Performance of CI Engine Operating with Hydrogen Supplement Co-combustion with Jojoba Methyl Ester

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Abstract
The present work experimentally investigates the performance of compression ignition (CI) engine which operates on hydrogen and Jojoba methyl ester (JME). The hydrogen is introduced by enriching air-intake manifold with hydrogen gaseous supplement at the atmospheric condition which is co-combusted in the presence of pilot flame that is initiated by jojoba methyl ester. The engine is supplied with a range of hydrogen-JME fuel mixture proportions to study the effect of hydrogen supplement on combustion and exhaust emissions. Under fixed flow rate of JME, the data shows that hydrogen supplement enhances diesel engine thermal efficiency while reducing specific fuel consumption. The gas emission results shows that as hydrogen supplement increases, NOx emission mildly increases while opacity majorly increases. Hydrogen supplement affects the internal combustion by causing an increase in internal pressure, pressure rise rate and maximum heat release rate. The experiment shows that hydrogen-JME duel fuel smoothly combust in compression ignition engine.

Keywords Hydrogen combustion; Jojoba Methyl Ester; compression ignition engine; Dual fuel engine; NOx; particulate matter

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Nomenclatures

CI	Compression ignition
$\frac{dQ}{d\theta}$	Heat release rate, kJ/degree
JME	Jojoba methyl ester
LHV	Lower heating value, MJ/kg
LPG	Liquefied petroleum gas
LPM	Liter per minute
$\dot{m}$	Mass flow rate, kg/s
PM	Particulate matter
Q	Integral heat release rate, kJ
$\dot{\mathrm{Q}}_{\mathrm{in}}$	Heat release rate, kW
SI	Spark ignition
sfc	Specific fuel consumption, $kg/\mathrm{kW} \cdot \mathrm{h}$
T	Torque, $N.m$
$\dot{W}$	Power, kW

Greek symbols

η	Thermal efficiency
ω	Angular velocity, Rad/s
θ	Crank angle, degree

Subscripts

in	Input
max	Maximum
out	Output

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1 Introduction

The sharp increase in energy demands has put stress on energy supply sources and types. Liquid diesel originating from crude oil is the most common fuel used in compression ignition engines. The continuous increase in crude oil demands has led scientists and engineers to explore the use of alternative possible fuels to run compression ignition engines such as LPG [1] and hydrogen [2, 3], in order to replace diesel or at least reduce its use as a fuel for engines. Co-combustion of hydrogen with diesel fuel is driven by numerous reasons [2] which are (1) enhancing the hydrogen to carbon ratio of the entire fuel supplied to the engine, (2) decreasing the heterogeneity of diesel fuel spray, and (3) reducing the combustion duration. The flame speed in stoichiometric hydrogen air mixture is seven times faster than the corresponding gasoline air mixture [4] which gives great advantage to internal combustion engines, leading to higher engine speeds and greater thermal efficiency [4]. Due to the fact that hydrogen has high heating value and is a carbon-free energy carrier, hydrogen is considered one of the most promising alternative fuels that can play great role in reducing dependence on fossil fuels.

The use of hydrogen in compression ignition (CI) engines shows brighter prospect when compared to spark ignition (SI) engines since the use of hydrogen as a fuel in spark ignition engine has showed a significant reduction in power output [6]. In addition, at high load multiple problems have been reported in spark ignition such pre ignition, backfire and knocking [6]. These problems have limited the use of hydrogen in SI engine [7, 8]. Recent published work [9] showed that in SI engine, the injection of hydrogen at the beginning of the compression stroke shows smooth engine running at stoichiometric air

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fuel ratio without abnormal burning while the danger of backfire is eliminated with a sequential timed multipoint injection. From literature, it is clear that more research work is needed to further clarify the features and benefits of hydrogen as a fuel for SI engines.

In compression ignition engines, hydrogen use has showed a significant increase in thermal efficiency [3, 8, 10-13] due to the fact that heat release rates from hydrogen-diesel co-combustion tend to be higher than those for diesel fuel combustion. Saravanan et al. [10] reported an increase by 20% in thermal efficiency [10] when compared to pure diesel combustion and an increase of 13% in NO_x emission. Hydrogen is combusted in CI engine as dual fuel since compression temperature is not enough to initiate the combustion due to hydrogen high self-ignition temperature [10]. In the dual fuel engine arrangement, the diesel fuel is used as the main fuel to initiate the ignition and combustion process while hydrogen is introduced as supplementary fuel through the air-intake manifold [3] or directly injected into the engine cylinders. Hence, major energy is obtained from diesel while the rest of the energy is supplied by the hydrogen. An increase of 30% in brake thermal efficiency has been reported in literature [11] for co-combusting hydrogen in the presence of diesel fuel in CI engine with compression ratio of 24.6. Lee et al. [8] has reported an increase in thermal efficiency of 22% for dual injection at low loads and 6% increase at high loads when compared to direct injection. Lee et al. [12] established that the stability and maximum power for dual injection are accomplished by direct injection of hydrogen. Das et al. [13] reported a maximum brake thermal efficiency of 31.3% at 2200 rpm with 13 N-m torque in which the study covered numerous techniques for adding the hydrogen such continuous carburation, continuous manifold injection, timed manifold injection and low pressure direct cylinder injection. Deb et a. [14]

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reported that the brake thermal efficiency increased by 3.94% with the addition of 11% hydrogen and by 16.80% with the addition of 42% hydrogen due to improved combustion and high flame velocity of hydrogen.

The co-combustion of a small amount of hydrogen (around 6%) can increase engine durability [16] since it shortens the diesel ignition delay and decreases the rate of pressure rise which provides better conditions for soft run of the engine. The hydrogen can be combusted as sole fuel in CI engine where glow plug is used to start the combustion [16]. A modeling study [16] has showed that using hydrogen as sole fuel in CI engine has actually better efficiencies than diesel fuel over all the load range and it shows better full load power and torque outputs running about same stoichiometry.

