Combustion characteristics of diesel HCCI engine: An experimental investigation using external mixture formation technique

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Combustion characteristics of diesel HCCI engine: An experimental investigation using external mixture formation technique

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HIGHLIGHTS

- ▶ Combustion characterization of diesel HCCI with external mixture formation.
- ▶ 'Diesel vaporizer' was successful in achieving HCCI combustion.
- ▶ Efficient HCCI combustion achieved up to medium load conditions.
- ▶ Combustion parameters are effectively controlled by EGR.
- ▶ Control of SOC and ROHR at higher loads remains challenging.

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ABSTRACT

In compression ignition engines, soot–NO_x paradox is an extremely challenging unresolved issue. Homogeneous charge compression ignition (HCCI) is one of the most promising solution that combines the advantages of both SI and CI combustion modes. It gives high thermal efficiency similar to compression ignition engines and resolve the associated issues of high levels of NO_x and PM simultaneously. In HCCI combustion, homogeneous mixture of air and fuel burns spontaneously throughout the combustion chamber, which reduces the total combustion duration due to very high rate of heat release. Determination of precise control parameters for controlling the 'rate of heat release' and 'start of combustion' are major research challenges in the development and deployment of this technology. In the present research, experiments were performed in a two cylinder engine, in which one cylinder is modified to operate in HCCI mode, while other cylinder operate in conventional CI mode. Homogeneous mixture preparation is the most challenging part for achieving diesel HCCI combustion. Low diesel volatility remains the main obstacle in preparing the homogenous fuel–air mixture therefore a dedicated device called 'diesel vaporizer' was developed. Exhaust gas recirculation (0%, 10% and 20%) was used for controlling the rate of heat release. To study the combustion behavior, experiments were performed at three different relative air–fuel ratios (λ = 4.95, 3.70 and 2.56). Enrichment of fuel–air mixture enhances the rate of heat release and the location of peak of in-cylinder pressure shift towards BTDC side due to earlier start of combustion. This was effectively controlled by EGR for leaner HCCI combustion conditions. Exhaust gases diluted the homogeneous charge and presence of non-reactive species reduce the rate of combustion. It controls the peak in-cylinder temperature, which is a responsible for extremely low NO_x formation. For richer fuel–air mixtures, EGR was relatively less effective due to dominance of 'rate of heat release', which was significantly high.

1. Introduction

Direct Injection Compression Ignition (DICI) and Spark Ignition (SI) engines are two basic technologies with established applications in automotive sector. SI and CI engines use fossil fuels and have their own pros and cons however DICI engines are becoming more popular due to their suitability for personal as well as commercial transportation in view of excellent fuel economy. However both diesel and gasoline engines are large contributors to urban air pollution. Carbon monoxide and unburned hydrocarbons emissions from these engines contribute to global warming. Oxides of nitrogen and hydrocarbons emitted from internal combustion engines react in atmosphere to form photochemical smog. Particulate emitted from diesel engines precipitate asthma and respiratory episodes. Due to adverse health effects of these pollutants, increasingly

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stringent emission norms are being adopted worldwide, which require simultaneous reduction of PM and NO_x emissions [1]. HCCI is an advanced combustion technology, which has potential for substantial and simultaneous reduction of NO_x and PM, with high efficiencies similar to diesel engines. In HCCI engines, combustion occurs as a result of spontaneous auto-ignition at multiple points throughout the charge volume. This unique characteristic of HCCI allows the combustion of very lean and dilute mixtures, resulting in relatively lower bulk temperatures and localised combustion temperatures, which significantly reduce engine-out NO_x emissions. Unlike CI combustion, the charge (fuel and air) is well mixed (homogeneous) in case of HCCI combustion. The absence of fuel-rich zones in HCCI combustion results in significantly large reduction in PM formation.

Initial efforts by Onishi et al. related to HCCI combustion involved gasoline-fueled engines where they attempted to increase combustion stability of two-stroke engines and this technology continues to be strongly pursued even today [2]. It is called Active Thermo-Atmosphere Combustion (ATAC). They discovered that a significant reduction in emissions and improvement in fuel economy can be obtained by creating conditions that lead to spontaneous ignition of the in-cylinder charge. They reported that stable HCCI combustion can be achieved between low and high load limits with gasoline at a compression ratio of 7.5:1 over the engine speed range from 1000 to 4000 rpm. On the basis of the results of two stroke engines, Najt and Foster [3] extended the work to four-stroke engines and attempted to gain additional understanding of the underlying physics of HCCI combustion. They used a CFR test engine with variable compression ratio and concluded that HCCI auto-ignition is controlled by low temperature chemistry (below 1000 K) and bulk energy release is controlled by high temperature chemistry (above 1000 K), which is dominated by CO oxidation. In the experiments, it was noted that HCCI combustion suffers from a lack of control of the ignition process and has a limited operating range. Thring [4] further extended the work of Najt and Foster [3] in four-stroke engines by examining the performance of a HCCI engine operated using fully-blended gasoline. He also found that the operating regime was restricted to part load operation, and control of the auto-ignition timing was a critical issue. Canakci and Reitz [5] performed gasoline HCCI experiments using double fuel injection strategy. Double injection was explored for emission control, and the engine was optimized using fully automated experiments and a micro-genetic algorithm optimization code. The variables changed during the optimization included intake air temperature, start of injection timing, and split injection parameters.

