EFFICIENCY AND CO_2 EMISSION OF HEAT ENGINES OPERATING WITH HYDROGEN RICH GAS (HRG) ADDITION

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Abstract

The effects of gas addition on efficiency and CO_2 emission were investigated with a compression ignition (CI) engine and a spark ignition (SI) engine operating with hydrocarbons-based fuels. Hydrogen rich gas (HRG) provided by an electrolyze, was aspirated into the air stream inducted in the engine cylinders. Investigation was conducted at engine light and medium loads and speeds, with relative low concentrations of HRG. For the CI engine, HRG addition up to 12.5% energetic fraction in fuel lowered CO_2 concentration (ppm) in the engine exhaust up to 8% and BTE up to 1%. CO_2 specific emission (g/kWh) was correspondingly lower with maximum 10%. For the SI engine fueled with gasoline, the effect of HRG addition was depending not only on engine load and speed, but also on the relative air-fuel ratio λ. It was found for CO_2 concentration a maximum decrease of 5% by HRG addition. BTE was improved in a limited domain of HRG fraction up to 20%, with maximum improvements between 2.5% and 21%. As a cumulative effect on CO_2 concentration an on BTE a maximum lowering by 30% of the CO_2 specific emission was reached. The effect of HRG supplementing LPG gas in the same SI engine was alternatively studied with similar results: CO_2 concentration reductions by maximum 3%, some possible improvements of BTE up to 33% at HRG fraction of 19% and finally a lowering of CO_2 specific emission by maximum 37%.

Key words: HRG, diesel engine, SI engine, CO_2 emission, BTE

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1. Introduction

Carbon dioxide (CO_2) is the most damaging greenhouse agent, its influence being evaluated by a warming effect of 1.7 W/m^2 (sq.m. of the Earth's surface). Once generated, CO_2 tends to remain in the atmosphere for a long time.

CO_2 is the inevitable by-product of fossil fuels burning. The abatement of CO_2 emission in the atmosphere is possible in these terms by improvement of the fuel burning efficiency. Decreasing the specific CO_2 emission of internal combustion engines (in gCO_2/kWh) is thus possible by the improvement of engine efficiency, expressed by the brake specific fuel consumption (g fuel/kWh); the vehicle specific CO_2 emission, expressed by $gCO_2/100km$ is similarly related to the vehicle specific fuel consumption (in L/100km). The European automakers have the task to reach the limit of 95 $gCO_2/100km$ in 2020; similarly, in USA the automakers are facing the task to reach a fuel average efficiency of 54.5 miles per gallon (mpg) (4.6 L/100km), in 2025.

Promotion of the low-carbon fuels is the other possible way to abate CO_2 emissions. Hydrogen, in the absence of carbon, appears as an alternative ideal fuel, producing by combustion only water. Hydrogen even by some other combustion characteristics seems to be an attractive fuel for spark ignition engines and a few prototype vehicles, operating with pure hydrogen, have been produced in the last decade. Due to

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economic technical reasons, vehicles fueled with pure hydrogen are nonetheless considered a challenge for the next future.

The use of hydrogen as an additive to the hydrocarbons – based fuels appears as an intermediate solution, which has been studied in numerous conditions and with varying results. It was found that the hydrogen addition can favorably influence the combustion process and the engine emissions under certain conditions (Bade Shresthva and Karim, 1999; Cracknell et al., 2002; Gerini et al., 1996; Haragopela and Bhakta, 1983; Ivanic et al., 2005; Kumar, 2003; Lilik et al., 2010; Ma et al., 2003). CO_2 emission reduction would depend on the amount of the hydrocarbons – based fuel substituted by hydrogen and on the engine brake efficiency.

Saravanan et al. (2007) have studied hydrogen addition to petroleum diesel on a single – cylinder, naturally aspirated, stationary, direct injection (D.I.) diesel engine with a rated power of 3.76 kW at 1500 rpm. Hydrogen was injected into the inlet port during the engine intake stroke. With optimized hydrogen injection duration and start of injection, the brake thermal efficiency at full load, with 12.9% (fuel energy) hydrogen was 29.4%, comparted to 23.6% for pure petroleum diesel (an increase of 14.6%). The CO_2 concentration in the exhaust gases decreased correspondingly by 79 %, from 11.6% to 2.4%. In another study of Saravanan et al. (2008) using the same engine and with 9.7% hydrogen addition by injection into the inlet port, the brake thermal efficiency at full load was 24.7%, with an increase of 4.7% compared to pure diesel fuel operation. The CO_2 emission decreased correspondingly by 9.6% from 0.727 g/kWh to 0.657 g/kWh.

Xiao et al. (2009) have investigated the effects of hydrogen addition on performance and emissions of an indirect injection diesel engine with a rated power of 5 kW at 1600 rpm. Hydrogen was introduced in the intake manifold using a mixing chamber for hydrogen and air. It was found that for the same total fuel energy input, the engine thermal efficiency for the dual fuel engine operating was slightly lower than for pure diesel operating. The relative difference in CO_2 concentration was increasing almost proportionally with the hydrogen concentration: at full load, with 42.5% (fuel energy) approximately hydrogen, CO_2 concentration was about 3.8%, compared to 7.2% with pure diesel fuel operation (a 47% decrease); at 75% load, with about 34% (fuel energy) hydrogen, the CO_2 concentration was lower by 34%.

Liew et al. (2010) have experimented hydrogen addition by aspiration on a heavy-duty diesel engine, with cooled EGR, rated at 265 kW for 1800 rpm. Tests according the 13-mode ESC cycle were carried out and for each mode the effects of the addition of 2% and 4% (vol. in air) hydrogen were measured. It was found that the addition of 2% decreased the integrated CO_2 specific emission from 639.8g/kWh for pure diesel operation to 579.6g/kWh (9.4% decrease) and the addition of 4% hydrogen decreased CO_2 concentration to 483.2 g/kWh (24% decrease).

The strategy of petroleum-based fuel partial substitution by hydrogen has been investigated also for spark ignition (SI) engines.

In a study performed by Conte and Boulouchos (2004) a blend of 21% H_2, 24% CO, 55% N_2 (vol.) was used. This simulates the output of an on-board reformer that partially oxidizes gasoline to provide a hydrogen rich gas. The experiments were carried out on a stationary SI engine with a rated power of 13.5 kW at 5000 rpm. Gasoline and the hydrogen-rich gas were introduced in the intake port by two injectors. The engine operation at 2000 rpm and 2 bar BMEP, with 21% (vol.) reformer gas resulted in an increase of brake thermal efficiency by 3%, with a decrease of the CO_2 specific emission by 4%, from 1250 g/kWh to 1200 g/kWh. Choi et al. (2005) have studied the effects of hydrogen enriched liquefied petroleum gas (LPG) on engine performance and emissions. Experiments were conducted on a single cylinder, 1.8 L, variable compression ratio SI engine. Operation at 1400 rpm and full load has shown that in the lean mixture zone, there is no difference on thermal efficiency with the addition of hydrogen. The CO_2 concentration in the exhaust gases decreases for lean mixtures and by hydrogen addition: for a relative fuel-air $\lambda = 1.1$. CO_2 concentration with pure LPG was 13%, and decreased to 12% with 10% (vol.) hydrogen addition (CO_2 decrease by 15%) and decreased further to 11% with 20% hydrogen addition (CO_2 decrease by 30%).

