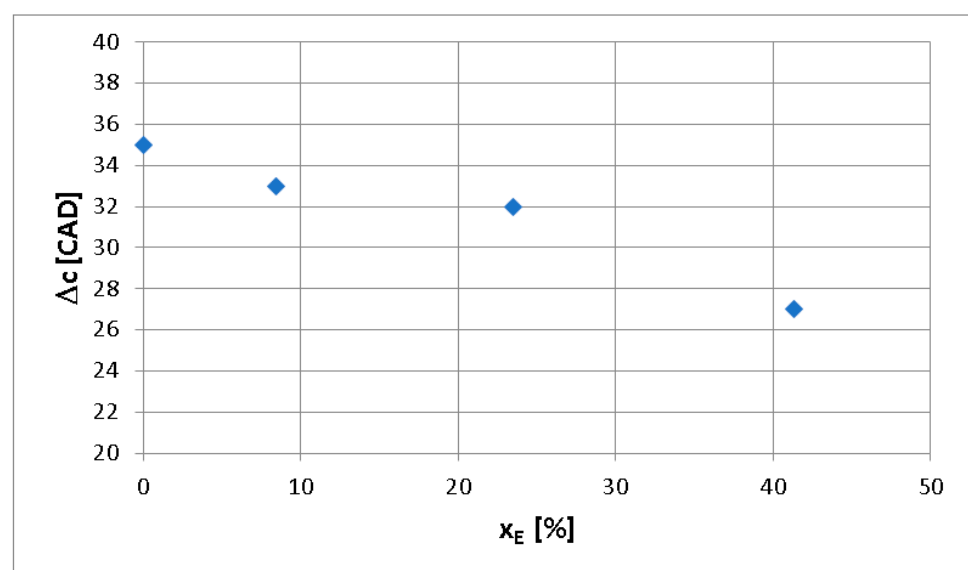


Figure 7. Total combustion duration for different hydrogen substitution ratios.

However, reducing combustion can positively influence engine efficiency, as shown in Figure 8. When using hydrogen, the reduction in the angle at which the maximum pressure appears per cycle, as shown in Figure 2, demonstrates the approach of the rapid combustion phase to the TDC, which leads to an increase in the share of isochoric combustion, which leads to an increase in thermal efficiency, as shown in Figure 8.

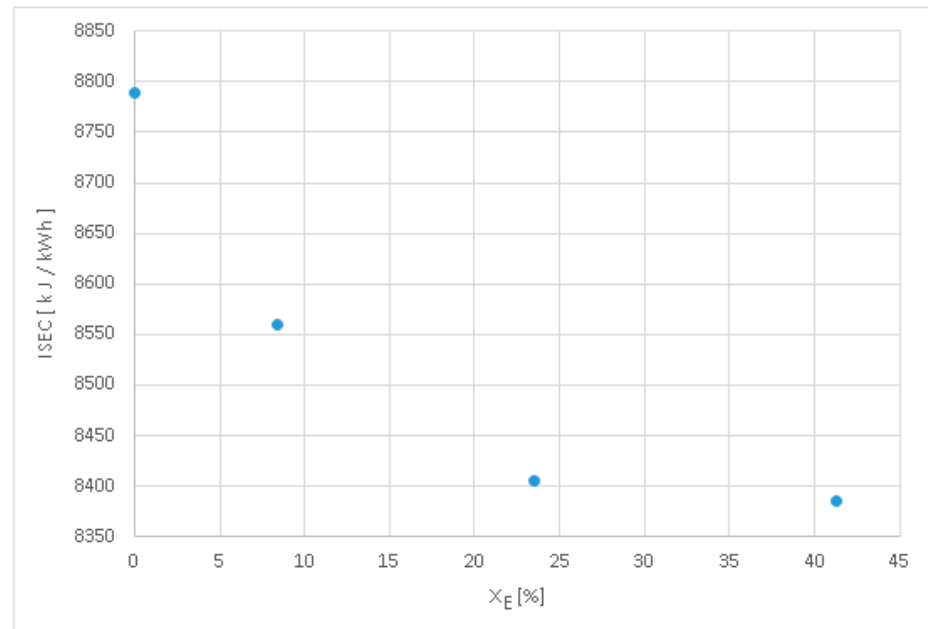


Figure 8. Indicated specific energy consumption for different energetic substitution ratios of the diesel fuel with hydrogen.

