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ORIGINAL PAPER

Conversion of a single-cylinder internal combustion engine to dual-mode homogeneous charge compression ignition engine

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Abstract

Homogeneous charge compression ignition (HCCI) technology has been a forerunner in improving efficiency and reducing emissions in conventional internal combustion engines. Despite significant research activity in the past few decades, engines operating on HCCI technology have not been commercially successful owing to practical engineering challenges. The current study attempts to convert a single-cylinder agricultural diesel engine to a gasoline HCCI mode. Dual mode of operation was employed to overcome the shortcomings of HCCI technology. The control algorithms were designed to switch between the gasoline HCCI mode and the diesel DI (direct injection) mode depending on load. A numerical simulation strategy was deduced to determine the initial experimental conditions. The HCCI mode could not be sustained above 40% load with the current control strategy which corresponds to 1.49 kW of brake power. In HCCI mode, a brake thermal efficiency of 23%, NO_x emissions of 1.4 g/kW h, and CO_2 emissions of 2200 g/kW h were achieved which was an improvement of approximately 15%, 80%, and 30%, respectively, at comparable loads in DI mode. Reduction in engine-out NO_x is an attractive feature of HCCI engines. However, at lower loads CO and HC emissions of 1600 g/kW h and 46 g/kW h were achieved which were higher than conventional diesel DI mode and would be tackled by after-treatment systems which are economical as compared to after-treatment of other air pollutants. Overall, the technology was found to be clean and economically viable. Modifications to achieve dual-mode operation holds a potential to commercialize it, as it can be cross-deployed in any engine of similar class, which is an important feature considering the impact of these engines on air quality and economy of a country.

Keywords HCCI · Gasoline · Emissions · Efficiency · CHEMKIN · PFI

Abbreviations

ATAC	Active thermo-atmosphere combustion
BTE	Brake thermal efficiency (%)
CAI	Controlled auto-ignition
CI	Compression ignition
CIHC	Compression ignited homogeneous charge
CO	Carbon monoxide (g/kW h)
CR	Compression ratio
DAS	Data acquisition system
DI	Direct injection
ECU	Engine control unit
EGR	Exhaust gas re-circulation
GDI	Gasoline direct injection
HCCI	Homogeneous charge compression ignition
IC	Internal combustion
MPFI	Multi-point fuel injection
NDIR	Non-dispersive infrared
NMEP	Net mean effective pressure (bar)
NO_x	Nitrogen oxides (g/kW h)
PID	Proportional integral derivative
PFI	Port fuel injection
PM	Particulate matter (g/kW h)
ROHR	Rate of heat release (J/deg)
RPS	Regulated power supply
SI	Spark ignition
SOC	Start of combustion
TDC	Top dead center
UHC	Unburned hydrocarbon (g/kW h)
VCR	Variable compression ratio

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Introduction

Internal combustion (IC) engines play a crucial role in various applications in the field of transportation and captive power generation such as automobile engines, marine engines, aircraft engines, and generator sets. In developing countries such as India, petroleum is the second largest source of energy preceded by coal, out of which about 40% energy is derived from diesel fuel (Central statistics office 2017). Moreover, high dependence on import of petroleum puts emphasis on developing an efficient and sustainable mode of engine operation. A large number of applications of IC engines as a source of energy ensure that a slight improvement in the efficiency and emissions would have large-scale implications on energy consumption and quality of air of a country. A report from World Energy Council reveals that in 2014 out of total primary energy supply (TPES) 81% was derived from fossil fuels. Moreover, 52% of TPES was obtained from oil and gas. Thus, the share of IC engines as a source of energy is significant and the impact of savings obtained from improvement in efficiency is substantial. The projected share of oil and gas in TPES in 2030 and 2060 is 56% and 52%, respectively, indicating the importance of developing technologies with improved efficiency of IC engines (Melany and Gerald 2016). Additionally, the upcoming stringent regulations after Euro VI are expected to lower the limits of carbon dioxide (CO_2) emissions, placing even more focus on engine efficiencies. It would also be necessary to further reduce the emission of noxious gases such as nitrogen oxides (NO_x) and carbon monoxide (CO) beyond those given in Table 1. In 2016, World Health Organization (WHO) reported that 91% of the world population was living in places where the air quality guidelines' levels set by WHO were not met. WHO has emphasized on policies and investments supporting cleaner transport, energy-efficient homes, power generation, industry, and better municipal waste management to reduce key sources of outdoor air pollution (WHO 2018). This has led to increasing focus on research in the field of newer combustion technologies for IC engines.

IC engines have conventionally been classified into two categories based on the method of ignition: spark ignition (SI) and compression ignition (CI). The compression ratio (CR) in SI engines is limited by knocking and is between 8 and 12, thus reducing efficiency. Moreover, high temperatures in the flame front lead to significantly high NO_x emissions in SI engines (Ryan and Callahan 1996). CI engines operate with high compression ratio exhibiting excellent efficiency. However, non-homogeneity of the charge leads to high emissions, necessitating expensive after-treatment systems (Pundir 2010).

Modifications in these conventional principles can aid in cleaner and more economical engine performance. Various advanced low-temperature combustion (LTC) modes may be adopted to achieve future emission norms with minimal after-treatment devices resulting in reduced cost of ownership (Dev et al. 2017). Comparison of engine performance in various LTC modes such as HCCI, reactivity-controlled compression ignition (RCCI), and premixed charge compression ignition (PCCI) reveals that the HCCI mode of combustion results in near-zero smoke emissions, lower HC and CO emissions compared to RCCI and PCCI modes (Murugesa and Anand 2017). HCCI combustion is a technology which provides a unique solution, promising to exploit the benefits of SI and CI engines, while avoiding the peak temperature zones which lead to higher NO_x formation. HCCI combustion is termed homogeneous as auto-ignition is expected to occur simultaneously at numerous locations within the combustion chamber, leading to efficient combustion of the fuel–air mixture without the formation of a distinct flame front as in SI or locally inhomogeneous mixtures as in CI engines (Pundir 2010). This reduces the local peak temperatures inside the chamber. NO_x formation increases drastically above 1800 K, and the capability of containing the peak combustion temperature below this limit in HCCI ensures ultra-low NO_x emissions (Pipitone and Genchi 2016). Also, as the mixture is uniform and lean, the formation of particulate matter is suppressed.

HCCI faces two major challenges. First, there is no direct control over the start of combustion (SOC). Second, high load operation of HCCI engines causes excessive knocking owing to extremely high heat release rates, while the low load operation of HCCI engines is limited by partial misfire (Zhao 2007). Hence, HCCI is operable only in part-load conditions and there is a need for smooth transition of the engine from HCCI to conventional SI or CI mode at higher loads. Additionally, during HCCI combustion, oxidation of CO to CO_2 is expected to be suppressed due to lower overall combustion temperature which suggests higher CO emissions (Dec 2002). A port fuel injection (PFI) system, wherein the fuel is injected into the intake manifold and a homogeneous mixture enters the combustion chamber during the intake stroke, has been

Table 1 European Stage V Non-Road Emission Limits (The International Council on Clean Transportation (ICCT) 2016)

Engine category	Equipment type	Power range (KW)	Engine type	CO (g/KWH)	HC (g/KWH)	NO_x (g/KWH)	PM (g/KWH)
NRE-v-1 NRE-c-1	Other non-road mobile machinery	$0 < P < 8$	CI	8.00	$HC + NO_x \leq 7.50$		0.40

widely used for homogeneous mixture preparation to run an engine in HCCI mode (Aoyama et al. 1996). However, due to the higher residence time of the unburnt fuel in the crevices of the combustion chamber typical of engines using a PFI system, unburned HC emissions are expected to be higher (Hasan and Rahman 2016). Though the HC and CO emissions may be higher in HCCI mode, these can be dealt with by using exhaust after-treatment systems.