After achieving positive results with HCCI combustions, it became imperative to investigate the possibility of extending the benefits of HCCI combustion using diesel. Experimental exploration of the potential for diesel–fueled HCCI started in the mid-1990s. However the problem of homogeneous mixture preparation was the biggest challenge for diesel HCCI combustion development. To achieve this, elevated temperatures are required for significant vaporization of fuel so that premixed homogeneous charge can be formed. Other challenge was that diesel exhibits significant cool-combustion chemistry, leading to rapid auto-ignition, once temperatures at the end of compression exceed above 800 K [6]. Several research efforts showed that early injection and late injection methodologies are very strong however generally these are recommended for gasoline like fuels. For diesel, port fuel injection is the most straight forward approach to obtain a premixed charge. Basically it is an external mixture preparation approach, which is based on fumigation of diesel at elevated temperatures [7]. This technique is excellent because it utilizes turbulent flow velocities at intake port to promote mixing and hence gives a very homogenous charge [8]. Ryan and Callahan [9] used port fuel injection to induct diesel into the intake air stream. An intake air heater was installed upstream of the fuel injector. Engine compression ratios were varied from 7.5 to 17 and EGR was used. Gray et al. found three key issues for diesel–fueled HCCI. First, very premature ignition and knocking occurred, when normal diesel compression ratios were used. Satisfactory results required compression ratios in the range of 8–13, depending on intake air temperature and amount of EGR used. Second, relatively high intake air temperatures (135–205 °C) were required to minimize the accumulation of liquid fuel on intake manifold surfaces. Third, unburned hydrocarbon emissions were very high. As a result of the poor combustion efficiency, reduced compression ratios and non-optimal combustion phasing, fuel consumption increased by approximately 28% over normal direct-injection (DI) diesel combustion [10]. Maurya and Agarwal [11,12] investigated port fuel injection for achieving HCCI for gasoline, alcohols and blends. Lu et al. [13] critically examined the HCCI combustion control using EGR and discussed effect of EGR on important combustion characteristics. First and second stage combustion phase retarded with EGR and total combustion duration increased with increasing EGR. Nakano et al. [14] performed a modal analysis for EGR controlled HCCI system and showed an excellent control capabilities using EGR. Komninos [15] and Kamninos and Kosmadakis [16] performed a multi-zone model analysis based on heat and mass transfer in HCCI combustion system. The multi-zone model included sub-models for heat transfer between different zones and to the cylinder wall along with mass transfer between the hotter and colder regions of the combustion chamber. Suyin et al. [17] reviewed diesel HCCI technology and discussed various control and performance parameters affecting HCCI. They concluded that for long-term development of HCCI diesel combustion systems and superior mixture formation and control, flexible fuel injection strategies and EGR control will be the most critical approaches.

Present experimental investigation is therefore aimed at development of methodology and hardware for preparing fully homogeneous mixture of diesel and air. This research explores the combustion characteristics of HCCI using an external mixing device "diesel vaporizer" [10,18,19]. It requires fewer hardware modifications in existing DI engine to convert it into HCCI engine. Diesel vapor produced by this device also contain very small diesel droplets in the form of mist, which can easily mix with air to form either homogeneous mixture or partly homogeneous mixture. Due to adequate mixing in the vaporizer, formation of mixture of air and partly vaporized fuel droplets takes place. Various experiments were performed to investigate HCCI combustion using different EGR conditions and engine load.

2. Experimental setup

The experimental setup is divided in five basic sub-systems, namely engine, diesel vaporizer, fuel injection system, exhaust gas recirculation system and data acquisition system. The schematic diagram of the experimental setup is shown in Fig. 1.

2.1. Engine

The experiments were performed in a constant-speed, two-cylinder, four-stroke, air-cooled, direct injection diesel engine (Make: Indec; Model: PH2). In this engine, one cylinder is modified to operate in HCCI combustion mode while the other cylinder operates in conventional diesel combustion mode. The engine is air-cooled and the cooling air is provided by a circular casing on the flywheel, which forces the atmospheric air around the cylinders. The engine is coupled with a single phase 9 kW, 220 volts AC generator. The generator's efficiency was measured experimentally

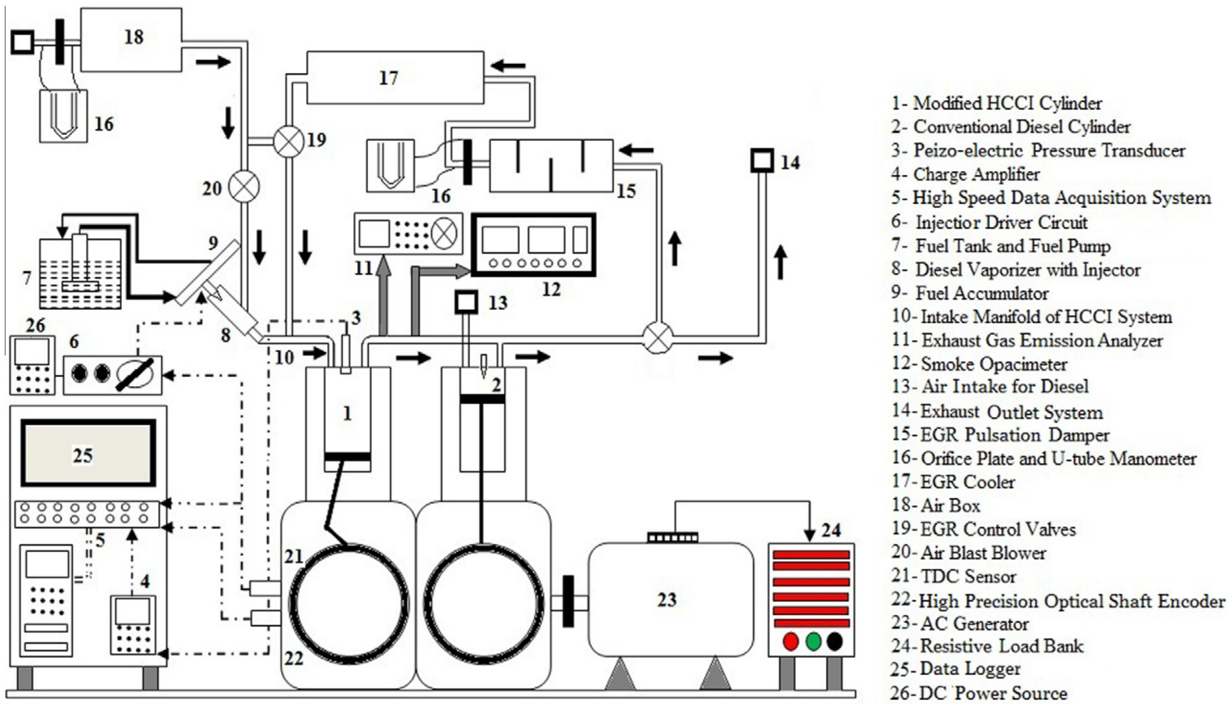

Fig. 1. Schematic of diesel HCCI experimental setup.

and the same was accounted for in the engine data analysis for the experiment. A load bank (10 kW having loading step of 0.5 kW) was used for loading the engine-genset system. The technical specifications of the unmodified engine are given in Table 1.

2.2. Diesel vaporizer

Homogeneous charge preparation is the most critical part in achieving diesel HCCI combustion. Low volatility of diesel is the main hurdle in formation of homogeneous mixture of fuel and air. In the present investigations, "diesel vaporizer", is designed for this purpose. The construction of diesel vaporizer is shown in Fig. 2.

