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Hydrogen as a Carbon Neutral Fuel for Automotives in Sustainable Transportation

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Abstract

The use of sustainable carbon-free energy sources is becoming a priority in the field of transport so that it becomes sustainable. Sustainable transport can also be achieved with vehicles equipped with diesel engines fuelled by alternative fuels that do not contain carbon, like hydrogen. The paper presents an analysis of the experimental results obtained at the fuelling with diesel fuel and hydrogen of a modern diesel engine, operating at 50% partial load and 2500 rev/min speed. For H_2 energy substitution degrees of up to 43%, the combustion process is improved: the specific energy consumption is reduced, the combustion duration is reduced, the heat release rate is increased, the maximum pressure is increased, the carbon-based pollutant emissions are decreased and the cyclic dispersion is reduced. For 33% H_2 energy substitution degree, the maximum pressure increases by 16.4%, the indicated mean effective pressure increases by 7.5%, the specific energy consumption is reduced by 5.36% and the level of greenhouse gases emission is reduced by 34.5% for carbon dioxide. In case of pollutant emissions, the smoke level is reduced by 58.6% and the unburned hydrocarbons level is reduced with 18%. For higher percentages of H_2, emissions reductions can be accentuated. At H_2 use, the combustion cyclic variability is reduced, the values of the COV variability coefficients determined for the parameters of interest and the combustion duration being reduced. As a novelty aspect, the optimal adjustment between engine load-speed-diesel fuel flow-hydrogen flow-maximum combustion pressure-smoke emission level-exhaust temperature level is presented. The use of hydrogen at the diesel engines can provide the beginning of sustainable transportation solutions in the future.

Keywords: sustainable transport; combustion; hydrogen; energetic fuel consumption; pollutant emissions; carbon-free fuel; cyclic variability; coefficient of variability; heat release rate; indicated mean effective pressure; carbon dioxide; autoignition delay; hydrogen

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1. Introduction

1.1. State of Art

Today, more than ever, the use of sustainable, carbon-free energy sources is becoming a priority in the field of transport so that it becomes sustainable [1–4]. The current

context of the use of alternative fuels, such as hydrogen, to improve the performance of the thermal engine, is one of a high interest [1,2,4]. Moreover, the study is interesting because there is not yet sufficient data's on the hydrogen use in diesel engine, most of the studies being for the spark ignition engine [1–3]. The spark ignition engine fuelled by H_2 already exists, being built by BMW or Toyota, but for the diesel engine, research is still in development [2–4]. Moreover, the current context of limiting the use of the diesel engine, may become favourable for the study of the hydrogen use at diesel engine, in order to reduce the level of polluting emissions and keep it in operation. Research on the use of hydrogen in diesel engines is not so numerous, being still in development, this represents a gap in research [1,2]. Therefore, any study on the functioning of the diesel engine fuelled with hydrogen is intended to increase the level of knowledge in the field. In diesel engine, the ignition of the air-hydrogen mixture must be done with the help of a diesel pilot, manifold or port hydrogen injection fuelling systems being under development [2,5]. The fuelling method, the adjustments and the hydrogen flow rates can have a beneficial influence for carbon emissions decrease, but the level of nitrogen oxides may increase [2–5]. The NO emission level can be controlled by variation of the excess air coefficient or exhaust gases after-treatment [2]. The ongoing research's developed mainly in two directions, manifold hydrogen injection or port fuel injection of hydrogen, present the influences of state, functional and sometimes constructive factors on the energy and pollution performance of the hydrogen-fuelled diesel engine [1–5]. The use of large quantities of hydrogen to restore engine power, especially for hydrogen direct injection, is difficult given the need to increase hydrogen flow rates and/or injector opening durations, which complicates the design of hydrogen injectors [4,5]. Therefore, a special attention is paid to establishing and adjusting the quantities of diesel fuel and hydrogen to ensure normal engine operation and an acceptable level of pollutant emissions and greenhouse gases [1–4]. The future prospects for hydrogen use in diesel engines can be good if problems related to H_2 abnormal combustion phenomena, hydrogen leaks in the engine crankcase, high cost price of hydrogen and issues of H_2 distribution infrastructure are solved [1–5]. From this point of view, an encouraging aspect is that the European Union requires member states to implement new hydrogen refueling stations for vehicles in the next 10–15 years [1–4]. More, the European Union reports define the automotive equipped with internal combustion engine fuelled by hydrogen as a "zero emission vehicle" [1,2]. Even in Asia, the use of buses equipped with hydrogen-internal combustion engines for urban transportation looks like an attractive solution due to possibility of emission reduction [2].

The use of alternative fuels can be a viable solution, but many countries hope that the electric vehicle will assure a sustainable transport of the future, especially in the main cities and in the urban areas [2,4,6]. But, for heavy vehicles, like trucks, buses, tractors, commercial vans and pick-up's, equipped with diesel engines, an electric power train substitute may be difficult to be find and to assure successful replacement [2,4–6]. These types of vehicles need space for people, materials, goods and merchandise not for large and heavy batteries [2,4–6]. Xu et al. [7] develop a study trying to find the answer to the question "An electric truck can be a viable alternative to diesel truck?," for roads in USA. In the journey on the highway, the traveling time increases with 32% because of the necessity of recharging stops, comparative to trucks equipped with diesel engines [7]. In terms of total costs, including batteries, in the case of an electric truck, the sums exceed \$5800, for a battery cost of over \$250/kWh, but for a truck equipped with a diesel engine the total cost remains below \$5000. Even in calculations with negligible battery prices, the operating price of an electric truck remains higher than that of a truck equipped with a diesel engine [7]. The activities allocated to the extraction of materials for the execution of batteries and the construction of batteries are large generators of CO_2 [5–8]. The use

of hydrogen for vehicles used for transportation may leads to the reduction of the carbon emissions [4,6]. If the transport sector with electric truck fleets becomes a large consumer of electricity in the future, then new problems arise, such as momentary collapse, which must be solved quickly, considering that the industry is already a large consumer of electricity [7]. Diesel truck cost per life span is $0.213/mile traveled, compared to battery cost/battery capacity per lifespan is $0.24/mile traveled [7,8]. In countries where electricity is obtained by burning coal, the advantages offered by electric vehicles are reduced and the lifetime emissions cycle is similar to other vehicles [3,5,7,8]. For example, according to the N.T. newspaper, the USA state of Nevada used 52% coal, 34% natural gas, and 7% hydropower to produce electricity in 2001. In 2017, the share of natural gas increased to 69%, and a new source of energy, solar, joined the mix, with a share of 12% [8]. Electric vehicles are not currently a panacea for climate change, the lifetime greenhouse gas emissions of electric vehicles can be similar to those of the most efficient gasoline or diesel vehicles, or even higher [4,8]. For electric vehicles, the limited range of action remain the main issues, according to studies by Albatayneh et al., Carbon Brief and others [4,8]. In Asia, the sustainable transport can be assured by accelerating the swich to the use of hydrogen and nocarbon fuel as renewable energy, according to Lee et al. [9]. The H_2 systems emit lower GHG emissions comparative to fossil fuel based systems [5,9]. The use of green H_2 can be a solution to the reduction with more than 50% of the GHG emissions and of the CO_2 emission, in particulary [3,5,9]. But more emissions are emited at H_2 production comparative to fossil fuel production and that why the production of the green hydrogen and its use must be promoted [3,4,9–11]. More CO_2 emissions are emitted during the battery fabrication for electric vehicles, according to Sipos et al. [5]. The level of CO_2 emissions related to the fabrication of the ICE fuelled with H_2 is similar with the level registered for fabrication of ICE fuelled by classic fuels [5]. The use of hydrogen fuelled automotives to transport materials or goods in the urban area can be a good solution to solve the pollution issues [4,6]. Is difficult to travel large distances for electric trucks [4], but hydrogen use may be a good alternative solution from this point of view. But H_2 cost and availability remains the main issue that must be solve in the future [4]. Besides electric vehicles, the hydrogen fuelled vehicles can contribute to the reduction of the GHG emission [6]. As a prediction, in 2050, the electrical vehicles may produce more GHG, around 200% per kilometre travelled, according to Hassouna et al. [6].

1.2. Literature Review

In small steps, the future of the sustainable transport can also be achieved with vehicles equipped with diesel engines fuelled by alternative fuels that do not contain carbon [2–4]. A privileged alternative fuel that does not contain carbon and can be used to fuel the diesel engine is hydrogen [1,3–5]. Hydrogen can be used in many areas of transportation like naval and roads [1–5]. Bayer et al. [10] develop a study for hydrogen, ammonia and diesel fuel use perspective, in terms of emissions during production and combustion. Bayer et al. [10] show that hydrogen has the shorten GHG emission life cycle and may produce only NO_x emission comparative to diesel fuel which produce CO_2, CO, NO_x, SO_x, CH_4 and N_2O or ammonia which produce CO, NO_x, CH_4 and N_2O, but in lower amounts. Bayer et al. [10] use hydrogen and ammonia to fuel a boat engine and obtain lower CO_2 emission level. Best performance results are obtained for 26% H_2 and 76% diesel fuel, comparative to other fuel combination use, like ammonia [10]. Future experiments will measure the combustion pressure, the NO_x and N_2O emissions level. Bayer et al. [10] assumes that ammonia may increase N_2O emission level [10]. Bayer et al. [10] also present the influence of H_2 type production, grey H_2, blue H_2 or green H_2, on GHG and CO_2 emissions. The

green H_2 is the most clean, with almost zero emissions, followed by blue H_2, with a 70% reduction in the emission level [10].

Abdelwahed et al. [11] use 0.5 L/min hydrogen flow to fuel an IC engine at the regime of 1000–3000 rev/min. For H_2 use, the engine power was increased with 3% [11]. At full load, the specific fuel consumption decreases with 10% and the thermal efficiency increases with 10%, at H_2 use. At low loads, the specific fuel consumption decreases with 9% [11]. The most important decrease in fuel consumption, of 19%, appears at 1500 rev/min, at full load [11]. At this regime, 1500 rev/min and full load, the brake thermal efficiency increases with 22% [11]. At H_2 use, comparative to diesel fuel use, the CO emission level decreases with 13% at medium loads and with 7% at full load [11]. The CO_2 emission level decreases with 17% at medium loads and with 7% at full load regime versus diesel fuel use [11]. The NO_x emission level decreases with 13–17% depending on the load regime [11]. Barreiros et al. [12] use diesel fuel-H_2-O_2 mixture to fuel a diesel engine generator, the H_2 and O_2 being injected into the inlet air [12]. At hydrogen use, a 7.6% reduction in fuel consumption was obtained, but the mechanical work decreases with 1% [12]. Due to higher H_2 combustion flame speed (290 cm/s for H_2 versus 30 cm/s for diesel fuel, at $\lambda = 1$) at hydrogen use the combustion duration decreases with 1.4 CAD, from 26.2 CAD to 24.7 CAD, for the 5–90% interval [12]. The IMEP decreases with 1% at H_2 and O_2 use, but the IMEP cyclic variability is reduced comparative to diesel fuel use [12]. The CO emission level decreases with 1.9%, hydrogen and oxygen use leading to the acceleration of the CO oxidation, but the CO_2 emission level is maintained at the same level [12]. The NO increases with 3.9% versus diesel fuel use [12]. Scrignoli et al. [13] develop a CFD simulation for the study of the diesel engine fuelled by hydrogen and diesel fuel. If small doses of H_2 are used, like 20% H_2, the engine brake thermal efficiency is improved. But it becomes difficult to use large H_2 doses, like 40–80% energy substitution [13]. At 2400 rev/min and 10 BMEP load, in-cylinder pressure, pressure rise rate, IMEP, 5%, 10%, 50% and 90% MFB are influenced by the increase of H_2 quantity [13]. Maximum pressure increases from 94 to 98 bar and the pressure rise rate increases with 0.2 bar per CAD at 80% H_2 and 20% diesel fuel use, from 2.9 bar/CAD to 3.1 bar/CAD [13]. The combustion rate is accelerated for 60% H_2, but for larger H_2 quantities the combustion rate is reduced [13]. For 40% H_2, the IMEP increases with more than 2,5%, from 10.6 bar to 10.9 bar, but for 80%H_2 the IMEP decreases from 10.6 bar to 8.5 bar [13]. Also, at this H_2 percent, the combustion duration was increased. In order to compensate these issues, the diesel fuel injection timing was modified in the domain 5–30 CAD [13]. Lower values of IMEP are registered for latter injection timings, at the increase of the H_2 dose [13]. The IMEP increases only at 20%H_2 use, at sooner injection timing [13]. The variation of the diesel fuel injection timing was limited in order to not exceed 150 bar in maximum pressure and 5 bar/CAD in pressure rise rate [13]. At full load, 4000 rev/min and 13 bar BMEP, the influence of H_2 percent on engine operation is similar and the maximum pressure increases from 139 bar to 182 bar, at 80% H_2. For hydrogen percent's until 40%, the combustion duration was decreased, but after this value, the combustion duration starts to increase [13]. Xu et al. [14] show the effect of the pilot quantity on combustion mechanism for a diesel engine fuelled with diesel fuel and hydrogen, at 1800 rev/min. For 30%H_2, the increase of the pilot mass, in domain 5–20%, leads to the increase of the heat release rate (HRR), maximum pressure and maximum pressure rise rate, a sooner achievement per cycle of the 10% MFB and 50% MFB and the decrease of the CO emission level [14]. The NO emission level is increased by the rise of the pilot quantity [14]. If the pilot timing is increased by 20 CAD, the CO emission level decreases and the NO_x emission level increases [14]. At this tune, the maximum pressure increases, the 10% MFB and 50% MFB are accelerated and the maximum of the HRR decreases [14]. If the pilot timing is increased by 40 CAD, then the maximum pressure

decreases, the peak of the HRR increases and fractions 10% MFB and 50% MFB appears later per cycle [14]. At this tune, the CO and the NO emission level are reduced versus the initial tune [14]. For lower injection timing, like 8 CAD, the NO emission level and the maximum pressure are increased and only CO and HRR peak are decreased [14]. Other pilot injection timing, like 6 CAD, tuned in order to decrease the NO level, are difficult to setup because the maximum pressure rise rate of 12 bar/CAD is exceed [14]. The HC and CO emissions levels have similar variations [14]. Bajerlein et al. [15] use diesel fuel-hydrogen blends and show that NO emission level decreases with 10–20% at hydrogen fuelling versus diesel fuel use, depending on engine load, 0–40 Nm [15]. Hydrogen is injected at high pressure to avoid the formation of microbubbles [15]. For the 20% decrease in NO level, the diesel fuel contains 0.3% hydrogen, which modify fuel dynamic and combustion [15]. The NO level decreases due to acceleration of the combustion, since H_2 is dissolved in the diesel fuel [15]. A shorten combustion duration leads to the reduction of the NO emission level [15]. The CO emission level increases with 50–150% comparative diesel fuel use, depending on engine load [15]. Leak of oxygen during injection of the fuel, influenced by the specific movement of air into combustion chamber, may affect the fuel oxidation and leads to the increases of the CO emission level, according to Bajerlein et al. [15]. At H_2 use, the HC emission decreases with 22% at 0% load, with 5% at 10% load, with 18% at 20% load, with 10% at 30% load and with 38% at 40% load comparative to classic fuel use due to improvement of the fuel atomization [15]. At 20% load, the PM emission level increases at H_2 use. For other load regimes, the PM emission level decreases with 2–20% [15]. Jamrozik [16] shows that hydrogen wider flammability limits (4–75%) and higher combustion speed (60 cm/s) leads to the improvement of the combustion efficiency. The use of 30% H_2 ensures an increase in maximum pressure, heat release rate and pressure rise rate during combustion [16]. The thermal efficiency is increased, but the combustion stability is affected. The COV of IMEP slightly increases, but the engine can operate stably [16]. For 30% H_2, the level of the CO, soot and CO_2 emissions is significantly reduced [16]. Only HC and NO_x emissions level is increased [16]. For 55% H_2 percent, the HC emission level starts to increase. Bellow 55%H_2, the HC level may be reduced with 40%, but there is no clear tendency for HC variation [16]. The HC variation depends on engine type, fuelling system design, engine tune or fuels quantities [16]. The specific energy consumption decreases at the increase of the hydrogen cyclic quantity [16]. Muhssen et al. [17] show that hydrogen use may leads to the reduction of the CO, HC and CO_2 emissions level, but may increase the NO_x emission level. Hydrogen use reduces the autoignition delay, increases the thermal efficiency, maximum pressure and HRR [17]. Rueda-Vazquez et al. [18] present a bibliographic study about commercial vehicles engines fueled by H_2, which shows reduction of the CO_2 emission level with 45%, if the diesel fuel is replaced by hydrogen. In addition, the HC and soot emissions levels are reduced at hydrogen use [18]. The increase of the hydrogen quantity leads to the increases of the in-cylinder maximum pressure and of the pressure rise rate [18]. The combustion duration is reduced at hydrogen use, with benefits on lower HC emission level, in many studied cases [18]. In conclusion, Rueda-Vazquez et al. study affirms that the decarbonisation of internal combustion engines remains an essential step in achieving global climate targets, especially in transport sectors where electrification faces economic or technical barriers [18]. In this context, hydrogen–diesel fuel-dual engines have gained increasing attention as an intermediate solution capable of reducing greenhouse gas (GHG) and carbon based emissions [18]. Wang et al. [19] present different types of internal combustion engines fuelled by hydrogen (HICE) designed by few companies in China for different application: automotives, car, planes, trucks, stationary engines. Hydrogen use leads to the increase of the maximum pressure and pressure rise rate at the beginning of the combustion [19]. This