This increase correlates with a tendency to reduce combustion duration, as shown in Figure 7. Thus, when using hydrogen fuelling, the indicated specific energy consumption (ISEC) decreases by 2.58% to $x_E = 8.4\%$, by 4.35% at $x_E = 23.5\%$, and by 4.58% at $x_E = 41.3\%$ compared to classic fuelling, $x_E = 0\%$. Similar results were obtained by other researchers [7,12].

High-speed combustion of homogeneous air–hydrogen mixtures contributes to reducing HC emission levels.

The HC emission level decreased by 4% at $x_E = 8.4\%$, by 14% at $x_E = 23.5\%$, and by 18% at $x_E = 41.3\%$, as Figure 9 shows. Similar results were obtained by other researchers [5,12,18].

The smoke emission level, assessed by the opacity value of the exhaust gases, was reduced when diesel was replaced with hydrogen, as shown in Figure 10. When using hydrogen, the smoke opacity decreased by 52.02% for $x_E = 8.4\%$, by 70.27% for $x_E = 23.5\%$, and by 45.27% for $x_E = 41.3\%$, as shown in Figure 10. At maximum x_E , the reduction in smoke opacity was less pronounced, probably due to the reduction in the amount of air available in the cylinder. Limiting the tendency for smoke emission to increase can be achieved by increasing the boost pressure, which will ensure an increase in the amount of air per cycle. Similar results were obtained by other researchers [4,5,12].

The NO emission level's variation is presented in Figure 11. When using hydrogen fuelling, the level of NO emissions increased due to the increase in cylinder temperature during combustion. For $x_E = 8.4\%$, the NO emission level increased by 18.58%. For $x_E = 23.5\%$ and $x_E = 41.3\%$, the NO level increased by 62.82% and 123.71%, respectively. If solutions such as tuning the injection advance, EGR rate modification, or using Ad-Blue are not applied to reduce NO, then in order to limit the increase in the NO level, limiting the energetic substitution ratio x_E to 41.3% can be considered. When using hydrogen fuelling, the NO emission level starts to increase because the combustion temperature increases.

The engine is equipped with an exhaust gas recirculation system, and a modification of the recirculated gas flow rate could be considered in further research, especially to limit the increase in NO emission levels. Similar results regarding the NO_x emission level's variation were obtained by other researchers [1–3,5,12,18,19].

The reduction in the CO₂ emission level, as shown in Figure 12, is attributed to the reduction in carbon content, as shown in Figure 13, when replacing diesel fuel with hydrogen.

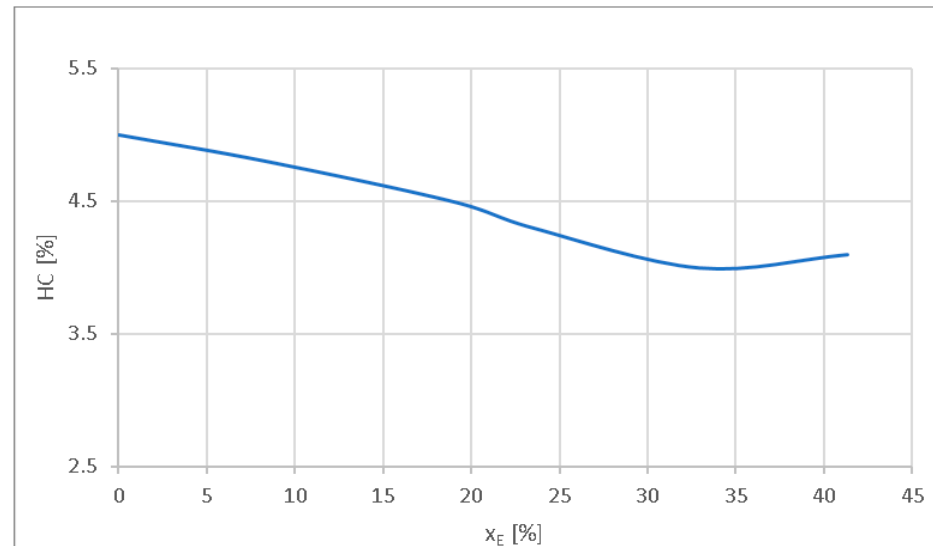


Figure 9. The HC for different energetic substitution ratios of diesel fuel with hydrogen.

