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Comparative study of laser ignition and conventional electrical spark ignition systems in a hydrogen fuelled engine

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ABSTRACT

Increasing population and demand for fossil fuels is leading to rapid depletion of petroleum reserves. Hydrogen has great promise as an alternative fuel for powering next generation internal combustion engines with improved thermal efficiency and reduced emissions. Laser ignition (LI) has emerged as an efficient ignition technique for delivering superior engine efficiency with lower emissions. Use of LI to initiate combustion in an engine fuelled with hydrogen–air mixtures can greatly help in reducing emissions, improving engine performance and tackling the problem of fossil fuel depletion. In this study, a laser ignited hydrogen fuel prototype engine has been developed and comparison of its performance, emissions and combustion parameters is done with baseline data generated using conventional spark ignition (SI) system.

Introduction

In the second half of twentieth century, global population began to explode. This led to phenomenal increase in demand for conventional fossil fuels to sustain increased transportation demand of a modern society. By 2030, global population will be 8 billion [1] and its primary energy demand will cross 700 EJ/year [2]. The fossil fuel reserves are rather limited, therefore more demand for fuel will straightway translate into higher energy prices, which will lead to fuel prices led inflation world over. In addition, world is facing serious challenges such as environmental degradation and climate change, which are reportedly because of pollution and harmful emissions caused by petroleum fuelled vehicles. In order to control and arrest this environmental damage, stringent emission norms have been adopted worldwide which are becoming more and more stringent with time. Therefore to avoid the rapid depletion of fuel reserves, slow down the increase in fuel prices and for stringent emission norm compliance, researchers are exploring alternate fuel options for vehicles. These alternate fuels can be adopted in current generation engines with minimal hardware modifications. Several alternative fuel candidates such as biodiesel, compressed natural gas (CNG), ethanol, hydrogen, hydrogen-CNG mixtures (HCNG), straight vegetable oils (SVO), liquefied petroleum gas

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(LPG) have been investigated for their engine performance in order to assess their technical feasibility in current generation engines. In a quest of running the internal combustion engines with improved thermal efficiency and lower emissions, hydrogen has emerged as a prominent alternate fuel candidate with ultra-low emission potential due to its extremely clean emission spectra.

Compared to other fuels such as gasoline, diesel, compressed natural gas (CNG), biodiesel etc., hydrogen offers numerous advantages as an alternate transportation fuel such as it can be produced using various renewable primary energy sources from water (through electrolysis), from coal and biomass (through gasification), from natural gas (using steam methane reforming and other ways) [3]. It does not contain carbon, hence does not emit any greenhouse gas upon combustion. Hydrogen has higher calorific value (w/w) resulting in higher power output upon combustion, higher auto-ignition temperature, low ignition energy, higher diffusivity and higher flame propagation speed compared to gasoline. In addition to several advantages, hydrogen has few disadvantages and limitations also. Due to very small ignition energy required to initiate combustion, chances of backfire and pre-ignition is higher in hydrogen fuelled engines. Density of hydrogen is very low, therefore significantly larger volume of hydrogen is required compared to conventional fuels for the same range of the vehicle. Among these, pre-ignition and backfire remain the biggest challenges for commercialization of hydrogen engine. Researchers have used several induction techniques, injection strategies, and water induction in the intake manifold for hydrogen fuelled engine in order to eliminate/control engine backfire. For controlling NO_x emissions, EGR and/or water injection into the intake manifold have been successfully attempted.

Das and Mathur [4] controlled NO_x emissions from a hydrogen fuelled engine using different EGR rates. Experiments were performed on a carbureted hydrogen engine. They obtained significant reduction in NO_x emissions using 15% EGR. Improvement in fuel consumption was also achieved. Brake specific fuel consumption (BSFC) decreased upon increasing EGR. Increasing the spark advance increased the temperature inside the combustion chamber, which increased the NO_x emissions from the engine. Mohammadi et al. [5] investigated performance and NO_x emissions by direct injection of hydrogen in a single cylinder SI engine and varied injection timings and ignition timings. Maximum BMEP of 6.5 bars was obtained, when hydrogen was injected into the combustion chamber during the intake stroke (300° BTDC). For this case, maximum brake thermal efficiency was found to be 35% at $\Phi = 0.5$. At $\Phi = 0.7$, NO_x emissions were roughly 8000 ppm. Injecting hydrogen during early stages of compression stroke (140° BTDC) increased BMEP to 9.7 bars with brake thermal efficiency of 38.9%. This led to reduction in NO_x emissions due to lean engine operation. Varde and Frame [6] performed experiments in a single cylinder SI engine to quantify the advantages of port injection of hydrogen over carbureted induction in terms of backfire and cyclic variations. Experiments were performed at 1800 rpm and 2100 rpm engine speeds. Improvements in lean-burn limit were observed for the port injection system. Thermal efficiency of port injection system in lean region was also higher. Maximum NO_x emissions were observed with mixture strength slightly leaner then stoichiometric. Flame speed for port injection was found to be higher than carburetion engine. At 20° BTDC, backfire occurred between $\Phi = 0.9$ to 1.15 for carburetor, whereas for port injection, backfire occurred between $\Phi = 1.1$ to nearly stoichiometric mixtures.

Another technique to overcome auto-ignition and backfire from hydrogen fuelled engine is by using electrode-less ignition system such as laser ignition (LI). In LI, a pulsating laser beam is converged at a point using a converging lens, which creates plasma at the focal point of the converging lens. Plasma is formed, when the energy density at the focal point increases beyond a threshold value. Apart from controlling backfire and auto-ignition, LI has proved to be a better ignition technique for improving engine performance and emissions. LI was successfully performed for the first time in an IC engine in 1978 by Dale et al. [7]. They compared the results of LI with baseline conventional spark plug ignition system and reported that at 300 mJ/pulse laser energy, peak cylinder pressure rise was higher for LI. Minimum air-fuel ratio limit, at which LI was possible, reduced to 27.8:1 compared to 22.5:1 for conventional ignition system. At a particular engine operating condition, 17% improvement in BSFC was observed, while using 17% EGR. Herdin et al. [8] performed experiments in a large-bore engine fuelled with natural gas. They reported that increasing the in–cylinder pressure decreased the energy required for creating plasma inside the combustion chamber. Compared to conventional spark plug ignition system, ignition delay was shorter for LI. Liedel et al. [9] investigated LI using a Q-switched Nd: YAG laser with wavelengths of 1064 nm and 532 nm and a pulse duration of 6 ns in a GDI engine. Fuel consumption in LI system was significantly lower compared to the conventional ignition system. Exhaust emissions reduced by 20% for LI. By changing the wavelength of the laser, no significant effect on engine performance was observed.