increase can be reduced by EGR use [19]. Further use of EGR will affect the autoignition process [19]. The COV_{IMEP} is increased at lean mixture use and for 4% higher ignition timings [19]. However, for higher H_2 concentration, at 0–20% engine load, the COV_{IMEP} decreases [19]. Mancaruso et al. [20] use hydrogen port-injection method to fuel an optical access diesel engine at 1500 rev/min speed, with no EGR. The levels of the CO_2, HC and PM emissions are reduced at hydrogen use [20]. The highest performance, in terms of brake thermal efficiency, is recorded for 10 CAD BTDC as injection timing of diesel fuel, H_2 being admitted into the cylinder once with the inlet air [20]. Mancaruso et al. [20] try to maintain constant the IMEP without manifold backfiring, at lean mixture operation, by using H_2 injection after valve overlap and before compression stroke starts [20]. In their experiment, Wright et al. [21] try to use fuel blends with 10%e–40%e hydrogen. The CO_2 emission level was decreased by H_2 use, but the soot level may be affected by large doses of hydrogen [21]. Hydra trucks fuelled with 40%e hydrogen can provide a 40% reduction of the CO_2 emission level [21]. Incomplete combustion leads to the increase of the HC emission level, but if the hydrogen fuelling is achieved with the improvement of the combustion efficiency, then the HC emission level can be reduced [21]. Panthi et al. [22] show a correlation between reduction of the combustion duration and the decrease of the COV_{IMEP}. Hydrogen-air mixture combustion leads to an increase in maximum pressure and flame speed, around 19 m/s [22]. At lean mixture use, $\lambda = 2.5$, the combustion duration is reduced [22]. Hosseini et al. [23] show that hydrogen fuelling can assure the reduction of the carbon based emissions at a diesel engine, but knocking and NO increased level may appears. Sharkey et al. [24] use hydrogen to fuel a 6 cylinder diesel engine and obtain: a 90% reduction of CO and CO_2, a 37% decrease of brake specific fuel consumption (BSFC), at 40% H_2 use. The NO level increases with 70%, but the use of water injection can decrease the nitrogen oxides level [24]. At the increase of the hydrogen share, the engine power, IMEP, mechanical efficiency and thermal efficiency are also increased [24]. Kiss et al. [25] observe the reduction of the CO_2 emission level with 30% and important reduction of the CO level are also registered. As prediction, at hydrogen use, the effective efficiency may increase and the combustion duration may decrease [25]. Akhtar et al. [26] develop a numerical simulation for hydrogen fuelling at different cylinder dimensions. The increase of the stroke leads to the increase of the combustion pressure, heat release rate and brake thermal efficiency, at high loads [26]. At low loads and bore reduction, the brake thermal efficiency is decreased because the combustion velocity is reduced [26]. The reduction of the cylinder bore leads to the reduction of the NO level [26]. The level of the CO, HC and soot emissions increases with 100% [26]. Karvounis [27] uses a 20% H_2 energy fraction share to fuel a marine diesel engine by port injection method. Karvounis [27] shows the increase of the in-cylinder pressure and temperature during air-hydrogen mixture combustion, due to the increase of the heat release rate at hydrogen use [27]. The maximum pressure increases with 8% and the angle of maximum pressure appears with almost 10 CAD sooner per cycle, for lower initial temperatures, around 300 K [27]. For higher initial temperatures, 370 K, the increase in maximum pressure is reduced, the peak pressure appears later on cycle, but the second stage of combustion seems to be increased at the hydrogen use [27]. According to Karvounis [27] the use of hydrogen leads to the reduction of the autoignition duration, but the combustion temperature and the NO level are increased. Large H_2 doses may assure a later pressure peak during combustion [27]. Menaa et al. [28] use H_2 to fuel a single cylinder engine at 1500 rev/min. Menaa et al. [28] affirm that is more convenient to use hydrogen at low and medium engine loads (20–60%) comparative to high loads, because the level of the emissions is lower. For 20% H_2 use the CO_2 and PM emissions level are reduced [28]. Menna et al. [28] also use natural gas for dual fuelling and observe that only the HC and NO emissions level was increased at high loads, hydrogen use being

more advantageous [28]. The hydrogen amount can be restricted in order to limit the increase of maximum pressure and HRR [28]. According to Candelaresi et al. study, the vehicles equipped with hydrogen engines are excellent solution for the decarbonisation of the transport sector [29]. In terms of life cycle profile, internal combustion engines fuelled by H_2, HICE, produce less CO_2/km (GWp = 4.343 × 10^2 [kg CO_2 eq·km^{-1}]) comparative to fuel cell electric vehicles (GWp = 5.601 × 10^2 [CO_2 eq·km^{-1}]) [29]. The automotives equipped with ICE fuelled by hydrogen, even they required large hydrogen quantities, are better involved in life-cycle environmental performance from the point of view of carbon, energy and acidification footprints [29]. The most adequate for environmental is the internal combustion engine fuelled with H_2 [29]. A disadvantage is represented by high consumption of H_2 [29]. In that case, the use of dual-fuelling, with H_2 and a fossil fuel, can be a solution that can offer a H_2 economy and to assure 1000 km rage of driving in automotive operation [29]. Candelaresi et al. [29] study, that compares fuel cell electric vehicles, hybrid electric vehicles, H_2 fuelled engines, CNG and hythane fuelled engines, conclude that the automotive fuelled by H_2 remains a suitable decarbonisation solution for medium term.

1.3. Novelty Aspects of the Current Research

The paper presents an analysis of the experimental results obtained for a modern diesel engine fuelled with diesel fuel and hydrogen. A difficult aspect of using H_2 at diesel engine is the construction of the experimental engine, which represents the basis of the experiment. Compared to other studies, a new aspect is related to the engine construction, which does not have the simultaneous opening period of the intake and exhaust valves. The intake valve opens at 9 CAD after TDC and the exhaust valve closes at 27 CAD before TDC, as can be seen in Table 1. Thus, there is no hydrogen leakage from the intake to the exhaust and there is no possibility of hot exhaust gas streams flowing back to the intake and igniting hydrogen in the engine intake system. Another new design aspect is represented by a crankcase gas recirculation system that limits hydrogen leakage to the engine crankcase and eliminates the possibility of H_2 ignition in the lower crankcase and reducing oil quality. The ventilation cylinder that extracts the crankcase gases allows them to be returned to the cylinder so that any H_2 fractions that escape into the crankcase can be returned to the cylinder for combustion. The solutions presented ensure proper engine operation at H_2 fuelling, avoiding uncontrolled crankcase ignition phenomena, intake and exhaust misfires. The hydrogen fuelling system is electronically controlled by an open-type computer, interconnected with the engine's electronic control unit to establish a correlation between the opening duration of the diesel and hydrogen injectors. For a more detailed analysis of the influence of the amount of hydrogen on the combustion process, the duration of the autoignition delay and the speeds of soot formation and oxidation are determined by theoretical calculation. The duration of the autoignition delay, the component phase of the combustion process in the diesel engine, is influenced by the processes of cold flames and blue flames [30–32], processes that are influenced by the presence of hydrogen. In order to explain the trend of variation of the smoke emission level from the exhaust gases, the influence of the H_2 cyclic dose on the soot formation speed and on the soot oxidation speed is analyzed. The novelty is defined by the analysis of the influence of the hydrogen dose on the nucleation process, which produces soot. The laws of soot formation and oxidation, determined by integrating the associated formation and oxidation speeds, allow the estimation by difference of the final smoke emission level from the exhaust gases [32]. Finally, these trends correlated with the values of the engine functional parameters can establish the scope of use of hydrogen doses that substitute diesel fuel. Thus, the novelty aspects of the paper are: (I) the experimental engine does not have overlapping opening

duration of the intake and exhaust valves, and thus there are no reverse hydrogen flows; (II) hydrogen leaks to the engine crankcase are taken over by a crankcase ventilation system and returned to the cylinder; (III) hydrogen fuelling system is electronically actuated and connected back to back with the diesel fuel system thru an open ECU, (IV) for the operating regime of 2500 rev/min and 50% load, the optimal adjustment between engine load-engine speed-diesel fuel flow-hydrogen flow-maximum combustion pressure-smoke emission level-exhaust temperature level has been achieved, the ESD maximum value being limited at 43%; (V) the connection between the specific parameters to the combustion study (maximum pressure, maximum pressure angle, reaction heat release rate, indicated average pressure) with the cyclical variability of combustion, the duration of combustion and with the positioning of combustion in the area of maximum turbulence, around TDC; (VI) the connection between the specific parameters to the combustion study (maximum pressure, maximum pressure angle, reaction heat release rate, indicated average pressure) with the duration of multi-stage processes at low temperatures specific to cold flame—blue flame reactions, which define the duration of the autoignition delay (AID); (VII) the correlation between the speeds of soot formation and oxidation, the hydrogen quantity and the trend of smoke emission variation.

Table 1. Technical parameters of the H_2 experimental diesel engine [33].

Parameter	Value
Number of cylinders	4
Power/speed [kW]/[rev/min]	50/4000
Torque/speed [Nm]/[rev/min]	210/2100
Supercharging pressure [bar]	1.8
Bore [mm]	76
Stroke [mm]	80.5
Length of the connecting rod	133.75
Compression ratio	18.5
Number of valve per cylinder	2
Intake Valve Opening [CAD before TDC]	9
Exhaust Valve Closing [CAD before TDC]	27
Valves overlap [CAD]	0
Engine cooling system	liquid
Diesel fuel fuelling system	Common rail 1800 bar
Hydrogen fuelling system	Manifold injection 1.5 bar

2. Experimental Investigations

2.1. Experimental Investigations Setup

The Figure 1 presents the schema of the experimental test bench, especially designed for the use of hydrogen at diesel engine. In order to achieve safe medium conditions during the experimental investigations, the test bed is equipped with an emergency sensor designed by Drager (Lubeck, Germany) which detects hydrogen leaks and, if necessary, sends acoustic and light signals to the test cell and shuts down the hydrogen fuelling system.

On the test bench, the diesel engine is mounted and equipped with all the equipment necessary for the experimental investigation. In the Table 1, the main technical characteristics of the engine are presented.

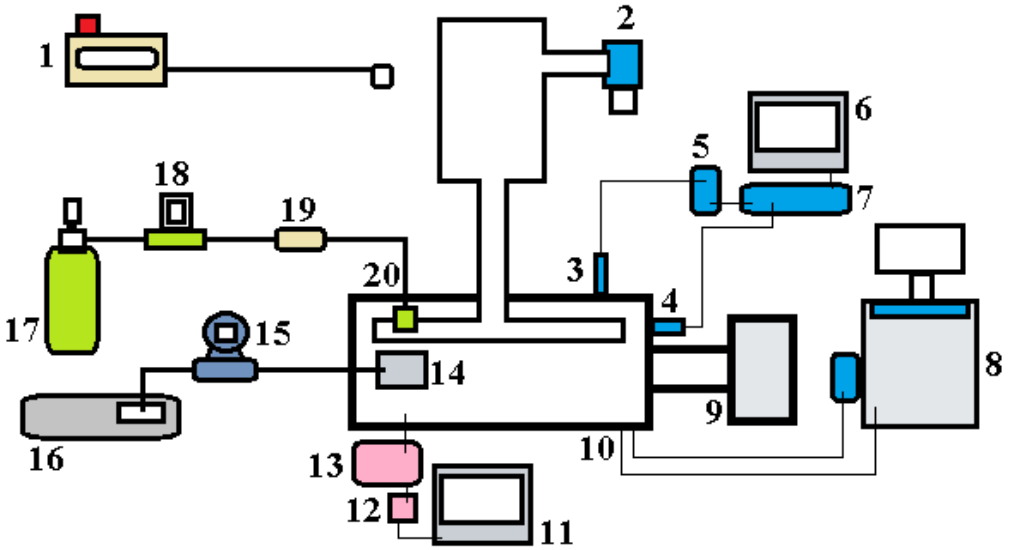


Figure 1. The experimental stand organization diagram. 1—Drager UniGard LT emergency sensor for hydrogen leaks (Lubeck, Germany), 2—Krohne H250 intake air flowmeter (Duisburg, Germany), 3—AVL GU12P pressure transducer (Graz, Austria), 4—AVL 365CC angle encoder (Graz, Austria), 5—AVL MicroIFEM piezo indicating amplifier (Graz, Austria), 6—data acquisition system PC, 7—AVL Indimodul 621 (Graz, Austria), 8—AVL 450 MDS gas analyzer (Graz, Austria), 9—Horiba Schenck E90 electric dynamometer (Oberursel, Germany), 10—diesel engine K9K, 11—open-ECU PC, 12—electric driver of open-ECU, 13—open-ECU Dastek unichip Q (Pretoria, Republic of South Africa), 14—high pressure pump of diesel fuel common rail injection system, 15—Krohne Optimass 3300C flowmeter for diesel fuel (Duisburg, Germany), 16—diesel fuel tank, 17—gaseous hydrogen tank, 18—Alicat Scientific MC50 flowmeter for hydrogen (Tucson, Arizona, USA), 19—flame trap, 20—hydrogen injectors.

The diesel engine is equipped with a secondary fuelling system used for hydrogen fuelling. The hydrogen fuelling system consists of an electromagnetic hydrogen fuel injector, model Milano (Gujarat, India) operated by an electronic control unit, flame arrestor, hydrogen flowmeter, hydrogen tank and pressure reducers. The electronic control unit Dastek Unichip Q (Pretoria, Republic of South Africa) is an open type ECU and is connected to the engine's electronic control unit so that the diesel injectors can be controlled and commanded simultaneously with the hydrogen injectors. An Alicat Scientific MC50 flowmeter (Tucson, United Stated of America) is used to determine hydrogen consumption. For operational safety and to protect the hydrogen fuelling system, a flame arrestor is installed between the H_2 injectors and the hydrogen tank. The hydrogen fuelling line is equipped with two pressure reducers in order to control the hydrogen pressure in two steps, in the 150–0.5 bar domain. In the first step of pressure reduction, the hydrogen pressure is decreased from 150 bar to 10 bar. For the second step of pressure reduction the hydrogen pressure is tuned from 10 bar to 0.5 bar. The hydrogen fuelling pressure in front of the hydrogen injectors was set up at 1.5 bar.