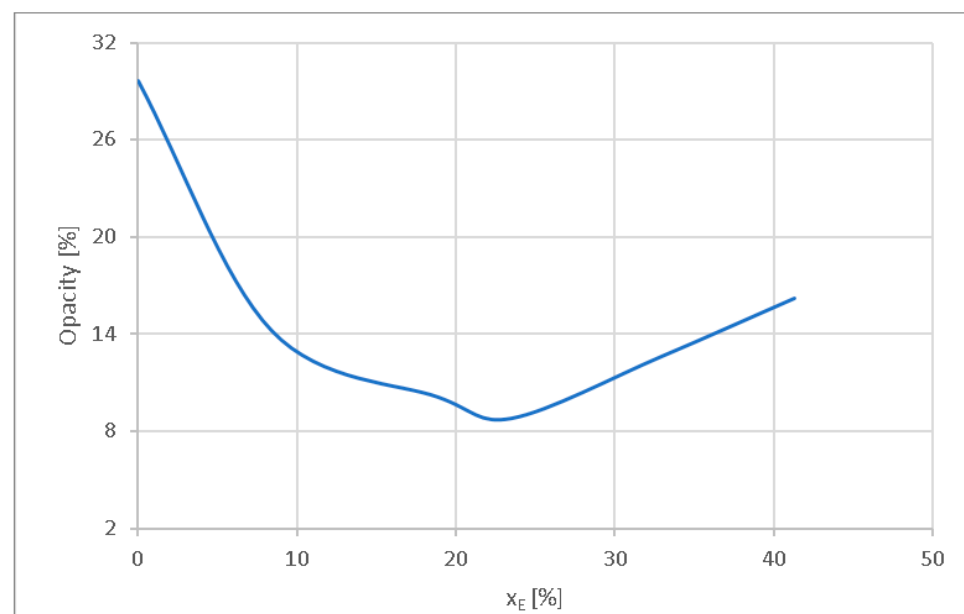


Figure 10. Smoke opacity for different energetic substitution ratios of the diesel fuel with hydrogen.

Compared to diesel fuelling, the CO₂ emission level was reduced by 11.13% at $x_E = 8.4\%$, by 26% at $x_E = 23.5\%$, and by 37.44% at $x_E = 41.3\%$. The reduction in the CO₂ emission level is related to the reduction in the carbon content when using hydrogen. The in-cylinder mixture contained 85.7% carbon when using diesel fuelling, but for hydrogen use, the carbon content decreased to 78.5% at $x_E = 8.4\%$, to 65.5% at $x_E = 23.5\%$ and to 50.3% at $x_E = 41.3\%$, as Figure 13 shows. When using hydrogen, the mixture's carbon content decreased by 8% for $x_E = 8.4\%$, by 23% for $x_E = 23.5\%$, and by 41% for $x_E = 41.3\%$. Similar results were obtained by other researchers [2,5,12,15].