Both McMillian et al. [10] and Srivastava and Agarwal [11] performed experiments using SI and LI system in a natural gas fuelled single cylinder engine. MacMillian et al. [10] performed experiments to determine misfire limit and knock limit of LI system. They reported increased misfire limit, and decreased ignition delay for LI compared to SI. Srivastava and Agarwal [11] carried out a comparative study between LI and SI. They also varied laser plasma position to further investigate the effect of plasma position on the engine performance. Superior combustion and performance parameters were obtained for LI compared to SI. Also slight increase in BSFC and power output was obtained upon moving plasma position deeper inside the constant volume combustion chamber (CVCC).

Experiments on LI of hydrogen–air mixture were performed by Srivastava et al. [12] however in a CVCC. A Q-switched Nd:YAG laser was used for the experiments. Pressure-time curves were obtained for both LI and SI by varying air-fuel ratio, while maintaining constant initial chamber filling pressure and initial chamber temperature. Variations in minimum laser pulse energy required to initiate combustion were determined by varying initial chamber filling pressure and air-fuel ratio. It emerged that peak pressure in both cases i.e. LI and SI was comparable but pressure rise rate was relatively higher for LI. Also for a fixed air-fuel ratio, MPE for lower chamber pressure was higher. MPE increased upon using leaner fuel-air mixture.

Several studies on laser ignition of hydrogen–air mixture have been carried out but almost all of them are in CVCC. This research is done to study the laser ignition of hydrogen–air mixture in a laser ignited internal combustion engine. In the research reported in this paper, comparative study between conventional SI system and LI system were carried out to investigate the technical potential of using LI system in a prototype hydrogen fuelled engine. Engine performance, emission and combustion characteristics for the two ignition systems are compared.

Experimental setup

For performing these experiments, a four-stroke single cylinder diesel engine was modified to develop a prototype hydrogen engine for this research and the modified engine was operated as a spark ignition (SI) engine. A DC dynamometer was used for loading the engine. A customized port fuel injection system was developed for injecting hydrogen directly into the intake manifold using custom-built high flow rate solenoid injector (Alternative Fuel Systems, Gs-60-05-5 H), which was specially designed for hydrogen applications. A customized triggering circuit was assembled to trigger the injector at desired crank angle position and control the injector opening duration. Since the solenoid injector was a peak and hold current type, a driver module (Alternative Fuel Systems) was used to convert the output triggering pulse into desired peak and hold current output, which opens solenoid valve of the injector. The specifications of the modified test engine are given in Table 1.

In order to measure the fuel flow rate into the combustion chamber, a mass flow meter (Emerson, CMF010M) was used. Two safety devices were designed and fabricated. A flame trap (also called water trap) was fabricated with the objective of trapping the flames propagating backwards to the hydrogen cylinder. Additionally, a flame arrestor was fabricated, which absorbs the heat and extinguishes the flames trying to propagate backward though it. In addition, fuel injection strategy in them manifold was devised such that it avoids backfire. In the injection strategy, when intake valve opens initially, only air enters into the combustion chamber and scavenges and cools the residual hot spots in the combustion chamber during the valve overlap. Colder intake air dilutes the hot residual exhaust present inside the combustion chamber instantly. Only after a time delay after intake valve opening, fuel injector is actuated, and hydrogen is injected into the intake manifold for induction into the combustion chamber. Since the intensity of hot spots is reduced by this time due to cooling, chances of backfire reduce significantly.

Table 1 – Specifications of the test engine.

Engine characteristics	Specifications
Model	Kirloskar DM-10 (modified)
Injection type	Port fuel injection
Number of cylinders	One
Ignition type	Spark ignition
Bore/Stroke	0.102 m/0.116 m
Displaced volume	0.948 L
Connecting rod length	0.232 m
Compression ratio	12:1
Inlet valve opening time	4.5° before top dead center
Inlet valve closing time	35.5° after bottom dead center
Exhaust valve opening time	35.5° before bottom dead center
Exhaust valve closing time	4.5° after top dead center
Cooling system	Water cooled

A piezoelectric pressure transducer (Kistler, 6013C) was used for acquiring the in–cylinder pressure data from the engine combustion chamber. To amplify weak charge signals generated by the pressure transducer, a charge amplifier (Kistler, 5015) was used. For acquiring pressure data from the engine combustion chamber, a high speed combustion data acquisition system (Hi-Techniques, MeDAQ) was used. A raw exhaust gas emission analyzer (Horiba, MEXA 584L) was used to measure regulated pollutants namely CO, CO_2, THC and NO.

Two different ignition systems were used for the comparative study, namely SI system and LI system. For performing experiments with SI system, a variable spark timing system (Altronic Inc., CD200) was installed. This variable spark timing system changes the spark timings using an ignition module. Schematic of the experimental setup with conventional electric spark plug is shown in Fig. 1.