Zhao (2007) and Yao et al. (2009) have extensively reviewed the published literature on HCCI combustion. They have discussed challenges faced by various researchers in achieving HCCI combustion using various techniques along with the different control strategies used by them to tackle these challenges. Use of gasoline as fuel with port fuel injection and intake air heating was found to be a straightforward approach to achieve HCCI combustion.

The first notable contribution in gasoline HCCI research came in 1979 from Onishi et al. (1979). Controlled auto-ignition (CAI) on a two-stroke petrol engine was studied and termed as active thermo-atmosphere combustion (ATAC). Najt and Foster (1983) were the first to demonstrate HCCI on a 4-stroke engine by using fuel mixtures representative of gasoline which they termed as compression-ignited homogeneous charge (CIHC) combustion.

Aoyama et al. (1996) observed the in-cylinder combustion of the premixed lean mixture using a high-speed camera for an engine with a CR of 17.4 with gasoline injected into the inlet port. Oakley et al. (2001) achieved gasoline HCCI combustion in a 4-stroke VCR engine with a CR of 11.5 and elevated intake temperature (593 K). Yang et al. (2002) built an HCCI–SI dual-mode engine and optimized the engine performance using waste thermal energy in the exhaust gases and coolant to heat the intake air as well as cam phasing to control the valve overlap and the effective compression ratio. Dilution of the charge by re-introduction of exhaust gas re-circulation (EGR) was found to increase the burn duration by 34% expanding the operating range of HCCI engines (Lawler et al. 2017). Large eddy simulations were performed to analyze the effects of thermal stratification of the charge. Heat release rates were found to depend on the bulk thermal stratification than spatial distribution of thermal stratification (Sofianopoulos et al. 2018). Investigations on various compression ratios using primary reference fuels (PRFs) indicate that the efficiency increased with increase in compression ratio owing to the expansion work. However, the increase in efficiency is offset by higher heat transfer loss and lower ratios of specific heats at higher compression ratios (Yang et al. 2018).

Gasoline injected by the PFI system, along with the in-cylinder injection of secondary fuel to overcome the difficulty in combustion phasing control, has also been employed to demonstrate HCCI combustion by various researchers. Kim et al. (2004) studied the combustion and emission characteristics of a partial HCCI engine with gasoline PFI and diesel DI. Kim and Lee (2006) compared the performance of gasoline, diesel, and *n*-heptane as the premixed fuels in a partial HCCI engine with a direct injection of diesel. They found that gasoline as premixed fuel was most effective in reduction in NO_x and soot emissions. Yang et al. (2011) employed methanol stratification to a PFI-based single-cylinder gasoline HCCI engine for extending the high load limit by improving control over combustion phasing.

HCCI with PFI has also been demonstrated with model fuels, gaseous fuels, alcohols, and their blends such as *n*-heptane (Liu et al. 2011), acetylene (Swami Nathan et al. 2010a), and wet ethanol (Mack et al. 2009), respectively. It has also been demonstrated with various mixtures, such as *iso*-octane and *n*-heptane (Collin et al. 2003), ethanol and *n*-heptane (Olsson et al. 2003), and bioethanol and water (Megaritis et al. 2008). Table 2 depicts the summary of research work on HCCI with various fuels. An extensive review and benchmarking of alternative fuels in conventional and advanced engine concepts with emphasis on efficiency, CO_2, and regulated emissions (Tuner 2016) reveals that fossil fuels cannot be replaced by a single alternate fuel and it is advisable to use their combinations. Moreover, cost and availability of alternate fuel are the major driving factors since the relative variation in benefits offered by them is small. A summary of merits and demerits of alternate fuels is provided in Table 3. Coskun et al. (2018) investigated the effect of two-stage direct injection strategies in a DI–HCCI engine fuelled with ethanol–gasoline blends. Experimental results showed that the variation of second injection timing and injection ratios had significant effect on peak pressure as compared to the variation of first injection. Turkcan et al. (2018) performed similar experiments and concluded that the combustion phase may be most effectively controlled by changing the second fuel injection timing. Moreover, addition of alcohol to gasoline results in reduced NO_x emissions and stable maximum cylinder gas pressure.

Majority of the investigations in HCCI are performed on stationary engines with an objective of extending the know-how to automotive engines. The significance of application of variable valve timing (VVT) and various fuel injection strategies was realized in the early 2000s. Since then, the research has been focused on achieving favorable conditions in combustion chamber using various fuels and their blends. However, claims regarding commercialization of HCCI were not found owing to cost, availability, and viability of alternate fuels.

With the current state of research, HCCI seems to be a promising candidate for stationary constant speed engines than automotive engines. It was realized that part-load operation of HCCI using gasoline as fuel has been found to be better than both SI and CI systems in terms of NO_x and PM emissions. Efficiencies better than SI engines and

Table 2 Summary of research work of HCCI with various fuels

Engine	Fuel	Key objectives/findings	References
1 Cylinder, 4 stroke, 825 cc, CR 12.4, 600 RPM, optical access	*n*-heptane	Effect of shape of various combustion chambers on temperature inhomogeneity is studied. Combustion process can be modulated by turbulent intensity in combustion chamber	Liu et al. (2011)
1 Cylinder, 4 stroke, 553 cc, CR 16, 1500 RPM	Acetylene	Engine operated at different power outputs with change in charge temperature and EGR. NO and smoke level was low, HC level was high, and BTE was better than CI mode	Swami Nathan et al. (2010a)
4 Cylinder, 4 stroke, 1900 cc, CR 17, 1800 RPM	Wet ethanol	Effect of ethanol–water fraction from 100% ethanol to 40% ethanol in water on engine operating limits was studied. Increase in water fraction leads to incomplete combustion, excessive intake temperatures, and high HC and CO emissions	Mack et al. (2009)
1 Cylinder, 4 stroke, 500 cc, CR 12, 1200 RPM, optical access	*iso*-octane and *n*-heptane	OH- and formaldehyde-LIF signals were measured simultaneously. 20 CAD BTDC low-temperature reactions are present and formaldehyde is formed. Thereafter, OH is formed in the areas from which the formaldehyde has disappeared and the OH signal is present to some 20 CAD ATDC	Collin et al. (2003)
6 Cylinder, 4 stroke, 11,705 cc, CR 18, 600-1600 RPM, turbocharged	Ethanol and *n*-heptane	Effect of cooled EGR on multi-cylinder HCCI engine was studied. CO, HC, and NO_x emissions were found to decrease with EGR compared to lean burn. Combustion efficiency, based on exhaust gas analysis, increases with EGR due to lower emissions of CO and HC	Olsson et al. (2003)
1 Cylinder, 4 stroke, 447 cc, CR 12.5, 1500 RPM	Bioethanol and water	Effect of inlet valve timing and water blending on bioethanol HCCI combustion using forced induction and residual gas trapping was studied. Significantly retarded or advanced inlet valve event decreases the required lambda for stable combustion compared to the optimal timing, resulting in potentially higher NO_x emissions Increasing the water content to 20% drastically reduces the available load range and lambda required for combustion	Megaritis et al. (2008)