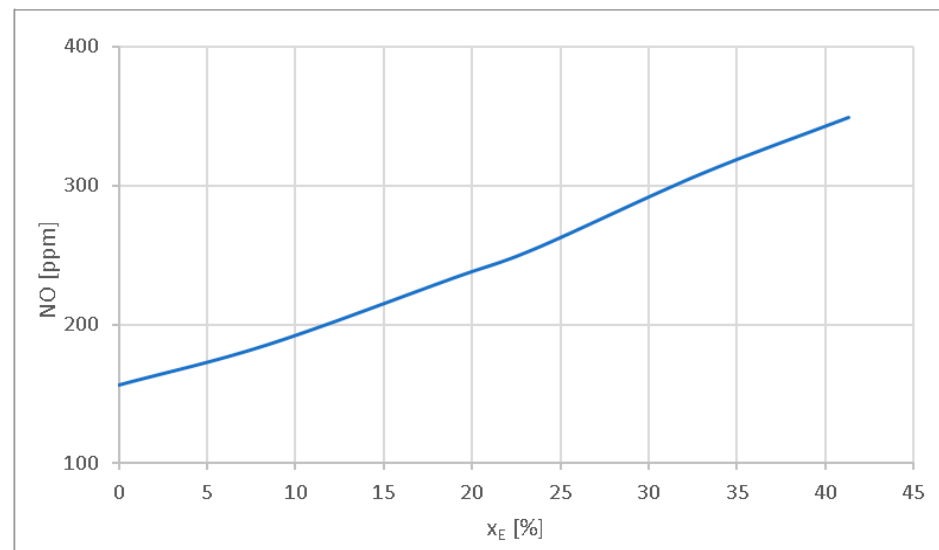


Figure 11. The NO for different energetic substitution ratios of diesel fuel with hydrogen.

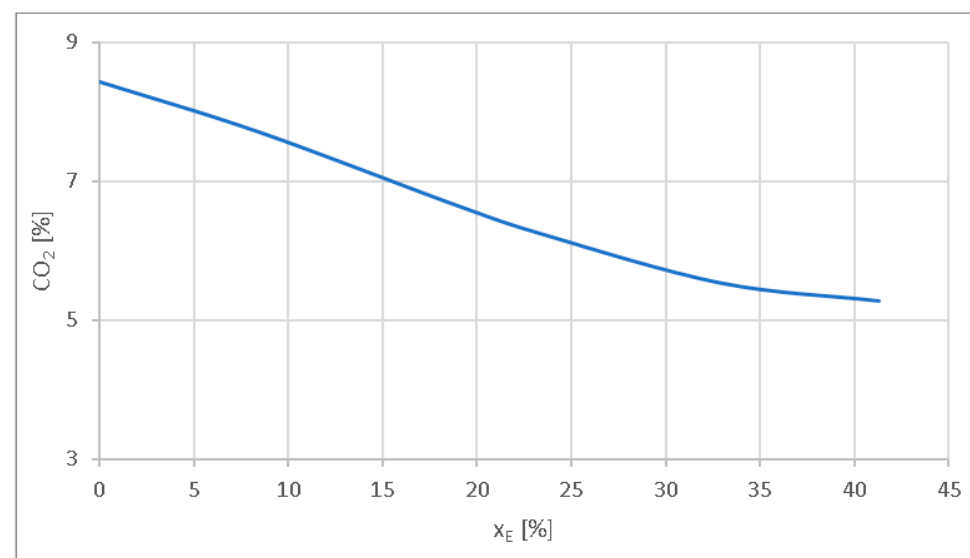


Figure 12. The CO₂ for different energetic substitution ratios of diesel fuel with hydrogen.

Regarding the variation in the level of carbon-based emissions with the variation of the quantity of cyclic H₂, the reduction in CO₂ emissions can be attributed to the reduction in the amount of carbon in the mass composition of the final mixture when replacing diesel fuel with hydrogen. The lower amount of carbon in the composition of the air–H₂–diesel fuel vapour mixture compared to diesel fuel actually implies a reduction in the initial amount of carbon available in the cylinder at the beginning of the combustion process, which subsequently leads to lower amounts found in the exhaust gases.

By partially substituting diesel with H₂, a 35% reduction in the amount of carbon available for combustion can thus be obtained.

The correspondence of the reduction in the initial carbon percentage with the increase in the H₂ dose can be observed in the variation in CO₂ emissions. Thus, at low substitution degrees up to 41.3%, the CO₂ emission was reduced by 37% compared to diesel fuel use.

The reduction in HC and smoke is attributed to the high-speed combustion of homogeneous air–H₂ mixtures, which ensure a reduction in HC emissions when their proportion in the cylinder increases. Reducing the diesel fuel dose and carbon content allows for a reduction in the amount of soot formed, and the presence of H₂ in the air–diesel fuel

vapour mixture leads to the acceleration of the oxidation of the soot formed. However, for $x_E > 20\%$, HC and smoke begin to increase slightly as a result of the reduction in the amount of oxygen, which itself is a result of a reduction in the amount of air available in the cylinder when the dose of H_2 in the intake air increases. In future studies, corrections can be made by increasing the amount of air by increasing the boost pressure when using higher cyclic doses of H_2 .