Diesel vaporizer has a main vaporizing chamber made of copper. Cylindrical surfaces of this vaporizing chamber are covered externally by a band heater, which provides required heating for fuel vaporization. Band heater is controlled by a PID temperature controller. Detailed specifications of the diesel vaporizer are given in Table 2.

Fuel injector sprays the atomized fuel into the heated diesel vaporizer chamber. Fuel droplets absorb latent heat of vaporization. Air supplied at high velocity from the blower forces these fuel vapors/mist containing tiny fuel droplets to mix homogeneously. This mixture is supplied to the combustion chamber through the intake valve located in inlet manifold.

Table 1
Technical specification of the test engine.

Characteristics	Specifications
Make/model/type	Indec/PH2/diesel engine
Injection type	Direct injection
Number of cylinders	Two
Bore/stroke	87.3/110 mm
Power output/cylinder	4.85 kW @ 1500 rpm
Compression ratio	16.5:1
Displacement/cylinder	659 cc
Fuel injection timing (SOI)	24° before TDC
Fuel injection pressure	210 kg/cm^2 @ 1500 rpm
Oil sump capacity	6.8 l

2.3. Fuel injection system

The fuel injection system of diesel HCCI engine consists of a fuel pump, fuel tank, fuel accumulator, fuel injector and an injector control circuit. An electrical fuel pump (12 V DC) installed inside the fuel tank is used to supply diesel from the fuel tank to the fuel accumulator. Fuel injector operates on 12 V Transistor-Transistor Logic (TTL) pulse generated by fuel injection control circuit, which controls the 'start of fuel injection' and 'injection duration'. Fuel injection control circuit takes input from the TDC sensor to determine the 'start of fuel injection'. Other important specifications of injection control circuit are given in Table 3.

2.4. Exhaust gas recirculation system

A part of the exhaust gas is cooled and recirculated back into the combustion chamber. It reduces maximum in-cylinder temperature thereby reducing NO_x emissions significantly. The exhaust gas from the two cylinder engine is however highly pulsating in nature and it makes the accurate flow measurement difficult therefore a damper is inserted in the circuit to reduce the exhaust gas pulsations. A U-Tube manometer is used to measure the flow rate of EGR across a calibrated orifice plate, mounted downstream of the damper.

2.5. Data acquisition system

Main components of data acquisition (DAQ) system are peizo-electric pressure transducer, charge amplifier, DAQ board, TDC sensor, and a precision optical shaft encoder. Data acquisition system contains inbuilt data acquisition cards (National Instruments) and software (LabVIEW Based) for data acquisition and processing respectively. The peizo-electric pressure transducer measures the instantaneous in-cylinder pressure and gives proportional voltage signal to DAQ system via charge amplifier. For the crank angle measurement, a precision optical shaft encoder (Make: Encoder India Limited, Faridabad, Model: ENC58/6-720ABZ/5-24V) is used,

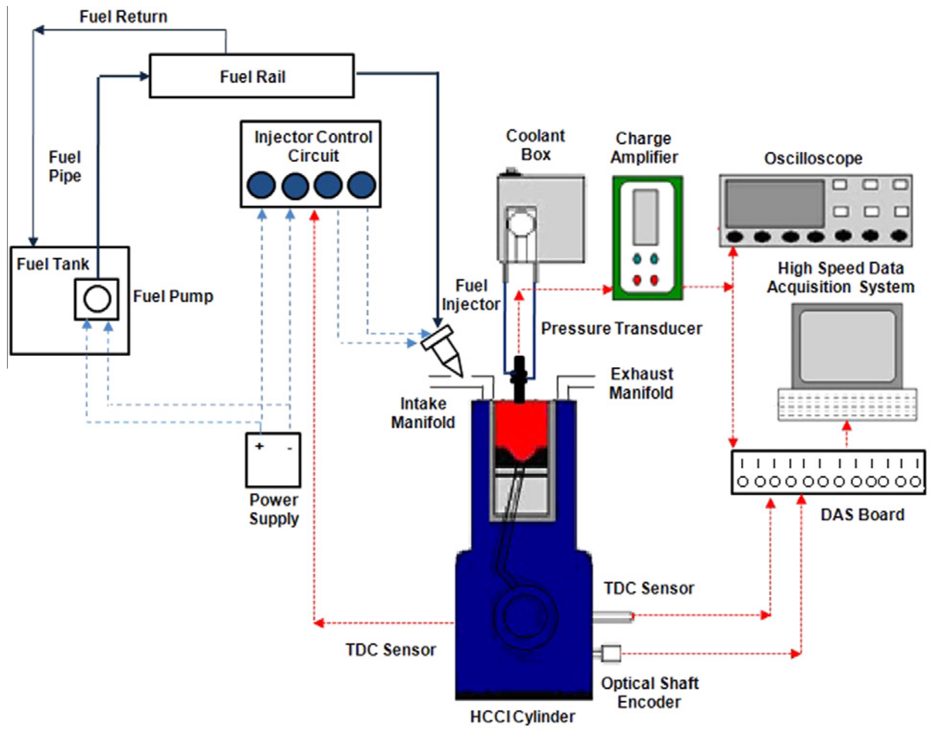

Fig. 2. Diesel vaporizer.

Table 2
Technical specifications of diesel vaporizer.

Characteristics	Specifications
Heater capacity	750 W
Vaporizer diameter	30 mm
Vaporizer length	150 mm
Warm-up time	6 min
Cut-off temperature	160 ± 10 °C
Fuel injection pressure	3.0 bar

Table 3
Specifications of injection control circuit.

Characteristics	Specifications
Input signal	5.0 V TDC pulse
Output signal	12 V TTL square pulse
Power supply	12 V DC
Min. pulse duration	3 ms
Max. pulse duration	24 ms
Step size	1 ms

Table 4
Specifications of high speed data acquisition system.

Characteristics	Specifications
Input signal	Charge signal (pC)
Voltage power supply	<42 V
Conversion factor	10 Bars/Volt
No. of channels	8
Data storage	10000 cycles
Reference signal	0.5 CAD

which gives one signal in every 0.5° CAD. The cycle reference signal is obtained from inductive TDC pickup. Other important specifications of data acquisition system are given in Table 4.

The data acquisition system digitizes and stores the input voltage signals, triggered by an external clock. Data acquisition hardware consists of a National Instrument BNC 2090 adapter, and National Instrument MIO16E1 Data acquisition card. BNC 2090 can be configured to use eight differential inputs or 16 single ended analog input channels. It is designed for input voltages less than 42 V. A data acquisition and analysis program was developed using National Instrument LabVIEW software (V8.6). The program allows simultaneous monitoring, processing and recording of various data sets from the engine. A combined diagram showing fuel injection sub-system and data acquisition sub-system is shown in Fig. 3.