The use of hydrogen fuel, as a potential supplement fuel that can reduce the use of liquid diesel fuel, comes with a drawback of the increase in NO_x emission. Thus the need for techniques to reduce NO_x become more vital for engines operating with dual hydrogen-diesel fuel. One common method to reduce NO_x emission in diesel engine is thru injecting steam to the combustion [17]. Another way to reduce NO_x is by operating the engine with lean mixtures. Lean mixture results in lower temperature that would slow the chemical reaction, which weakens the kinetics of NO_x formation [18,19]. As reported by Talibi et al. [20], the NOx emission can radically increase when the when the combustion temperatures resulting from H_2-diesel fuel co-combustion exceeded the threshold temperature for NOx formation temperatures.

One of the main advantages of hydrogen combustion over diesel fuel is that it does not produce major pollutants such as hydrocarbon (HC), carbon monoxide (CO), sulphur dioxide (SO_2), smoke, particulate

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matter, lead, and other carcinogenic compounds. This is due to the fact that only water is produced out of the complete hydrogen combustion in air, in addition to the generated NOx due to the presence of Nitrogen in the air [21]. So, the hydrogen- operated engines' main disadvantage is the NO_x emissions. Under the clearly high combustion temperatures, supported further by the combustion of the Hydrogen in the overall fuel supplied to the engine, the nitrogen present in the air reacts with oxygen to form NO_x. The study conducted by Lilik [22] of hydrogen-assisted diesel combustion shows a modest increase of NOx with increasing levels of hydrogen addition. A particularly interesting finding [22] is that NO_2 emissions tended to increase with increasing hydrogen enrichment, while NO emissions tended to decrease.

Hydrogen has an considerably lower cetane number than diesel fuel [23] and is not simply ignitable by means of compression engines, and therefore it requires an ignition source. In the case of a compression engine, the hydrogen can either be aspirated into the engine or injected directly into the cylinder. Pilot flame initiated by diesel fuel has been widely used to ignite hydrogen [23, 24].

A significant number of studies [26-27] have reported a reduction in PM emissions while using hydrogen as a supplement fuel in a compression engine that run one diesel fuel. Different reasons caused this reduction in PM which are: (1) the increase in oxidation of particulates due to higher in-cylinder temperatures that result from increased heat release rates, (2) the reduction of the overall carbon-hydrogen ratio of the combined hydrogen-diesel fuel, and (3) the formation of hydroxyl radical from hydrogen combustion which also enhances PM oxidation. A reduction in smoke to near zero levels is reported in by Tomita et al. [28] at all diesel injection timings and at all equivalence ratios of hydrogen.

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Lilik et al. [24] reported an increase in PM emissions at high engine loads due to the fact that hydrogen displaces air when it is introduced through the air-intake manifold which slows the rate of carbon oxidation reactions, resulting in a rise in PM emissions. Hence more experimental studies are needed to clearly explain such emissions trends.

The use of JME as a diesel fuel substitute has been investigated by different research groups [29, 30]. Radwan et al. [29] indicated that JME is safe fuel with no sulphur content and ability to reduce the engine wear. Shah et al. [30] reported that JME has higher kinematic viscosity when compared to soybean oil methyl esters (SME).

The current study addresses the advantage of using hydrogen supplement in CI engine that co-combusted with Jojoba methyl ester as the pilot flame. The study focus on using the engine at partial load (low torque) with no added techniques to reduce NOx other than using hydrogen. Also the study reports the effect of hydrogen on emission. The hydrogen is introduced through enriching the air-intake manifold with hydrogen at atmosphere condition. The study reports the effect of hydrogen supplement fuel on the engine efficiency, torque, specific fuel consumption, exhaust temperature, heat release rate, NOx emission and PM emission.

2 Experimental Setup

Fig. 1 shows a schematic diagram of the experimental setup used to test the co-combustion of hydrogen with diesel fuel. A Ricardo E6 research engine is used in the test setup which is a single cylinder

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compression ignition engine. The engine is fully equipped with instrumentations for measurements of all engine operating parameters. No modifications is done on the engine to work with hydrogen since hydrogen is introduced through the air-intake manifold at room pressure as shown in Fig. 1. The engine is equipped by an electrical dynamometer rated power of 22 kW and rated voltage of 400/230 V. The torque of the engine is measured through force transducer that is connected to the electrical dynamometer which has uncertainty of ±0.1 N. The liquid fuel flow rate is calculated by measuring the time needed to consume fix amount of JME liquid fuel. The engine specifications are shown in Table 1. The chemical characteristics of the primary fuel JME (0.1) and the supplement fuel (hydrogen) are listed in Table 2. In this study JME (0.1) means the mixture is formed by mixing a 10% methyl alcohol to raw Jojoba oil volumetric ratio (0.1 mole alcohol/mole of raw jojoba oil). More details about the fuel JME (0.1) are published by Selim et al. [31].

As shown in Fig. 1, hydrogen gas is injected into the air-intake manifold at atmosphere pressure. A pressure regulator, a volumetric rotameter (with ±0.6 LPM uncertainty) and a throttle value are used to control the hydrogen flow rate and outlet pressure. The flow rate of air is measured using a calibrated orifice meter with pressure transducer arrangement. The pressure transducer has uncertainty of ±0.1 Pa. The JME flow rate is measured by recording the required time to consume a fixed volume of JME with uncertainty of ±0.1 ml/s. The measurement of combustion pressure, engine speed, engine output torque, and crank angle are collected using a high speed data acquisition system. A labVIEW interface program has been written to collect the data at a rate of 60,000 points per second and to store the data.

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The main objective of the conducted experiments is to understand the effect of hydrogen supplement once co-combusted with pilot flame using Jojoba methyl ester liquid fuel. To fully understand the effect of hydrogen on the performance of a dual fuel single cylinder CI engine, numerous tests are conducted under different conditions as listed below:

1) Test (1): Range of hydrogen enrichment flow rates. Under same JME flow rate, different hydrogen contents ranging from 0 to 20 LPM of hydrogen are used to evaluate the effect of hydrogen enrichment.
2) Test (2): Range of JME pilot fuel flow rates. Under same hydrogen flow rate of 12 LPM and fixed engine speed of 1260 rpm, different amount of JME pilot fuel are tested. The JME flow rates are varied from 0.4 to 1.2 L/hr.
3) Test (3): Range of ignition timing. Under fix flow rate of JME-H_2 fuel flow rate, different injection timings ranging from 20 to 46 degree are evaluated.