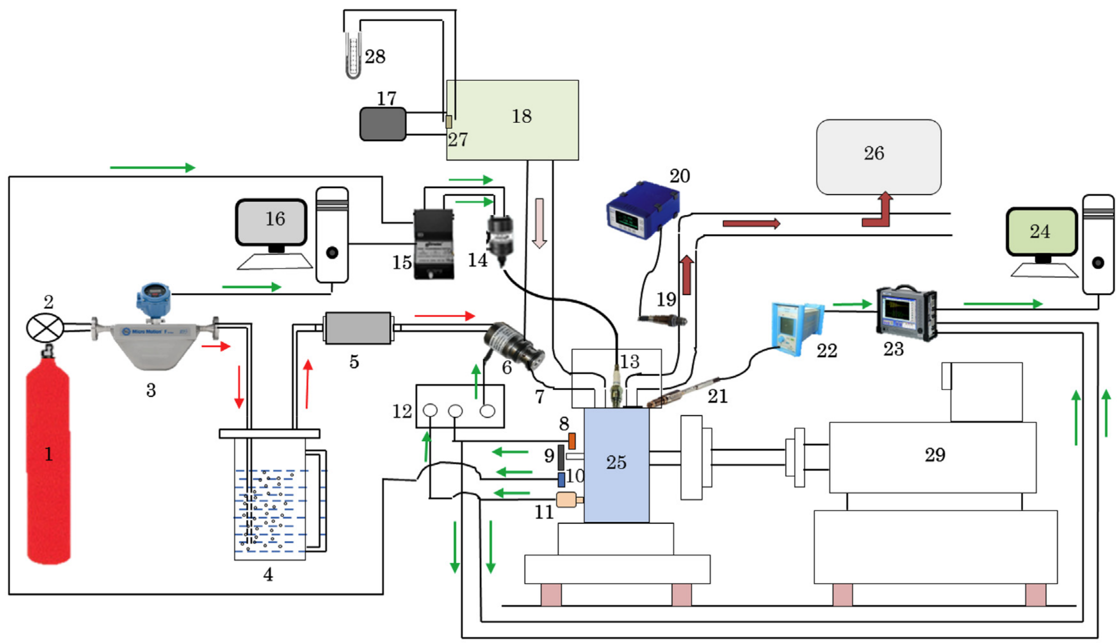
After performing the baseline experiments using spark plug, this experimental setup was modified to perform experiments with LI system. The schematic for the engine experimental setup with laser spark plug ignition system is shown in Fig. 2. For LI, a Q-switched Nd:YAG laser (Litron, Nano L-200-30) was used. A laser spark plug developed in ERL IIT Kanpur by Srivastava and Agarwal [18] was used for focusing laser beam into the combustion chamber. A separate laser triggering circuit to trigger laser at desired crank angle position was developed. For mounting the laser, a laser stand was designed in ERL, IIT Kanpur by Srivastava and Agarwal [18]. It consists of a beam reflector, and a unit having diverging and collimating lenses. Laser beam first passes through a diverging and collimating unit, where beam diameter is increased. The enlarged beam is then reflected by a beam reflector. Angle of the beam reflector is adjusted in such a way that laser beam is aligned to the axis of laser spark plug, which is installed in the engine cylinder head. Experiments were repeated again with LI system at identical engine operating conditions in order to carry out comprehensive comparative study between SI system and LI system.

Results and discussion

For comparing the performance of LI and SI systems for igniting hydrogen–air mixtures, experiments were performed under similar engine operating conditions using both ignition systems.

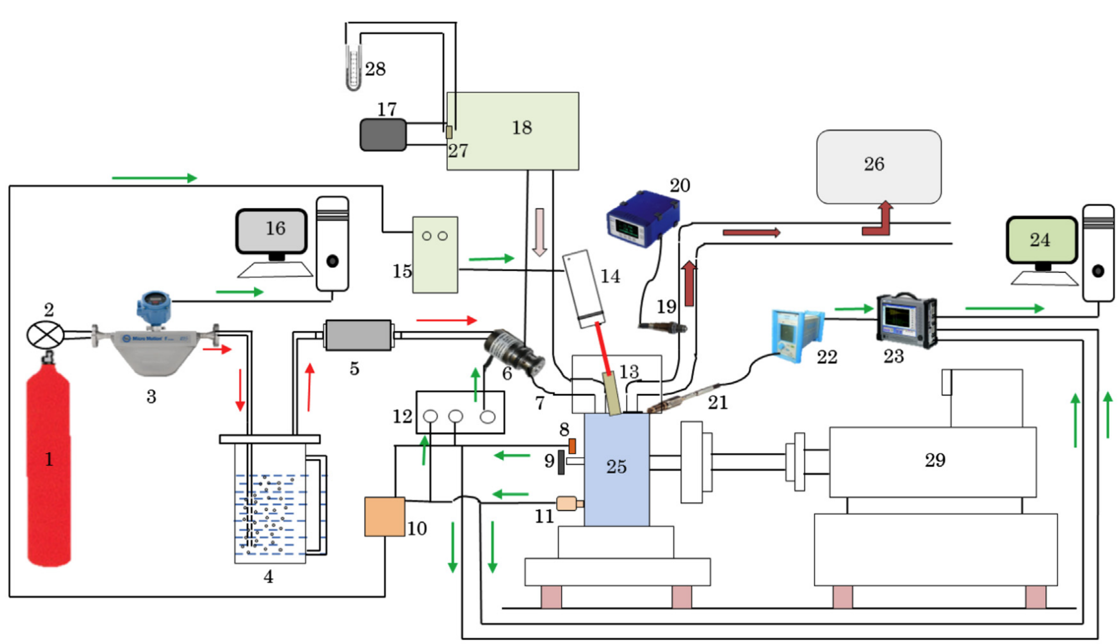
Effect of engine load variations

Experiments were performed at a fixed spark timing of 23° BTDC for both ignition systems. BMEP was varied from 0.66 to



1. Hydrogen bottle, 2.Pressure regulator, 3. Hydrogen mass flow meter, 4.Flame trap, 5.Flame arrestor, 6.Injector, 7.Intake manifold, 8.TDC sensor, 9.Triggering disc, 10.Pickup sensor, 11.Shaft encoder, 12. Injector triggering circuit, 13.Spark plug, 14.Ignition Coil, 15.Spark timing control unit, 16.Computer for ignition control and hydrogen metering, 17.Air filter, 18.Air box, 19.Lambda sensor, 20.Lambda module, 21.Pressure transducer, 22.Charge amplifier, 23.High speed combustion data acquisition system, 24.Computer for combustion data acquisition, 25.Engine, 26.Raw exhaust gas emission analyzer, 27. Orifice plate, 28. U-tube manometer, 29. DC dynamometer

Fig. 1 – Schematic of experimental setup for spark ignition system.