Table 3 Summary of merits and demerits achieved by alternate fuels (Tuner 2016)

Alternate fuels	Merits and demerits
Rapeseed methyl ester, ethanol, methanol, dimethyl ether (DME), methane gases	Reduced green house gas (GHG) emissions.
	Reduced health risks such as cancer and negative impact on environment
Fatty acid methyl ester (FAME), hydro-treated vegetable oil (HVO)	Alternative to diesel fuel
	Reduced GHG and soot
	No improvement in efficiency
Alcohols	Reduced GHG and regulated emissions, improved engine efficiency
	Require anti-corrosion measures
Compressed natural gas (CNG), biogas, methane	Reduced GHG

comparable to CI engines can also be achieved using gasoline HCCI. But there is a need for developing efficient control systems and duel-operation systems to extend operation to full loads. In this work, a CI engine is modified to operate in dual mode (gasoline HCCI and diesel DI) on the go without stopping the engine, which seems to be the best way to achieve better efficiency, lower emissions, and overcome the shortcomings of HCCI mode of operation. Port fuel injection was preferred as a method of injection over direct injection owing to better homogeneity offered by a PFI system. The engine modified in this work is a stationary, constant speed, diesel CI engine which has wide application in agriculture and industries. Such engines are quite relevant in developing economies. Hence, it is ensured that the nature and the extent of the modification are easily applicable to such engines. Building on our previous work (Awate et al. 2015) and using a custom-made engine control unit (ECU) (Jeeva et al. 2015), this study presents successful employment of a load-based rpm control system on a gasoline HCCI engine with a PFI system and compares its performance and emissions characteristics with conventional diesel mode of combustion. The overall efficiency of the system was improved by approximately 8% through this technology, additionally leading to cleaner operation of the engine by reducing air pollution in terms of CO_2 and NO_x. Since these engines are used widely, commercialization of the modifications stated in the work may exploit their reach to achieve greater impact on air quality and economy of a country. The modifications applicable to this engine may be cross-deployed to any engine of similar class.

Numerical investigations

Preliminary numerical investigations were carried out using CHEMKIN (Reaction Design 2013) to determine the values of parameters such as CR, inlet air temperature, and equivalence ratio that would lead to successful implementation of HCCI combustion. A homogeneous zero-degree reactor was considered for simulations, wherein properties of the reactants are held uniform throughout the volume. It idealizes the HCCI combustion process as one occurring in a variable volume fixed-mass reactor (Awate et al. 2013). The engine parameters utilized in the simulations were identical to the experimental setup being used. An adiabatic model was selected to simplify the analysis. This was justified since this was only a preliminary analysis for the selection of initial parameters. A surrogate fuel (50% *iso*-octane + 35% toluene + 15% hexene) was used as a replacement for gasoline fuel for the simulations (Mehl et al. 2011). A chemical kinetic mechanism considering 1389 species and 5935 reactions as reported by Mehl et al. was used in this study. The thermodynamic parameters used were obtained from the Lawrence Livermore National Laboratory database (Mehl et al. 2011). The results from the numerical simulations are briefly described in this section.

Effect of compression ratio

The first set of simulations was conducted to estimate an ideal CR to initiate the experimental studies. Figure 1a shows the behavior of the in-cylinder pressure with varying CR with an inlet temperature and an equivalence ratio of 423 K and 0.3, respectively, for all simulations. The mechanical limit of pressure for the engine used for experiments was 80 bar for continuous operation. It was observed that a lower CR of 12 resulted in engine misfire, while higher CRs such as 18 and 20 resulted in very high peak pressures which could lead to serious damage to the engine. A CR of approximately 14 was selected to be ideal as the initial choice for the experiments.

Effect of inlet air temperature

The second set of simulations was conducted to identify the initial value of inlet temperature for the CR of 14. Figure 1b

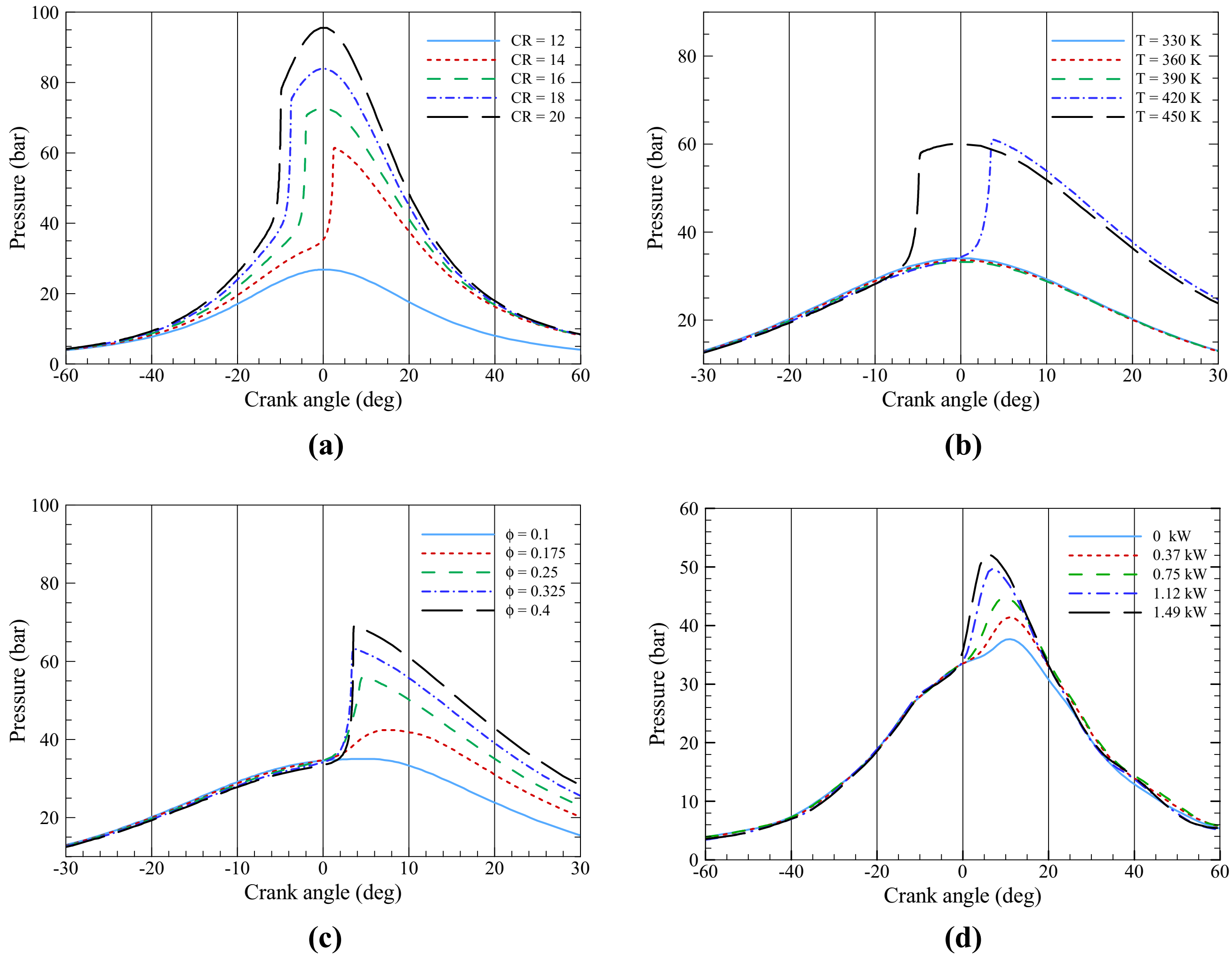

Fig. 1 Variation of the in-cylinder pressure trace with CR (**a**), with inlet air temperature (**b**), and with equivalence ratio (**c**) obtained from simulations; **d** Variation of the in-cylinder pressure trace with load obtained from experiments

shows the behavior of the in-cylinder pressure with varying inlet air temperature for a compression ratio and equivalence ratio of 14 and 0.3, respectively, for all the simulations. It was observed that the engine misfired at lower inlet air temperatures such as 330 K, 360 K, and 390 K, while at a higher inlet air temperature of 450 K, combustion was significantly advanced which was undesirable. Hence, the initial choice for the inlet air temperature was estimated to be 420 K.