The engine efficiency and specific fuel consumption are calculated using equations (1) and (2) respectively:

$$\eta = \frac{\dot{W}_{out}}{\dot{Q}_{in}} = \frac{T \cdot \omega}{(\dot{m} \times LHV)_{JME} + (\dot{m} \times LHV)_{H_2}} \quad (1)$$

$$sfc = \frac{\dot{m}_{fuel}}{\dot{W}_{out}} \quad (2)$$

$$\dot{m}_{fuel} = (\dot{m})_{JME} + (\dot{m})_{H_2} \quad (3)$$

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The lower heating value is used in equation (1) for the efficiency calculation since no vapor is condensed during the experiment. The density of hydrogen is calculated at the air-intake condition; namely at atmosphere pressure and room temperature.

All data reported in this study are the average results of all acquired cycles according to the following relations:

$$P_{max} = \frac{\sum P_{max,\ per\ cycle}}{No.of\ cycles} \tag{4}$$

$$T_{exhaust} = \frac{\sum T_{exhaust,\ per\ cycle}}{No.of\ cycles} \tag{5}$$

$$\eta = \frac{\sum \eta_{per\ cycle}}{No.of\ cycles} \tag{6}$$

$$\frac{dP}{d\theta}_{max} = \frac{\sum \frac{dP}{d\theta}_{max,\ per\ cycle}}{No.of\ cycles} \tag{7}$$

$$\frac{dQ}{d\theta} = \frac{\gamma}{\gamma-1} P \frac{dV}{d\theta} + \frac{1}{\gamma-1} V \frac{dP}{d\theta} \tag{8}$$

$$\frac{dQ}{d\theta}_{max} = \frac{\sum \frac{dQ}{d\theta}_{max,\ per\ cycle}}{No.of\ cycles} \tag{9}$$

$$Q = \int \frac{dQ}{d\theta} d\theta = \sum \frac{\Delta Q}{\Delta\theta} \Delta\theta \tag{10}$$

The exhaust emission are measured using a gas analyzer known as VARIO plus SE instrumentation which is manufactured by MRU Instruments, Inc. The gas analyzer uses electrochemical sensors to measure the gas component concentrations in flue gases with accuracy of ±6 ppm for NOx. The unit is calibrated with certified calibration gasses for NO (100 ppm of NO in NO+N_2 mixture) and NO_2 (500

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ppm of NO_2 in NO_2+N_2 mixture). Each set of experimental set of experiments, the unit was flushed with regular room air before start recording any measurements.

The particulate matter (PM) emission measured by reporting exhaust opacity through the use of AVL Opacimeter which is a dynamic partial-flow measuring instrument. The AVL Opacimeter provides a continuous measurement of exhaust gas opacity through the use of a measuring chamber of defined measuring length and non-reflecting surface that is filled homogeneously with the exhaust gas. The loss of light intensity between a light source and a receiver is measured and from the light loss the opacity of the exhaust gas is calculated based on the Beer-Lambert law.

The uncertainty of the measurements are estimated by the uncertainty of the used measuring device which is shown in Table 3.

3 Results and Discussion

The effect of hydrogen enrichment on CI engine performance running on Jojoba methyl ester pilot fuel is investigated under different testing conditions which are as follow:

3.1. Range of hydrogen enrichment flow rates.

3.2. Range of JME pilot fuel flow rates.

3.3. Range of ignition timing.

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3.1. Range of hydrogen enrichment flow rates

The effect of amount of hydrogen supplement when it is burned with fixed amount of JME (1.67×10^{-4} kg/s) is shown in Fig. 2 where the pilot fuel (JME) is injected at 35 degree btdc. In this test a fixed amount of JME fuel flow rate is consumed by keeping engine speed fixed at 1260 rpm while variable amount of hydrogen is introduced through the air intake. The results show that as hydrogen supplement increases the engine efficiency increases which is expected since hydrogen presence will upsurge the combustion temperature and enhances mixing due to the fact that flame move faster in hydrogen when compared to diesel, keeping in mind that hydrogen is burned as an additional fuel. As shown in Fig. 2a and for engine speed of 1260 rpm, the thermal efficiency increases with the increase of hydrogen flow rate from 0 to 20 LPM. The presence of extra fuel (hydrogen) supplement increases the torque as expected as shown in Fig. 2b. Figure 2c shows that as hydrogen supplement flow rates increases the exhaust gas temperature will increases which is due to the fact that more fuel is burned and that hydrogen has higher heating value when compared to JME. The specific fuel consumption for fixed engine speed of 1260 rpm and different hydrogen flow rate values are shown in Fig. 2d and it is clear that as hydrogen flow rate increases, the specific fuel consumption (sfc) decreases. This reduction in sfc is expected since the lower heating value (LHV) of hydrogen is two and half times higher than LHV of JME (see table 2).

The increase in combustion temperature (indicated by the exhaust gas temperature) tends to increase NO_x emission, as shown in Fig. 2d, since NOx is produced when nitrogen and oxygen are present at

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elevated temperatures. The addition of hydrogen causes exhaust temperature to increase which in return has mildly increased the NOx emission as shown in Fig. 2e. The combustion of hydrogen with JME shows lower NOx when compared to diesel [3] using same engine setting, this could be due to the fact that JME has more water content [32] compared to diesel which would produce lower flame temperature. The JME and diesel water content % by volume is 0.5 and 0.1, respectively [32].