In the experiments, commercially available diesel with cetane number 45 is used. For HCCI experiments, higher cetane fuel would causes shorter ignition delay hence will deliver superior combustion characteristics. Higher viscosity of the test fuel could be another issue, which could possibly adversely affect spray pattern therefore air–fuel mixing. Abrasive solid dust particles or soluble organo-metallic additive compounds may also be present in diesel, which might form a layer inside the vaporizer surfaces and reduce its effectiveness.

3. Results and discussion

Experiments were performed according to experimental matrix with varying relative air–fuel ratio and EGR ratio. There are two operating limits namely ‘knocking limit’ and ‘misfire limit’, which are relevant to HCCI engine operation. When the rate of pressure rise is unacceptably high, the limit is termed as ‘knocking limit’ whereas the limit corresponding to unacceptably high cyclic variations is termed as ‘misfire limit’. First limit describes the safety issues related to experimental setup because at higher engine load conditions, heavy knocking, which may cause excessive wear and other detrimental engine effects. Second limit defines lower load

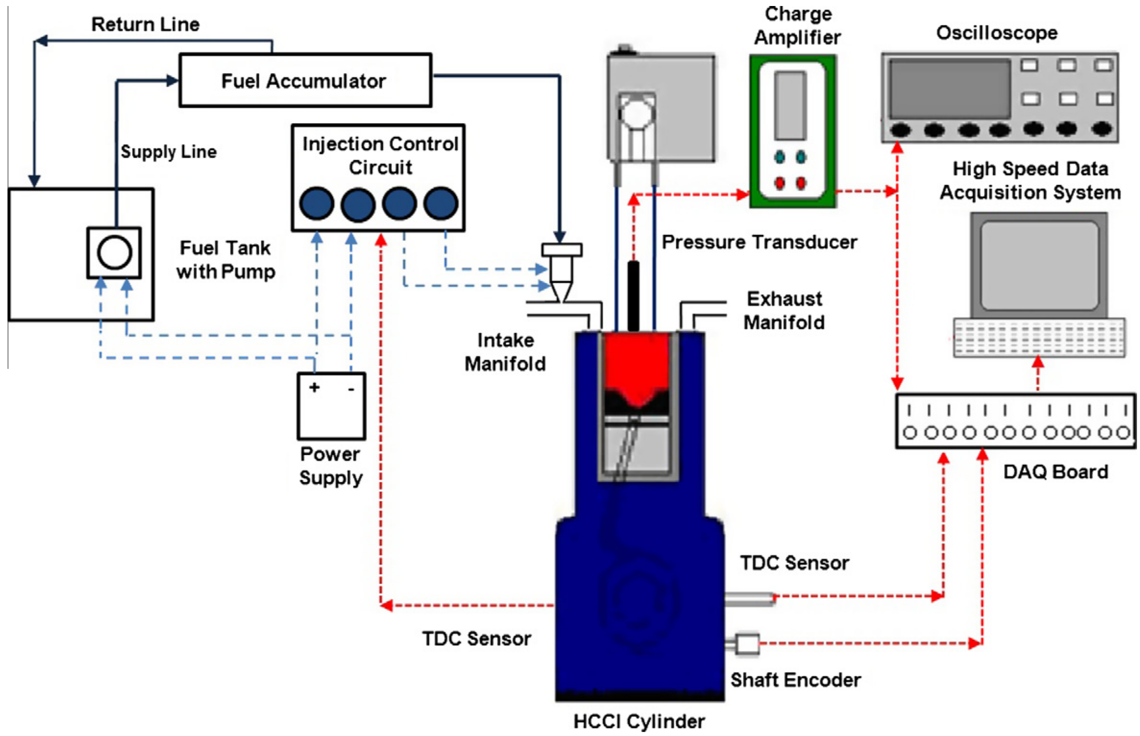

Fig. 3. Fuel flow and data acquisition sub-systems.

limit where the fuel quantity decreases hence the net heat release also decreases. Misfire limit affects emission parameters significantly [12]. Generally HCCI combustion is preferred for low (up to 1 kW) and medium (1–2.5 kW) engine loads because at higher (2.5–4 kW) load conditions, it gives abnormal combustion due to relatively higher rate of heat release. In the present research, experiments are performed for 1 kW (Misfire limit higher than $\lambda = 4.96$) to 4.0 kW (Knocking limit lower than $\lambda = 2.56$) range. Effi-

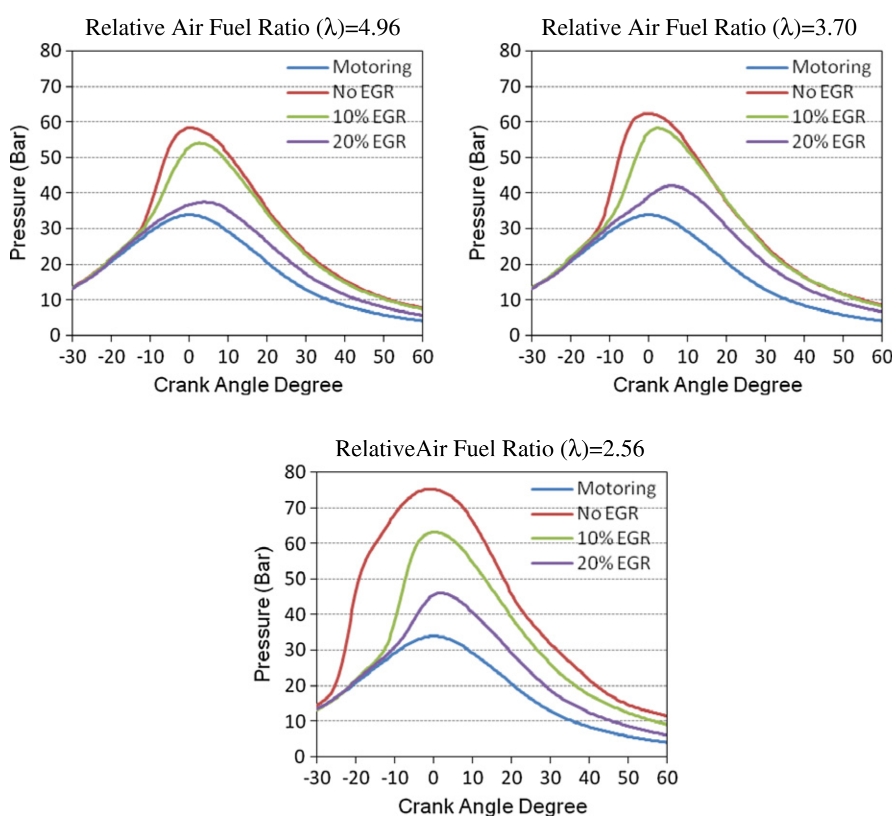

Fig. 4. In-cylinder pressure variation at different load and EGR conditions.