1. Hydrogen bottle, 2. Pressure regulator, 3. Hydrogen mass flow meter, 4.Flame trap, 5.Flame arrestor, 6.Injector, 7.Intake manifold, 8.TDC sensor, 9.Triggering disc, 10.Laser triggering circuit, 11.Shaft encoder, 12. Injector triggering circuit, 13.Laser Spark plug, 14.Laser, 15.Laser power supply, 16.Computer for ignition control and hydrogen metering, 17.Air filter, 18.Air box, 19.Lambda sensor, 20.Lambda module, 21.Pressure transducer, 22.Charge amplifier, 23.High speed combustion data acquisition system, 24.Computer for combustion data acquisition, 25.Engine, 26.Raw exhaust gas emission analyzer, 27. Orifice plate, and 28. U-tube manometer, 29. DC dynamometer

Fig. 2 – Schematic of experimental setup for laser ignition system.

3.31 bar and the engine combustion, emission and performance results are reported in following sub-sections.

Combustion analysis

Fig. 3 shows the in–cylinder pressure (P) and rate of pressure rise (RoPR) w.r.t. crank angle degree (CAD) for both LI and SI of hydrogen–air mixture with varying BMEP.

From Fig. 3, it is evident that maximum in–cylinder pressure (P_{max}) and maximum RoPR increase with increasing BMEP. Crank angle position of P_{max} shifts towards TDC with increasing BMEP. Reason for higher P_{max} and RoPR with increasing BMEP is relatively earlier start of combustion (SOC) at high loads for hydrogen–air mixtures. At higher BMEP, combustion starts earlier, which leads to shifting of pressure peak towards TDC. This results in improved combustion and higher combustion efficiency. For LI, P_{max} and RoPR is higher because of relatively earlier SOC compared to SI. Earlier SOC is related to the energy density at the ignition point. There is a basic difference in the energy absorption mechanisms between laser plasma and spark plasma. In LI, optical frequency of laser beam is higher than the plasma frequency. Therefore laser beam can propagate into the interiors of plasma, where it gets absorbed at high efficiency near the focal point [13]. SI operates at a frequency lower than the plasma frequency, therefore spark cannot propagate inside the plasma and plasma is sustained by absorption within a thin layer near the plasma surface [13]. Therefore the pressure and temperature conditions generated by plasma of LI are of the order of 10^3 bar and 10^6 K respectively [13–15], whereas for plasma of SI, they are of the order of 10^2 bar and 10^4 K [16,17] respectively. Such intense conditions with orders of magnitude higher temperatures and pressures at the plasma location in LI lead to relatively earlier SOC of the hydrogen–air mixture in the combustion chamber, leading to relatively higher combustion efficiency vis-à-vis SI. Similar results of in–cylinder pressure for LI and SI are reported by Dale [7] and Srivastava and Agarwal [18]. Higher combustion efficiency also results in higher heat release rate (HRR) and cumulative heat release (CHR) for LI compared to SI for hydrogen–air mixtures (Fig. 4).

CHR for both LI and SI of hydrogen–air mixtures is almost similar (Fig. 4). However HRR is higher for LI compared to SI. Increasing BMEP increases CHR due to higher amount of fuel inducted into the engine combustion chamber. Hydrogen quantity required for LI is lesser compared to SI for maintaining same BMEP however CHR remains almost same. This is due to higher combustion efficiency for LI, which leads to higher heat release per unit mass of fuel inducted.

Fig. 5 shows mass burn fraction (MBF) and combustion duration (CD) for both LI and SI of hydrogen–air mixtures. Three different MBF shown are namely 10%, 50% and 90% MBF. In this investigation, SOC is taken as 10% MBF and EOC as 90% MBF.

For both LI and SI combustion, increasing BMEP advances SOC as depicted by 10% MBF (Fig. 5). Also combustion starts earlier for LI compared to SI. Ignition delay decreases with

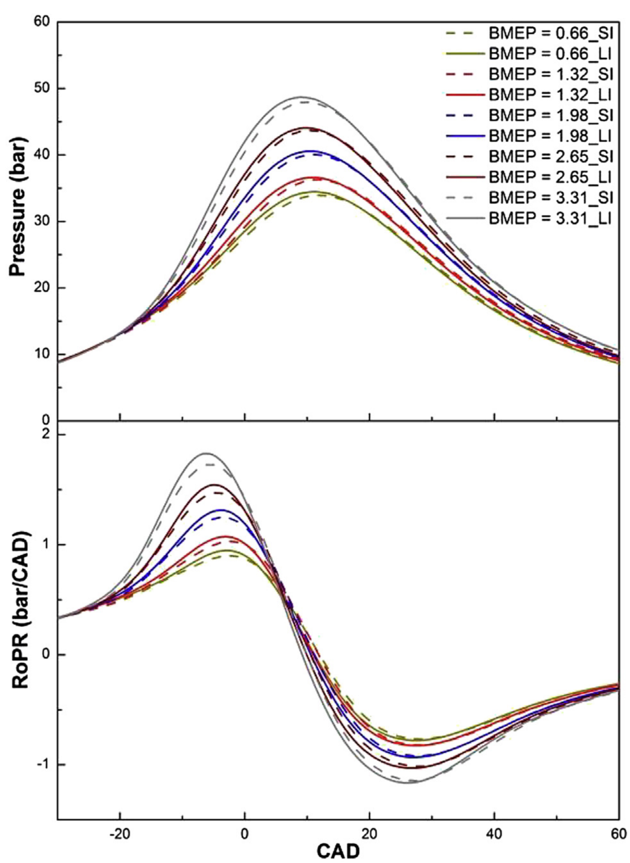


Fig. 3 – In–cylinder pressure (P) and rate of pressure rise (RoPR) for LI and SI of hydrogen–air mixture.