These works are generally focused on the effects of relatively high concentration of hydrogen. Investigations conducted with smaller amounts of hydrogen added to the main fuel are also of interest for a vehicle engine; this would correspond to the hydrogen output of an onboard water electrolyze or gasoline reformer.

The present contribution describes the results of an investigation on the effects of using HRG (Hydrogen Rich Gas) mixture generated by water electrolysis as additive to the base petroleum fuel. Besides the main components, molecular hydrogen and oxygen, the gas resulting from an advanced electrolysis process would possibly contain up to 5% other components, promoting combustion (Santilli, 2006). In the present work, HRG was considered as a stoichiometric gas mixture of only molecular hydrogen 66.5% and molecular oxygen 33.5%. Experiments were conducted on two engines, a tractor diesel engine and a car SI engine, with HRG gas aspirated with the air in the intake manifold. The effects on efficiency and CO_2 emission were emphasized.

2. Experimental details

The tested diesel engine was naturally aspirated direct injection, conventional tractor engine, with 4 cylinders having a total displaced volume 3759 cm^3 and a rated power of 50 kW at 2400 rpm. The schematic of the experimental set up is shown in Fig. 1 (Birtas et al., 2011).

HRG gas was delivered by an electrolyzer produced by ROKURA with a maximum output of 30 l/min. The electrolyzer (GG) was equipped with instrumentation for the measurement of power consumption, state parameters of the gas and the gas flow rate delivered to the engine. The induction system connecting the electrolyzer to the engine intake was also provided with safety equipment and an electronic precision mass flow meter (MFM) to measure the gas HRG flow rate.

Concerning the engine operating parameters, the instant torque and speed were controlled and measured by an eddy current dynamometer (Dyno), type AVL Alpha 160. Engine air consumption was measured by a volumetric flow meter (AFM) type SCHLUMBERGER FLUXI 2000/TZ. The diesel fuel consumption was measured with a dynamic fuel meter (DFM) – AVL type 733 S. The exhaust emissions (CO_2, CO, NO_x, THC) were measured by an exhaust gas analyzer (GA) type HORIBA MEXA 7170 D, which calculates also the relative air-fuel ratio λ. All the global engine operation parameters were measured with adequate devices and registered by the equipment (DACS) type AVL Puma Open v.1.4.

Pressure traces have been recorded for two cylinders by an AVL data acquisition system, with quartz pressure transducers (PT) type AVL GM 12 D, crank angle encoder (CAE) type AVL 365 C and data acquisition module (DAM) type AVL Indiset 620 provided with the AVL Indicom 1.6 software for data collection and processing. For each operating point during the test program have been registered 200 consecutive cycles. Test operation conditions were in the part-load domain (40% and 60%) and two engine speeds: 1400 rpm (maximum torque speed) and 2400 rpm (nominal power speed). Fuel injection timings were kept the original ones. The spark ignition engine was a conventional carbureted car engine with 4 cylinders with 1397 cm^3 total displaced volume and a rated power of 40 kW at 5500 rpm.

The experimental test bed was correspondingly equipped for the control of the engine operation and the collection of all the relevant parameters characterizing the electrolyzer and the engine (Niculae et al., 2013). The original engine fueling system with gasoline was supplemented with two additional systems, for LPG and HRG gas: LPG gas was introduced in the original carburetor, adequately modified, HRG gas was introduced upstream the carburetor (Fig. 2). The systems for gaseous fuels were provided with the necessary equipment's selected to ensure safety operation. In this sense, the HRG gas system was provided with three flame arrestors (F_1, F_2, F_3), with one-way valve (UV) and electro-valve (EV). The LPG line was provided with a vaporizer (V), manual valve (MV) and electro valve (EV) and rotameters (RT) for gas flow measurement. The engine was coupled with a Hoffman eddy current dynamometer for the control and the measurement of the engine torque and speed.
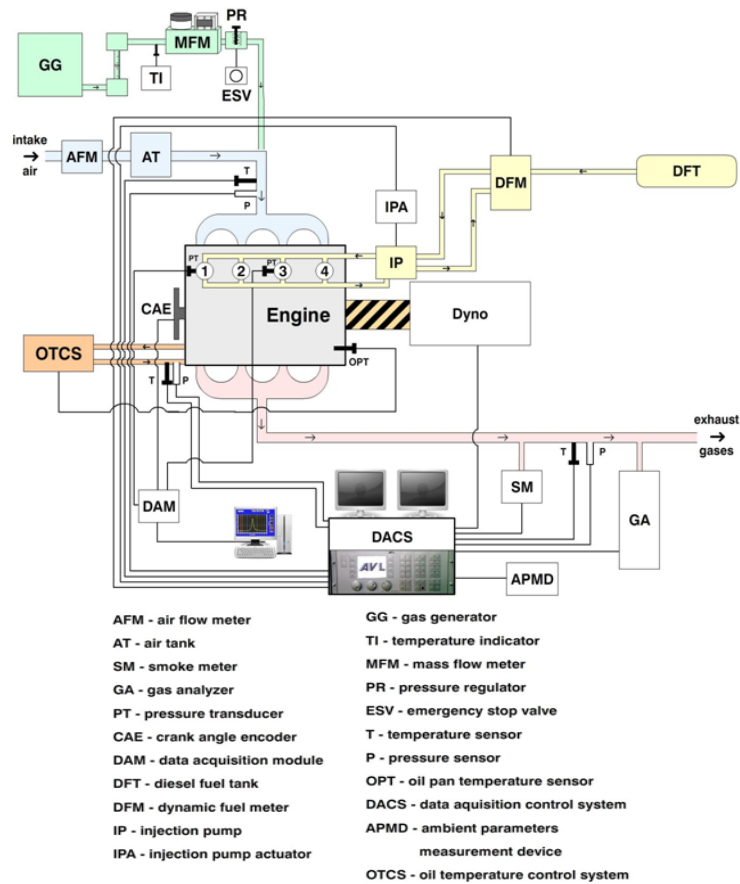

AFM - air flow meter
AT - air tank
SM - smoke meter
GA - gas analyzer
PT - pressure transducer
CAE - crank angle encoder
DAM - data acquisition module
DFT - diesel fuel tank
DFM - dynamic fuel meter
IP - injection pump
IPA - injection pump actuator