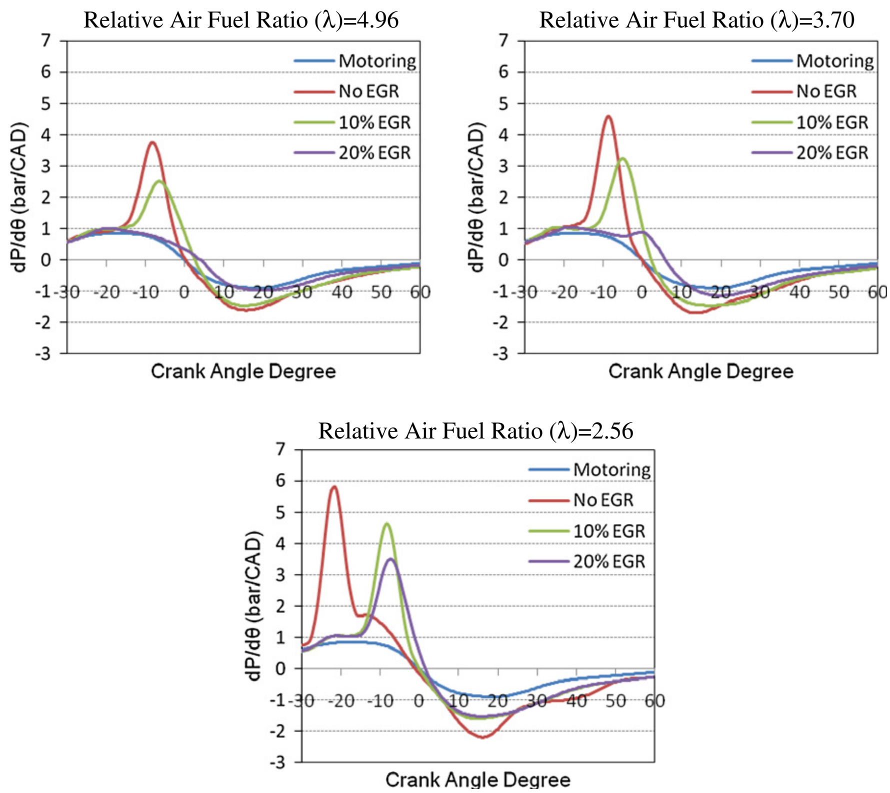

Fig. 5. Rate of pressure rise at varying λ and EGR.

cient HCCI combustion (with quiet and smooth engine operation) could be achieved up to $\lambda \geqslant 3.70$, while at higher load conditions ($\lambda < 3.0$), it gives noisy engine operation. The results are divided in following subcategories.

3.1. In-cylinder pressure analysis

In-cylinder pressure data analysis is the most effective way to analyze the engine combustion behavior because in-cylinder pressure history directly influences the power output, combustion characteristics and emissions from an engine. In present research, in-cylinder pressure data is recorded vis-a-vis crank angle. In order to avoid the cyclic variations and to account for measurement errors, average pressure data for 50 consecutive cycles is used for analysis. P-θ diagram provides information about start of combustion, rate of pressure rise, and maximum cylinder pressure. Fig. 4 shows the in-cylinder pressure variation with respect to crank angle for different mixture strengths (λ) and EGR rates.

This figure clearly shows the occurrence of auto-ignition (during HCCI combustion) in all three cases. Rise in slope of pressure curve vis-a-vis motoring curve shows the 'start of combustion'. As the mixture become richer (lower λ values), start of combustion shifts towards BTDC side. It happens due to early start of combustion during the compression stroke. Presence of sufficient fuel at higher temperature and pressure favors earlier auto-ignition of charge. For $\lambda = 2.56$ at no EGR, the combustion starts well before TDC (25° BTDC). Here early combustion of charge is controlled by EGR, which retards the start of combustion significantly (Fig. 4). Peak cylinder pressure also decreases with increasing EGR due to reduction in rate of combustion at lower in-cylinder temperature. In all three EGR conditions (0%, 10% and 20%), peak cylinder pressure occurs near TDC which shift away from TDC with increasing EGR. It happens because of delayed combustion due to mixture dilution by recirculated exhaust gas. As EGR rate increases, level of dilution also increases, increasing the delay period, which pushes the peak of pressure curve towards the ATDC side. EGR rate

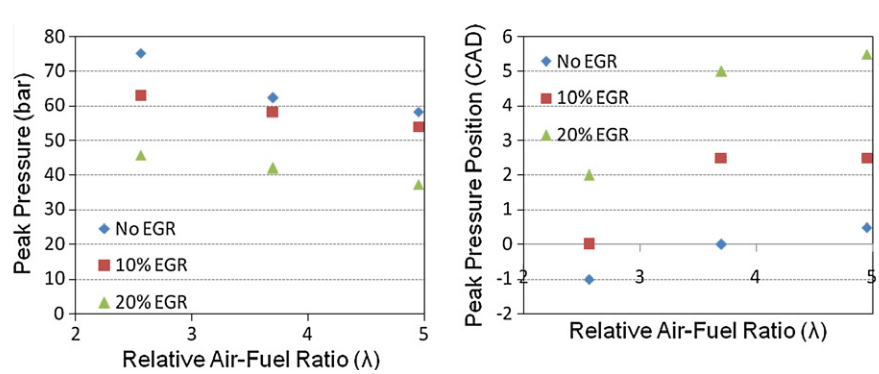

Fig. 6. Peak in-cylinder pressure and its position for varying λ and EGR.