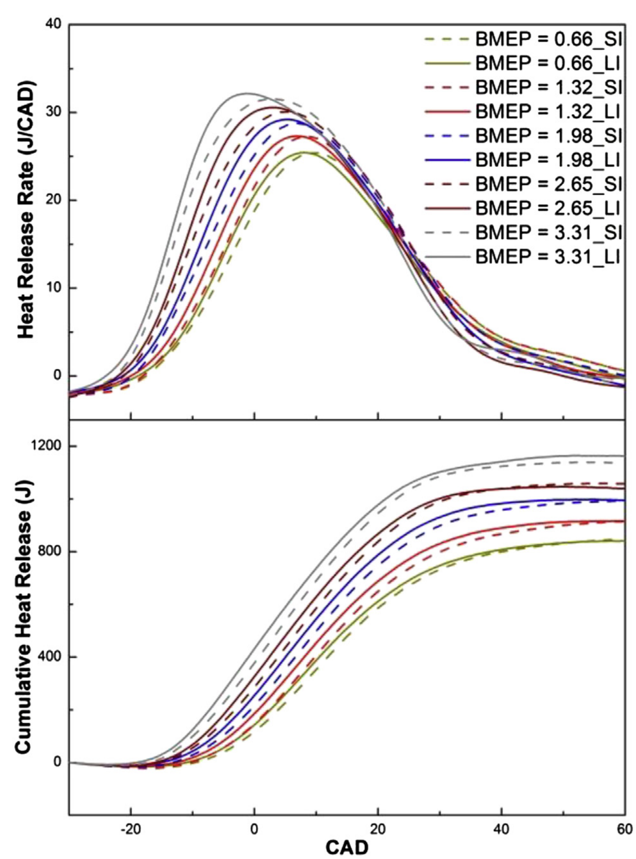


Fig. 4 – Heat release rate (HRR) and cumulative heat release (CHR) for LI and SI of hydrogen–air mixtures.

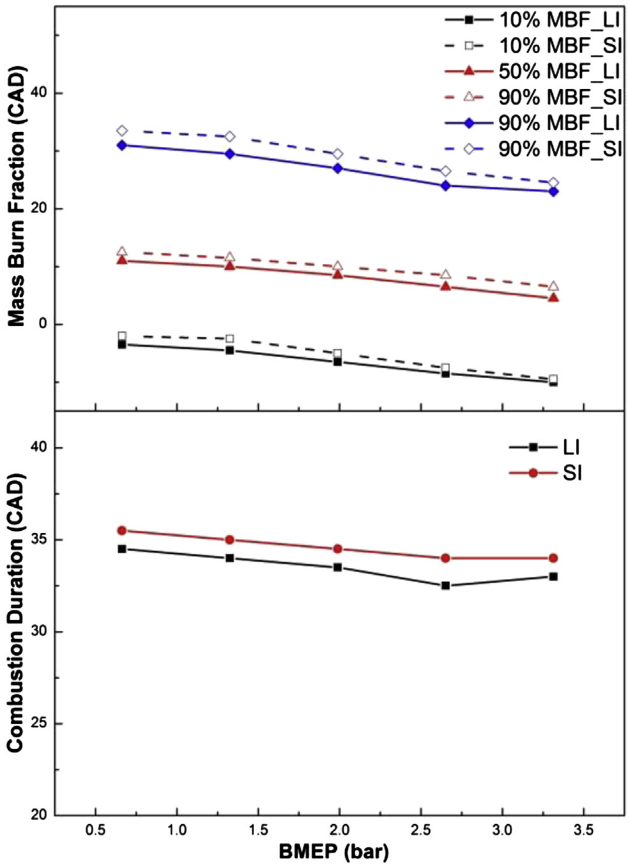


Fig. 5 – Mass Burn Fraction (MBF) and combustion duration (CD) for LI and SI of hydrogen–air mixtures.

increasing fuel quantity, causing earlier ignition of air-fuel mixture for higher engine loads. For higher BMEP, higher fuel quantity is required, which decreases relative air-fuel ratio (λ), leading to an increase in richness of the fuel-air mixture. This leads to relatively earlier SOC at higher BMEPs. Also the temperature and pressure of the ignition volume for LI is significantly higher compared to SI, which means that higher intensity energy is available at the point of ignition in case of LI, leading to relatively earlier SOC. Similar results were also reported by Herdin et al. [8].

For LI, increasing BMEP decreases CD (Fig. 5). Further increasing BMEP beyond a certain value increases the CD slightly. Similar results on CD for hydrogen were reported by Yamin et al. [19]. SI engine has three main stages of combustion namely ignition lag, flame propagation and after-burning. For lower BMEP, lesser amount of fuel is used. Therefore fuel burns completely in the flame propagation zone. Increasing BMEP increases the amount of fuel inside the combustion chamber. If mixture becomes rich then the combustion process continues till after-burning stage, which increases the CD. In this case, when BMEP is less than 3.31 bar, entire fuel burns in flame propagation stage and at BMEP 3.31 bar, mixture becomes richer, therefore combustion is extended to after-burning stage, leading to increase in CD (Fig. 5).

Performance analysis

Fig. 6 shows the variations of engine performance parameters such as brake thermal efficiency (BTE), air-fuel ratio (λ) and exhaust gas temperature (EGT) w.r.t. BMEP for both LI and SI of hydrogen–air mixtures.

BTE increases with increasing BMEP for both LI and SI. Higher BTE is related to superior combustion of hydrogen–air mixture inside the combustion chamber. Energy density generated by laser beam for ignition is more intense than the electrical spark plug. This leads to superior combustion inside the combustion chamber resulting in higher combustion efficiency and higher BTE for LI compared to SI.