GG - gas generator
TI - temperature indicator
MFM - mass flow meter
PR - pressure regulator
ESV - emergency stop valve
T - temperature sensor
P - pressure sensor
OPT - oil pan temperature sensor
DACS - data aquisition control system
APMD - ambient parameters measurement device
OTCS - oil temperature control system

Fig. 1 Schematic of the diesel engine test bed

The exhaust emissions (CO_2, CO, NO_x, THC) were measured and the relative air/fuel ratio, λ was calculated by a gas analyzer AVL Di Gas 4000. A data acquisition system with pressure transducer Kistler 601 A, charge amplifier Kistler 5001 C, data acquisition card AVL Indimeter 617 and crank angle encoder AVL 364 was used for measurement and acquisition of in-cylinder pressure. For each tested operation point, pressure traces for 500 consecutive cycles were registered. The operation conditions were light load, 1600 rpm considered as representative for the city traffic and medium load, 2500 rpm, as representative for the inter-city traffic. The load was identified by the pressure existing in the inlet manifold: $p_{ca} = 540$ mbar for light load and $p_{ca} = 370$ mbar for medium load. Final data comparisons concerning the effect of HRG addition at various λ and engine speed and load, were made for optimized ignition timings (minimum for best torque, MBT). Brake thermal efficiency for the engine operation with two fuels was calculated with Eq. (1):

$$BTE = \frac{P_{ec} \cdot 3600}{C_{MF} \cdot LHV_{MF} + C_{AF} \cdot LHV_{AF}} \cdot 100 \quad (1)$$

where P_{ec} – corrected brake power, kW; C_{MF}, C_{AF} – mass consumption of main fuel (diesel fuel, LPG, gasoline and additional fuel (HRG), kg/h; LHV_{MF},

LHV_{AF} – lower heating value for the main and additional fuel, respectively, kJ/kg. The addition of HRG was quantified by percentage of the total fuel energy. CO_2 specific emissions expressed in gCO_2/kWh were analyzed in terms of CO_2 concentrations in the exhaust (ppm), as a direct result of the fuel composition and combustion process, and the engine BTE.

3. Experimental results and discussion

3.1. Diesel engine operating with addition of HRG

Carbon dioxide concentration (ppm) in the exhaust decreases at constant relative air-fuel ratio by enrichment with HRG, as a result of the partial substitution of carbon in the main fuel by hydrogen. The decrease was more important at the lower speed of 1400 rpm, where higher HRG concentrations were reached for the same gas output of the electrolyzer. At 40% load, CO_2 was lower by 1.8 % with 3.4 % (fuel energy) HRG, by 5.3 % lower with 7.8% HRG and by 8% with 12.5% HRG, compared with 44800 ppm CO_2 for pure commercial diesel fuel. At 60% load, a generally higher CO_2 concentration, corresponding to lower excess of air, was also decreasing by addition of HRG: by 3% lower with 5.6 % HRG and by 5% with 9.4% HRG, compared with 62000 ppm CO_2 for pure diesel fuel (Fig. 3a).
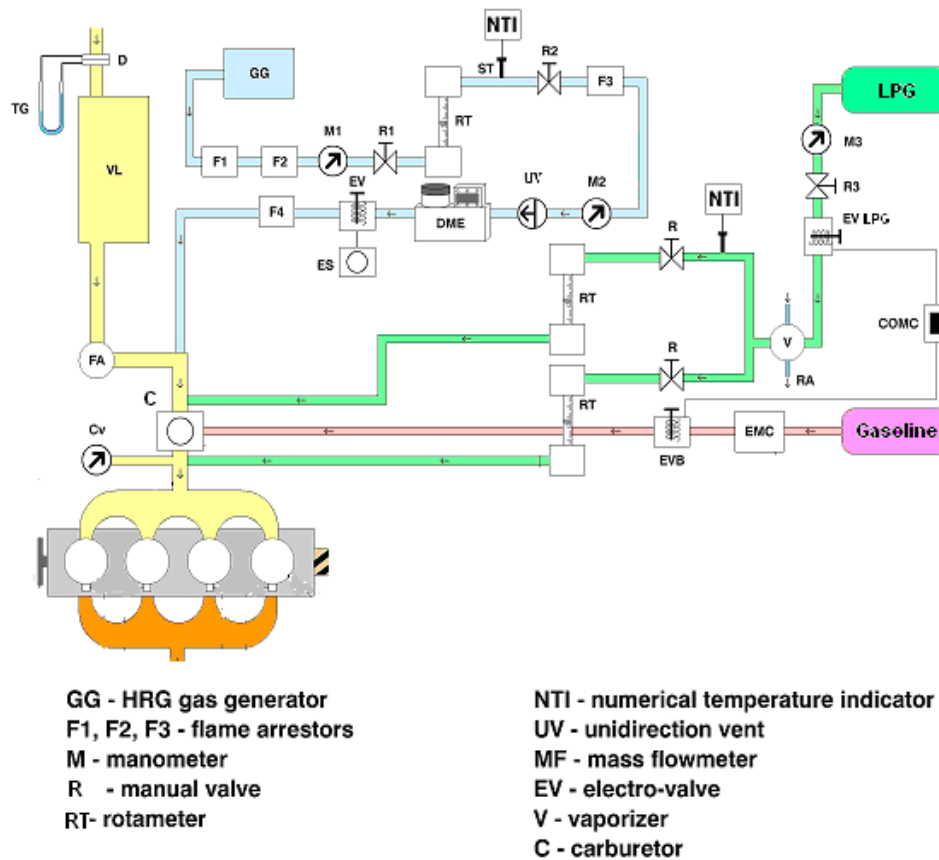

GG - HRG gas generator
F1, F2, F3 - flame arrestors
M - manometer
R   - manual valve
RT- rotameter

NTI - numerical temperature indicator
UV - unidirection vent
MF - mass flowmeter
EV - electro-valve
V - vaporizer
C - carburetor

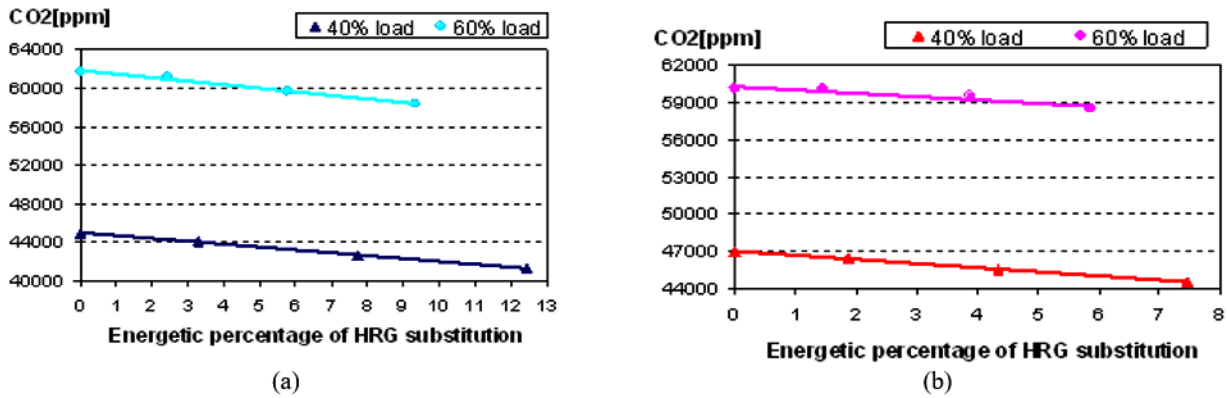**Fig. 2.** Schematic of the fueling system with LPG and HRG