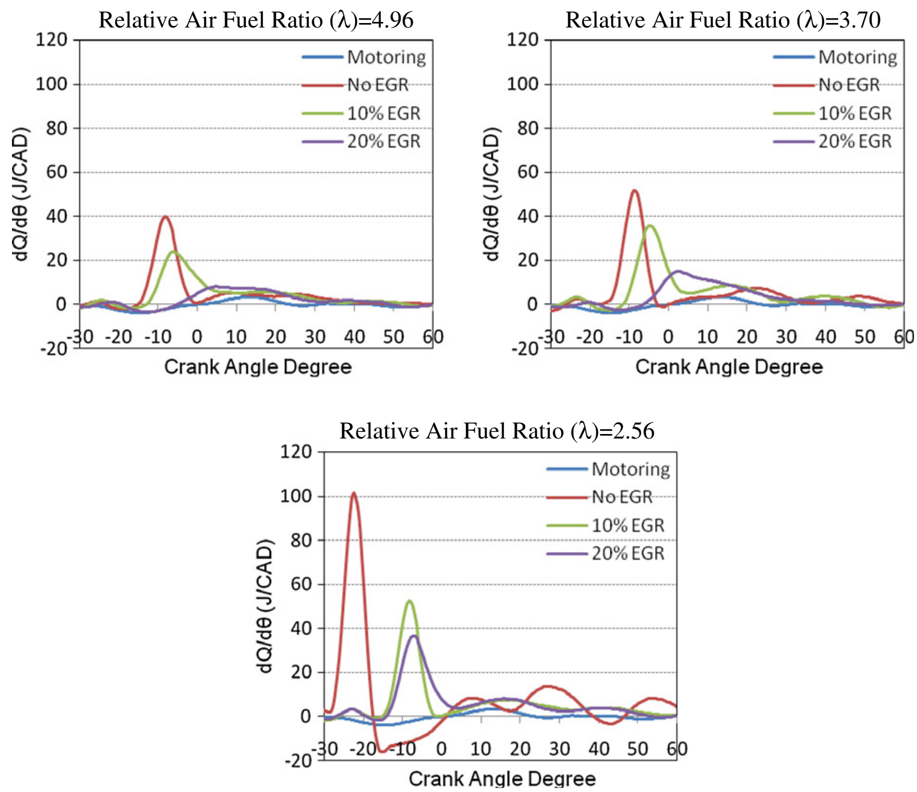

Fig. 7. ROHR for varying λ and EGR.

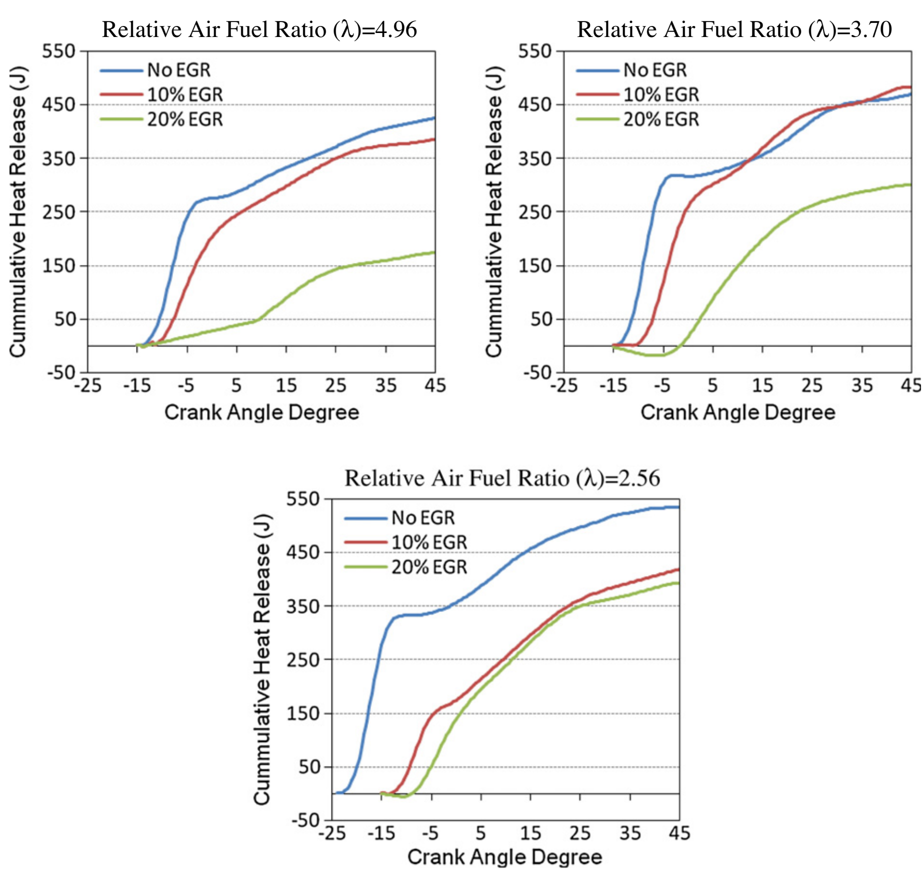

Fig. 8. Cummulative heat release for varying λ and EGR.

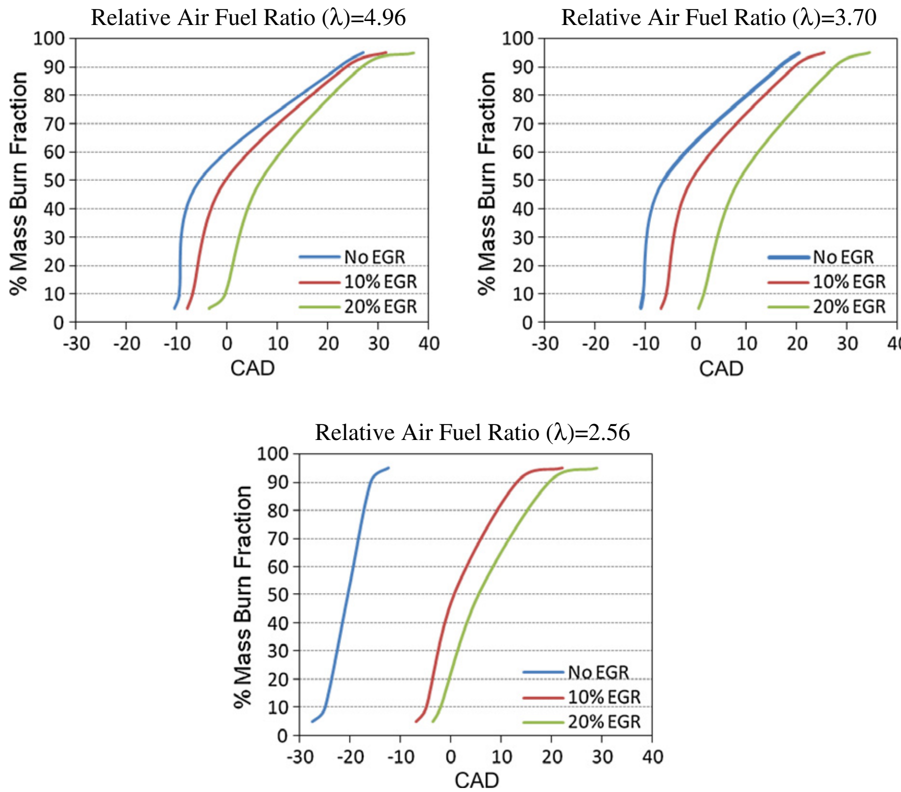

Fig. 9. Mass burn fraction for varying λ and EGR.