2.2. Experimental Investigations Methodology

The methodology of the hydrogen and diesel fuel fuelling requires the control of the hydrogen and diesel fuel cyclic quantities. The operating regime of 19 kW, 70 Nm and 2500 rev/min, is a regime of interest, being an operating regime frequently used in engine operation. The hydrogen flow was set up in the domain of 0.42–0.44 slpm in order to achieve the values of the Energy Substitution Degree (ESD) of 19%, 33% and 43%, percentages at which normal engine operation is ensured. The ESD is established by using

cyclic masses and Lower Heating Values (LHV) of both fuels, the calculation is made with the Formula (1):

$$\mathrm{ESD} = \frac{\mathrm{HC_{H_2}} \cdot \mathrm{LHV_{H_2}}}{\mathrm{HC_{H_2}} \cdot \mathrm{LHV_{H_2}} + \mathrm{HC_{df}} \cdot \mathrm{LHV_{df}}} \times 100 \quad (1)$$

where HC_{H2}—hourly Consumption of hydrogen, HC_{df}—hourly consumption of diesel fuel LHV_{H2}—Lower Heating Value of hydrogen, LHV_{df}—Lower Heating Value of diesel fuel.

The reference of the experiment is setup for ESD = 0% when the engine is fuelled only with diesel fuel. At dual fuelling, the diesel fuel cyclic quantity is reduced and the cyclic quantity of the hydrogen is increased until the engine power is reestablished to the reference value. The tune of the fuels cyclic quantities is established thru the modification of the injectors opening durations via DastekUnichip ECU (Pretoria, Republic of South Africa). A number of 200 consecutive cycles are measured and registered for each ESD equal to 0%, 19%, 33% and 43%. Diesel fuel and hydrogen consumptions, level of the exhaust emissions, supercharging pressure and temperatures were measured as set of five three measurements was recorded for each ESD. The error of the experimental results [30] was evaluated with the Formula (2):

$$\mathrm{Er} = \frac{\frac{1}{x} \cdot \sum_{j=1}^{x} \left(d_j - \frac{\sum_{j=1}^{x} d_j}{x} \right)^2}{\frac{\sum_{j=1}^{x} d_j}{x}} \times 100\ [\%] \quad (2)$$

where: d—the data of interest, j—the current value of the data, x—the number of data's inside the measurement sample. The values of Er do not exceed 1%.

The calculation relationship for the coefficient of cyclic variability COV related to the variability of the combustion parameter [30] is of the form:

$$(\mathrm{COV})_{CP} = \frac{\sqrt{\frac{1}{N_C} \cdot \sum_{k=1}^{N_C} \left(CP_k - \frac{1}{N_C} \cdot \sum_{k=1}^{N_C} CP_k \right)^2}}{\frac{1}{N_C} \cdot \sum_{k=1}^{N_C} CP_k} \cdot 100\ [\%] \quad (3)$$

where: CP—combustion parameter and successively represents maximum pressure (p_{max}), angle of maximum pressure (AMP), maximum pressure rise rate (MPRR) and indicated mean effective pressure (IMEP), k—current value in the consecutive cycle, N_C—number of the combustion cycle.

The combustion control in the diesel engine is done by controlling the autoignition delay. The smoke emission is defining for the acceptance of the exploitation of the diesel engine. Thus, the two parameters, the autoignition delay and the smoke emission level are of particular interest. In the study methodology, the experimental study is completed by the theoretically evaluation of the autoignition and the soot emission level, in order to explain the physical processes.

The experimental autoignition delay (AID) is evaluated by the Wolfer (4) Formula [32]:

$$\mathrm{AID} = 0.44 \cdot p^{-1.19} \cdot e^{\frac{4650}{T}}\ [\mathrm{ms}] \quad (4)$$

where the p and T are averaged values for pressure and temperature during autoignition phase of combustion.

The evaluation of the soot emission level is based on the soot forming speed (SFS) and soot oxidation speed (SOS), are calculated with the Hiroyasu-Morel-Karibel (5) and (6) formulas [32]:

$$\mathrm{SFS} = 2.5 \cdot 10^{-3} \cdot m_f \cdot p^{0.5} \cdot e^{-\frac{52298}{R \cdot T}} \left[\frac{mg}{s} \right] \quad (5)$$

$$SOS = 9.55 \cdot 10^{-4} \cdot m_s \cdot p^{0.5} \cdot e^{-\frac{5000}{T}} \left[\frac{mg}{s}\right] \quad (6)$$

where m_f is the mass of diesel fuel, m_s is the mass of soot, p and T are in-cylinder pressure and temperature and R is the ideal gas constant.

The calculated soot emission level is determined by the difference between the soot forming and soot oxidation speeds.

The accuracy of the equipment's used during experimental session reflects the precision of the measurement and is presented in the Table 2.

Table 2. Precision of the measurements on the test bench equipment's [34–40].

Parameter	Precision
Engine power [kW]	±0.2%
Engine torque [Nm]	±0.2%
Diesel fuel flow [kg/h]	±0.1%
Hydrogen flow [slpm]	±0.2%
Intake air [m^3/h]	±0.3%
HC emission level [ppm]	±1 ppm vol.
Smoke opacity [%]	±0.1%
NO emission level [%]	±1 ppm vol.
CO_2 emission level [%]	±0.65% vol.
In-cylinder pressure [bar] (including p_{max}, MPRR, IMEP)	±0.05%
Angle of maximum pressure [CAD]	±0.1%

Before the experimental investigations, all equipment at the stand was calibrated so that the measured values were correct. The engine operating parameters, in-cylinder pressure in consecutive combustion cycles, fuel consumptions, exhaust gases level, COV's were measured, recorded and calculated.

The paper presents an analysis of the experimental results obtained at the fuelling a modern diesel engine with diesel fuel and H_2, at 50% partial load and a speed of 2500 rev/min regime. This operating regime is often used in engine operation, the speed being close to the maximum engine torque regime, the load being usually used when driving a loaded vehicle in urban and extra-urban conditions. The influence of hydrogen dose on combustion, efficiency, carbon based emissions and mechanical operation is presented.

3. Results

The combustion of the air-hydrogen homogeneous mixture leads to the increase of the maximum pressure comparative to the diesel fuel use case, Figure 2.

At the increase of the hydrogen quantity the maximum pressure also increase. As average value for 200 consecutive combustion cycles at diesel fuel use the maximum pressure is 103.75 bar and increases with 19% till 123.66 bar as average value due to reduction of the combustion duration at hydrogen use comparative to classic fuelling, Figure 2. For 19%ESD the average maximum pressure increases with 11.35% and for 33%ESD the average maximum pressure increases with 17.2% versus diesel fuel use, 0%ESD. At diesel fuel and hydrogen fuelling, the maximum pressure increases, and at 43%ESD in some individual cycles, peak values of 126.54 bar and 128.56 bar, respectively, are reached, but these values do not exceed the permissible range that ensures the normal operation of the diesel engine. However, the tendency of maximum pressure increase may

be a criterion for limiting the amount of hydrogen used at 43%ESD in order to maintain the reliability and normal wear of the engine. Other researchers obtained similar tendency or results [13,14,16–19,22,26–28].

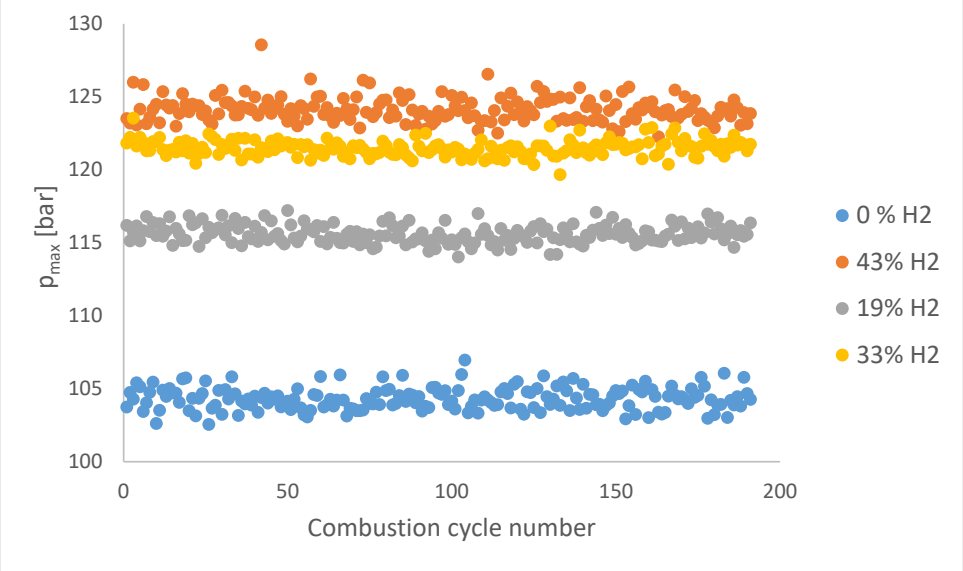


Figure 2. Maximum pressure for different energy substitution degrees.

The heat release rate (HRR) increases at dual fuelling with diesel fuel and hydrogen, Figure 3. When using the amount of hydrogen 19% ESD and 33%ESD, the peak value of the heat release rate increases by 12.5%, reaching 1 CAD faster per cycle, which shows an acceleration of the combustion process. For 43%ESD the peak of HRR increases with 11% comparative to classic fuelling. The HRR curves tends to become slightly vertical at hydrogen use, which would correspond to the acceleration of the combustion process in the presence of air-hydrogen mixtures, compared to diesel fuel. It also seems that at hydrogen use, combustion seems to end earlier. Similar results were obtained by other researchers [14,16,26].

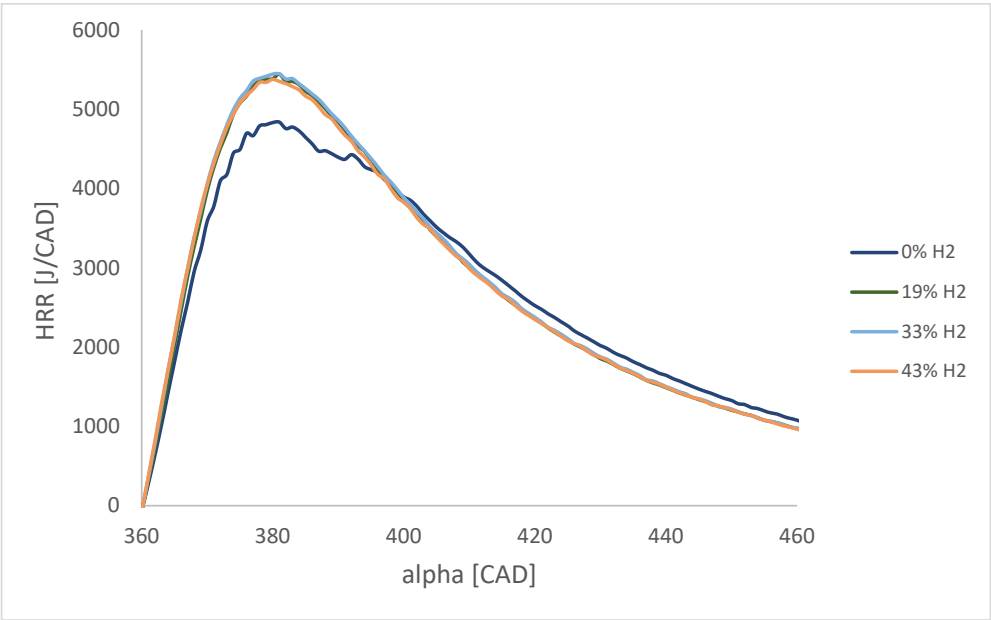


Figure 3. Raw heat release rate for different energy substitution degrees.

Due to higher value of the LHV of the hydrogen comparative to diesel fuel, at hydrogen use the mechanical work increases. The indicated mean effective pressure (IMEP) increases with 4.95% at 19%ESD, with 7.5% at 33% ESD and with 4% at 43%ESD comparative to diesel fuel use, 0%ESD, Figure 4. Similar results were obtained by other researchers [13,24].

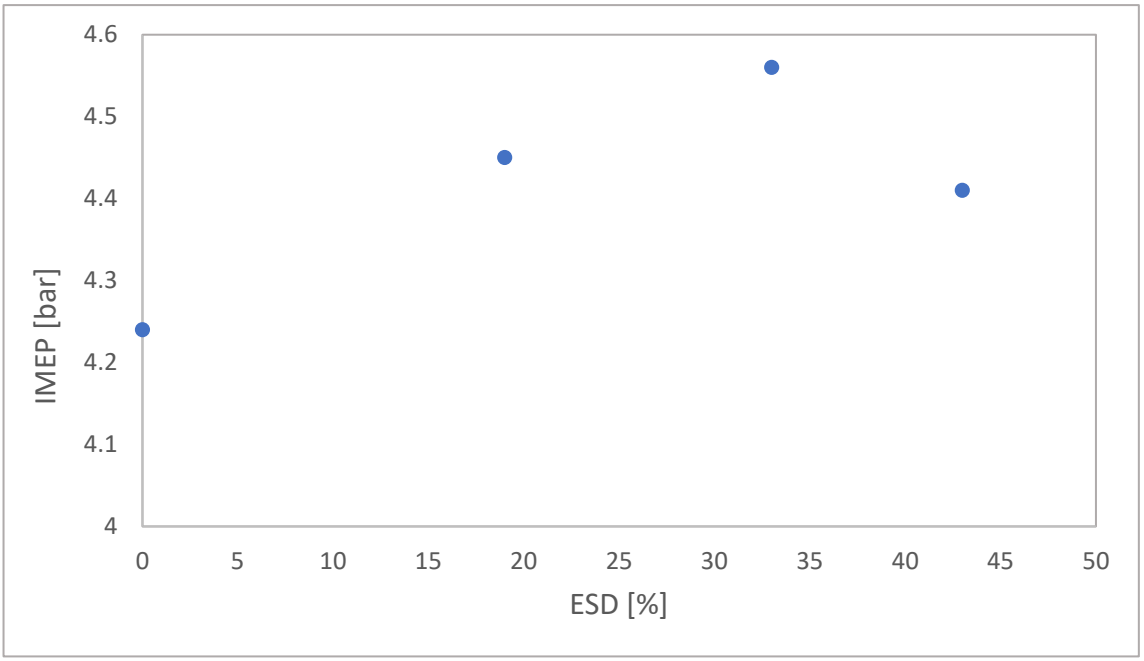


Figure 4. The IMEP for different for different energy substitution degrees.

For dual fuelling the specific energy consumption (SEC) decreases with 2.94% at 19%ESD, with 5.36% at 33%ESD and with 3.73% for 43%ESD. At hydrogen use, the engine economy increases, especially for the ESD range of 19–33%, Figure 5.

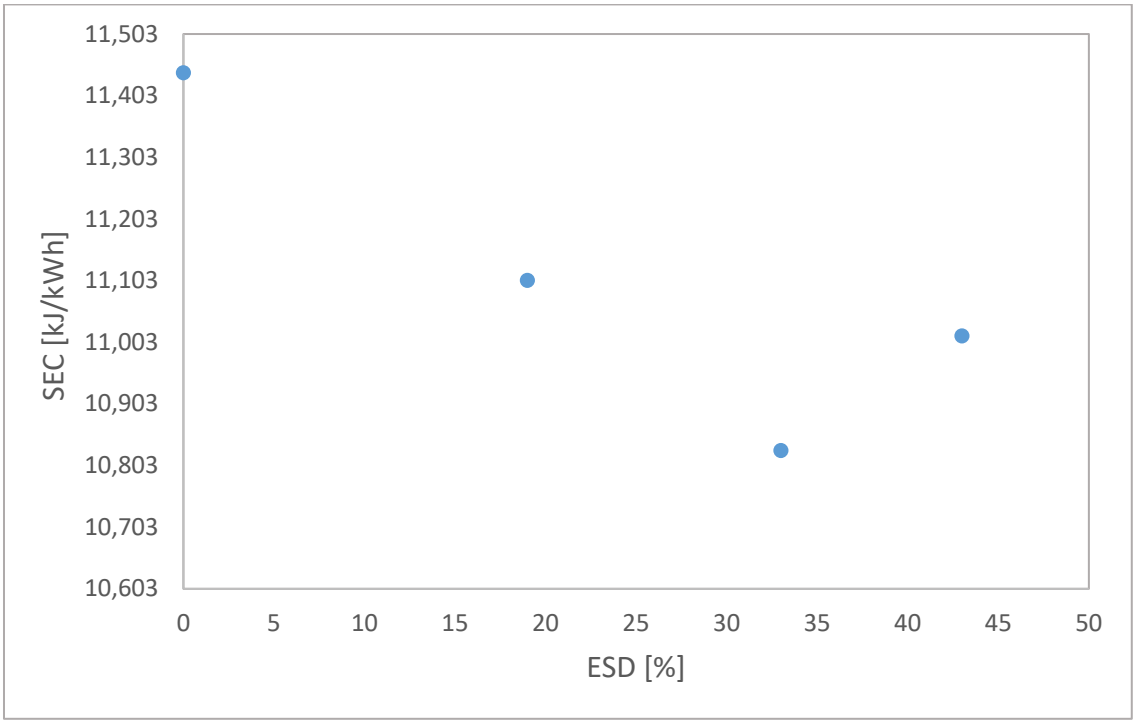


Figure 5. Specific Energy Consumption for different energetic substitutions degrees.