Leaner the air-fuel mixture, higher is its relative air-fuel ratio (λ). Increasing BMEP decreases λ (Fig. 6). At higher BMEP, more hydrogen is required, therefore mixture becomes relatively richer hence λ decreases. For LI, λ is slightly higher because of lesser fuel requirement to produce same BMEP due to relatively superior combustion efficiency of LI. BSFC curve (Fig. 7) depicts the same.

From Fig. 6, it is evident that EGT increases upon increasing BMEP for both LI and SI combustion. For LI, EGT is relatively lower compared to SI. The reason for this behavior

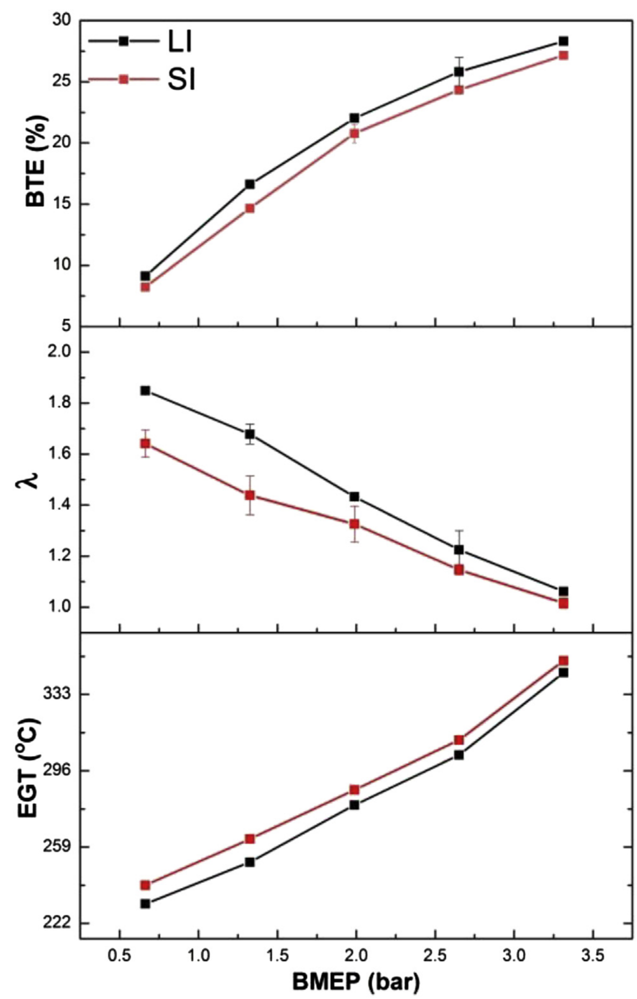


Fig. 6 – Brake thermal efficiency (BTE), relative air-fuel ratio (λ) and exhaust gas temperature (EGT) for SI and LI of hydrogen–air mixtures.

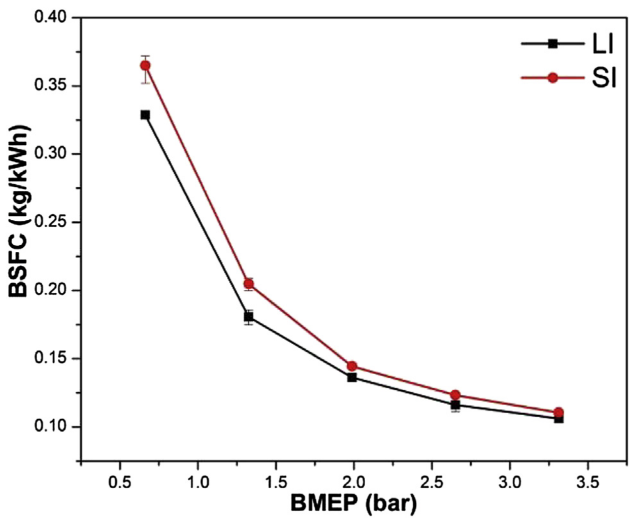


Fig. 7 – Brake specific fuel consumption (BSFC) variations for LI and SI of hydrogen–air mixtures.

is that the flame front propagation is quite different for LI combustion and SI combustion. In SI combustion, flames propagate in the form of a flame front starting from the spark plug, propagating radially outwards. Highest temperature is observed in the reaction zone i.e. which is just inside the flame front. On the other hand, for LI of hydrogen–air mixture, flame front propagation is caused by multiple explosions inside the flame kernel [20]. Therefore flame front propagates volumetrically and temperature distribution is relatively more uniform in the combustion chamber. Additionally, in case of SI, spark plug electrodes are located close to the cylinder head, therefore the flames have to travel relatively longer distance, and combustion continues for relatively longer time. This leads to heat being released deeper into the expansion stroke, therefore higher exhaust gas temperature are experienced. On the other hand, in LI, the plasma is generated relatively centrally, away from the cylinder head, therefore maximum distance of flame travel is relatively shorter therefore CD is also shorter (Fig. 5). Due to volumetric flame front propagation originating from a relatively more central location in the combustion chamber, temperature inside the combustion chamber for LI is relatively lower than SI.

Fig. 7 shows trend of BSFC for both LI and SI. It can be seen that increasing BMEP reduces the BSFC, which means that at higher engine loads, lesser hydrogen is required to produce unit power output. Also BSFC is relatively lower for LI compared to SI combustion. This is related to BTE. At higher BMEP, BTE is also higher, therefore lesser hydrogen quantity would be required to produce unit power output at higher loads. These results correlate well with the results reported by Liedel et al. [9].