also affects volumetric efficiency of the HCCI combustion engine. At no EGR condition, intake air temperature is quite low, which results in higher volumetric efficiency. As EGR increases, it causes increase in intake charge temperature therefore affecting volumetric efficiency adversely. Fig. 5 shows the variation in 'rate of pressure rise' with respect to crank angle for varying mixture strengths and EGR.

From Fig. 5, it can be seen that the rate of pressure rise increases with increasing engine load (decreasing λ). It happens mainly because of higher rate of combustion due to mixture enrichment. When charge becomes richer, it favors earlier start of combustion due to dominance of cold combustion chemistry of diesel [6]. As the EGR rate increases, delayed combustion is observed for richer mixtures thereby affecting the HCCI combustion positively.

Fig. 6 shows the effect of λ and EGR on peak cylinder pressure and its crank angle position. As the mixture become richer, peak in-cylinder pressure increases whereas increasing EGR lowers the peak in-cylinder pressure. Fuel enrichment of mixture shifts the peak in-cylinder pressure to the left, while increasing EGR shifts the peak to the right. Higher fuel quantity is responsible for higher in-cylinder pressure and earlier start of combustion. Similar results were also obtained by Kanda et al. [20], where high rate of EGR (54%) is used to retard the ignition timing toward TDC thus improving the IMEP.

3.2. Heat release analysis

Heat release analysis is another characterization tool for HCCI combustion. It is calculated from the acquired cylinder pressure data using "zero dimensional heat release model" [21]. HCCI heat release pattern is different from conventional combustion modes due to occurrence of combined phenomenon of simultaneous ignition of homogeneous mixture using compression (SI and CI). Higher rate of heat release (ROHR) in HCCI combustion creates problem in controlling the combustion rates and affects safety and structural integrity of the engine. The ROHR for varying λ and EGR is shown in Fig. 7.

As λ decreases and mixture becomes rich, ROHR significantly increases (Fig. 7). At relatively richer condition ($\lambda = 2.56$), ROHR becomes higher than 100J/CAD, which is quiet high compared to conventional combustion modes [22]. In HCCI combustion, EGR is used for combustion control i.e. controlling the ROHR. It reduces the mixture reactivity by addition of non-reactive gaseous species. A large fraction of heat is absorbed by the recirculated combustion products (mainly CO_2). Therefore higher EGR provide lesser mixture reactivity, i.e. reduced ROHR. Two-stage combustion can be also clearly observed from Fig. 7. In all three cases, heat release curve posses two peaks, one for low temperature heat release region and the other one for high temperature heat release region [23–25]. The first stage of heat release curve is associated with low temperature kinetic reactions. Time delay between first and main heat release is attributed to the "Negative Temperature Coefficient (NTC) regime", which is located between the two heat release stages. In this NTC regime, the overall reaction rate decreases though the in-cylinder temperature increases, leading to lower reactivity of the system [23–25]. Cumulative heat release curves (Fig. 8) show total heat released during combustion. Effect of mixture enrichment and EGR is clearly seen in the cumulative heat release curves.

Fig. 8 shows that lower values of λ gives higher cumulative heat release at a higher rate (higher slope of curve) due to higher fuel energy input in every engine cycle. Cumulative heat release decreases by increasing EGR due to absorption of heat by relatively colder recirculated gas species. EGR consists of CO_2 and other high heat capacity gases, which absorbs large fraction of combustion heat release. Phenomenon of delayed combustion can also be understood by this cumulative heat release curve. Increasing EGR also reduces the slope of cumulative heat release curve.

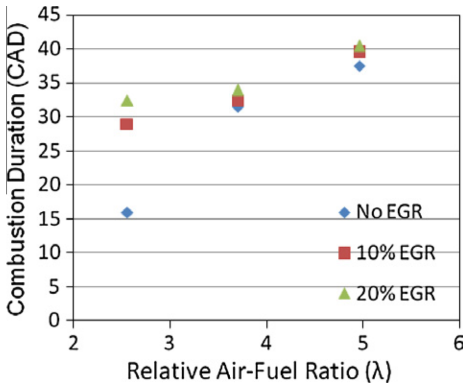

Fig. 10. Combustion duration for varying λ and EGR.

3.3. Mass burn fraction analysis

In this section, rate of combustion reaction progression is analyzed on the basis of mass burn fraction. Mass burn fraction is calculated from Rassweiler and Withrow method [26], which is a well-established method for estimating MFB. It was calculated as

$$(\Delta p_c) = p_i - p_{i-1}\left(\frac{V_{i-1}}{V_i}\right)^{\gamma}$$

where Δp_c is the increase in in-cylinder pressure due to combustion and γ is the polytropic exponent.

MFB can now be computed as

$$\frac{m_{b(i)}}{m_{b(\text{total})}} = \frac{\sum_{j=0}^{i}\Delta(p_c)_j}{\sum_{j=0}^{N}\Delta(p_c)_j}$$

where it is assumed that sample 0 is between inlet valve closing and the start of combustion, and that sample N is after the combustion is completed.

Fig. 9 shows the spontaneous combustion characteristics of the HCCI engine. For all three mixture strengths, nearly 50% of fuel burns in early stage of combustion, while remaining part burns in the later stage of combustion. As the mixture strength increases, early stage mass burn fraction also increases due to higher rate of combustion. Here, the role of EGR is also clear and it can be seen that it reduces reaction rates hence reducing the mass burn fraction in first stage of combustion.

This analysis shows variation in combustion duration with varying λ and EGR (Fig. 10). When the charge becomes relatively richer, combustion duration tends to decrease due to relatively higher rates of combustion, however with increasing EGR, combustion duration increases due to slower combustion reactions. It happens mainly due to variation in chemical reactivity of the combustible mixture. When EGR is introduced, it reduces in-cylinder temperature and finally rate of reaction slows down.