The SEC decreases at H_2 use is due to the higher LHV of H_2, higher burning speed and wider flammability limits of H_2 versus diesel fuel. Similar result of specific energetically fuel consumption variation were obtained also by others researchers [11,12,16,24].

In a diesel engine, the level of HC emission is significantly influenced by the poor formation of the air-fuel mixture [30,32]. When diesel fuel is substituted with hydrogen, the amount of liquid fuel that must be sprayed, vaporized and mixed with air is reduced. Thus, is reduced the amount of diesel fuel in the preformed mixture and the amount of the mixture that burns diffusively. At the same time, the wide flammability limits of air-hydrogen mixture ($\lambda s = 0.13$–$\lambda i = 10.08$ for H_2 versus $\lambda s = 0.34$–$\lambda i = 1.68$ for diesel fuel) and its higher burning speed (250 cm/s for hydrogen and 116 cm/s for diesel fuel, at $\lambda = 1.5$), reduce the probability of flame extinguishing in the gas mass or at the wall [2,30–32]. The higher speed of combustion of air-hydrogen mixtures favours the reduction of the HC emission level, due to acceleration of the flame propagation, Figure 6. Thus, the level of HC emission is reduced by 10% at 19%ESD, by 18% at 33%ESD and by 22% at 43%ESD. Similar results were obtained by other researchers [14–16,18,20,21,23].

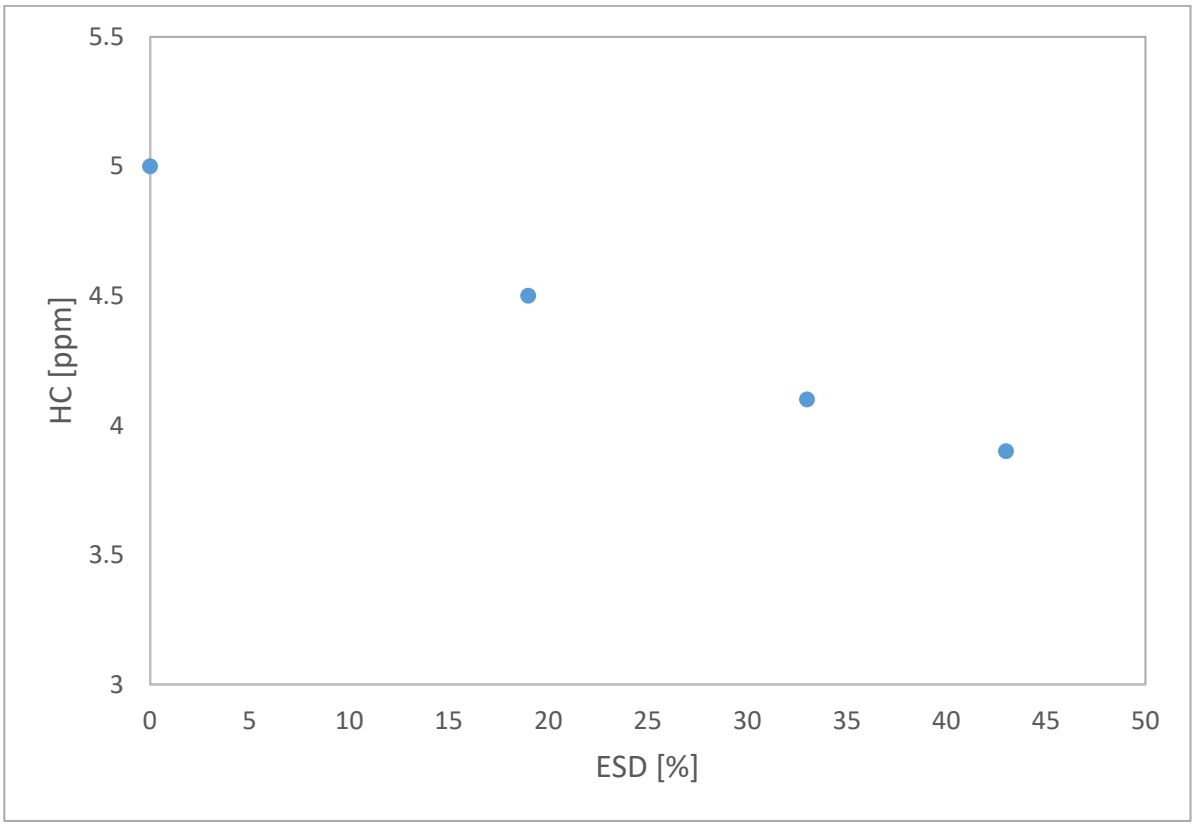


Figure 6. The HC emission level versus energetic substitution degrees.

Hydrogen, H_2, is a carbon neutral fuel, with 0% carbon, and its use in substitution of the diesel fuel, $C_{16}H_{34}$, which contains 85.7% carbon, leads to the reduction of the carbon content into the in-cylinder mixture. The exhaust smoke of the diesel engine is formed from soot particles and liquid particles of unburned or partially oxidized fuel. The following types of smoke are distinguished by their appearance:

– white smoke (cold smoke) present when the engine is started, consisting of suspended liquid fuel particles, the colour being due to the droplet diameter of over 0.8 μm, wavelength corresponding to blue light, thus the drops appear white colour [30];

- blue smoke, which appears later after startup, the engine heats up and the smoke turns blue as a result of the droplet diameter being reduced to below 0.8 μm and the drops appear colored in blue [30];
- greyish-blue smoke which appears when oil enters the combustion chamber and is not related by hydrogen use or not [30];
- black smoke (hot smoke) is formed by carbon particles resulting from the incomplete combustion because in the high temperature area, carbon complexes appear through a cracking process, which together with aromatic hydrocarbons and tars, through agglomeration, produce flakes that coagulate and form hot soot. The use of hydrogen may assure the decrease of the level for all types of smoke [30].

At hydrogen use, the quantity of the diesel fuel droplets that burns diffusively is reduced. Thus, the diffusive combustion process is improved, which can have the effect of reducing the amount of soot formed in the area of the jet where the diesel fuel is injected into the flame. At replacing diesel fuel with hydrogen, the nucleation phenomenon that produces soot is reduced; also, the soot oxidation is accelerated due to the higher combustion speed of the air-hydrogen mixture. The acceleration of soot oxidation under conditions where the initial amount of soot is lower at H_2 use, leads to a reduction of the smoke emission level. At H_2 fuelling, the smoke opacity is reduced by 44.8% at 19%ESD, by 58.6% at 33%ESD and by 65.5% at 43%ESD comparative to diesel fuel fuelling case, Figure 7. Other researchers obtain similar results [15,16,18,20,23,28].

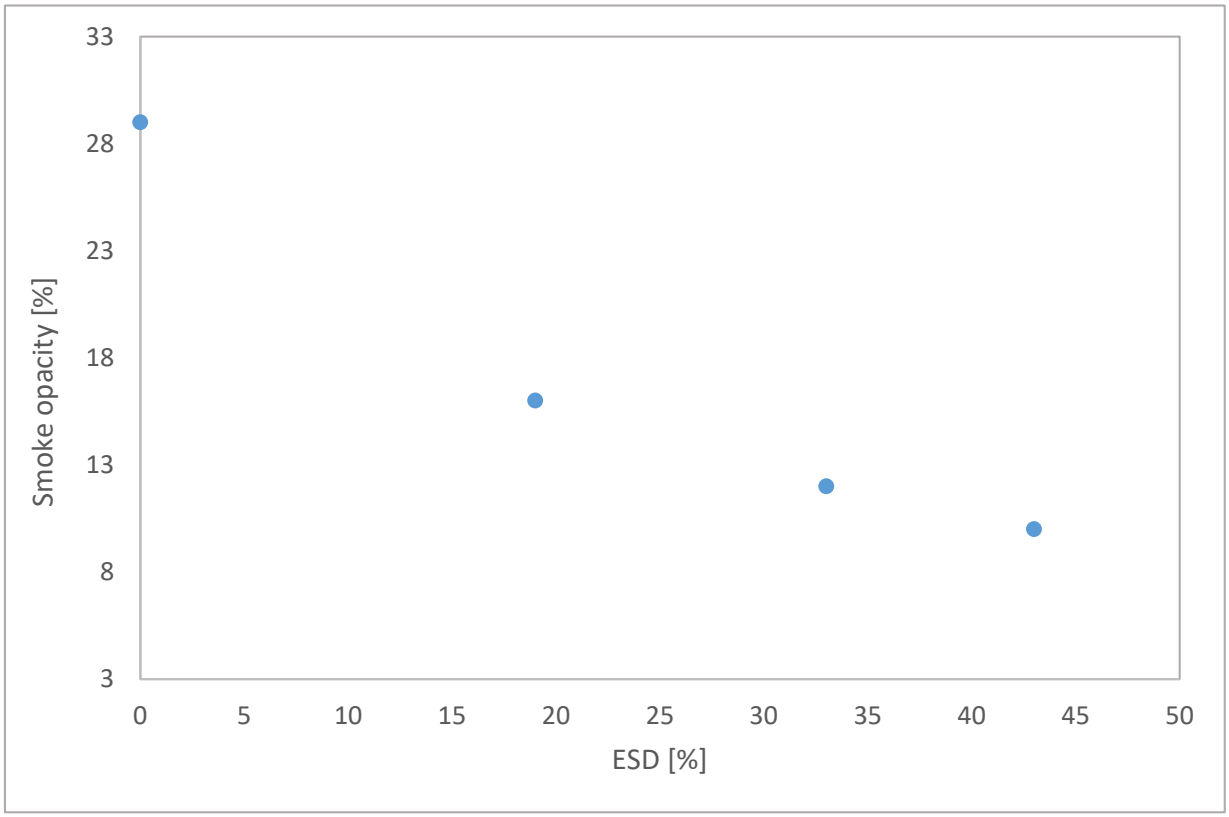


Figure 7. The smoke opacity level versus energetic substitution degrees.

The NO emission level slightly increases at H_2 use, Figure 8. At dual fuelling with hydrogen and diesel fuel, the NO emission level increases with 18.5% at 19%ESD, by 33.3% at 33%ESD and by 49.3% at 43%ESD comparative to diesel fuel fuelling, Figure 8.

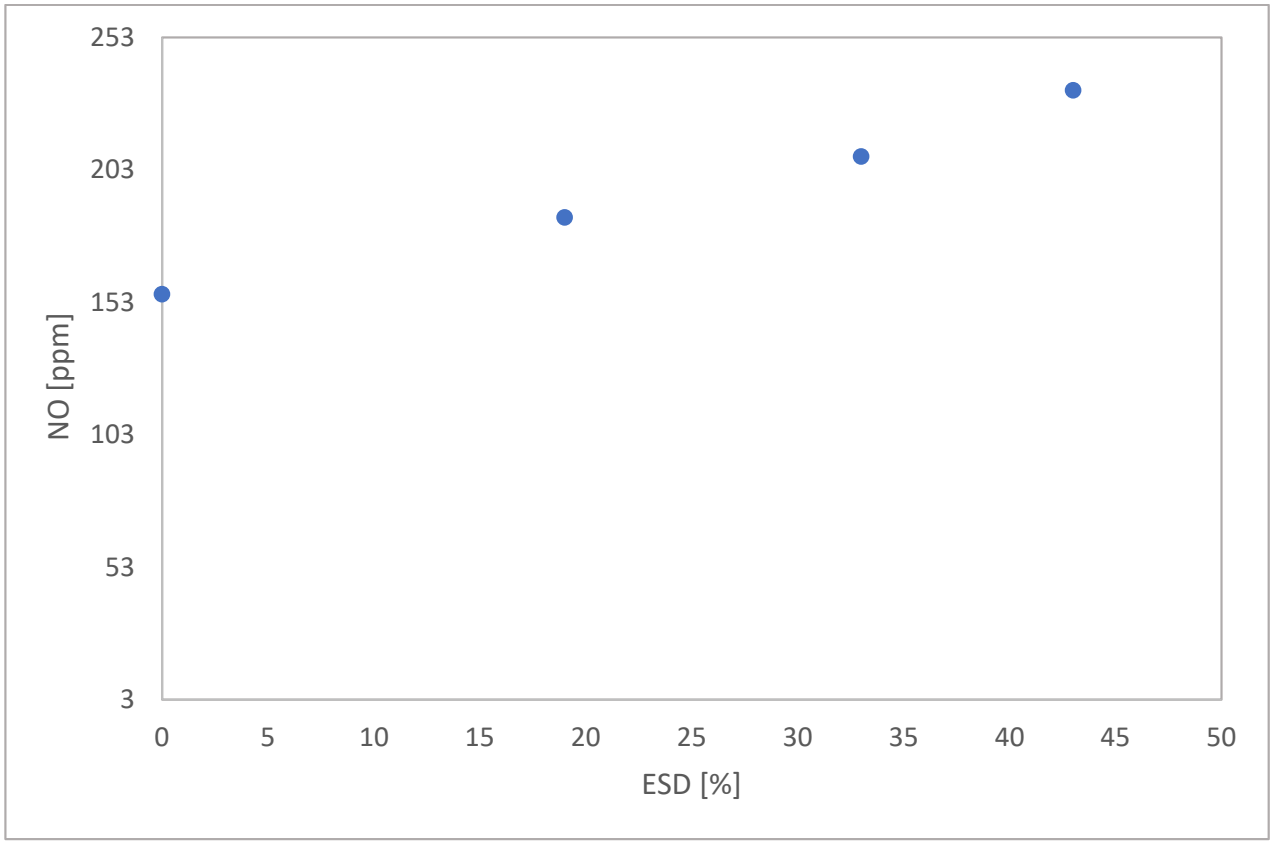


Figure 8. The NO emission level versus energetic substitution degree.

The NO emission level increases at H_2 fuelling as result of the increased combustion temperature. The increase of the NO emission level is related with the increase of the oxygen amount, the coefficient of air-excess λ being increased at H_2 use from 1.9 at 0%ESD to 2.13 at 19%ESD, to 2.19 at 33%ESD and to 2.2% at 43%ESD. Similar results have been obtained by other researchers [10,12,14,16,17,24,27,28]. Further reductions of the NO can be assured by increasing the amount of EGR and, if necessary, by using the AdBlue method.

At the use of carbon neutral fuels such as hydrogen, the carbon content of the in-cylinder mixture is reduced, which also ensures a reduction in the level of CO_2 emission.

When diesel fuel is replaced by hydrogen, the in-cylinder mixture carbon content is reduced to 69.4% carbon for 19%ESD, to 57.4% carbon for 33%ESD and to 48.8% carbon for 43%ESD. Thus, the CO_2 emission level is reduced by 28.6% at 19%ESD, by 34.5% at 33%ESD and 43%ESD, Figure 9. The reduction in carbon content also influences the reduction in HC and smoke emissions, Figures 6 and 7. The higher amount of oxidant required for hydrogen combustion could limit the CO_2 decrease (the oxygen required for complete combustion is 0.2500 kmol O_2/kg H_2 for hydrogen versus 0.1043 kmol O_2/kg $C_{16}H_{34}$ for diesel fuel; the minimum amount of air required for complete combustion is 1.1900 kmol air/kg H_2 for hydrogen versus 0.4966 kmol air/kg $C_{16}H_{34}$ for diesel fuel) [29–32]. But at the same time, at the increase of the H_2 quantity, the carbon content also decreases and the reduction in carbon content is greater than the increase in H_2 amount and the need to increase the amount of air. In future studies, the amount of air per cycle, respectively the amount of oxygen, can be increased to see if the reduction in CO_2 levels can be further enhanced. Similar results of CO_2 reduction were obtained also by other researchers [10,11,14,16,20,21,23–25,28,29].