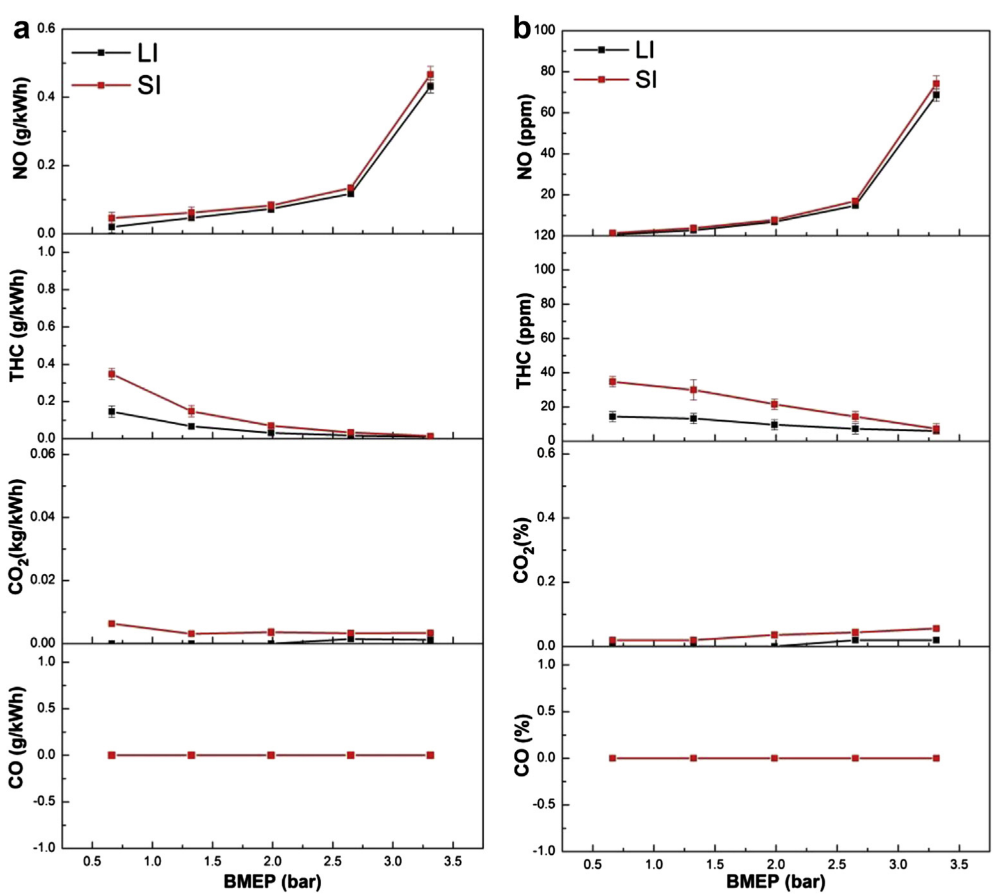


Fig. 8 – (a) Mass emissions and (b) Raw emissions for LI and SI of hydrogen–air mixtures.

Emissions analysis

Fig. 8 shows both specific emissions and raw emissions for LI and SI. NO emissions for SI are slightly higher than LI. NO emissions are generally dependent on peak temperature inside the combustion chamber. Since in case of LI, temperature is distributed more uniformly inside the combustion chamber, unburned charge is not exposed to localized extreme temperatures. For SI, extreme temperatures are available inside the flame front. Therefore tendency of NO formation is relatively higher because the combustible charge is in direct contact with the high temperature flame front. Bihari et al. [21] also reported lower NO_x emissions for LI compared to SI, which is in line with the observations of this study.

Total unburned hydrocarbons (THC), carbon dioxide (CO_2) and carbon monoxide (CO) emissions are very low in the hydrogen fuelled engine. CO emissions are almost zero and THC and CO_2 emissions are also insignificant. This is due to complete absence of carbon in the fuel. Small amount of THC and CO_2 emissions are observed because of combustion of lubricating oil, which gets thrown into the combustion chamber due to piston ring dynamics. CO_2 emission for LI is also almost zero. Therefore, all these regulated emissions are insignificant and their relative comparison is not worth.

Effect of spark timing variations

Spark timing was varied for both LI and SI in order to observe the effect of spark timings on combustion of hydrogen–air mixtures. For these experiments, a fixed value of air-fuel ratio (λ) = 1.4 was maintained for all spark timings. Four different spark timings were used in this experiment (14, 17, 20 and 23° BTDC).

Fig. 9 shows the P-θ and RoPR variations for LI and SI under varying spark timings for combustion of hydrogen–air mixtures.

On advancing the spark timing, P_{max} and RoPR increased for both the LI and SI. Combustion starts relatively earlier in the combustion chamber upon advancing the spark timing. This leads to combustion of higher quantity of charge relatively earlier in the expansion stroke, when the piston is closer to TDC and traveling downwards. This affects the combustion of hydrogen–air mixture positively [22]. Hence P_{max} increased upon advancing the spark timing. This also increased RoPR because of higher charge burning near TDC.

Fig. 10 shows the HRR and CHR for LI and SI. HRR increased upon advancing the spark timing. On advancing the spark timing, higher quantity of charge burns closer to TDC, during the expansion stroke. This leads to higher HRR for advanced spark timings. At a constant BMEP, HRR for LI is relatively higher because of higher combustion efficiency for LI compared to SI. Since the hydrogen quantity is same in both LI and SI and combustion efficiency of LI is higher, therefore cumulative heat release is higher for LI compared to SI (Fig. 10).

Fig. 11 shows MBF (10%, 50% and 90%) and CD for LI and SI at different ignition timings.

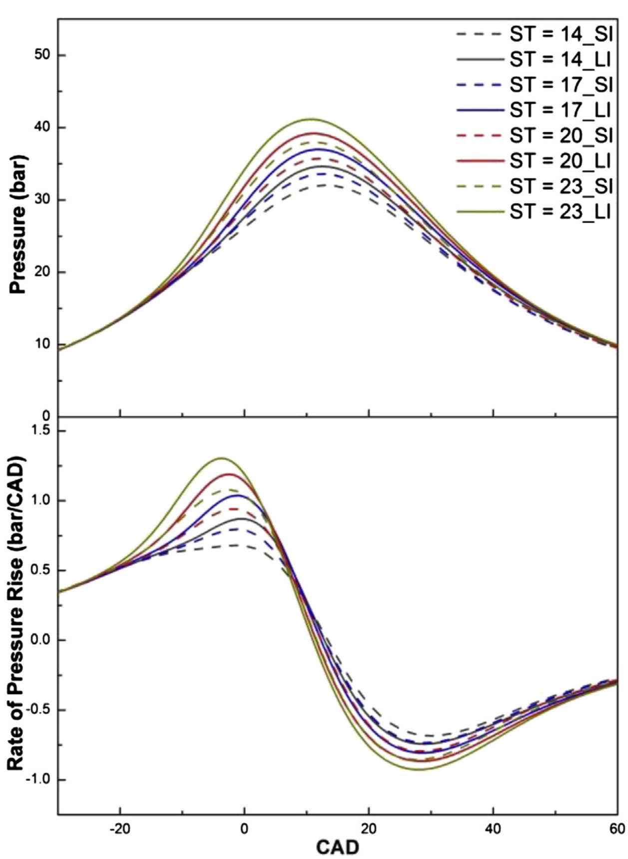


Fig. 9 – P-θ and RoPR curves for LI and SI of hydrogen–air mixtures with varying spark timings.