4. Conclusions

In the present study, combustion characterization of diesel HCCI with external mixture formation was investigated for different EGR conditions. External charge preparation technique gives improved mixture homogeneity compared to other methods. ‘Diesel vaporizer’ proved successful in achieving diesel HCCI combustion. Diesel HCCI combustion is highly sensitive to λ and EGR. Experiments were performed at different mixture strengths from $\lambda = 4.96$ (lean or misfire limit) to $\lambda = 2.56$ (rich or knocking limit). Efficient HCCI combustion (with quiet and smooth engine operation) can be achieved up to $\lambda \geqslant 3.70$, while at higher load conditions ($\lambda < 3.0$), it gives noisy engine operation. As mixture becomes richer, peak in-cylinder pressure and rate of heat release increase rapidly due to higher rate of combustion and start of combustion shifts towards left. Start of combustion is very sensitive to λ. For richer mixture conditions, it can be controlled by EGR also. Two stages of heat release due to (i) low temperature chemistry of diesel and (ii) high temperature chemistry, are observed for diesel HCCI combustion. At higher engine loads, rate of heat release is quite high (up to 100J/CAD), which is effectively controlled by EGR. As mixture becomes rich, combustion duration decreases rapidly (from 40 CAD for $\lambda = 4.96$ to 16 CAD for $\lambda = 2.56$), whereas EGR increases combustion duration due to delayed combustion. Diesel HCCI combustion offers several advantages such as higher efficiency and smooth engine operation however the operating window is relatively small.

References

[1] Mose FX, Sams T, Cartellieri W. Impact of future exhaust gas emission legislation on the heavy duty truck engine. SAE Paper 2001-01-0186; 2001.
[2] Onishi S, Jo SH, Shoda K, Jo PD, Kato S. Active thermo-atmosphere combustion (ATAC) – a new combustion process for internal combustion engines. SAE Paper 790501; 1979.
[3] Najt PM, Foster DE. Compression-ignited homogeneous charge combustion. SAE Paper 830264; 1983.
[4] Thring RH. Homogeneous-charge compression ignition (HCCI) engines. SAE Paper 892068; 1989.
[5] Canakci M, Reitz RD. Effect of optimization criteria on direct-injection homogeneous charge compression ignition gasoline engine performance and emissions using fully automated experiments and micro genetic algorithms. J Eng Gas Turb Power 2004;126:167–77.
[6] Dec JE, Kelly-Zion PL. The effects of injection timing and diluents addition on late-combustion soot burnout in a DI diesel engine based on simultaneous 2-D imaging of OH and soot. SAE Paper 2000-01-0238; 2000.
[7] Stanglmaier R, Roberts C. Homogeneous charge compression ignition (HCC): benefits, compromise and future engine applications. SAE Paper 1999-01-3682; 1999.
[8] Alperstein M, Swim WB, Schweitzer PH. Fumigation kills smoke – improves diesel. SAE Trans 1958;66:574–95.
[9] Ryan III TW, Callahan TJ. Homogeneous charge compression ignition of diesel fuel. SAE Paper 961160; 1996.
[10] Gray III AW, Ryan III TW. Homogeneous charge compression ignition of diesel fuel. SAE Paper 971676; 1997.
[11] Maurya RK, Agarwal AK. Experimental study of combustion and emission characteristics of ethanol fuelled port injected homogeneous charge compression ignition (HCCI) combustion engine. Appl Energy 2011;88:1169–80.
[12] Maurya, RK, Agarwal AK. Experimental investigation on the effect of intake air temperature and air-fuel ratio on cycle-to-cycle variation of hcci combustion and performance parameters. Appl Energy 2011;88(4):1153–63.
[13] Lu XC, Chen W, Huang Z. A fundamental study on the control of the HCCI combustion and emissions by fuel design concept combined with controllable EGR. Part 2: Effect of operating conditions and EGR on HCCI combustion. Fuel 2005;84:1084–92.
[14] Nakano M, Mandokoro Y, Kubo S, Yamazaki S. Effects of exhaust gas recirculation in homogeneous charge compression ignition engines. Int J Engine Res 2000;1:269. http://dx.doi.org/10.1243/1468087001545173.
[15] Komninos NP. Investigating the importance of mass transfer on the formation of HCCI engine emissions using a multi-zone model. Appl Energy 2009;86(7-8):1335–43.
[16] Komninos NP, Kosmadakis GM. Heat transfer in HCCI multi-zone modeling: validation of a new wall heat flux correlation under motoring conditions. Appl Energy 2011;88(5):1635–48.
[17] Suyin Gan, Hoon Kiat Ng, Kar Mun Pang. Homogeneous charge compression ignition (HCCI) combustion: implementation and effects on pollutants in direct injection diesel engines. Appl Energy 2011;88(3):559–67.
[18] Nakagome K, Shimazaki N, Niimura K, Koui7bayashi S. Combustion and emission characteristics of premixed lean diesel combustion engine. In: International Congress and Exposition. SAE Paper 970898; 1997.
[19] Hasegawa R, Yanagihara H. HCCI combustion in a DI diesel engine. SAE Paper 2003-01-0745; 2003.
[20] Kanda T, Hakozaki T, Uchimoto T, Hatano J, Kitayama N, Sono H. PCCI operation with early injection of conventional diesel fuel. SAE paper 2005-01-0378; 2005.
[21] Heywood JB. Internal combustion engine fundamentals. New York: McGraw-Hill Book Company; 1988.
[22] Sinha S, Agarwal AK. experimental investigation of the combustion characteristics of a biodiesel (rice-bran oil methyl ester)-fuelled direct-injection transportation diesel engine. J Aut Eng 2007;221:92.
[23] Torres GM, Jose F, Aguilar JE, Lencero T. Experimental Study of the Performances of a Modified Diesel Engine Operating in Homogeneous

Charge Compression Ignition (HCCI) Combustion Mode versus the Original Diesel Combustion Mode. Energy 2009;34:159–71.
[24] Furutani M, Ohta Y, Komatsu K. Onset behavior of low-temperature flames caused by piston compression. JSAE Rev 1993;14(2):12–8.
[25] Pucher GR, Gardiner DP, Bardon MF, Battista V. Alternative combustion systems for piston engines involving homogeneous charge compression ignition concepts – a review of studies using methanol, gasoline and diesel fuel. SAE Paper 962063; 1996.
[26] Rassweiler G, Withrow L. Motion pictures of engine flames correlated with pressure cars. SAE Paper 380139; 1938.