Fig. 3 Variation of CO$_2$ concentration with HRG (fuel energy fraction) substituting diesel fuel: (a) 1400 rpm; (b) 2400 rpm

A similar effect was found at 2400 rpm. For 40% load, CO$_2$ concentration decreased by 3% with 4.3% HRG and by 5% with 7.3% HRG, compared with 47000 ppm CO$_2$ for pure diesel fuel; at 60% load, CO$_2$ reduction by 1.9% with 3.7% HRG was found which compares with 60400 ppm for pure diesel fuel (Fig. 3b).

The brake thermal efficiency BTE decreases slightly by the addition of HRG. At 1400 rpm and 40 % load, BTE decreases from 32.1% with pure diesel fuel to 31.8% with 7.8% HRG (decrease by 0.9%); at 60% load, BTE decreases from 35.8% with pure diesel fuel to 35.1% (by 1%) for 9.4% HRG (Fig. 4.a). At 2400 rpm, the effect of diesel fuel substitution by HRG, on BTE is similar but smaller: at 40% load, BTE decreases from 25.4% to 25.09% (1% decrease) (Fig. 4b). Differences on BTE are commonly correlated with combustion process alterations. The combustion characteristics corresponding to HRG addition in various fractions were compared to pure diesel fuel condition by the heat release characteristics resulting from cylinder pressure data (Fig. 5). The characteristics show same slight alterations by HRG addition.

The start of combustion, α_i usually considered as the moment when the rate of heat release becomes positive was insignificantly affected by the presence of HRG, with a maximum difference of 0.8°CA found at 1400 rpm and 60% load. The duration of the premixed phase of combustion conventionally identified with the release of the first 5% heat release ($\Delta\alpha 5\%$) appears also unaffected by HRG addition. This lack of influence of HRG can be associated with the lack of hydrogen influence on the low-temperature mechanism of hydrocarbons self-ignition and thus on the fuel ignition delay. As it is known, the autoignition temperature of diesel fuel is lower compared with the autoignition temperature of hydrogen. On the other hand, hydrogen from HRG makes a homogeneous mixture with air before the diesel fuel self-ignition, having a concentration well below the lean limit of flammability (4% vol.). During the first phase of combustion participates thus only that fraction of hydrogen, originating from the volumes where the added diesel fuel prepared for combustion could generate mixtures within the flammability limits. The

insignificant influence of HRG addition on the first phase of combustion is thus possible.

Duration of the main (diffusion) phase of combustion, with heat release up to 90% ($\Delta\alpha 5$-90%) was slightly increased by HRG addition; by 3.6% at 1400 rpm, 60% load and 9.2% HRG and by 2.9% at 2400 rpm, 40% load and 7.5% HRG (Fig. 5). An increase of this phase was unexpected, since the presence of hydrogen should be favorable for the development of the diffusion combustion. A negative impact on the rate of mixing process could explain the increase of this phase duration. This slight increase of the combustion duration correlates anyway with the lowering of BTE.

A contribution to the lower BTE registered by HRG addition could be the effect of higher heat losses to the combustion chamber walls compared with commercial hydrocarbon fuels. The hydrogen-air mixtures have characteristically thinner quenching distances, promoting thus a higher heat transfer during combustion. (Karim, 2005; Toshio, 2007)

The calculated CO$_2$ specific emission, in g/kWh reflects the influence of its concentration in ppm and of BTE (Fig.6). At 1400 rpm and 40% load, CO$_2$ appears lower by 2.5% with 3.4% (fuel energy) HRG and by 10% with 12.5%, compared with 872 gCO$_2$/kWh with pure diesel fuel. At 60% load, CO$_2$ decreases by 4.3% with 5.6% HRG and by 7.8% with 9.4% HRG, compared with 800 gCO$_2$/kWh without HRG. At 2400 rpm and 40% load, the decrease was by 3% with 4.3% HRG and by 6% with 7.3% HRG from 1100 g/kWh without HRG. At 60% load a decrease by 4.6% was obtained with 5.8% HRG. The maximum HRG fractions which were tested correspond to the limit output of the electrolizer, of 30 L/min.