One would expect that as exhaust temperature increases, the opacity would decreases however this did not happen as shown in Fig. 2f which is due to the fact that as hydrogen is added the amount of JME is kept fixed which pushed the combustion toward the rich fuel combustion zone. The increase in opacity is due to the fact that hydrogen competes with JME when it comes to reacting with oxygen and most of the time it is more favorable to burn hydrogen than JME since it has higher flame speed and hydrogen burns in the engine as pre-mixed flame. Also the increase in soot has been dramatic at such partial load due to the fact that hydrogen is added by enriching air-intake manifold with hydrogen which make hydrogen compete with the presence of oxygen which increases the chance of soot formation.

As shown in Fig. 2g, as hydrogen flow rate increases the air-fuel ratio decreases since air decreases as hydrogen content in the air manifold increases. This is expected since hydrogen is taking the place of air forcing less air to be inhaled to the piston as well as that hydrogen contribute to the increase in overall fuel flow rate. Simply hydrogen will displace air causing a reduction in air-fuel ratio.

Figure 3a shows that as hydrogen flow rate increases from 0 to 20 LPM, the maximum combustion pressure slightly increases from 42 bar to 47, respectively. The increase is expected since the hydrogen flame speed moves faster than inside gasoline mixture [4] which surly move faster than diesel flame [4].

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Correspondingly, the maximum of the in-cylinder pressure rise rate is shown in figure 3b which shows that addition of hydrogen will increase the maximum pressure rise rate dramatically from pure JME case compared to JME with 4 LPM of hydrogen. The maximum combustion pressure rise rate keep increasing as more hydrogen supplement is added to the combustion however the rate become less dramatic between 4 LPM and 20 LPM. As the combustion pressure rise rate increases the pressure tends to reach the maximum pressure while during the compression stroke which tends to increase the maximum pressure shown in Fig. 3a.

Fig. 3c shows how the maximum heat release rate change with respect to crank angle. It is clear that as amount of hydrogen increases the amount of hear release increases which mean more energy is released. This is expected since hydrogen supplement increases the amount of the fuel available for combustion.

The integral release heat curves with respect to crank angle are shown in figure 4 and are calculated as shown in equation (10). As expected the addition of hydrogen will increase the heat release value and extend the heat release time which explain the rise in exhaust temperature as discussed in earlier figure. It is clear that the hydrogen supplement which has high heating value compared to JME will provide measurable amount of heat release as shown in Fig. 4 which cause the integral heat release curves to shift up for all cases.

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3.2. *Range of JME pilot fuel flow rates*

The effects of amount of JME pilot fuel flow rate on engine performance while being supported with hydrogen supplement are shown in Fig. 5. In this test, a fixed rate of hydrogen flow rate with fixed engine speed of 1260 rpm is tested under different JME pilot flow rates. As shown in Fig. 5a, at engine speed of 1260 rpm with hydrogen supplement of 12 LPM, the engine efficiency dramatically increases in the early stages of JME fuel flow rate hikes, however at certain point around JME flow rate of 0.76 L/hr, the increase in efficiency become less pronounced. This means that the combustion is getting richer and the increase in output power is going in less rate compared to the increase in the JME fuel flow rate. As shown in Fig. 5b, the torque keeps increasing linearly with JME fuel flow rate which is expected since as more fuel is burned a higher torque is expected however this happen at the expense of efficiency as shown Figure 4a which continue increases however at slower rate. The increase in JME flow rate means more fuel is burned and more smaller flames are produced by the pilot fuel combustion which causes an increase in the exhaust gas temperature as shown in Fig. 5c. Due to the fact that engine efficiency keeps increasing with the increase in JME pilot fuel flow rate, one expects that the sfc is going to which is observed in Fig. 5d. The drop in sfc is more dramatic in early stage of JME flow rate hikes however later this drop become less dramatic as combustion become more richer.

The effect of ramping JME flow rate on NOx emission and on opacity are shown in Fig. 4e and Fig. 5f, respectively. As shown in Fig. 5e, the effect of increasing the JME fuel flow rate on NOx is weak and the amount of NOx did not change a lot as JME flow rate varies. In general, one can observe a mild

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down trend in NOx emission as JME flow rate increase. This trend is due to the fact that as more JME fuel injected to the engine, the flame temperature is expected to decrease due to the energy consume by the evaporation JME liquid fuel. On the other hand the opacity will increase dramatically with the increase of JME flow rate since that fuel provide more carbon particles which increase the chance of soot formation as shown in Fig. 5f. The increase in JME means the air-fuel ratio will decrease as shown in Fig. 5e which also support the increase in opacity since the combustion tilt toward rich mixture. As shown in Fig. 5e, as pilot fuel flow rate increases the air-fuel ratio decrease since air flow rate is kept constant while pilot fuel flow rate increases.

3.3. Range of ignition timing

The effects of JME fuel injection timing on engine performance while being supported with hydrogen supplement are shown in Fig. 6. In this test, a fixed rate of hydrogen flow rate with fixed engine speed of 1260 rpm is tested under different injecting timing. As shown in Fig. 6a, at engine speed of 1260 rpm with hydrogen supplement of 12 LPM, the engine efficiency has beak value around 25 degree. Overall the efficiency decreases as injection timing advances from 25 degree to 40 degree btdc. Early injection will cause too much pressure rise before end of compression stroke which reduces output power and hence reduces engine efficiency. Late injection (around 20 degree btdc), means the fuel is combusting while the piston is moving downward which reduce the engine efficacy due to late combustion and reduced pressures in the cylinder. The engine torques is shown in figure 6b which follow similar trend as

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the efficiency curve for the same reasons explained earlier for Fig. 6a. The exhaust gas temperature is shown in Fig. 6c. As shown in Fig. 6c as injection timing advances, the cylinder pressure is expected to increase causing an increase in the combustion temperature however this increase is small (around 60 ^{o}C). The specific fuel consumption for fixed engine speed of 1260 rpm and flow of 12 LPM of hydrogen supplement fuel is shown in Fig. 6d. The specific fuel consumption increases as injection timing is advanced since as stated earlier advancing injection timing will reduce output power.