Effect of equivalence ratio

Once the CR and the inlet temperature were set to 14 and 420 K, respectively, the effect of equivalence ratios was estimated to determine the performance of the engine at various loads. Figure 1c shows the behavior of the in-cylinder pressure with varying equivalence ratios, representing different loads. It was observed that the engine misfired at the lowest equivalence ratio of 0.1. An initial value of 0.25 was estimated to be a suitable choice for the preliminary experiments.

In conclusion, a combination of a CR of 14, an inlet temperature of 420 K, and an equivalence ratio of 0.25 was selected to be the starting point for the experiments of gasoline HCCI combustion.

Experimental setup

Engine specifications

Experiments were performed on a commercially available agricultural single-cylinder 4-stroke single-speed diesel engine manufactured by Kirloskar Oil Engines Ltd. The engine specifications are stated in Table 4.

The engine has a design CR of 16.5. It was modified into a variable compression ratio (VCR) engine by M/S

Legion Brothers. The CR may be altered by changing the clearance volume using an adjustable setting provided on the top of the cylinder head. The engine was loaded by an adjustable eddy current dynamometer (Powermag). In-cylinder pressures were measured by a pressure transducer (Cityzen) with an operating range of 0–200 bar. Crank angles were measured by using an encoder (Kübler, Sendix 5000) with an operating range of 0–6000 rpm. A data acquisition system (DAS) was installed to collect data from these sensors for further analysis. The pressure–crank angle data were logged for 50 consecutive cycles and processed appropriately for analysis. A calibrated multi-gas analyzer (AVL Ditest Gas 1000) was used to measure exhaust gas emissions such as NO_x, CO, CO_2, and UHC. The CO, CO_2, and UHC emission measurements were based on the non-dispersive infrared (NDIR) principle, while the NO_x emission measurement was based on the electrochemical principle.

Table 4 Engine specifications

S. No.	Parameter	Value
1	Make	Kirloskar Oil Engines Ltd.
2	Model	AV1
3	Bore	80 mm
4	Stroke	110 mm
5	Displacement	553 cc
6	CR	16.5:1
7	Rated power	5 HP at 1500 rpm
8	Cooling	Water cooled

Modifications for the implementation of HCCI combustion

Multiple modifications were implemented to run the engine interchangeably in the diesel DI mode as well as in HCCI mode as shown in Figs. 2 and 3. The descriptions of the labels used in Figs. 2 and 3 are stated in Table 5.

The modifications made to the engine are discussed below.

Intake air heating system

A heating system consisting of 2 electric heaters, 3.5 kW each, was installed to preheat the intake air. The heaters were placed in a rectangular casing, which was provided with baffles to increase the heat transfer from the heaters to the intake air. An on–off control system was used to maintain the intake air temperature at the desired value. This temperature was selected as 433 K for a CR of 14.4 based on the preliminary results obtained from the analysis using CHEMKIN as discussed in Sect. 2. Wall wetting caused by condensation of fuel droplets on the wall near the injector location may pose a problem in engines with PFI system as it can cause significant changes in the amount of fuel inducted into the engine in the subsequent cycles. A surface heater was installed just after the injector to avoid wall wetting. The temperature of the surface heater was set to 393 K. A thick layer of glass wool was used for insulating the heater casing and connecting pipes present in the intake air system. The temperatures of the inlet air, exhaust gases, and cooling water were monitored using thermocouples installed at pertinent locations. Internal EGR was not used, as it was

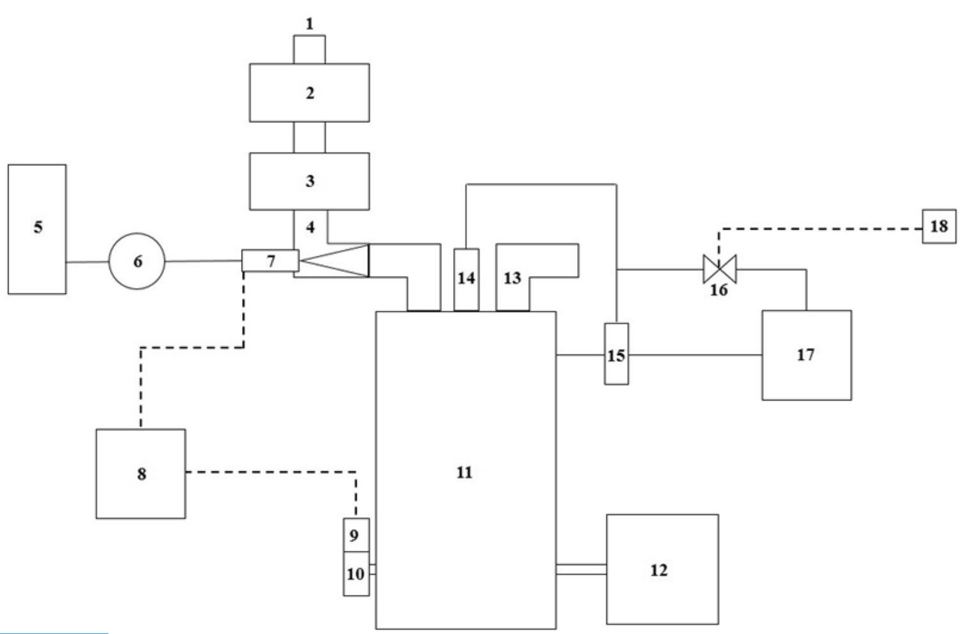

Fig. 2 Diesel engine modified for gasoline HCCI operation (Awate et al. 2015)