3.2. SI engine operating with gasoline and HRG

CO$_2$ concentration in the exhaust gases, for a given depression in the inlet manifold and engine speed, is variable with the relative air-fuel ratio λ and the concentration of HRG. At 540 mbar, 1600 rpm and rich mixture λ = 0.92, CO$_2$ concentration was 13% with pure gasoline and decreases with addition of HRG, by 2% for 4.8% HRG and by 4.6% for 11.3% HRG.
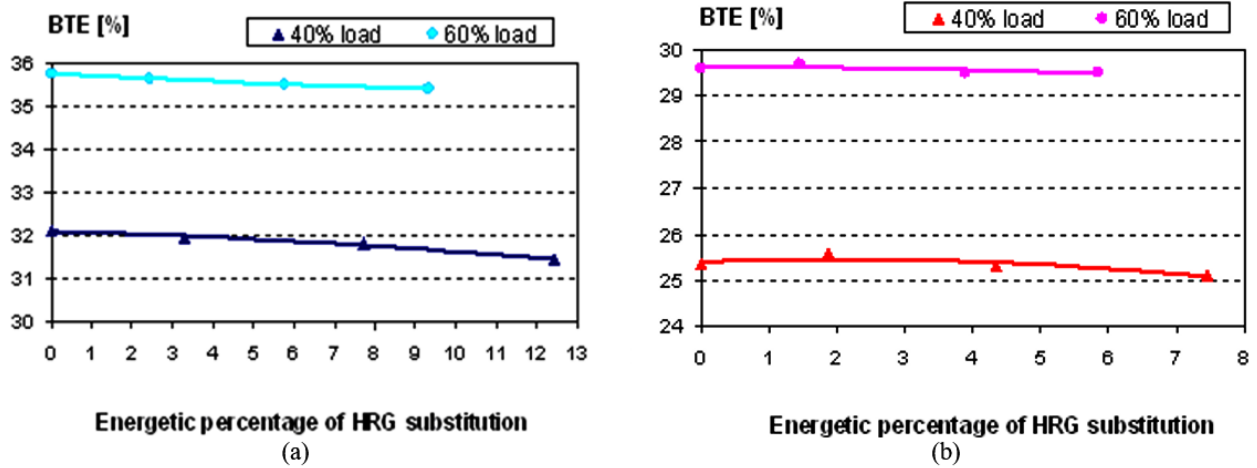
Fig. 4. Variation of BTE with HRG (fuel energy fraction) substituting diesel fuel: (a) 1400 rpm; (b) 2400 rpm
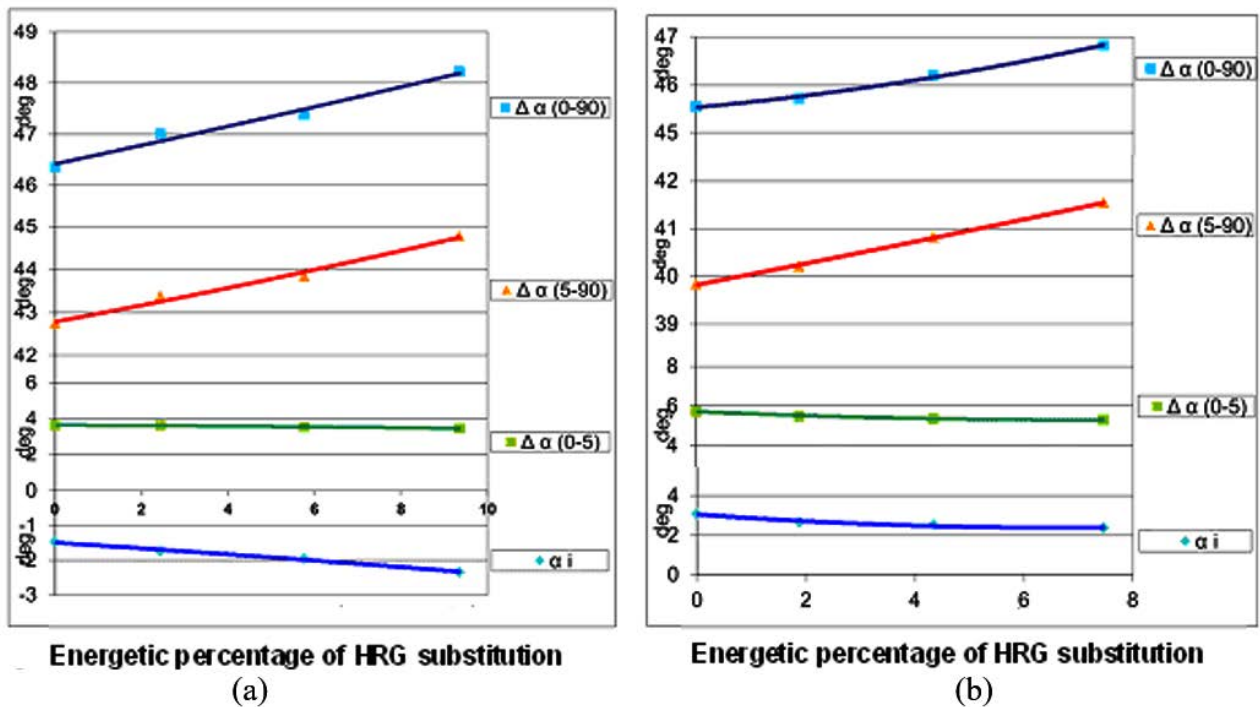

Fig. 5. Variation of the heat release characteristics with HRG (fuel energy fraction) substituting diesel fuel:
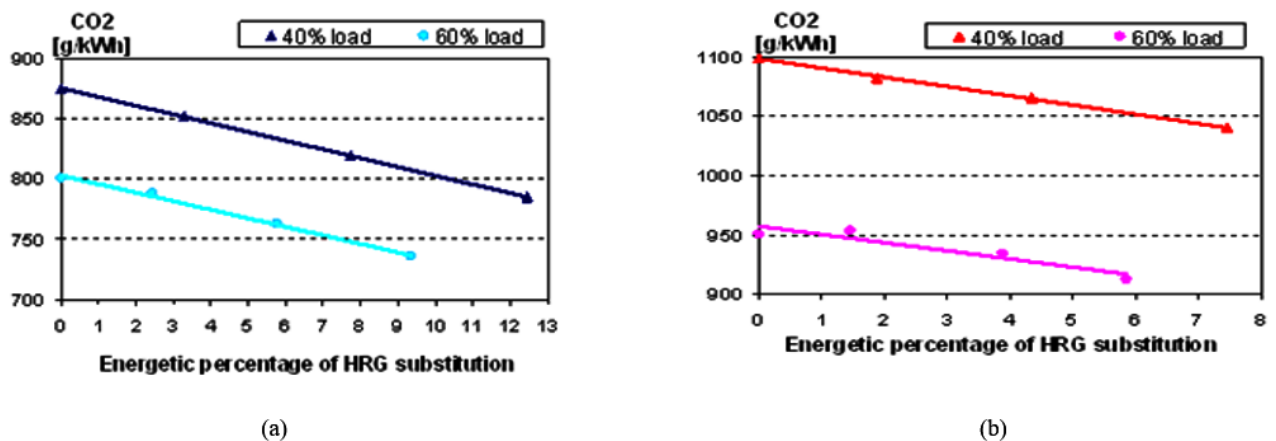(a) 1400 rpm, 60% load; (b) 2400 rpm, 40% load

Fig. 6. Variation of CO_2 specific emission (g/kWh) with HRG (fuel energy fraction) substituting diesel fuel:
(a) 1400 rpm; (b) 2400 rpm

With a slightly lean mixture $\lambda = 1.18$, CO_2 concentration is generally higher, correlating with a lower concentration of CO and unburned hydrocarbons emissions in the exhaust; the advanced oxidation of these species is stimulated by HRG addition. CO_2 concentration is lower by 0.8% for pure gasoline, compared with the rich mixture, decreases from 12.9%, without HRG, by 2% with 9.3% HRG, and by 4% with 16% HRG (Fig. 7a).

At 380 mbar and 2500 rpm, CO_2 concentration shows a smaller dependency on HRG: a decrease by 2% with 3.5% HRG, and an increase by 1.5% with 7.8% HRG at $\lambda = 1.06$; for a leaner mixture, $\lambda = 1.18$, the CO_2 variation becomes insignificant (Fig. 7b). BTE is improved in a limited domain of HRG concentrations. At 540 mbar, 1600 rpm and $\lambda = 0.92$, BTE peaks with an increase by 9% at 4.8% HRG and for $\lambda = 1.18$ peaks at 9% HRG, increasing by 20% compared with pure gasoline (Fig. 8a). At 380 mbar, 2500 rpm, BTE shows a similar, also slighter dependency on HRG: at $\lambda = 1.06$, BTE peaks at 3.5% HRG, with an increase by 3%. At $\lambda = 1.18$, a BTE peak is not apparent, BTE is higher by 2.5% at 5.3% HRG, and by 3.7% at 9% HRG (Fig. 8b). The beneficial effect of HRG addition on BTE correlates with a shortening of the initial phase (0-10%) and the main phase (10-90%) of heat release on the combustion characteristics as it is exemplified in Table 1. The partial substitution of gasoline by HRG, when the engine operation point is kept constant, has a supplemental effect on the engine power; the total heat value of the fuel increases for a higher HRG fraction. The engine power is lower at $\lambda = 1.18$ and CO_2 specific emission is accordingly higher at $\lambda = 1.18$ in comparison with $\lambda = 0.92$. As the result of all these effects, CO_2 emission, in g/kWh, decreases by HRG addition. At 540 mbar and 1600 rpm, CO_2 shows a minimum, roughly corresponding to BTE peak.