The engine NO_x emission is shown in Fig. 6e and soot emission is shown in Fig 6f. As shown in Fig 6e, as injection timing is advanced, the NO_x increases which is due to the high rise in the peak temperature and pressure of the engine during the compression stroke. Overall NOx drop as opacity increases which typical since as late combustion occur there is high chance that the combustion is not complete forcing the formation of more soot and less NOx formation. The existence of a peak injection timing correspond for similar peak in NOx emission. As the injection timing becomes more advanced, the pressure and temperature at time of injection becomes less and less. This tends to increase the delay period of the diesel fuel and hence more mass of fuel is being injected without burning. This tends to increase the smoke formation in the exhaust as shown in Fig. 6f, however it reduces the NOx formation as shown in Fig 6e.

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4 Conclusions

In this work, an experimental investigation has been conducted to examine the effect of the co-combusting hydrogen supplement on the performance of compression ignition engine that run on JME as dual fuel. The hydrogen is introduced to the engine at atmospheric conditions by injecting the hydrogen to the air-intake manifold. The results show that under same loading condition, the hydrogen supplement boosts efficiency and cuts specific fuel consumption (sfc) which is desirable since it reduces the dependence on liquid fuel. The data shows that the use of hydrogen supplement with JME will increase exhaust gases temperature with mild effect on NOx emission and dramatic increase in opacity. Controlling pilot fuel flow rate is very important combustion parameter, which has strong impact on thermal efficiency, exhaust temperature, opacity emission and has mild effect on NOx emission. The results show an increase in the maximum heat release rate and overall increase in the amount of integral heat release values. Overall, it is clear that hydrogen supplement is a great potential fuel to be combusted as duel fuel with JME in compression ignition engine.

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Table 1 Ricardo E6 Engine specifications

Number of cylinders	1
Bore	76.2 mm
Stroke	111.1 mm
Swept Volume	0.607 liters
Max. Speed	60 rev/sec (3000 rpm)
Max. Power, Diesel (CR = 20.93)	9.0 kW, Naturally Aspirated
Compression Ratio (CR)	Max. CR 22
Injection Timing	Variable, 20°- 46° btdc

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Table 2 Fuel Properties

Fuel Propriety	JME (0.1) [31]	Hydrogen
Chemical Formula	Percentage per weight: 7.96%H, 78.96%C, 11.0%N and 2.1% rest	H_2
Density, kg/m^3	860	0.08988
Molecular Weight, kg/kmole	170	2.016
Lower Heating Value, MJ/kg	42.6	119.96
Stoichiometric air-fuel ratio, kg/kg	14.6	34.3
Ignition temperature, ^{o}C	366	600
Adiabatic flame temperature, ^{o}C	1720	2210
Sulphur content by weight, %	0.6	0

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Table 3: Instrumentations brand names and uncertainty.

Measurement	Sensor	Uncertainty
Temperatures	Omega k-Type thermocouples	0.5 ℃
Engine speed	Speed sensor	1 rpm
Engine force	Omega load cell LCMAD-25	0.1 N
Gaseous H2 volume flow rate	Volumetric rotameter	0.6 LPM
Diesel flow rate	Stop watch and 10 ml volume	5 ms
Air volume flow rate	Pressure transducer model 407910/EXTECH instrument	0.1 Pa
Cylinder pressure sensor	AVL type ZI31 0.1 bar	0.1 bar
NOx Emissions	VARIO plus SE	5 ppm
Opacity	AVL 439 Opacimeter	0.01 %

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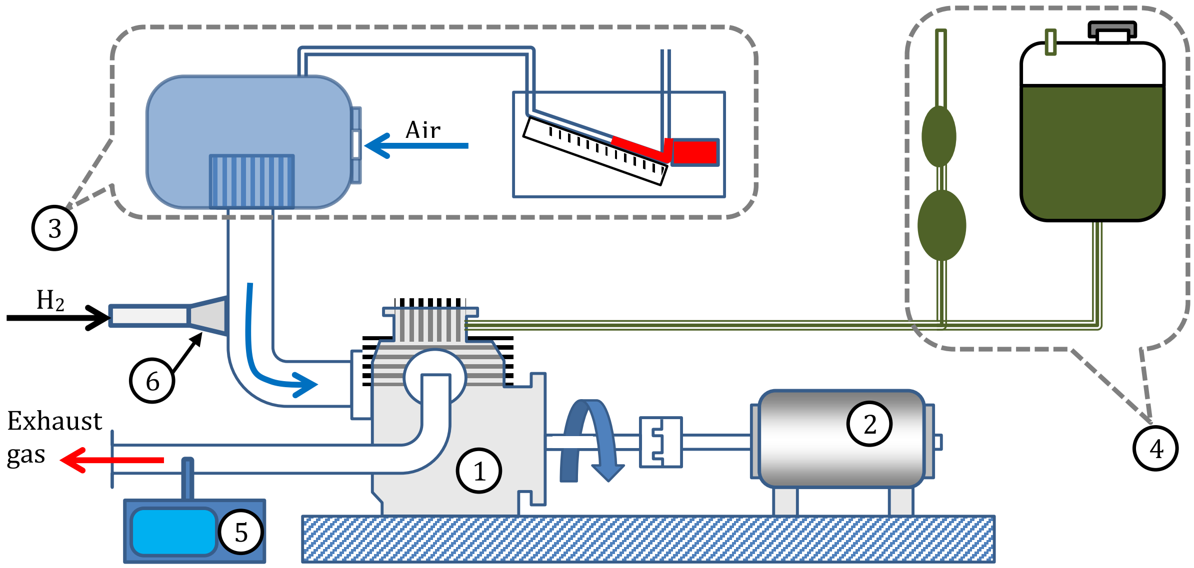

1: Ricardo E6 Engine 2: Electrical Dynamometer 3: Air intake metering system 4: Diesel Fuel metering System 6: Gas Analyzer & Opacity meter 6: Hydrogen supply