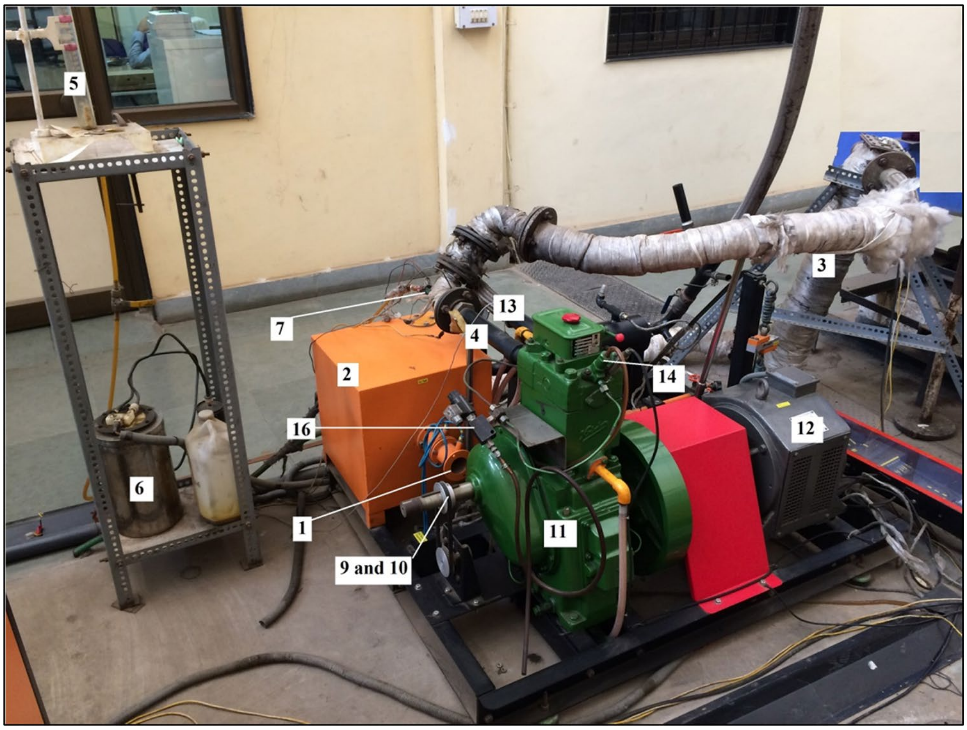

Fig. 3 Diesel engine modified for gasoline HCCI operation (actual image)

Table 5 Description of labels for Figs. 2 and 3

S. No.	Name	S. No.	Name
1	Air intake	10	Modified gear on cam shaft
2	Surge tank	11	Kirloskar CI engine
3	Electric heaters	12	Dynamometer
4	Inlet manifold with surface heater	13	Exhaust manifold
5	Gasoline burette	14	Diesel injector (mechanical)
6	Gasoline pump	15	Diesel pump (mechanical)
7	Gasoline port fuel injector	16	Diesel bypass with solenoid valve
8	Engine control unit (ECU)	17	Diesel storage tank
9	Magnetic pickup unit	18	Switch to operate bypass valve

reported that internal EGR does not reduce the intake air heating requirement significantly (Kozarac et al. 2014).

Fuel delivery system

The fuel delivery system in the DI mode running on diesel consisted of a storage tank, a mechanical pump, a mechanical injector, and a mechanical governor. The mechanical governor was equipped to control the rpm of the engine to the desired value in the range of 1400–1700 in the DI mode. A separate fuel delivery system for gasoline was installed for the operation in HCCI mode. Pressurized gasoline was injected by a solenoid-operated injector (Valeo, 348002) into the port, where it was expected to evaporate and mix with the heated intake air to form a homogeneous mixture. Both the gasoline pump and the gasoline injector were powered by a regulated power supply (RPS). To ensure no diesel was injected into the combustion chamber during the HCCI mode, a high-pressure solenoid-operated directional poppet valve (Spica, Y04147-002, DTDA-MCN-212) was installed in the diesel supply line between the fuel pump and the injector to bypass the diesel back to the diesel storage tank. When this solenoid valve was actuated, fuel followed the path of least resistance which was open to atmospheric pressure and thus no fuel was injected by the diesel injector. A 12 V battery was used to power this bypass valve.

Load-based control using ECU

The injection timing and the injection duration of the gasoline injector were controlled by a custom-made ECU. A detailed description of the development process of this ECU was reported by Jeeva et al. (2015). The objective of load-based rpm control in a constant rpm engine is to control the amount of fuel injected to maintain the rpm when the load is changed. This was achieved by modifying the ECU program to perform proportional integral derivative (PID) control as per the algorithm in Fig. 4.

$r(t)$ = target rpm, $y(t)$ = sensed rpm (feedback), $e(t)=r(t)-y(t)$ = error in rpm, $u(t)$ = injection duration, K_P, K_D, K_I = proportionality constants for PID controller.

The parameters, K_P, K_I, and K_D, were tuned appropriately through experiments to ensure smooth operation of the engine along with minimum fluctuations in rpm.

It may be observed that all modifications are external in nature and can be applicable to any stationary engine to achieve dual-mode HCCI operation. Economic viability of these modifications may encourage their commercialization.

Switching between diesel DI and gasoline HCCI modes

At the beginning of the experiment, the engine was started in the diesel mode and the heaters were switched on to raise the temperature of the intake air to 433 K. Once this temperature was reached, the gasoline injector and the bypass solenoid valve were switched on simultaneously to switch the engine to gasoline HCCI mode. After the required data were collected, the gasoline injector and the bypass valve were switched off simultaneously to switch the engine back to diesel DI mode.

Results and discussion

During an experiment, readings were taken twice for every load and three such experiments were conducted to address repeatability. The error bars on the data in HCCI mode represent one standard deviation of the data for the three sets of experiments. The data from the HCCI mode were compared with the data from an optimized Kirloskar diesel engine with a stroke volume of 661 cc, CR of 14.4, and other parameters held similar to those of the engine used for experiments in this study. The results have been presented in terms of the combustion characteristics, the performance characteristics, and the emission characteristics in this section.

Combustion characteristics

In-cylinder pressure

The variation of in-cylinder pressure with crank angle for different loads is plotted in Fig. 1d. It was observed that the peak pressure increased with load in agreement with the results of the simulations discussed in earlier section, where the increase in the equivalence ratio was representative of the increase in the load. The selection of inlet air temperature as 433 K was found to be an appropriate choice as the SOC occurred near TDC (top dead center) for every load.

Figure 5a demonstrates the variation of the in-cylinder peak pressure (P_{max}) with load for the HCCI mode and the conventional diesel mode. P_{max} was observed to be higher at every load in HCCI mode compared to the conventional diesel mode. The higher $dP/d\theta_{max}$ owing to faster rates of combustion in the HCCI mode as shown in Fig. 5b caused the P_{max} to be higher than the conventional diesel mode. P_{max} and $dP/d\theta_{max}$ were found to increase with load fairly linearly for both the modes because of the combustion of a larger amount of fuel at higher loads. The $dP/d\theta_{max}$ values for every load were less than the knocking criterion stated by various researchers, namely 20 bar/deg (Chen et al. 2001) and 10 bar/deg (Aroonsrisopon et al. 2002). Moreover, Coefficient of Variation (COV) of P_{max} and $dP/d\theta_{max}$ was found