It was thus obtained, at the operating point of 540 mbar, 1600 rpm and $\lambda = 0.92$, from 1287 g/kWh for pure gasoline, a decrease by 22% with 4.8% HRG and by 31% with 11.3% HRG; with $\lambda = 1.18$, CO_2 emission was 1520 g/kWh for pure gasoline, and decreased by 21% with 20.3% HRG (Fig. 9a). Addition of HRG at 380 mbar, 2500 rpm, with $\lambda = 1.06$ decreased CO_2 specific emission from 670 gCO_2/kWh by 9% for 5.5% HRG and by 17% for 7.7%. For $\lambda = 1.18$, CO_2 emission decreased from 650 g/kWh by 7.7% with 9% HRG (Fig. 9b).

3.3. SI engine operating with LPG and HRG

The results obtained by the substitution of gasoline with liquefied petroleum gas are similar. CO_2 concentration is slightly influenced by the enrichment with HRG. At 540 mbar, 1600 rpm, $\lambda = 1.06$, CO_2 decreases by 3% with 15% HRG; at 380 mbar, 2500 rpm, $\lambda = 1.28$, CO_2 decreases by about 1% with 6.8% HRG (Fig. 10). CO_2 concentration for LPG is correspondingly lower than for gasoline.
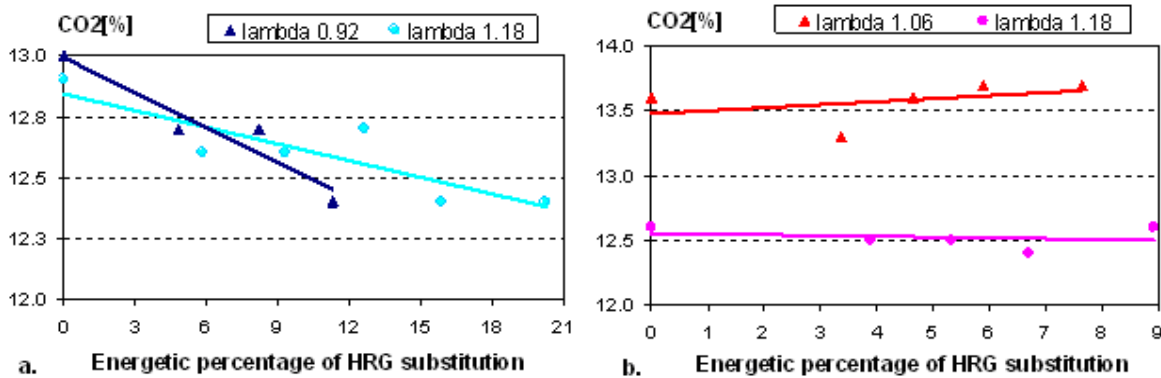

Fig. 7 Variation of CO_2 concentration with HRG (fuel energy fraction) substituting gasoline:
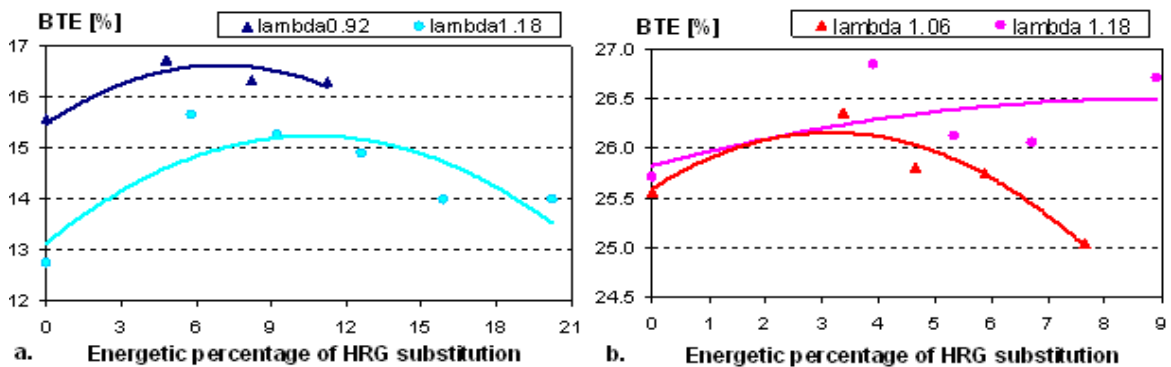(a) 540 mbar, 1600rpm; (b) 380 mbar, 2500 rpm

Fig. 8 Variation of BTE with HRG (fuel energy fraction) substituting gasoline: (a) 540 mbar, 1600rpm; (b) 380 mbar, 2500 rpm
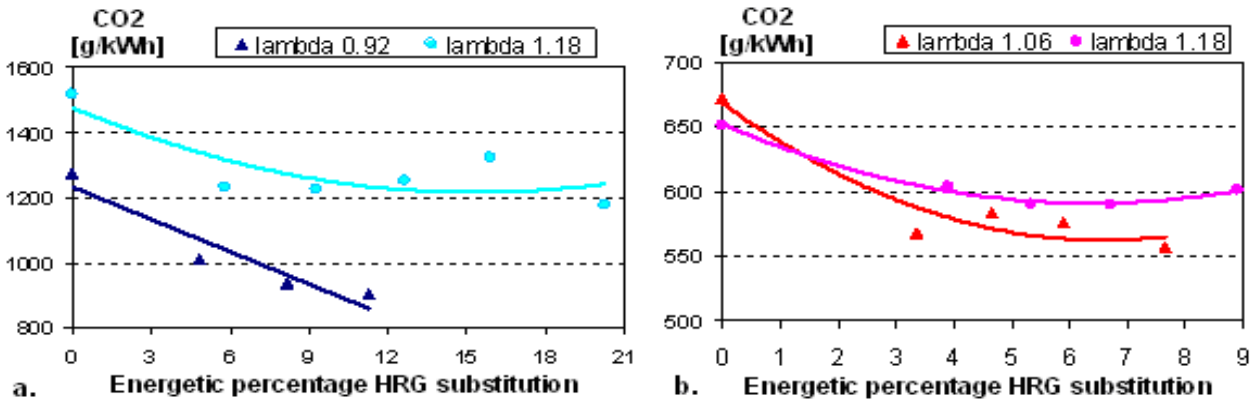
Fig. 9 Variation of CO_2 specific emission, in g/kWh, with HRG (fuel energy fraction) substituting gasoline:
(a) 540 mbar, 1600rpm; (b) 380 mbar, 2500 rpm