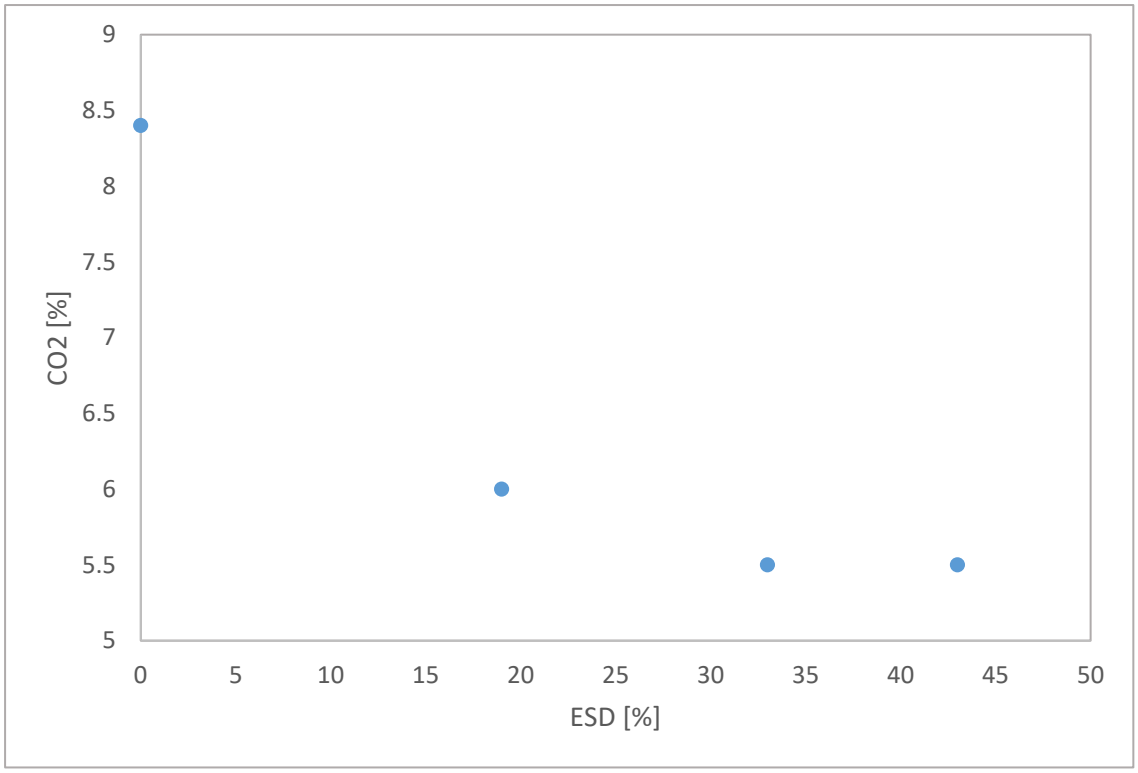


Figure 9. The CO_2 emission level versus energetic substitution degree.

Fuelling a diesel engine with hydrogen offers the advantage of reducing carbon based emissions, increasing economy and energy performance, but it must be verified if the normal operation of the engine is ensured. It is therefore necessary to study the cyclic variability of the combustion process at H_2 fuelling. The evaluation of the cyclic variability during the combustion process is usually done using two parameters, coefficient of variability for the maximum pressure, $(COV)_{Pmax}$, and coefficient of variability for the indicated mean effective pressure, $(COV)_{IMEP}$. Cyclic variability can be well evaluated by the values of these two coefficients, which are correlated with the engine's traction adaptability. The normal vehicle (engine) handling is satisfied if the values for $(COV)_{Pmax}$ and $(COV)_{IMEP}$ do not exceed 10% [30–32].

The coefficients of cyclic variability are defined for a number of successive cycles, N_C, in accordance with the hypothesis of a normal distribution for the values of maximum pressure and indicated mean effective pressure.

The COV of the maximum pressure is usually used to evaluate the cyclic variability during engine operation at the maximum brake torque speed (MBT speed) or at regimes close to this regime, especially when adjustments are made to the injection timing. Since the investigated operating regime is that of the MTB speed, but at partial load, 70 Nm, the determination of the $(COV)_{Pmax}$ at different ESD values is also made. The variation of $(COV)_{Pmax}$ with ESD is shown in Figure 10.

At hydrogen use, the $(COV)_{Pmax}$ is continuously reduced at the increase of the H_2 cyclic dose. At diesel fuel use, 0%ESD, the $(COV)_{Pmax}$ is 0.73% and is reduced to 0.53% for 19%ESD, to 0.44% for 33%ESD and to 0.68% at 43%ESD. The $(COV)_{Pmax}$ reduction shows the decrease in the cyclic dispersion between the maximum pressure values from one cycle to another, which can also be observed in Figures 11 and 12. At maximum dose of hydrogen, the dispersion between the maximum pressure values, Figure 12, is smaller than in the case of diesel fuel use, Figure 11.

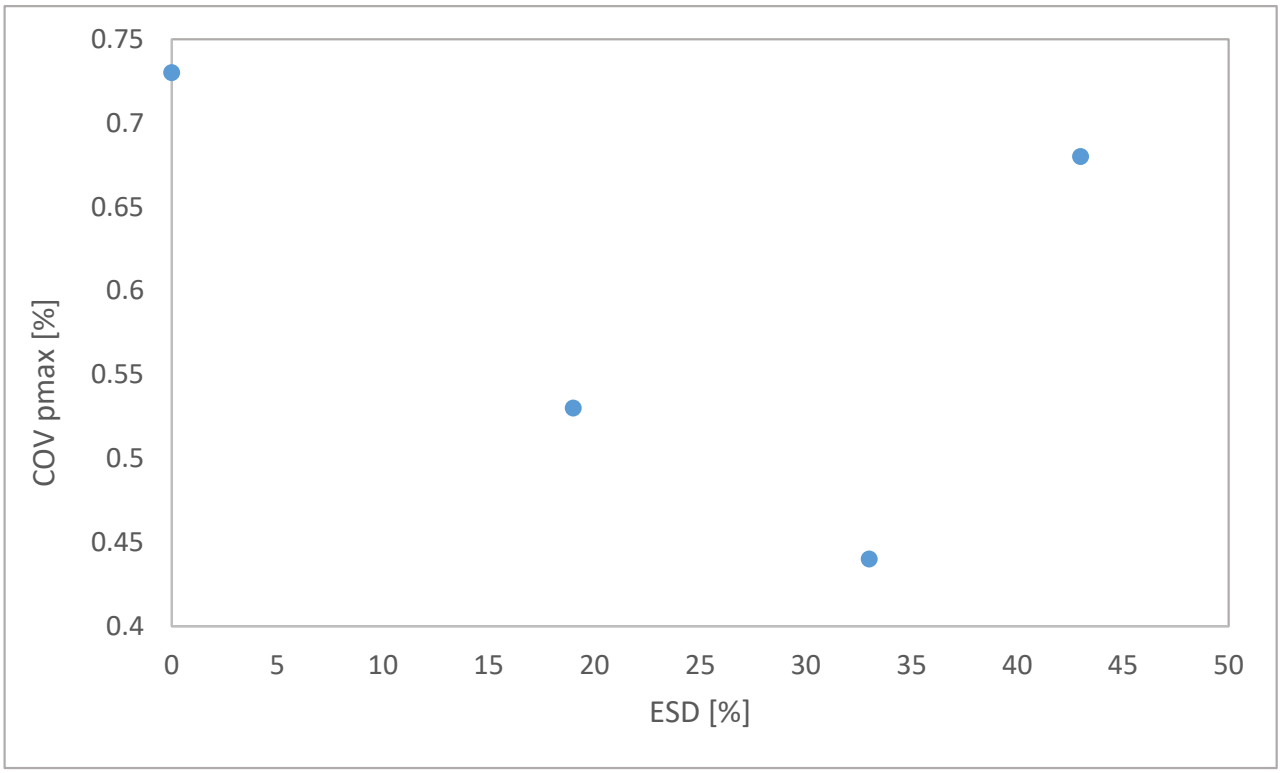


Figure 10. The COV of maximum pressure versus energetic substitution degree.

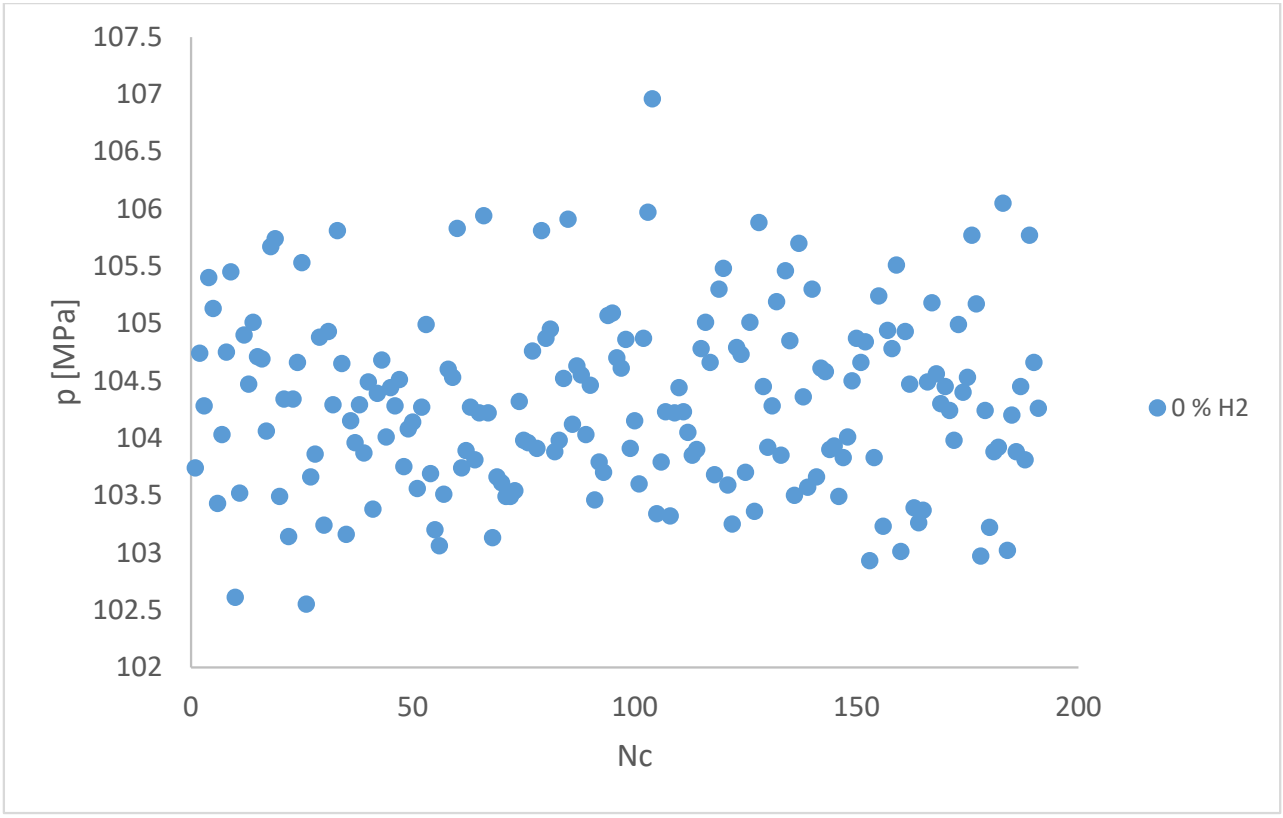


Figure 11. Cyclic dispersion between values of maximum pressure in consecutive combustion cycles for diesel fuel use, at 0% H_2.

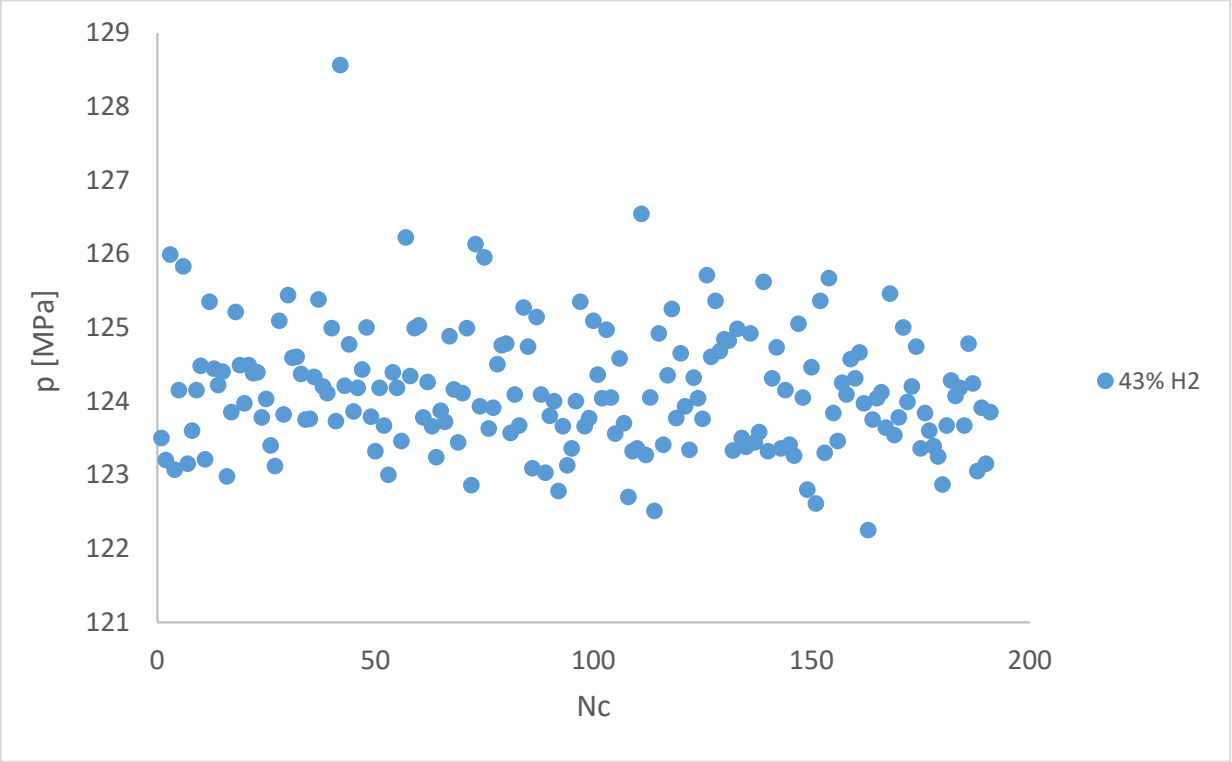


Figure 12. Cyclic dispersion between values of maximum pressure in consecutive combustion cycles for hydrogen and diesel fuel use, at 43% H_2.

The cyclic variability for the angle of maximum pressure (AMP) is reduced for substitution degrees in the range of 19–33%, but for higher substitution degrees, 43%ESD, the COV of AMP increases, Figure 13.