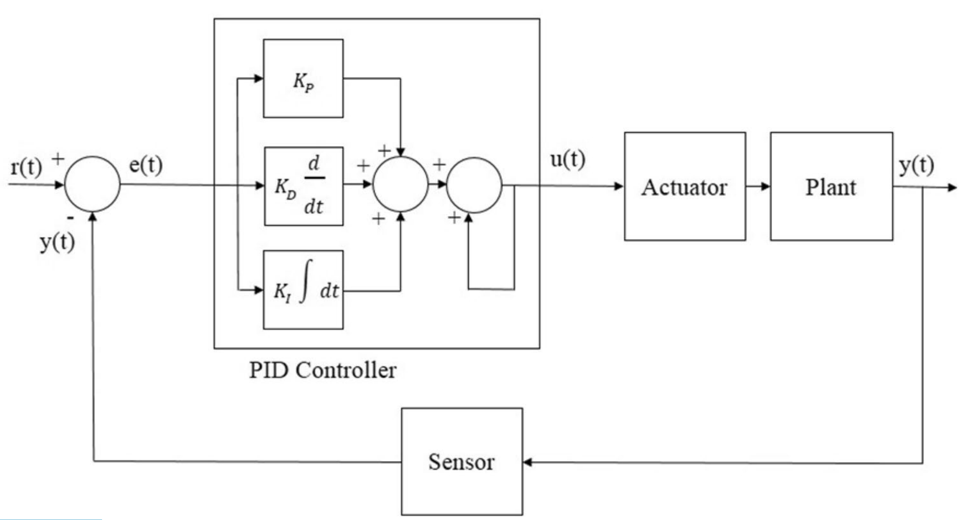

Fig. 4 PID algorithm used for load-based rpm control

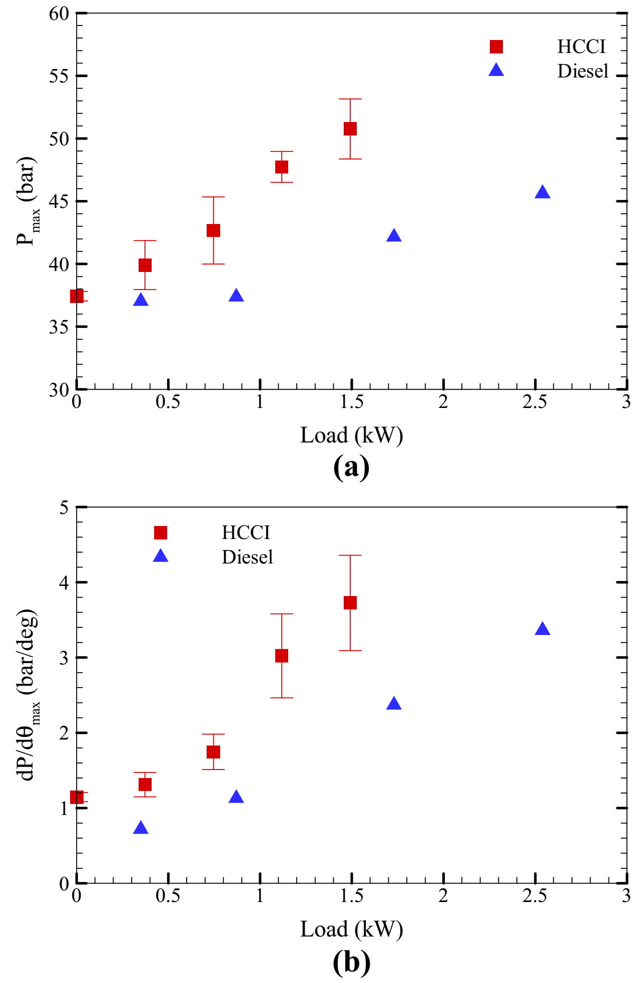

Fig. 5 Variation of P_{max} (**a**) and $dP/d\theta_{max}$ (**b**) with load

to decrease with load indicating more stable combustion (Li et al. 2017).

For the values of loads above 1.49 kW, which correspond to 40% of the maximum load, the intensity of engine noise was found to be unacceptable. Additionally, the values of P_{max} and $dP/d\theta_{max}$ were found to be close to the knock criteria determined in the previous work (Awate et al. 2015) for the values of load above 40% of maximum load.

Peak in-cylinder temperature (T_{max})

T_{max} is an important parameter to be analyzed given its impact on the nature of engine emissions. The in-cylinder temperatures are ideally expected to be uniform given the homogeneous nature of HCCI combustion, and hence the maximum mean in-cylinder temperature (T_{mean}) may be assumed to be representative of the T_{max} occurring inside the cylinder (Maurya and Agarwal 2011). T_{mean} for HCCI mode was calculated using the ideal gas equation as shown in Eq. (1):

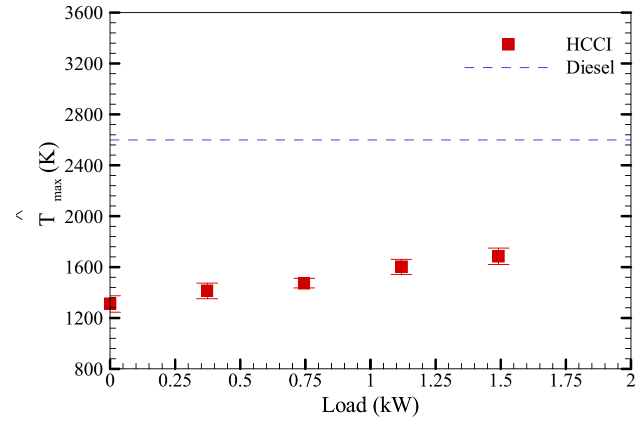

Fig. 6 Variation of the T_{max} with load

$$T_{mean}(\theta) = \frac{P(\theta)V(\theta)}{mR}, \quad (1)$$

where $P(\theta)$ is the in-cylinder pressure, $V(\theta)$ is the volume, m is the mass, and R is the specific gas constant.

In diesel mode, T_{max} was estimated to be around 2600 K which is the adiabatic flame temperature for stoichiometric air–fuel composition owing to the presence of near stoichiometric combustion zones (Dec 2009). T_{max} was found to be lesser than 1800 K for the HCCI mode as shown in Fig. 6, which is in agreement with the hypothesis that HCCI combustion leads to lower peak temperatures than conventional combustion.

Rate of heat release (ROHR)

The ROHR was calculated from the in-cylinder pressures acquired during experiments by applying the first law of thermodynamics with appropriate assumptions as per the formulation in Eq. (2):

$$\frac{dQ}{d\theta} = \frac{1}{\gamma - 1} V \frac{dP}{d\theta} + \frac{\gamma}{\gamma - 1} P \frac{dV}{d\theta}, \quad (2)$$

where Q is the amount of heat released, θ is the crank angle, V is the volume, P is the in-cylinder pressure, and γ is the ratio of specific heats.

Heat loss to the cylinder walls was neglected, and a value of 1.35 was assumed for γ for the calculation of ROHR. Figure 7 shows the effect of load on the ROHR plotted against crank angle. It was observed that the peak heat release rate increased with load. The increase in P_{max} and $dP/d\theta_{max}$ with load contributed to this increase in the peak heat release rate. Also, the crank angle at which peak heat release occurred was found to decrease as the load increased. Heat release was found to occur in a single stage, as significant heat release from low-temperature reactions is typically not observed for