Fig. 1, Hamdan and Selim

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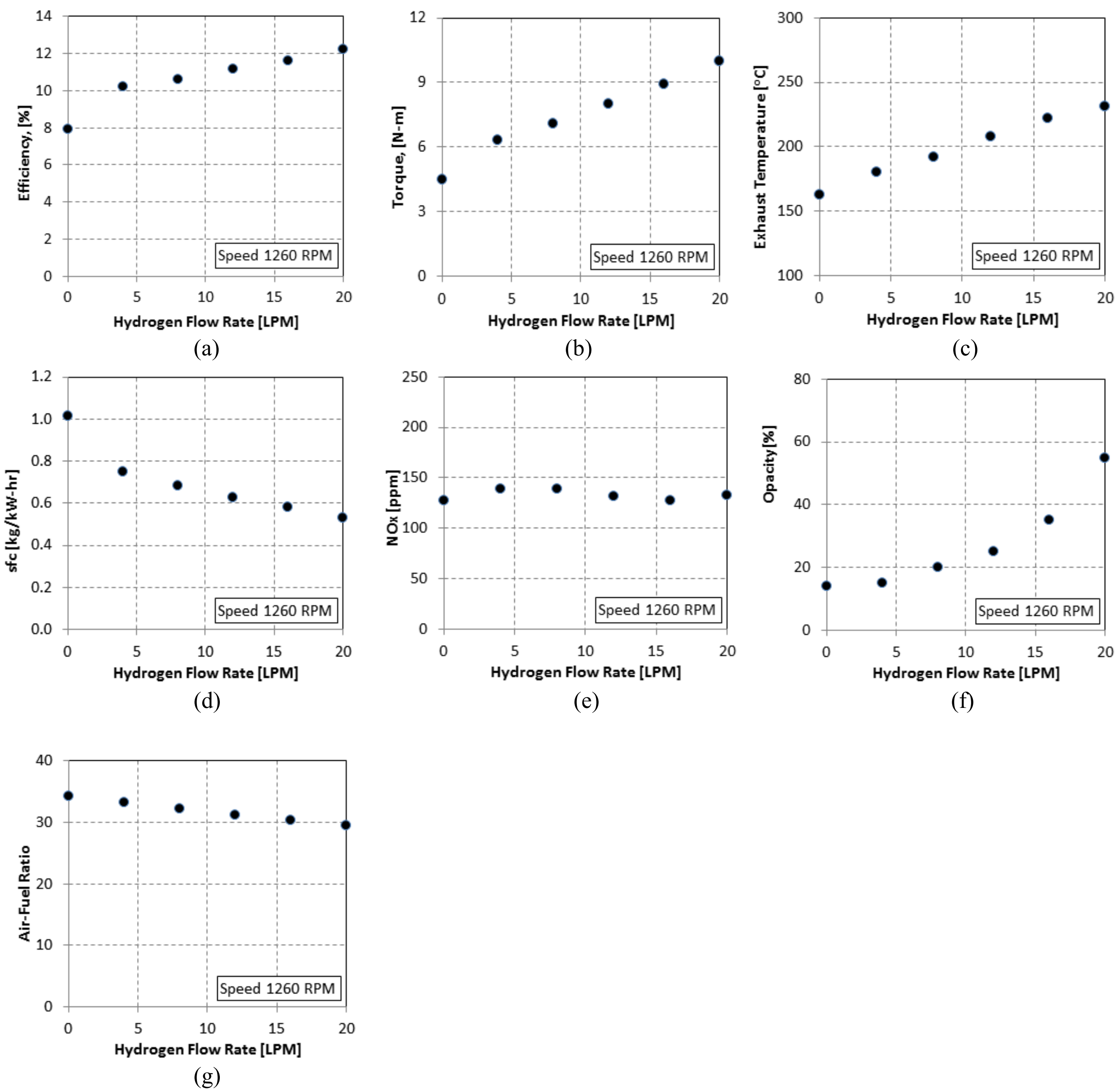

Fig. 2, Hamdan and Selim

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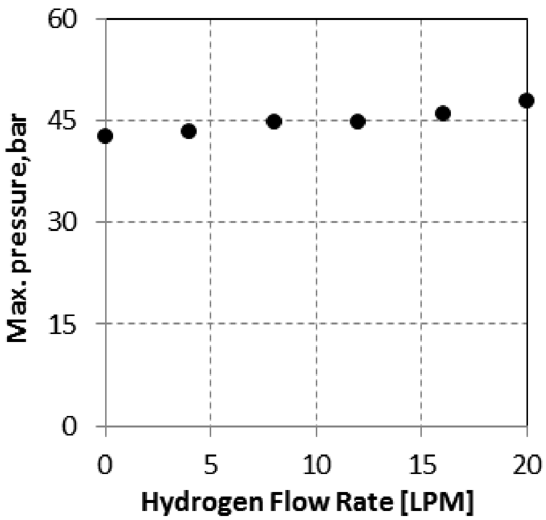

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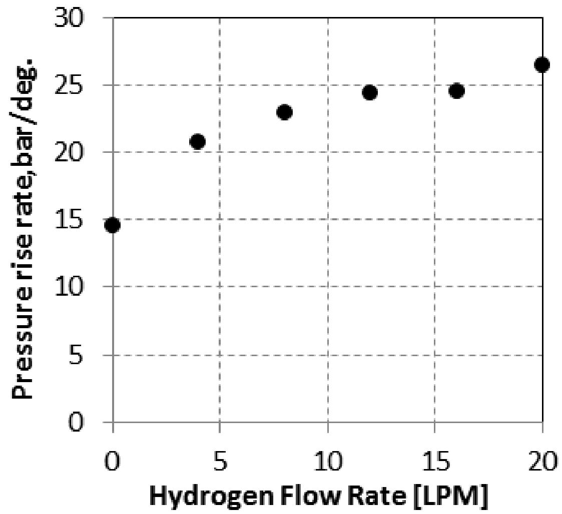