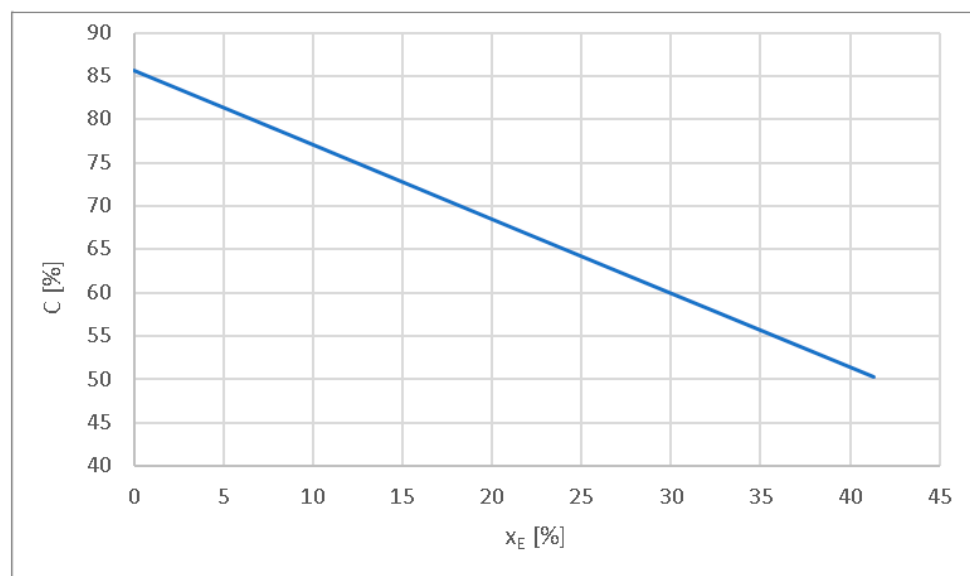


Figure 13. The carbon content in cylinder mixture for different energetic substitution ratios of diesel fuel with hydrogen.

The level of CO_2 emissions is influenced to a greater extent by the carbon content of the mixture in the cylinder, which is reduced when using H_2 , and the emissions of HC and smoke are more strongly influenced by the reduction in the O_2 content, as an oxidant, because they are more sensitive to the reduction in the amount of air necessary for combustion when the dose of H_2 is increased. The level of CO_2 emission is also influenced by the energetic performance of the engine cycle, which increases when using H_2 ; the level of CO_2 emission decreases, and the influence of the dose of H_2 on the energetic performance of the cycle is stronger than the influence of the reduction in available oxygen.

The results derived from this research are intended to inform the development of optimized dual-fuel strategies that effectively balance performance improvements with environmental considerations for the investigated operating regime. In order to maintain power performance at 37% load and 2500 rev/min and to maintain the advantages of improved engine efficiency and reduced exhaust emissions, the dual-fuel strategy—which effectively balances performance improvements with environmental considerations—involved limiting the energetic substitution ratio to 41.3%.

Optimal dual-fuel strategies that effectively balance performance improvements with environmental considerations require limiting the energetic substitution ratio to a value of 23.5%. At this setting, the increasing trend of smoke emissions is limited, and the significant reduction of 70.27% in smoke emissions is maintained by limiting the hydrogen flow rate to 0.396 slpm. In these settings, with dual fuelling, an energy consumption level of 8405 kJ/kW h can be obtained, which is 4.35% lower than for diesel fuelling. With this limitation, the HC emission level is reduced by 14%.

The maximum pressure can be limited to an increase of 14.7%, without exceeding values of 120 bar, which brings advantages in terms of reliability and combustion noise during operation. The combustion noise can be limited to a percentage increase of 50.93%;

thus, the value of the maximum pressure rise rate can be limited to 6.5 bar/CAD. With these settings, the heat release rate would not be significantly affected, and the combustion duration would be reduced by 8.5%, by 2 CAD.