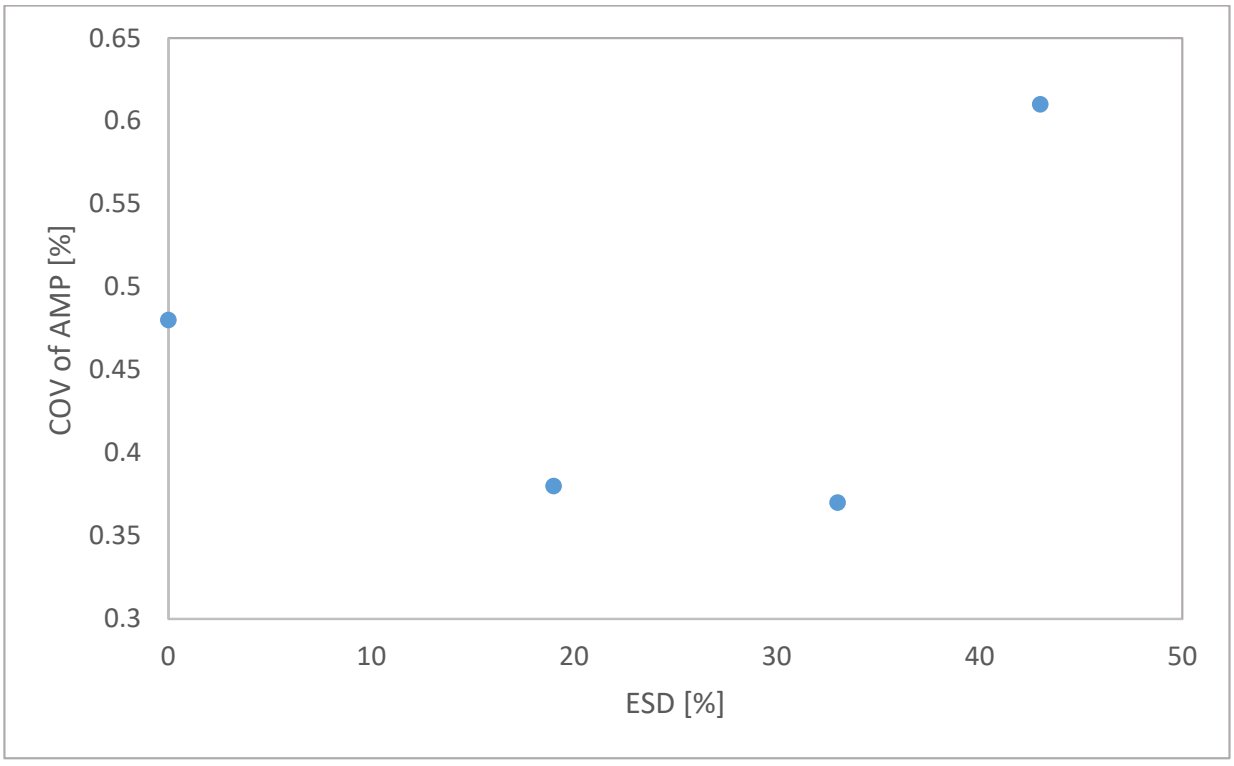


Figure 13. The COV for angle of maximum pressure versus energetic substitution degree.

The cyclic variability of the maximum pressure rise rate during combustion is reduced at hydrogen use, compared to classic fuelling, Figure 14. The coefficient of variability for the maximum pressure rise rate, COV of MPRR, decreases from 14.57% for 0%ESD to lower values such as 11.63% for 19%ESD, 6.97% for 33%ESD and 11.02% for 43%ESD.

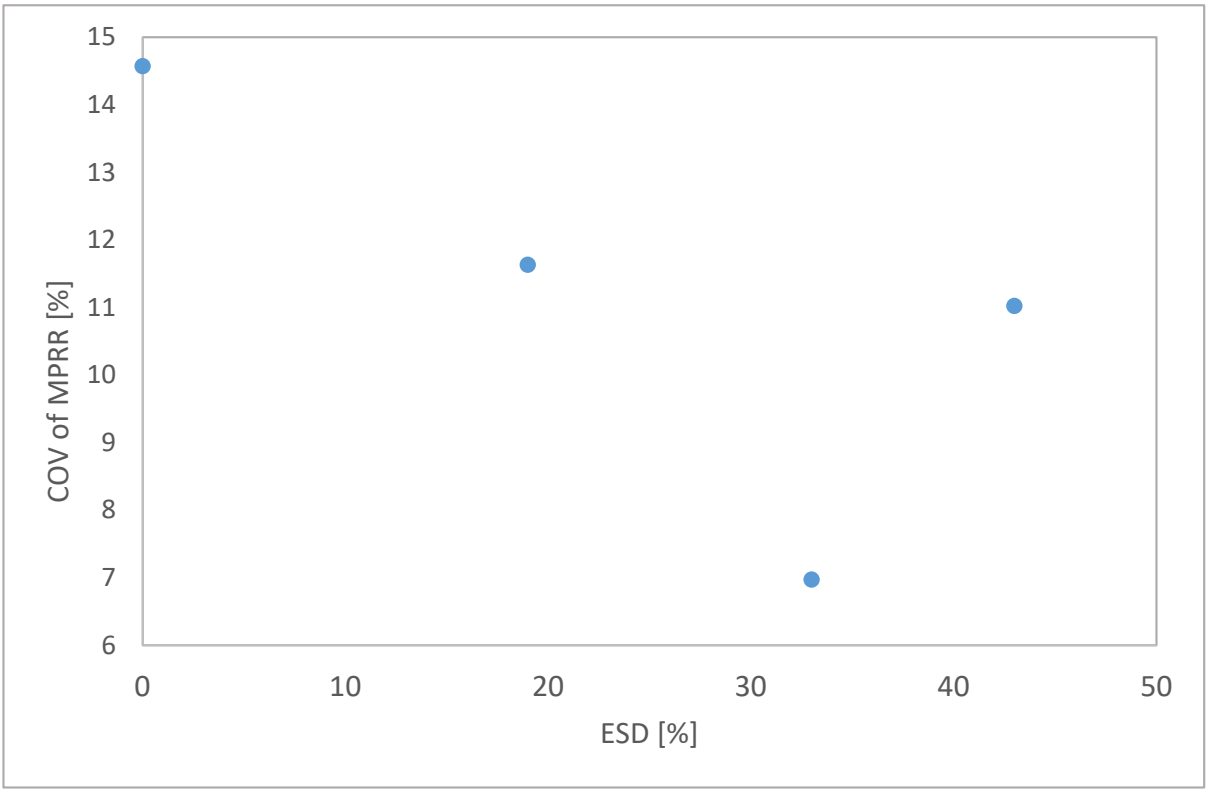


Figure 14. The COV of maximum pressure rise rate versus energetic substitution degree.

The engine response to combustion variability is better evaluated based on the coefficient of cyclic variability of the indicated mean effective pressure, $(COV)_{IMEP}$. The $(COV)_{IMEP}$ shows the engine's ability to reproduce the combustion phases from cycle to cycle when a new fuel, such as hydrogen, is used. By analyzing the $(COV)_{IMEP}$ variation, the instability of the autoignition and combustion process is better evaluated, observing the irreproducibility of the combustion phases in successive cycles. The variation of $(COV)_{IMEP}$ at different energetic substitution degrees is presented in Figure 15.

At dual fuelling with hydrogen and diesel fuel, the cyclic variability coefficient of IMEP decreases from 1.13% to 0.68% for 19%ESD, 0.67% for 33%ESD and 1.07% for 43%ESD, Figure 15. At hydrogen use, due to wider flammability limits and higher burning rate of air-hydrogen mixtures, the instability of the autoignition and combustion process is reduced. Thus, is ensures the reproducibility of the combustion phases, the acceleration of autoignition, rapid and diffusive combustion in successive cycles, by increasing the reaction rate as a result of the modification of the activation energy of the air-fuel mixture in the presence of H_2 [31,32]. This leads to the reduction of the cyclic variability of IMEP, especially at percentages of 19% and 33% H_2.

At higher H_2 percentages, 43%H_2, the establishment of ultra-lean mixtures in the cylinder, before the start of combustion, can leads to an increase in the $(COV)_{IMEP}$ value, Figure 15. The $(COV)_{IMEP}$ value assigned to 43%ESD is below the value assigned to operation with diesel fuel. Generally, higher combustion speed of air-hydrogen mixtures (2.37 m/s at $\lambda = 1$) versus air-diesel fuel mixtures (0.25–0.35 m/s at $\lambda = 1$) leads to the reduction of the COV values due to reduction of the combustion duration [31,32].

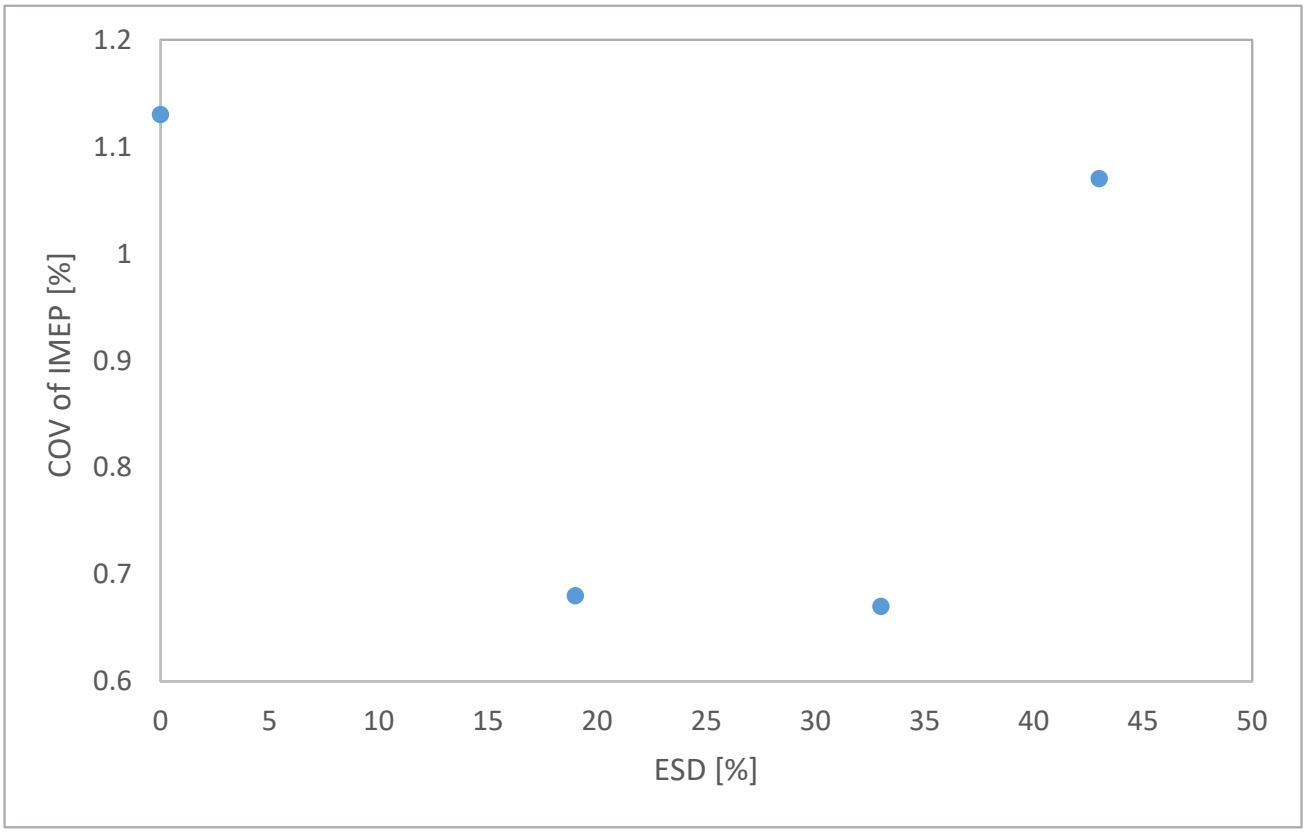


Figure 15. The COV of indicated mean effective pressure versus energetic substitution degree.

Other researchers obtained similar results regarding the COV variation [12,19,22].

The duration of the combustion process is reduced at hydrogen fuelling compared to diesel fuel fuelling, Figure 16. The higher combustion speed of the air-hydrogen mixture leads to a reduction in the combustion duration by 14.24 CAD for 19%ESD, by 12.23 CAD for 33%ESD and by 9.39 CAD for 43%ESD. The reduction in the combustion duration at H_2 use is correlated with the increasing trend of the maximum pressure, Figure 2. The reduction in the combustion duration is correlated with the decreasing trend of the HC and smoke emissions, Figures 6 and 7, and of the coefficients $(COV)_{Pmax}$ and $(COV)_{IMEP}$, Figures 10 and 15. Other researchers obtained similar results regarding the combustion duration variation [12,22,25].

The combustion phase's duration is influenced by the autoignition delay duration. The autoignition of the diesel fuel has a catenary-thermal character, the beginning of the reaction becoming catenary, later superimposed by a thermal effect [31,32]. The time elapsed from the appearance of the first reactive elements until reaching the critical reaction speed, called autoignition delay, is influenced by the presence of hydrogen in the air-diesel fuel vapour mixture. The minimum ignition energy of hydrogen is lower comparative to classic fuel (0.018 mJ for H_2 versus 0.2–0.3 mJ for classic fuels) [4,29–32]. The flammability limits are larger for air-H_2 mixture (λ = 0.13–10.2 for H_2 versus λ = 0.33–1.68 for diesel fuel), favouring faster ignition and a shortening of the autoignition delay duration [31,32]. The higher the activation energy is, the longer the induction period will be. By reducing the pressure and temperature, the autoignition delay increases. The combustion of air-hydrogen mixtures has the opposite influence.

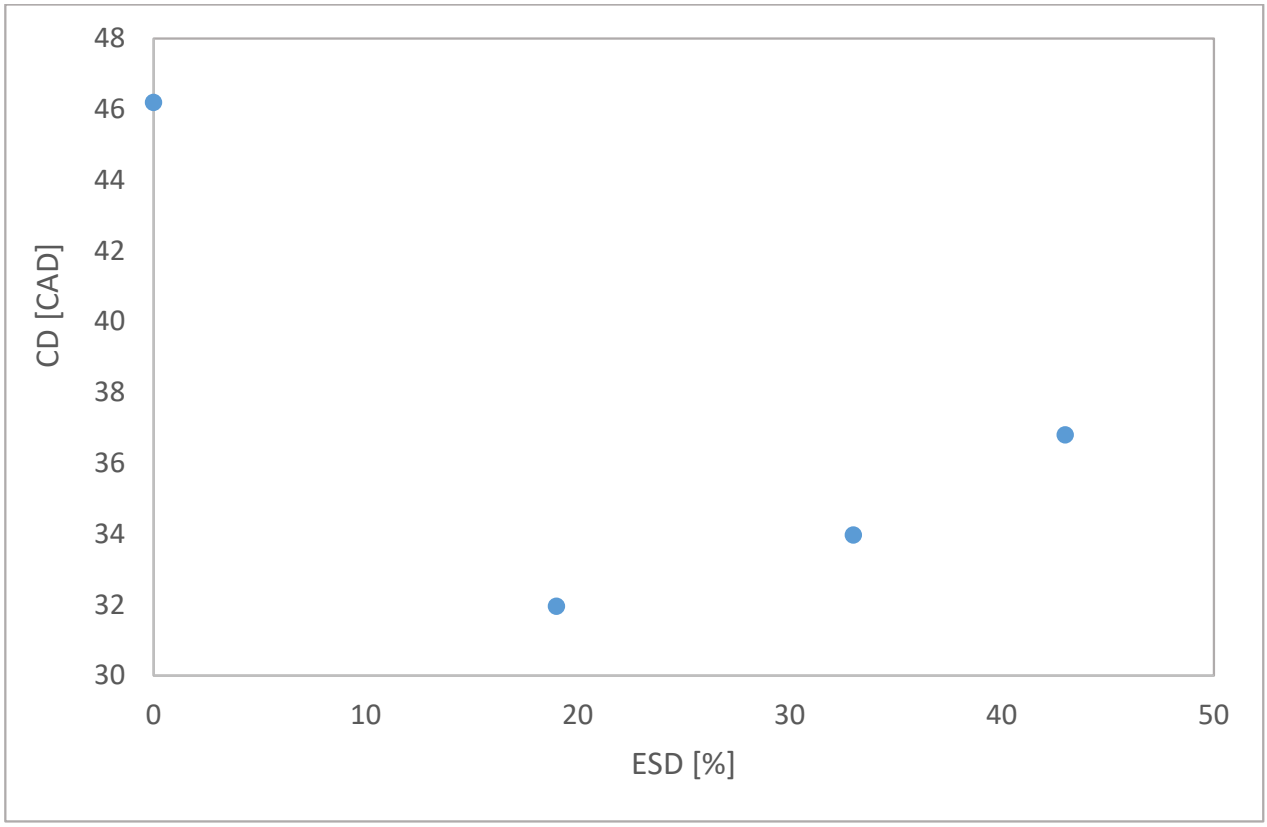


Figure 16. The combustion duration versus energetic substitution degree.