Table 1. Effect of HRG addition on BTE

Engine operating condition	a – lambda 0.92				b – lambda 1.18			
Energetic percentage of HRG (%)	0	4.8	8.2	11.3	0	3.9	5.3	6.7
Initial combustion phase (CAD)	23.5	20.1	20.5	20.3	31.4	31.4	26.0	24.5
Main combustion phase (CAD)	21.5	20.6	20.5	20.4	21.9	18.8	19.3	19.4

The addition of HRG has a contradictory effect on BTE. At 540 mbar and 1600 rpm, for $\lambda = 1.06$, BTE shows a decrease of 5% for about 7% HRG and a decrease of 12% for 15% HRG. For $\lambda = 1.28$, BTE increases by 31% with 9% HRG and by 27% with 19% HRG (Fig. 11a). At 380 mbar and 2500 rpm, addition of HRG showed a negative influence (Fig. 11b), which was not explained. As a general result, CO_2 specific emission, in g/kWh decreasing with HRG addition is varying in large limits with the engine operating condition. At 540 mbar, 1600 rpm, $\lambda = 1.06$, CO_2 of 1300 g/kWh for pure LPG decreases by 7.6% with 7.7% HRG and by about 13% with 15% HRG. For $\lambda = 1.28$, CO_2 of 1860 g/kWh with pure LPG decreases by 24% with 9% HRG and by 38% with 19% HRG (Fig. 12a). At 380 mbar, 2500 rpm, $\lambda = 1.06$, CO_2 of 625 g/kWh is lower by 12% with 6.9% HRG; for $\lambda = 1.28$, CO_2 of 580 g/kWh decreases by 1.7% with 5% HRG (Fig. 12b).

4. Conclusions

The present study has investigated BTE and CO_2 specific emissions, of a tractor diesel engine and a passenger car SI engine operating in dual fuel mode. The main fuel was commercial diesel fuel for the first engine and gasoline or LPG for the second engine; the additional fuel was HRG gas, generated by water electrolysis. HRG gas was aspirated in the induction manifold. The main conclusions of the study are the following:

1. Addition of HRG is efficient in decreasing CO_2 specific emissions, in g/kWh of diesel engine with a slight decrease of BTE. At 40% load, 1400 rpm, CO_2 was lower by 8% with 12.5% HRG and BTE was reduced by 2%; at 60% load, 2400 rpm, CO_2 decreased by about 5%, with 5.8% HRG and a reduction of BTE by 1%.
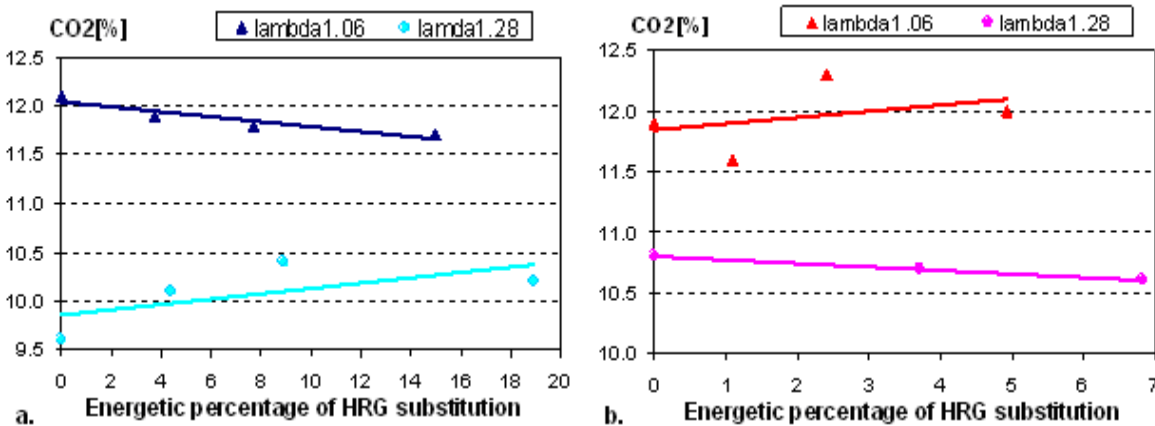

Fig. 10 Variation of CO_2 concentration with HRG (fuel energy fraction) substituting LPG:
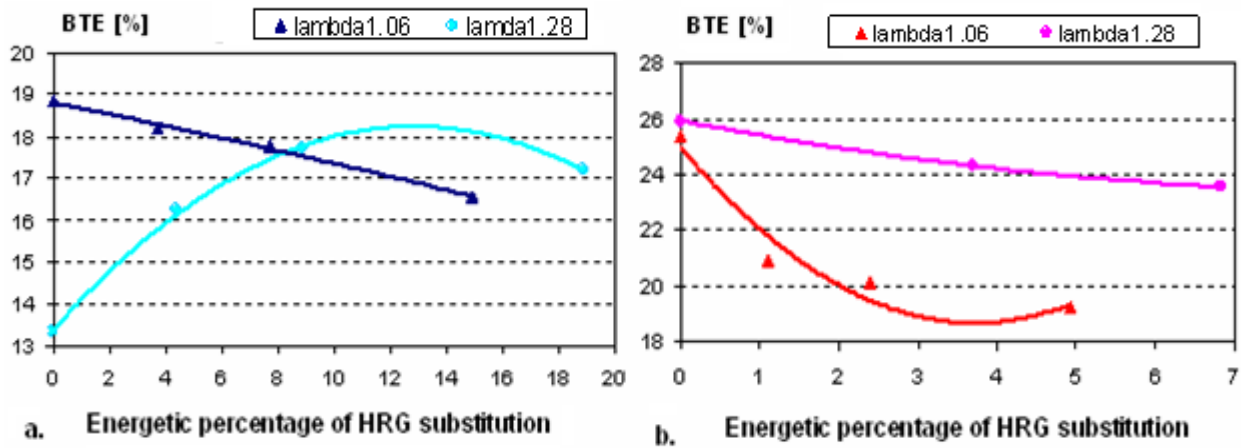(a) 540 mbar, 1600rpm; (b) 380 mbar, 2500 rpm