For these settings, the tendency of the NO_x emission level to increase would be limited, but AdBlue technology can be used to decrease the NO_x emission level. With these settings, the emission of greenhouse gas, CO_2 , would be reduced by 26% compared to classic fuel use.

4. Conclusions

The use of hydrogen to substitute diesel fuel has the following effects.

- The carbon content of the in-cylinder mixture decreases to 78.5% at $x_E = 8.4\%$, to 65.5% at $x_E = 23.5\%$ and to 50.3% at $x_E = 41.3\%$, from 85.7% carbon with diesel fuelling. When using hydrogen, the mixture's carbon content decreases by 8% for $x_E = 8.4\%$, by 23% for $x_E = 23.5\%$, and by 41% for $x_E = 41.3\%$.
- The CO_2 emission level is reduced by 11.13% at $x_E = 8.4\%$, by 26% at $x_E = 23.5\%$, and by 37.44% at $x_E = 41.3\%$. The reduction in the CO_2 emission level is related to the reduction in the carbon content when using hydrogen.
- The HC emission level decreases by 4% at $x_E = 8.4\%$, by 14% at $x_E = 23.5\%$, and by 18% at $x_E = 41.3\%$, due to high-speed combustion of homogeneous air–hydrogen mixtures.
- Due to a reduction in the carbon content, the smoke emission level, which is assessed by the opacity value of the exhaust gases, decreases by 52.02% for $x_E = 8.4\%$, by 70.27% for $x_E = 23.5\%$, and by 45.27% for $x_E = 41.3\%$. At maximum x_E , the reduction in smoke opacity is less pronounced, probably due to the reduction in the amount of air available in the cylinder. Limiting the tendency for smoke emissions to increase can be achieved by increasing the boost pressure, which will ensure an increase in the amount of air per cycle.
- The NO emission level increases because of temperature rise during combustion. For $x_E = 8.4\%$, the NO emission level increases by 18.58%. For $x_E = 23.5\%$ and $x_E = 41.3\%$, the NO level increases by 62.82% and 123.71%, respectively. If solutions such as tuning the injection advance, EGR rate modification, or using Ad-Blue are not applied to reduce NO, then in order to limit the increase in the NO level, limiting the energetic substitution ratio x_E to 41.3% can be considered.

The reduction of the carbon content, CO_2 , HC, and smoke opacity can be assured in by the following factors.

- In-cylinder pressure increases to a maximum value which tends to approach the TDC, which may correspond to faster, more brutal combustion in the presence of air–hydrogen mixtures established in the cylinder at the moment of autoignition of the diesel dose. This trend is amplified by the increase in the cyclic hydrogen dose, when x_E increases. In some individual combustion cycles, it can be observed that at maximum $x_E = 41.3\%$, the pressure oscillations produced by the brutal combustion are amplified.
- The angle of maximum pressure appears sooner per cycle, with 1 CAD at $x_E = 8.4\%$, with 2 CAD at $x_E = 23.5\%$, and with 4 CAD at $x_E = 41.3\%$. The approach of the maximum pressure angle to the TDC can be correlated with the increase in the share of the premixed phase of combustion among the preformed mixtures involved in hydrogen fuelling.
- The value of maximum pressure increases by 5.98% at $x_E = 8.4\%$, by 14.77% at $x_E = 23.5\%$, and by 19.11% for $x_E = 41.3\%$ compared to $x_E = 0\%$. The higher combustion speed of air–hydrogen mixtures compared to that of air–diesel vapor mixtures

at the same excess air coefficient can lead to a reduction in the duration of combustion, meaning that the use of hydrogen has an increasing influence on efficiency.