Inside the diesel engine cylinder, favourable conditions are created for the multistage ignition of the air-fuel mixture. The initial reactive elements created by oxidizing hydrocarbon molecules with the formation of peroxides, oxygen-rich substances. After the period of peroxide formation, in which large amounts of peroxides have accumulated, reaching a critical concentration, they decompose explosively, forming aldehydes and radicals. The reactions are exothermic, so the amount of heat released determines the increase in pressure and temperature, the luminescence that occurs during this period being characteristic of the radiation of formaldehyde molecules, reactions positively influenced by the combustion of air-H_2 mixtures [31,32]. These reactions have been called cold flames and take place during the cold flame period.

By oxidizing aldehydes, a new type of peroxide is formed which, through accumulation, decomposes explosively when they reach the critical concentration, forming a new type of flame, called secondary cold flame [31,32]. This process is developed with heat release, increase in pressure and temperature. The time allocated to these reactions, in which radicals and carbon monoxide, CO, appear as decomposition products, is called blue flame, due to the specific luminosity of carbon monoxide [31,32].

The large number of radicals present together with the oxidation of CO determines the appearance of a hot flame, in which the entire amount of heat is released, and the pressure and temperature increase [32].

The combustion of air-H_2 mixtures leads to an increase in pressure and temperature, accelerating the appearance of a hot flame. Thus, autoignition at low temperatures is a succession of chemically different stages, a process called multistage autoignition. The reactions specific to these stages take place in the entire volume of the mixture, the flame nucleus appearing where the conditions are favourable, the chemical reaction speed being the highest [32]. All these processes seems to be accelerated by the presence of H_2, the autoignition being reduced, Figure 17.

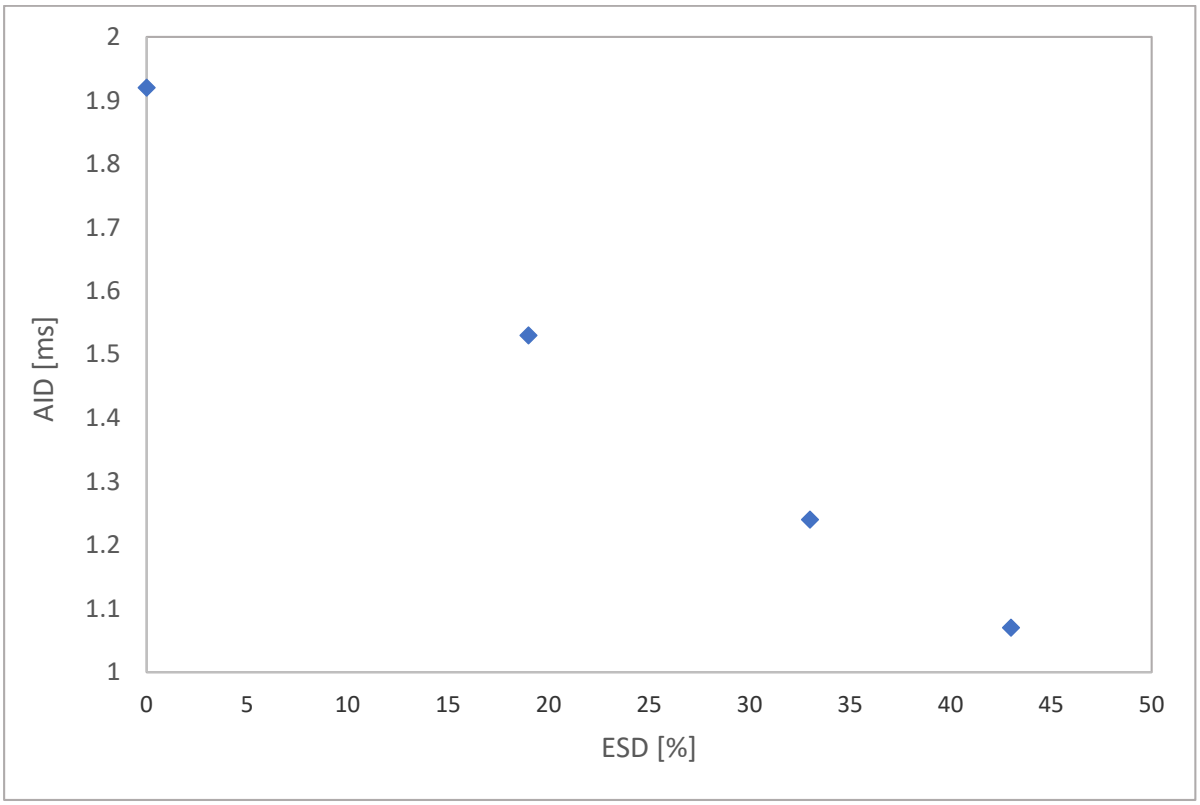


Figure 17. The autoignition duration versus energetic substitution degree.

In correlation, the decreasing trend of the values of the cyclic variability coefficients, $(COV)_{Pmax}$ and $(COV)_{IMEP}$, shows that the engine fuelled with diesel fuel and hydrogen can operate properly, even with lower cyclic variability for the substitution degrees of 19% and 33%. In order to limit the increasing trend of the maximum pressure during combustion, of the cyclic variability of maximum pressure and IMEP, but also to maintain the reduction of SEC, it is possible to consider limiting the degree of energetic substitution to the value of 43%. At this limit value of 43%ESD, a 22% reduction of the HC emission level, a 65.5% reduction of the smoke emission level and a 34.52% reduction of the CO_2 emission level are obtained.

Regarding the obtained results, at the increase of the hydrogen cyclic quantity the maximum pressure increases due to the reduction of the combustion process, knowing that air-H_2 mixtures offers large limit of inflammability and higher combustion speed, comparative to air-diesel fuel mixture. Higher combustion flame speed developed in air-H_2 mixture (250 cm/s for air-H_2 versus 116 cm/s for air-diesel fuel at lean dosage $\lambda = 1.5$) leads to the reduction of the combustion duration and of the cyclic variability [32]. As the burning duration decreases, the cyclical variability decreases, the cyclical variability coefficients reducing their values. The interval 19–33%ESD it seems to be closer to an optimal range of hydrogen use due to the limitation of the maximum pressure, around 120 bar, and limitation of the IMEP decrease. More importantly, for this range, HRR increases, SEC decreases and COV values continuously decrease. For this range, 19–33%ESD, continuous reductions in CO, HC, smoke, and CO_2 emissions levels are noted.

Limiting the amount of hydrogen to 43%ESD would be necessary to limit the increase in the level of NO emission. Of course, in further investigations, the level of NO emission can be reduced by using the AdBlue method and increasing the EGR flow rate, but for this study, limiting the NO level increase requires limiting of the H_2 cyclic dose at 43%. By limiting the ESD to 43%, it is ensured: limiting the increase in SEC, limiting the reduction

of IMEP, even if the values for 43%ESD are superior to the operation with diesel fuel. By limiting the ESD to 43%, the increase in maximum pressure above ~130 bar is limited, the increase in combustion duration is limited, if the injection timing remains unchanged. More, the increase in $(COV)_{IMEP}$ and $(COV)_{AMP}$ values is limited, the cyclic variability of combustion being thus controlled.

The reduction of the combustion duration is influenced by the reduction of autoignition delay (AID) at hydrogen use, Figure 17. At hydrogen use, the autoignition delay, expressed in milliseconds, decreases by 20.3% at 19%ESD, by 35.4% at 33%ESD and by 44.2% at 43%ESD, as Figure 17 shows. Thus, the autoignition delay, expressed in crank angle degrees (CAD), decreases from 28.8 CAD at 22.9 CAD at 19%ESD, at 18.6 CAD for 33% ESD and at 16 CAD for 43%ESD. The autoignition delay in CAD is reduced by 20% at 19%ESD, by 35% at 33%ESD and by 44% at 43% ESD, which also influences the reduction of the combustion duration. The reduction of the autoignition delay in CAD is related with the reduction of the combustion process duration, Figure 16. At H_2 use, the activation energy of the mixture is modified and the ignition process is accelerated (hydrogen minimum ignition energy: 0.018 mJ versus classic diesel fuel minimum ignition energy: 0.2–0.3 mJ) [31,32]. Hydrogen is more reactive than diesel fuel and its use leads to the increase of the chemical reaction speed [31]. The duration of the appearance of aldehydes, specific to the formation of cold flames, is reduced. The oxidation of carbon monoxide is accelerated and the specific stage for blue flames formation is reduced. At the same time, the increase in pressure and temperature during the combustion of air-H_2 mixtures favors an increase in the chemical reaction speed, an acceleration of the multi-stage mechanism of autoignition and a reduction in the autoignition delay duration. Thus, preformed and diffusive combustion phases are placed closer to TDC, to the area of maximum turbulence, which leads to a reduction in the overall duration of combustion, an increase in maximum pressure and a reduction of specific energy consumption. In the presence of air-H_2 mixtures, combustion approaches more closely to an isochoric transformation, which ensures an increase in the effective efficiency and a reduction in the specific energy consumption.

The soot forming speed (SFS), at differed ESD [31,32] is presented in Figure 18.

The soot formation process starts from the formation of nucleus that subsequently generate soot particles, the process being influenced by the chemical nature of the fuel, but also by the pressure and temperature during combustion. The soot formation speed (SFS), which establishes the rate at which soot nucleus are formed, is influenced by the substitution of diesel fuel with hydrogen. Obviously, the higher soot formation speed is associated with engine operation at diesel fuel use, 0% ESD, Figure 18. The reducing of the diesel fuel quantity, due to increase of the H_2 dose, the soot formation speed is reduced by 17% for 19% ESD, 30% for 33% ESD, 38% for 43% ESD, Figure 18. Reducing the cyclic dose of diesel fuel leads to a reduction in the density of the soot nucleus that are formed, the content of hydrocarbons subjected to dehydrogenation and condensation processes being diminished. By integrating the soot formation speeds, the soot formation laws are obtained.

The soot oxidation speed (SOS) is presented in the Figure 19. The speed at which particles from the soot nucleus are oxidized is influenced by the speed of a chemical process controlled by the speed of adsorption and desorption of oxygen, respectively of the reaction products from the envelope of the gaseous diesel fuel droplet [32]. This process is influenced by the number of carbon atoms, a number that is reduced when diesel fuel is substituted with H_2. Hydrogen being more reactive, accelerates the soot oxidation reaction and the oxidation rate increases by 8.8% at 19% ESD, by 4.58% at 33% and by 1.75% at 43%. High-speed combustion of air-hydrogen mixtures can favour an increase in the oxidation speed of soot particles.

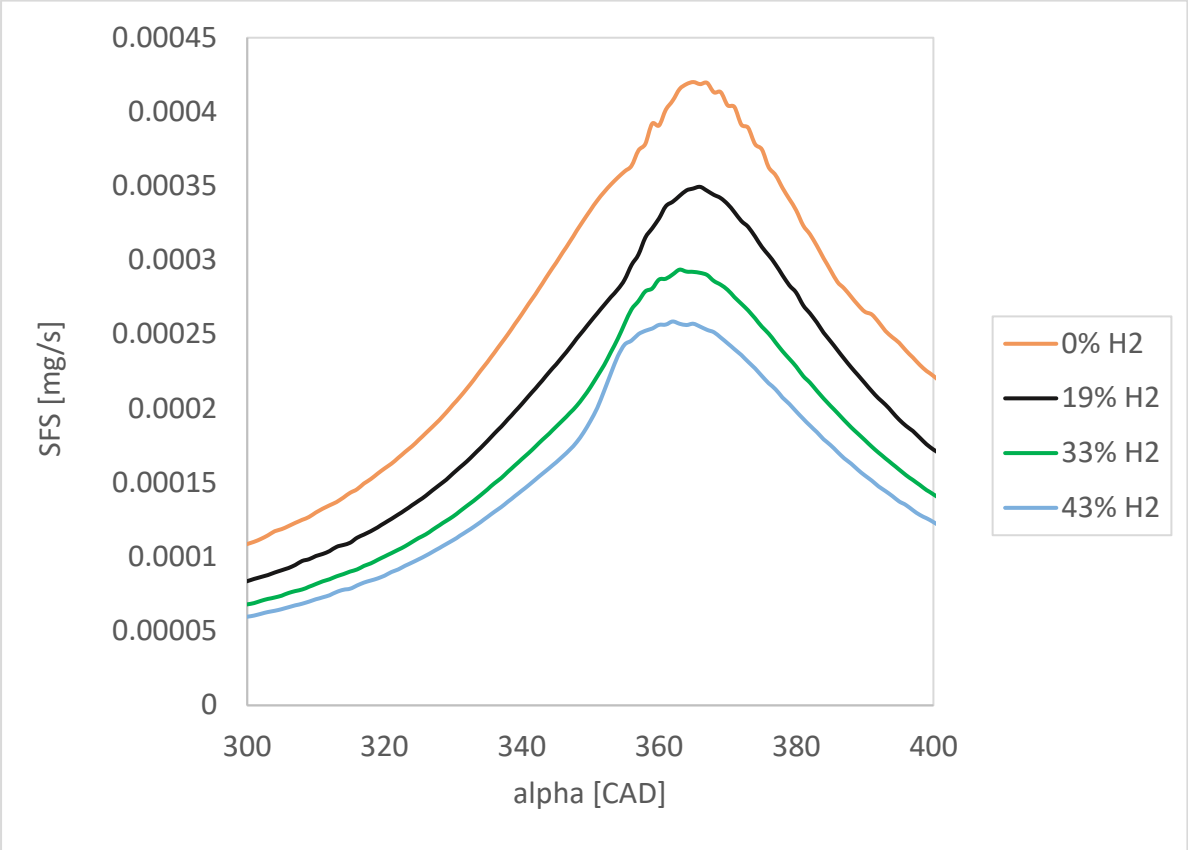


Figure 18. The soot formation speed at different energetic substitution degree.

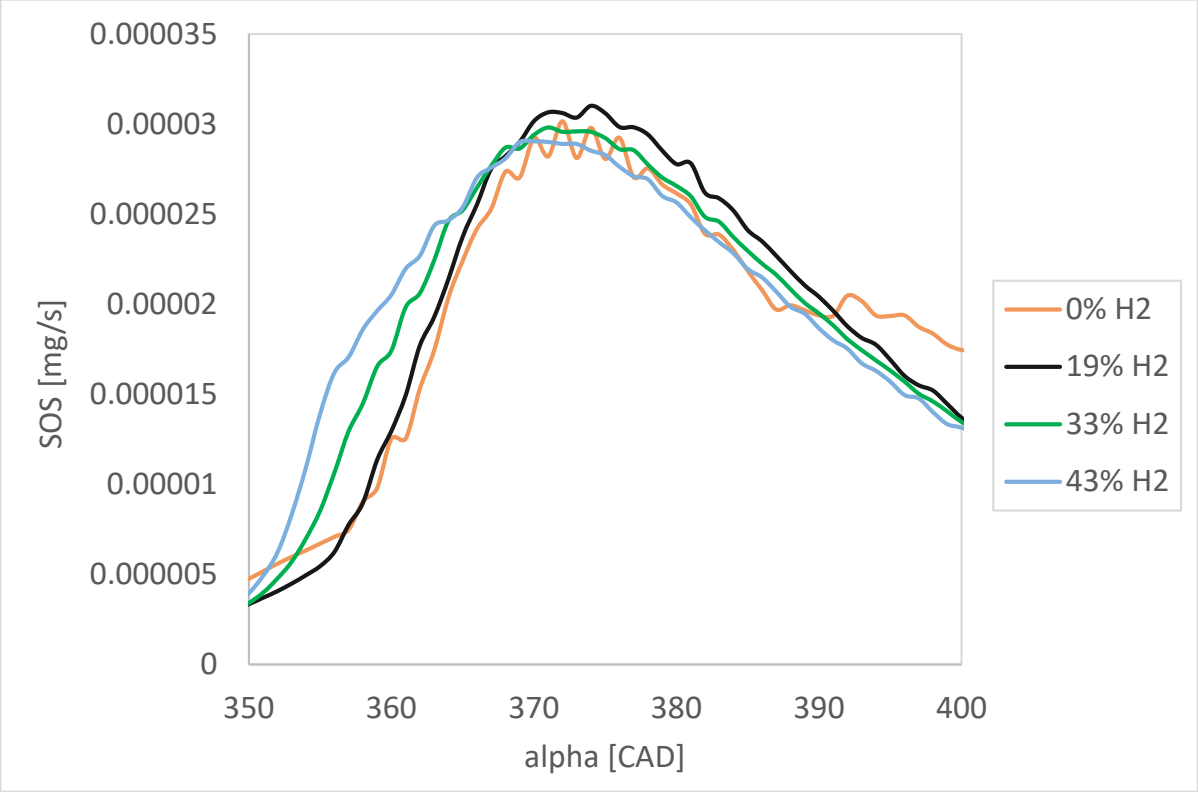


Figure 19. The soot oxidation speed at different energetic substitution degree.