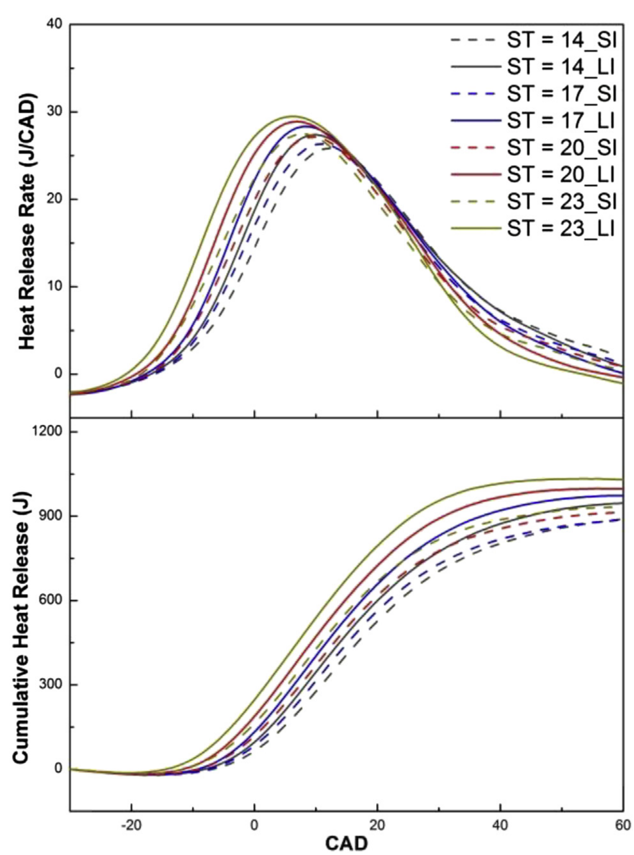


Fig. 10 – HRR and CHR for LI and SI of hydrogen–air mixture with variable spark timings.

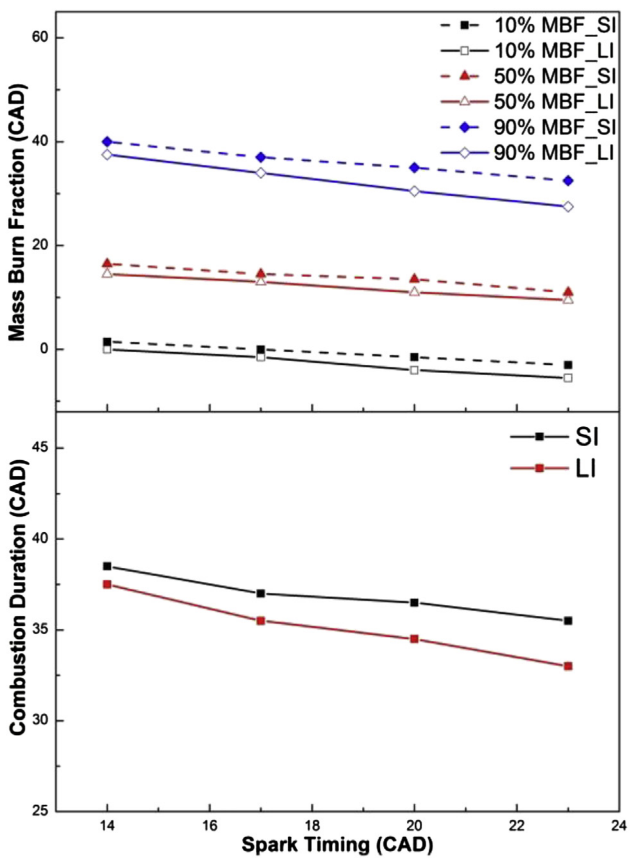


Fig. 11 – MBF and CD for LI and SI of hydrogen–air mixture with variable spark timings.

As seen from 10% MBF curve, combustion starts relatively earlier for LI compared to SI for fixed spark timing due to higher energy density at the point of ignition. From CD curve (Fig. 11), it can be seen that advancing the spark timing shortens CD. Also for LI, CD is shorter compared to SI. On advancing the spark timing, combustion predominantly takes place closer to TDC. Due to higher compression near TDC position, flame propagates quickly, shortening CD. For LI, combustion starts relatively earlier, therefore CD shortens by a greater degree for LI compared to SI at the same spark timing.

Conclusions

This study was carried out to investigate the advantages of laser ignition (LI) system over conventional electrical spark ignition (SI) system for a hydrogen–air mixture fuelled prototype engine. Following conclusions can be drawn from the experimental results:

- At all loads, P_{max}, RoPR, HRR in case of LI were relatively higher compared to SI.
- Combustion started relatively earlier for LI compared to SI. Combustion duration (CD) for LI was also relatively shorter than SI.
- Engine performance parameters for LI were superior than SI.
- Exhaust gas temperature for SI were relatively higher, leading to higher NO formation. All other regulated emissions were insignificant in both cases, due to complete absence of carbon in the test fuel.
- At a fixed λ, advancing the ignition timing improved combustion parameters for both LI and SI. At the same ignition timing, combustion parameters for LI were found to be superior to SI.

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