- The maximum pressure rise rate increases by 6.51% at $x_E = 8.4\%$, by 50.93% at $x_E = 23.5\%$, and by 23.94% at $x_E = 41.3\%$, compared to $x_E = 0\%$. There is a correlation between the trends of decrease in the maximum pressure angle, increase in the maximum pressure, and increase in the maximum pressure rise rate. Although the recorded values do not exceed the range of usual values specific to diesel engines, the increasing trend recorded in the maximum pressure values and the pressure increase rate may be a criterion for limiting the cyclic hydrogen dose to $x_E = 41.3\%$ in order to maintain engine reliability.
- The peak of the increase in heat release rate was 45.77% at $x_E = 41.3\%$. For the energetic substitution ratios $x_E = 8.4\%$ and $x_E = 23.5\%$, similar values were registered, but a tendency to accelerate the combustion process in the presence of hydrogen was observed for all energetic substitution ratios, with the peak of HRR occurring earlier in the cycle due to the higher value of the LHV of hydrogen, large inflammability limits, and a higher flame speed comparative to diesel fuel.
- The angle at which the maximum value of the HRR occurs was reduced by 1 CAD at $x_E = 8.4\%$, by 5 CAD at $x_E = 23.5\%$, and by 9 CAD at $x_E = 41.3\%$. The reduction in the angle of maximum HRR correlates with the reduction in the angle at which the maximum pressure occurs.
- The heat release law becomes more aggressive when the hydrogen cyclic dose is at its maximum, at $x_E = 41.3\%$, and the rapid phase of the combustion process increases its share at this value of $x_E = 41.3\%$.
- The total duration of combustion is reduced when using hydrogen due to H_2 's higher flame speed and wider inflammability limits. The combustion duration decreases by 2 CAD at $x_E = 8.4\%$, by 3 CAD at $x_E = 23.5\%$, and by 8 CAD for $x_E = 41.3\%$ versus classic fuelling, $x_E = 0\%$. The combustion process reduces its duration by 5.7% at $x_E = 8.4\%$, by 8.5% at $x_E = 23.5\%$, and by 22.8% for $x_E = 41.3\%$ compared to diesel fuelling. Reducing the duration of combustion leads to an increase in maximum pressure and maximum pressure rise rate when using H_2 ; when using hydrogen, the reduction in the angle at which the maximum pressure appears per cycle indicates the approach of the rapid combustion phase to the TDC, which leads to an increase in the share of isochoric combustion, which itself leads to an increase in thermal efficiency. This increase correlates with the tendency to reduce the duration of combustion.
- The indicated specific energy consumption decreases by 2.58% to $x_E = 8.4\%$, by 4.35% at $x_E = 23.5\%$, and by 4.58% at $x_E = 41.3\%$ compared to classic fuelling, $x_E = 0\%$.

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Abbreviations

AVL	Anstalt für Verbrennungskraftmaschinen List automotive research institute Graz Austria
BTE	Brake Thermal efficiencyEfficiency (or η_e)
BTDC	Before Top Dead Centre
BSFC	Brake-Specific Fuel Consumption
BSEC	Brake-Specific Energetic Consumption
CFR	Cooperative Fuel Research
CAD	Crank Angle Degree (or $^{\circ}\text{CA}$)
C_h	Hourly Fuel Consumption
CO	Carbon Monoxide
CO ₂	Carbon Dioxide
DI	Direct Injection
DME	Dimethyl Ether
EU	European Union
ECU	Engine Control Unit
EGR	Exhaust Gas Recirculation
G	Gasoline
H ₂	Hydrogen
HC	Unburned Hydrocarbon
H _i	Lower Heating Value
HRR	Heat Release Rate
ICE	Internal Combustion Engine
IMEP	Indicate Mean Effective Pressure
J	Joule
K	Kelvin
L	Liter
LHV	Lower Heating Value
LPG	Liquid Petroleum Gas
lpm	Liter Per Minute
MTB	Maximum Torque Brake
m	Kinetic Parameter
m/s	Meter/Second
NO _x	Nitrogen Oxides (Or NO)
PFI	Port Fuel Injection
$p_{\max}$	Maximum Pressure
p	In-Cylinder Pressure
rpm	Revolutions Per Minute (Or Rev/Min)
SIE	Spark Ignition Engine (Or SI Engine)
SCR	Selective Catalytic Reduction
slpm	Standard Litre Per Minute

TDC	Top Dead Centre
T	In-Cylinder Temperature
λ	Coefficient Of Excess Air
x_E	Energetic Substitution Ratio

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