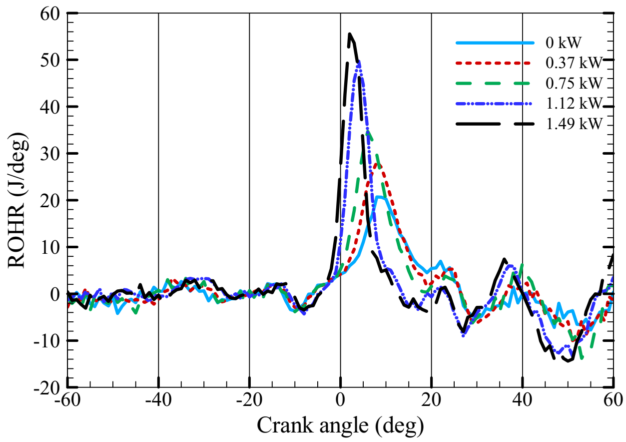

Fig. 7 Behavior of ROHR with varying load

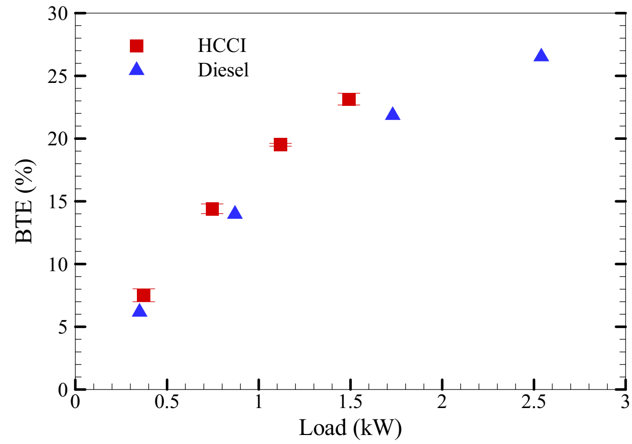

Fig. 8 Variation of BTE with load

a fuel such as gasoline with a high octane number (Sjöberg and Dec 2003). Heat release takes place quickly in approximately 10 CAD as the combustion starts at several sites in combustion chamber at the same time, indicating the extent of homogeneity achieved by PFI system.

Performance characteristics

Brake thermal efficiency (BTE)

Figure 8 shows the variation of BTE against load for the HCCI mode and the conventional diesel mode. The lowest value of temperature of exhaust gas was observed as 220 °C, whereas the lowest temperature of inlet air required to sustain HCCI mode of combustion was found to be 100 °C. A preliminary calculation shows that the additional power required for heating the inlet air may be extracted from the exhaust gas stream using an appropriate heat exchanger such as plate-type heat exchanger with counterflow. Hence, it was neglected in the calculation of BTE for HCCI mode. It was observed that at any given load, a higher BTE was achieved in the HCCI mode than the diesel mode during the experiments. Near-constant volume combustion similar to Otto cycle is expected in HCCI combustion (Gomes Antunes et al. 2008). For the same compression ratio, Otto cycle exhibits higher efficiencies than diesel cycle, providing an explanation for the observed trend (Heywood 1989). As expected, BTE was found to increase with an increase in load for the range of operation under consideration. It was found that BTE values were approximately 15% higher for comparable loads. The overall efficiency is calculated as per ISO 8178 test cycle using B-type test and engine-type D2. The efficiency achieved in diesel mode is 19.92%, whereas efficiency achieved in dual mode (HCCI mode for low loads and diesel mode for higher loads) is 21.46% resulting in overall improvement of 7.75% in efficiencies. The cost of modification of the engine is approximately INR 10,000, based on the cost of an air-to-air heat exchanger, a high-pressure solenoid valve, and associated tubing and electronics. A life cycle cost analysis of the engine, performed under the assumptions of daily operation of 10 h and 25 days per month, shows that the dual-mode operation results in savings of INR 4200 every year, which leads to a break-even period of two and a half years. The break-even period is short compared to the usual life cycle of these engines, which is approximately 10 years, making dual-mode modification economically viable.

Volumetric efficiency

The variation of volumetric efficiency of the engine with load is shown in Fig. 9. The volumetric efficiency was found to be low in HCCI mode because of two probable reasons. The first was the use of heated air, which resulted in inhalation of gases with lesser density and reduced the capacity of the total mass of air to be inducted. Additionally, the excess pressure drop caused by the resistance introduced because of the intake air heating system reduced the volumetric efficiencies in the HCCI mode. However, BTEs for HCCI mode were higher than the diesel mode despite lower volumetric efficiencies. The volumetric efficiency values were approximately constant over the range of loads for both the modes.

Emission characteristics

Pertinent emissions such as NO_x, HC, CO, and CO_2 were measured and compared for the HCCI mode and conventional diesel mode. Owing to the fundamentally less sooty nature of gasoline as compared to diesel, the comparison of PM emissions was omitted in this study, following various researchers working on gasoline HCCI (Aoyama et al. 1996; Oakley et al. 2001; Yang et al. 2002).

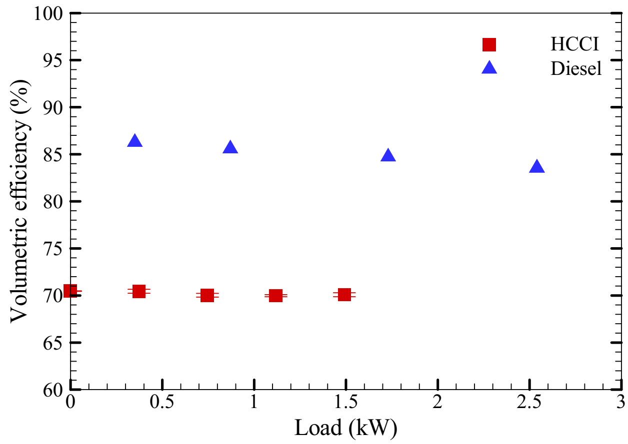

Fig. 9 Variation of volumetric efficiency with load

NO_x

Figure 10 depicts a comparison of NO_x emissions between HCCI mode and diesel mode with increasing load. It was observed that NO_x emissions were substantially lower in the HCCI mode as compared to the diesel mode. While NO_x was approximately 75% lower at 10% load, the gap widened to approximately 95% at 40% load. The reason attributed to this was the lower in-cylinder peak temperature as discussed in Sect. 4.1.2 which is a characteristic of HCCI combustion. The NO_x emissions can be further reduced by using strategies such as EGR by controlling the high heat release rates associated with HCCI mode of combustion (Olsson et al. 2003). Use of EGR may provide an added benefit of combustion control and, thus, smoother operation. Using biogas can also be a viable option to control the combustion (Swami Nathan et al. 2010b).

NO_x in IC engine emission is primarily thermal NO_x which is strongly dependent on in-cylinder temperatures (Oakley et al. 2001). Although the brake thermal NO_x values were found to be decreasing with load for HCCI mode, the absolute NO_x values were found to be increasing with load for both the modes owing to the increasing in-cylinder temperatures. Since the after-treatment of NO_x is complex and expensive (Pundir 2010), lower engine-out NO_x is an attractive feature of HCCI engine, for cleaner and economically sound engine operation.

HC

It is observed from Fig. 11 that the amount of HC emissions at all loads was much higher in the case of the HCCI mode as compared to the diesel mode. It was conjectured that in HCCI mode a PFI system was used, which typically leads to higher HC emissions, most of which come from the crevices (Brijesh et al. 2015). In PFI systems, air–fuel mixture enters the combustion chamber from the initiation of the suction stroke and stays there until the combustion takes place. Hence, the residence times are much higher than engines with DI. Higher residence times translate into a higher probability of the fuel being trapped into the crevices which remains unburnt and is released in the exhaust stroke. Reduction in the HC emissions with load in both the modes was owing to the higher in-cylinder temperatures during the exhaust stroke which promoted the oxidation of the HC (Pundir 2010).