For the analysis of the soot oxidation process, the oxidation speeds were integrated to obtain combustion laws. The global mass combustion laws thus obtained, expressed in relative values, the reference being given by the diesel fuel use, 0% ESD, provide the maximum values that are used to achieve the smoke emission balance. At the substitution of the diesel fuel with hydrogen, the acceleration of soot oxidation is noted, in conditions of lower amounts of formed soot. At H_2 use, all the soot oxidation speeds are superior to diesel fuel, especially at the beginning of the soot oxidation process. At higher percentages of hydrogen, 43% ESD, the soot oxidation speed is more reduced, probably as a result of the reduction of the oxygen content. This aspect can be taken into account in order to limit the cyclic dose of hydrogen, in order to avoid a possible increase in smoke emission. The trend of variation of the soot oxidation speed, SOS, correlates with the trend of acceleration of the heat release rate, HRR, Figure 3. The combustion of air-H_2 mixtures favours an earlier oxidation of the soot and higher peaks in oxidation speed. The acceleration of the soot oxidation speed also correlates with the trend of reduction of the autoignition delay duration.

The performed soot balance, regarding the formation and oxidation of soot in the engine cylinder, at diesel fuel and diesel fuel-hydrogen fuelling, allows the determination of the final, relative, amount of smoke in the exhaust gases, for each energetic substitution degree, ESD, as is shown in Figure 20.

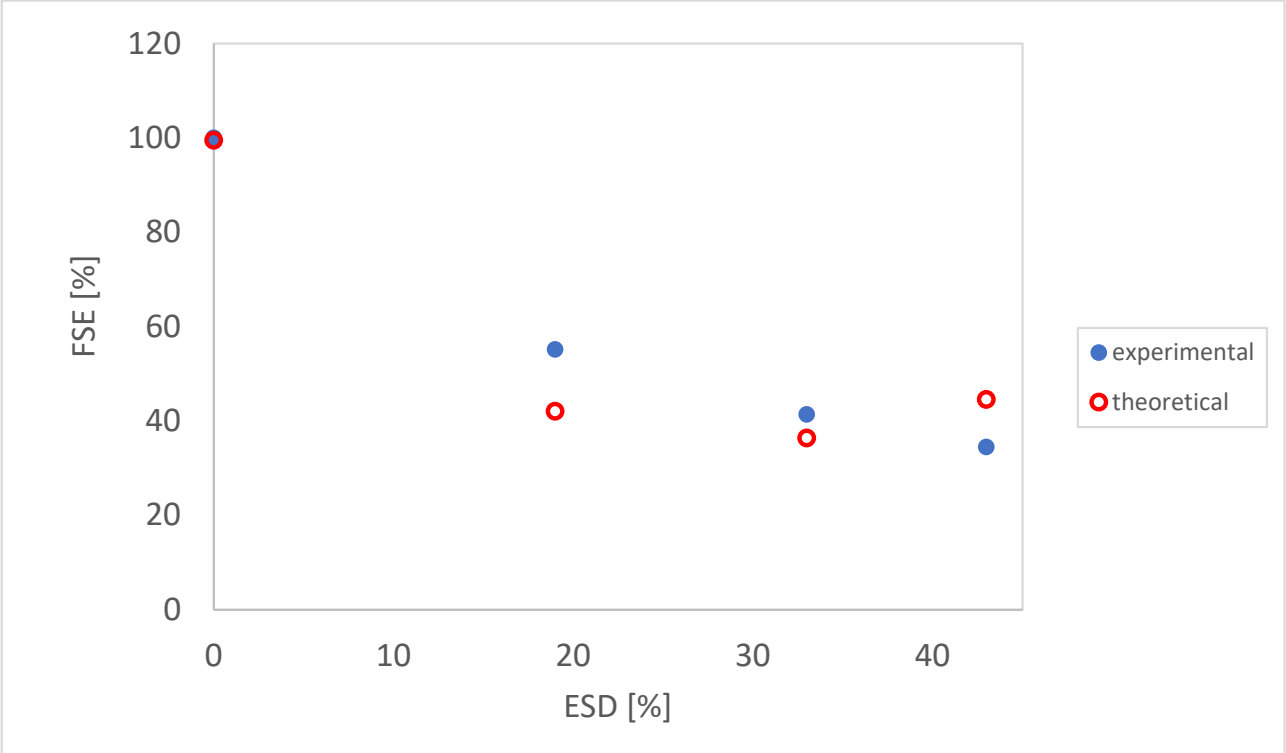


Figure 20. The final soot emission at different energetic substitution degree.

Figure 20 shows the variation of the relative final soot emission (FSE) in the exhaust gases for each energetic substitution degree, determined theoretically and compared with the experimental data. The dispersion between the theoretically and experimentally values is relatively small. The study complements the experimental values from the point of view of validating the smoke reduction trend, with the increase of the H_2 cyclic dose. Thus, by the speed of formation and oxidation of soot, it is possible to explain, depending on the variation of the pressure and temperature regime, the continuous reduction of the smoke emission level at hydrogen fuelling.

4. Conclusions

The substitution of diesel fuel with hydrogen, in 19–43% energy substitution degree, at the operating regime of 2500 rev/min and 50% load leads to the main influences:

At the increase of the hydrogen quantity, the maximum pressure increases with ~19%, from 103.75 bar until 123.66 bar, due to reduction of the combustion duration at hydrogen use comparative to classic fuelling. For 19%ESD the average maximum pressure increases with 11.35% and for 33%ESD the average maximum pressure increases with 17.2% versus diesel fuel use, 0%ESD. At diesel fuel and H_2 fuelling, the maximum pressure increases and at 43%ESD, in some individual cycles, peak values of 126.54 bar and 128.56 bar, respectively, are reached; these values do not exceed the permissible range that ensures the normal operation of the diesel engine. At hydrogen use as 19% and 33%ESD, the peak value of the heat release rate increases by 12.5%, reaching 1 CAD faster per cycle, which shows an acceleration of the combustion process. For 43%ESD the peak of HRR increases with 11% comparative to classic fuelling. Thus, the indicated mean effective pressure, IMEP, increases with 4.95% at 19%ESD, with 7.5% at 33% ESD and with 4% at 43%ESD comparative to diesel fuel use, 0%ESD. The value of the maximum pressure depends on the duration of the combustion and its placement in relation to TDC, the rate of heat release and the duration of the initial phase. Thus, the reduction of the duration of the initial phase, the increase of HRR and the approach of the maximum pressure angle to TDC are correlated with the tendency of maximum pressure increase. Moreover, these combustion parameters depend on the dosage, lambda varying from $\lambda = 1.9$ at 0%ESD, to 2.13 at 19%ESD, 2.19 at 33%ESD and 2.2% at 43%ESD. The dependence of these parameters on the specified factors is complex and involves the variation of the pressure, the MPRR, the maximum pressure angle and the IMEP. The maximum pressure and the angle at which it occurs per cycle depend on the variations of the HRR and the positioning of the rapid combustion phase around TDC.

The specific energy consumption, SEC, decreases with 2.94% at 19%ESD, with 5.36% at 33%ESD and with 3.73% for 43%ESD. For dual fuelling the engine efficiency increases, especially for the range of 19–33%.

Regarding the influence on the level of pollutant emissions, the HC emission level is reduced by 10% at 19%ESD, by 18% at 33%ESD and by 22% at 43%ESD due to higher speed of combustion of air-hydrogen homogeneous mixtures, 250 cm/s for hydrogen and 116 cm/s for diesel fuel, at $\lambda = 1.5$. The smoke opacity is reduced by 44.8% at 19%ESD, by 58.6% at 33%ESD and by 65.5% at 43%ESD comparative to diesel fuel fuelling case, due to acceleration of soot oxidation under conditions where the initial amount of soot is lower at H_2 use. Reduction of the diesel fuel amount at hydrogen use leads to the reduction of the HC and soot emission. The NO emission level increases with 18.5% at 19%ESD, by 33.3% at 33%ESD and by 49.3% at 43%ESD comparative to classic fuelling, 0%ESD, because of the combustion temperature increase. The increase of the coefficient of air-excess from 1.9 to 2.2 at H_2 use, influence the increase of the NO emission level. Further investigations require the use of AdBlue technology or an increased rate of EGR.

Regarding the influence on the GHG level, the CO_2 emission level is reduced by 28.6% at 19%ESD, by 34.5% at 33%ESD and 43%ESD due to reduction of the carbon content to 69.4% C at 19%ESD, to 57.4% C at 33%ESD and to 48.8% C at 43%ESD, comparative to 85.7% C at 0%ESD.

In generally the combustion variability is reduced by hydrogen use. The $(COV)_{Pmax}$ is 0.73% for classic fuelling and is reduced to 0.53% for 19%ESD, to 0.44% for 33%ESD and to 0.68% at 43%ESD. The $(COV)_{AMP}$ is reduced for 19–33% ESD, but increased for 43%ESD. The COV for the maximum pressure rise rate, $(COV)_{MPRR}$, decreases from 14.57% (at 0%ESD) to 11.63% (at 19%ESD), 6.97% (at 33%ESD) and 11.02% (at 43%ESD). The COV

of IMEP decreases from 1.13% to 0.68% for 19%ESD, 0.67% for 33%ESD and 1.07% for 43%ESD. For 43%ESD, the establishment of ultra-lean mixtures inside the cylinder before the combustion starts can leads to the increase of the $(COV)_{IMEP}$ value. The decreasing trend of the $(COV)_{pmax}$ and $(COV)_{IMEP}$, shows that the engine fuelled with diesel fuel and hydrogen can operate properly, even with lower cyclic variability for the substitution degrees of 19%ESD and 33%ESD.

At hydrogen use, the combustion duration decreases till 9.39 CAD at 43%ESD, due to higher combustion speed of the air-hydrogen mixture. The reduction in the combustion duration is related with the increase of the maximum pressure, reduction of the HC and smoke levels and reduction of the cyclic variability. At the same time, the duration of the autoignition delay is reduced, the reduction being correlated with the acceleration of processes related to cold and blue flames. The multi-stage mechanism of autoignition is accelerated in the presence of air-hydrogen mixtures that burn near by the droplets that are subjected to vaporization, mixing and autoignition processes.

If we look at the maximum value of the cyclic dose of hydrogen used for fuelling, then the engine's performance is influenced differently:

- if the hydrogen percent is limited at 33%ESD, then the maximum pressure is limited at average value of 121 bar, the IMEP at 0.456 MPa. In this case, at hydrogen fuelling, the HC decreases with 18%, the smoke opacity decreases with 58.6% and the CO_2 decreases with 34.52% versus diesel fuel fuelling. These exhaust emissions reduction are ensured under the conditions of reducing the specific energy consumption with 5.36% comparative to classic fuelling case. For this limitation at 33%ESD, the $(COV)_{Pmax}$ is limited at 0.44% and the $(COV)_{IMEP}$ at 0.67%;
- if the hydrogen share is limited at 43%ESD, then the maximum pressure is limited at 125 bar, the IMEP at 0.456 MPa. In this limitation case, at hydrogen fuelling, the HC decreases with 22%, the smoke opacity decreases with 65.5% and the CO_2 decreases with 34.52% versus diesel fuel use. These exhaust emissions reduction are ensured under the conditions of reducing the specific energy consumption with 3.73% comparative to classic fuelling case. For this limitation at 43%ESD, the $(COV)_{Pmax}$ is limited at 0.68% and the $(COV)_{IMEP}$ at 1.07%.

At the substitution of the diesel fuel with hydrogen, soot formation speed is reduced, due to the reduction of the diesel fuel amount per cycle. The combustion of air-hydrogen mixtures favours the acceleration of soot oxidation. The soot oxidation speed is also increased due to the increase of in-cylinder pressure and temperature. The combined effect between the decrease of the soot formation speed and the increase of the soot oxidation speed may explain the continuous decrease in smoke emission from exhaust gases, evaluated by smoke opacity.

These results were obtained for optimal correlation between 19 kW engine load–2500 rev/min engine speed–minimum 2.7 kg/h diesel fuel flow-maximum 0.68 kg/h hydrogen flow–130 bar maximum combustion pressure–10% minimum smoke emission level–350 °C exhaust temperature–43% ESD limit, in order to limit the increase in NO emission levels.

Hydrogen can be a viable alternative fuel for the diesel engine and its use can be done without major modifications of the engine design, offering the first steps for a sustainable solution of transportation in the future.

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A.C., C.P., N.N., L.N. and C.N.; writing—original draft preparation, A.P., A.C., C.P., N.N. and C.N.; writing—review and editing, A.P., A.C., C.P., N.N., C.N. and L.N.; visualization, A.C., C.P., N.N., D.F., C.N., L.N. and A.P.; supervision, C.P., N.N., C.N. and A.C.; project administration, C.P., N.N., A.C. and C.N.; funding acquisition, C.P., N.N., A.C., D.F., L.N. and C.N. All authors have read and agreed to the published version of the manuscript.

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Abbreviations

AVL	Anstalt für Verbrennungskraftmaschinen List automotive research institute Graz Austria
AMP	Angle Of Maximum Pressure
BTE	Brake Thermal Efficiency
BTDC	Before Top Dead Centre
BSFC	Brake Specific Fuel Consumption
C	Carbon
CAD	Crank Angle Degree
CD	Combustion Duration
CI Engine	Compression Ignition Engine
CH_4	Methane
C_16H_{34}	Diesel Fuel (Or Cetane)
CNG	Compressed Natural Gas
CO	Carbon Monoxide
CO_2	Carbon Dioxide
COV	Coefficient Of Variability
CP	Combustion Parameter
DI	Direct Injection
ECU	Engine Control Unit
EGR	Exhaust Gas Recirculation
ESD	Energy Substitution Degree
FSE	Final Soot Emission
GHG	Greenhouse Gases
H_2	Hydrogen
HC	Unburned Hydrocarbons
HICE	Hydrogen Internal Combustion Engine
HRR	Heat Release Rate
hythane	Hydrogen and Methane

ICE	Internal Combustion Engine
IC Engine	Ignition Compression Engine (Or Diesel Engine)
IMEP	Indicate Mean Effective Pressure
K	Kelvin
kW	Kilo Watts
kg/h	Kilograms Per Hour
L	Litre
LHV	Lower Heating Value
lpm	Litre Per Minute
MDS	Modular Diagnostic System
MFB	Mass Fraction Burnt
MTB	Maximum Torque Brake
MPa	Mega Pascal
mJ	miliJoule
m/s	Meter/Second
MPRR	Maximum Pressure Rise Rate
mm	Millimetre
m^3/h	Cubic Meters Per Hourly
Nm	Newton Metre
NO_x	Nitrogen Oxides (Or NO, N_2O)
PFI	Port Fuel Injection
p_{max}	Maximum Pressure
ppm	Parts Per Millions
rev/min	Revolutions Per Minute (Or rpm, RPM)
SIE	Spark Ignition Engine (Or SI Engine)
SCR	Selective Catalytic Reduction
SFS	Soot Forming Speed
Slpm	Standard Litre Per Minute
SOS	Soot Oxidation Speed
SO_x	Sulfur Oxide
TDC	Top Dead Centre
T	In Cylinder Pressure
λ	Coefficient of Excess Air
λi	Inferior Limit of Inflammability
λs	Superior Limit of Inflammability
μm	Microns

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