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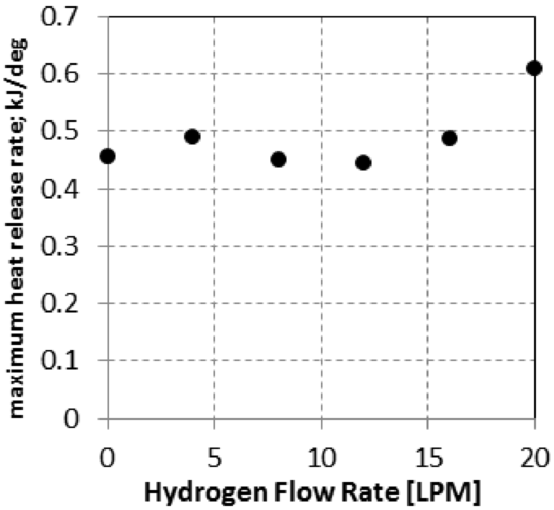

(c)

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Fig. 3, Hamdan and Selim

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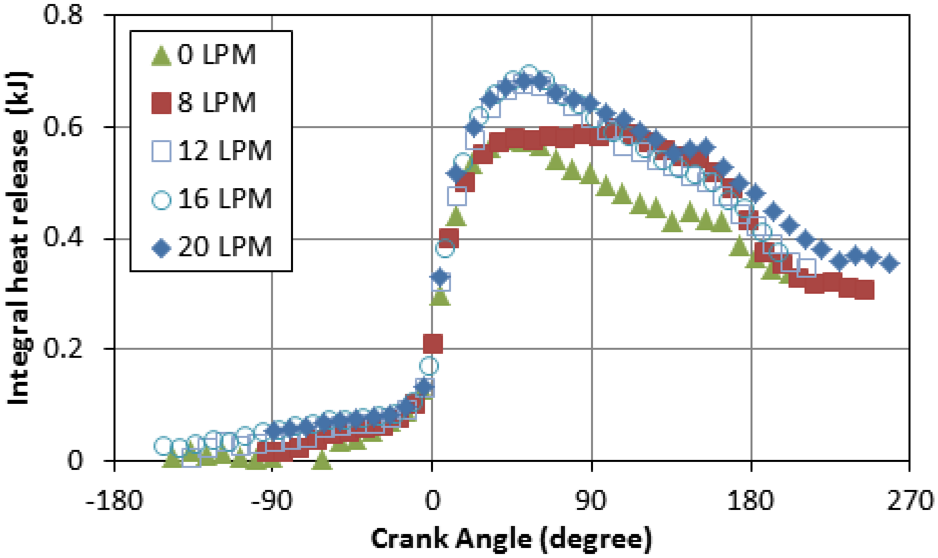

Fig. 4, Hamdan and Selim

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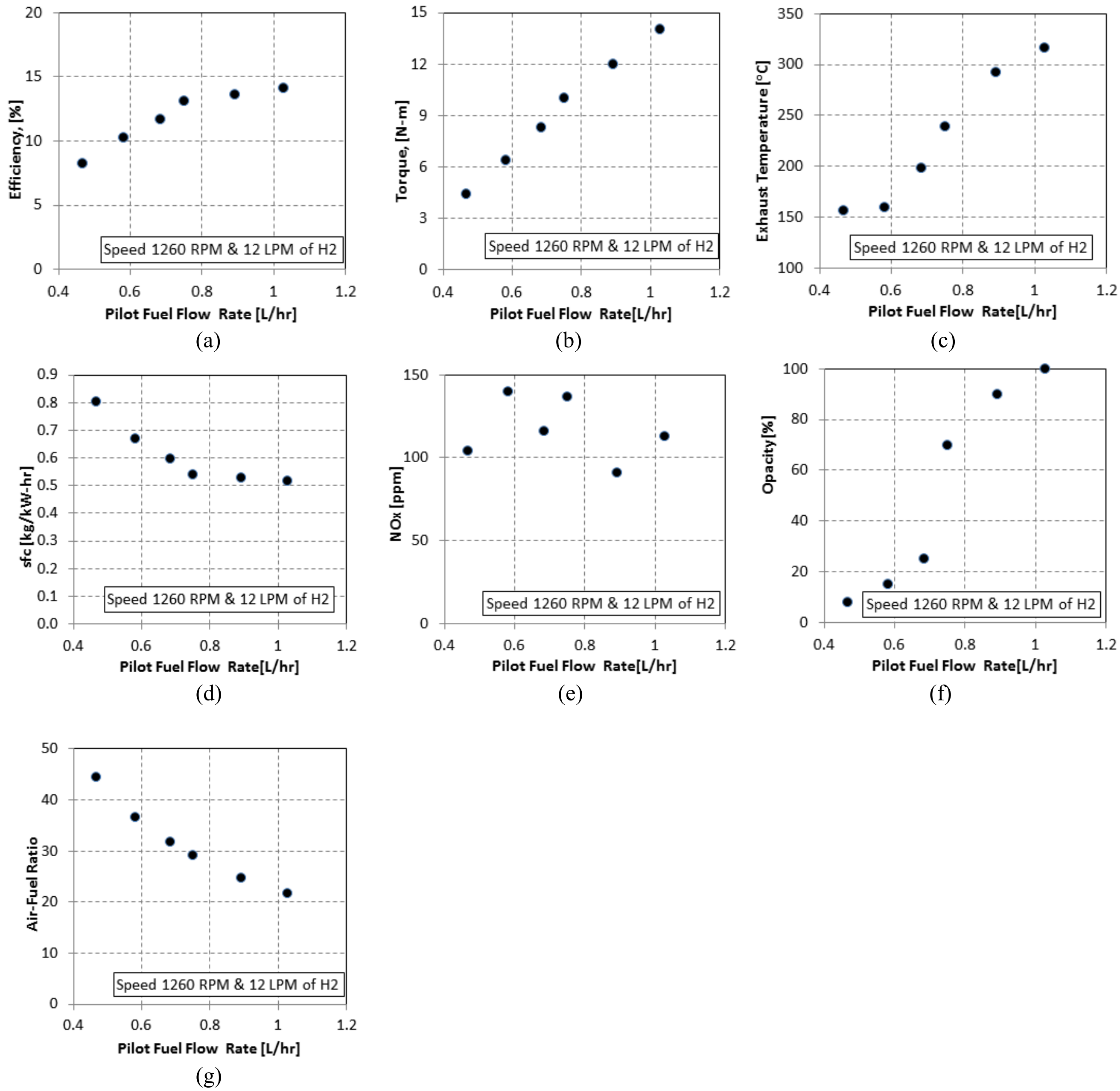