CO

CO emissions are generally associated with combustion occurring at equivalence ratios higher than stoichiometric values. As the mixture is homogeneous and lean in HCCI, CO emissions should not be ideally formed. However, the high values of CO emissions in HCCI mode reported by various research groups, especially at low loads, were

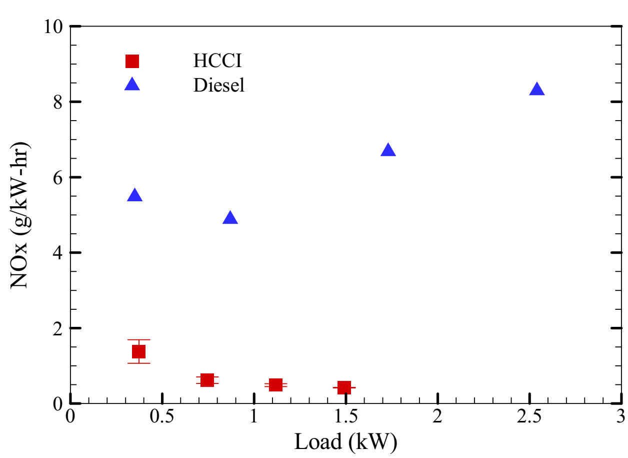

Fig. 10 Variation of NO_x emissions with load

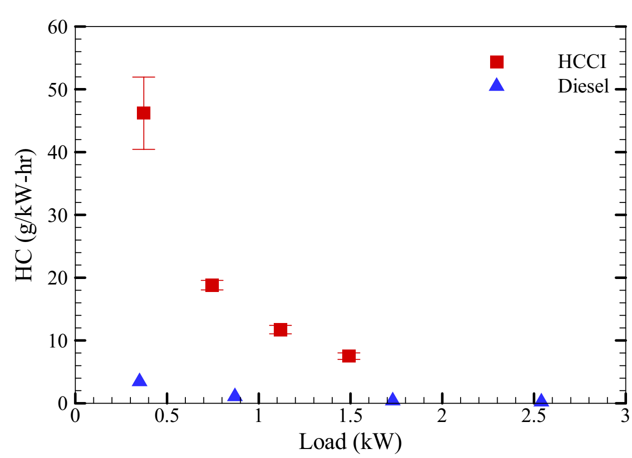

Fig. 11 Variation of HC emissions with load

attributed to the quenching of CO oxidation process in the lean mixture at lower combustion temperatures (Zhao 2007). Moreover, poor CO to CO_2 conversion owing to late heat release as observed in Fig. 7 results in higher CO emissions at lower loads (Sjöberg and Dec 2003). In the diesel mode, the CO emissions may be attributed to the presence of locally rich zones (Heywood 1989).

Figure 12 shows a comparison between the CO emissions of the HCCI mode and the conventional diesel mode. It was observed that the CO emissions were higher for the HCCI mode as compared to the conventional mode at all the loads. It was also observed that the difference between the emission values of the two modes reduced significantly at higher loads. CO emissions reduced with load for both the modes, which was in agreement with higher in-cylinder temperatures at higher loads promoting the oxidation of CO. Higher HC and CO emissions in HCCI engines need to be treated. It is noteworthy that the after-treatment of HC and CO is simpler and is not subjected to fuel consumption penalty as compared to NO_x and PM after-treatment.

CO_2

CO_2 emissions for the HCCI mode and the diesel model are shown against load in Fig. 13. It was observed that the amount of CO_2 emissions at all loads is lower in the HCCI mode as compared to the conventional diesel mode owing to the combustion of lower amount of fuel. The amount of brake-specific CO_2 emission was found to reduce for both the modes as the load increased which is in congruence with the increasing BTE with load as observed in Sect. 4.2.1.

Conclusions

Experiments were conducted to run a single-cylinder diesel DI engine in HCCI mode using gasoline as fuel, and stable operation was achieved under part-load conditions. It was found that the parameters predicted by the CHEMKIN simulations provided an excellent initial guess for the HCCI experiments. These parameters were further tuned to obtain smoother operation of the engine. Load-based rpm control was implemented using PID algorithm to run the engine at constant rpm, thus enabling comparison between conventional diesel and gasoline HCCI mode at different loads. The performance characteristics such as BTE, volumetric efficiency, ROHR, peak pressure, and peak temperature were evaluated and compared for both the modes. It was found that BTE values were 23% which are approximately 15% higher for comparable loads, while the volumetric efficiency values were approximately 20% lower at comparable loads for the operation in HCCI mode. Additionally, exhaust gas emissions were measured and compared for both the modes. It was found the HCCI mode had 1.4 g/kW h of NO_x emissions and 2200 g/kW h of CO_2 emissions which are lower by approximately 80% and 30%, respectively, at comparable loads. However, HC and CO emissions were drastically higher at lower loads than the conventional diesel mode. Although engine-out NO_x emissions are within specified limits, HC and CO emission limits specified by Euro Stage V standards could not be achieved which may be readily tackled by after-treatment to reduce tailpipe emissions. However, HC and CO after-treatment is simpler and less expensive as compared to NO_x and PM after-treatment. The engine could not be operated beyond 40% of the maximum load, owing to its tendency to knock, and was instead switched over to the diesel mode.

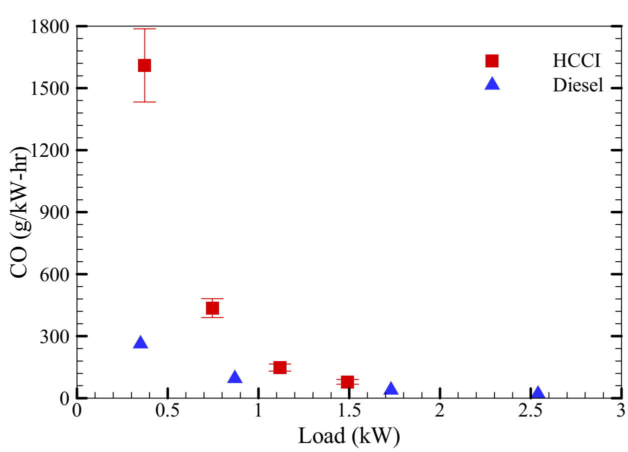

Fig. 12 Variation of CO emissions with load

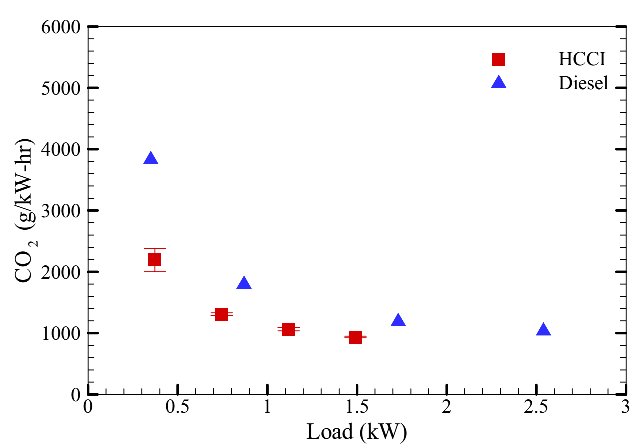

Fig. 13 Variation of CO_2 emissions with load

Thus, in pursuit of developing a cleaner technology to reduce air pollution and maintain higher fuel economy, operating the engine in HCCI mode for part-load conditions and switching to conventional mode for higher loads was found to be a sustainable solution. The choice of a PFI system to achieve homogenous mixture was found to be appropriate. The dual-mode HCCI operation was achieved successfully by modifying the engine. Moreover, the modifications were found to be economically viable and applicable to any engine of this class and have a great potential for commercialization.

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