Fig. 11. Variation of BTE with HRG (fuel energy fraction) substituting LPG: (a) 540 mbar, 1600rpm; (b) 380 mbar, 2500 rpm
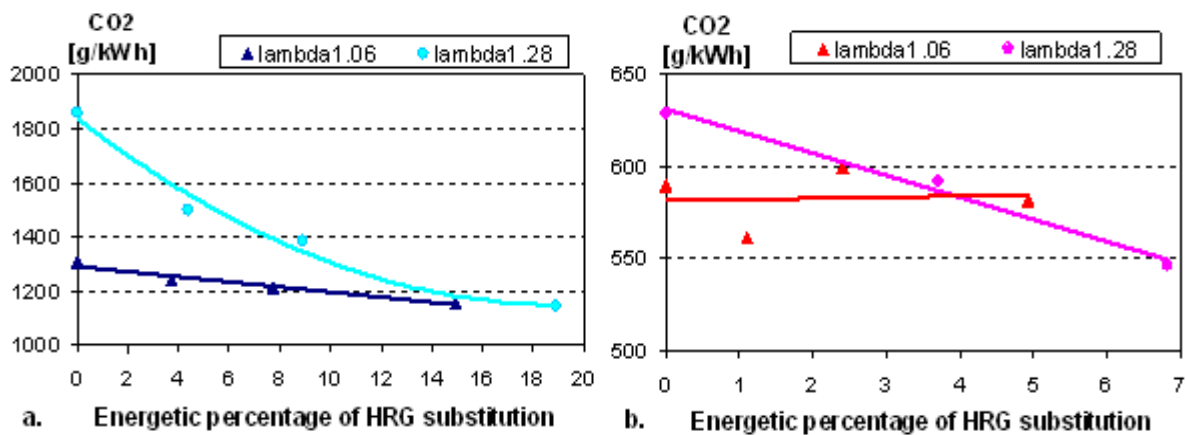

Fig. 12. Variation of CO₂ specific emission, in g/kWh, with HRG (fuel energy fraction) substituting LPG:
(a) 540 mbar, 1600rpm; (b) 380 mbar, 2500 rpm

2. Addition of HRG to gasoline, in a SI engine is more efficient on the abatement of CO₂ specific emissions than on a diesel engine, due to the generally favourable effect on BTE. At 540 mbar, 1600 rpm, λ = 1.18, CO₂ was lessened by 21% with 20.3% HRG and an increase of BTE by 9.5%. An optimum concentration of HRG of 9% was found for BTE, with a maximum increase by 21%.

3. The beneficial effect of HRG addition, in the abatement of CO₂ specific emission of SI engine was found also in the case of LPG as main fuel. At 540 mbar, 1600 rpm, λ = 1.06, CO₂ specific emissions decrease by 13% with 15% HRG and BTE was lower by 12%; a peak of BTE, with an increase by 33% was reached with 12.8 % HRG at λ = 1.28.

4. The effects on engine CO₂ specific emissions and BTE correspond to relative small amounts of HRG gas. Results should be further analyzed considering the associate effects on other engine emissions and the adverse impact of electrolyzer energy consumption.

References

Bade Shresthva S.O., Karim G.A., (1999), An investigation of the effects of the addition of dissociated water products to a gas fueled spark ignition engine, SAE Paper 1399-01-35016, On line at: https://www.sae.org/publications/technical-papers/content/1999-01-3516/

Birtas A., Voicu I., Petcu C., Chiriac R., Apostolescu N., (2011), The effect of HRG gas addition on diesel engine combustion characteristics and exhaust emissions, *International Journal of Hydrogen Energy*, **36**, 12007-12014.

Choi G.H., Chung Y.J., Han S.B., (2005), Performance and emissions characteristics of a hydrogen enriched LPG internal combustion engine at 1400 rpm, *International Journal of Hydrogen Energy*, **30**, 77-82.

Conte E., Boulouchos K., (2004), Influence of hydrogen-rich-gas addition on combustion, pollutant formation and efficiency of an SI engine, SAE Paper 2004-01-0972, On line at: https://www.tib.eu/en/search/id/BLSE%3ARN173554483/2004-01-0972-Influence-of-Hydrogen-Rich-Gas-Addition/.

Cracknell R.F., Alcock J.L., Shirvill L.C., Ungut A., (2002), Safety consideration in retailing hydrogen, SAE Paper 2002-01-1928, https://doi.org/10.4271/2002-01-1928.

Gerini A., Monnier G.A., Bonetto R., (1996), Ultra low emission vehicle using engine fuel, SAE Paper 961079, On line at: https://www.sae.org/publications/technical-papers/content/961079/.

Haragopela Rao B., Bhakta H.N., (1983), Hydrogen for dual fuel engine operation, *International Journal of Hydrogen Energy*, **8**, 381-384.

Ivanic Z., Ayala F., Goldwitz J., Heywood J.B., (2005), Effects of hydrogen enhancement on efficiency and NO_x emissions of lean and EGR diluted mixtures in a SI engine, SAE Paper 2005-01-0253, doi: 10.4271/2005-01-0253.

Karim G.A., (2005), Combustion in gas fuelled compression ignition engines on hydrogen combustion engines, *Journal of Engineering for Gas Turbines and Power,* **125**, 827.

Kumar Senthil M., Ramesh A., Nagalingam B., (2003), Use of hydrogen to enhance the performance of a vegetable oil fuelled compression ignition engine, *International Journal of Hydrogen Energy*, **28**, 1143-1154.

Liew C., Li H., Liu S., Besch M.C., Ralston B., Clark N., (2010), Exhaust emissions of a H_2-enriched heavy-duty diesel engine equipped with cooled EGR and variable geometry turbocharged, *Fuel*, **91**, 155-163

Lilik G.K., Zhang H., Herreros J. M., Haworth U.C., (2010), Hydrogen assisted diesel combustion, *International Journal of Hydrogen Energy*, **35**, 4382-4398.

Ma J., Su Y., Zhang Z., (2003), Simulation and prediction on the performance of a vehicle's hydrogen engine, *International Journal of Hydrogen Energy*, **28**, 77-83.

Niculae Gh., Chiriac R., Apostolescu N., (2013), Effects of HRG gas addition on performance and emissions of a SI engine fuelled with Liquefied Petroleum Gas, *Revista de Chimie*, **64**, 574-579.

Santilli R.M., (2006), A new gaseous and combustible from water, *International Journal of Hydrogen Energy*, **31**, 1113-1128.

Saravanan N., Nagarajan G., Dhanasekaran C., Kalaiselvan K.M., (2007), Experimental investigation of hidrogen port fuel injection in DI diesel engine, *International Journal of Hydrogen Energy*, **32**, 4071-4080.

Saravanan N., Nagarajan G., Sanjay G., Dhanasekaran C., Kalaiselvan K.M., (2008), Combustion analysis on a DI diesel engine with hydrogen in dual fuel mode, *Fuel*, **87**, 3591-3599.

Toshio Sh., (2007), Improving thermal efficiency by reducing cooling losses in hydrogen combustion engines, *International Journal of Hydrogen Energy*, **32**, 4285-4293.

Xiao F., Sohrabi A., Galai M., Karim G.A., (2009), The performance of a IDI diesel engine having low concentrations of hydrogen in the intake air, SAE Paper 2009-01-1830, On line at: https://www.sae.org/publications/technical-papers/content/2009-01-1830/