Fig. 5, Hamdan and Selim

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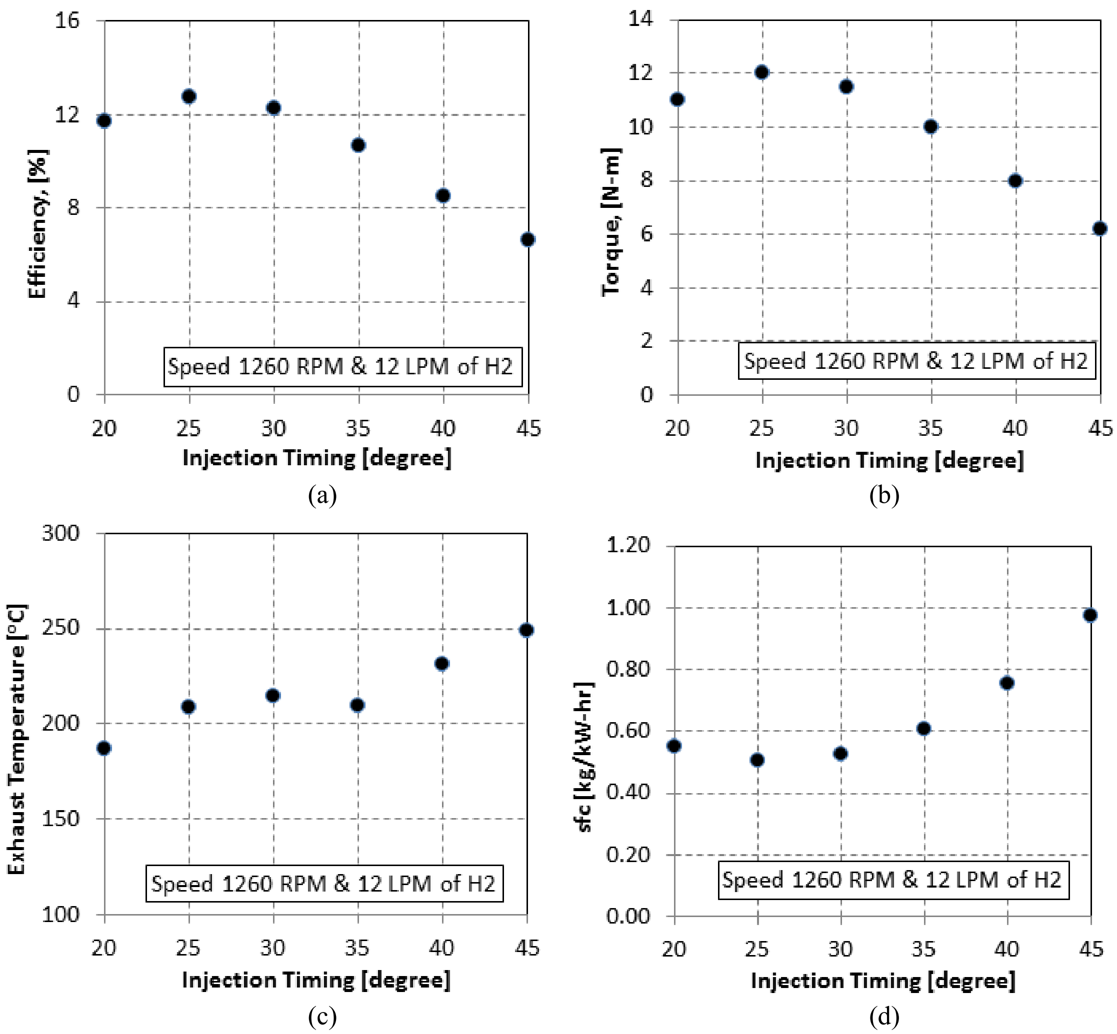

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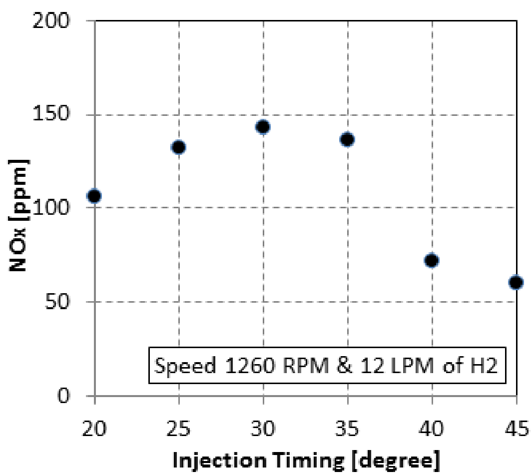

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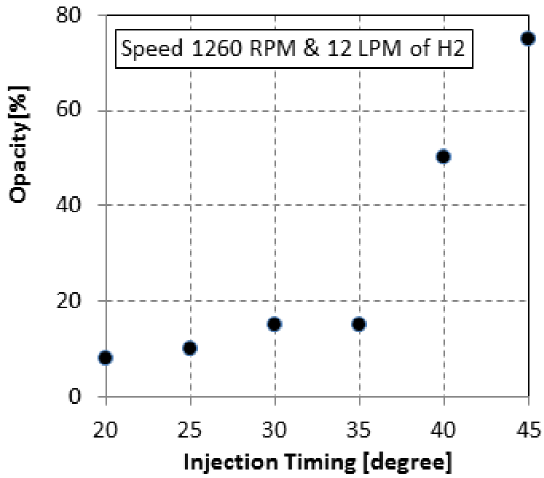

(f)

Fig. 6, Hamdan and Selim

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