A review of Homogeneous charge compression ignition (HCCI) engine

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Abstract- Homogeneous charge compression ignition (HCCI) engine uses a relatively new mode of combustion technology. In principle, there is no spark plug or injector to assist the combustion process, and the combustion starts at multiple spots once the mixture has reached its auto ignition temperature. The challenges over the operation of HCCI mode engines are the difficulties of controlling the auto-ignition of the mixture, operating range, homogeneous charge preparation, cold start, controlling knock and emissions of unburned hydro carbon (UHC) and carbon monoxide (CO), and need to be overcome these difficulties for achieve successful operation of HCCI mode engine. This paper reviews the implementation of HCCI combustion in direct injection diesel engines using early, multiple and late injection strategies and reviews the different strategies of Homogeneous mixture preparation by external (manifold injection) and in-cylinder mixture preparation methods. Although this review focus on governing factors in HCCI operations such as injector characteristics, injection pressure, piston bowl geometry, compression ratio, intake charge temperature and exhaust gas recirculation (EGR) are discussed in this review. And also review the effects of design, combustion control technique and operating parameters on HCCI mode engine emissions, particularly NOx and soot are investigated. For each of these parameters, the theories are discussed in conjunction with comparative evaluation of studies reported in the specialised literature. Form this study we will be concluded that the efficient method of homogeneous mixture preparation and will be identified the parameter to achieve the combustion in HCCI mode engine.

Keywords: Homogeneous charge compression ignition, knocking, performance, emission and combustion characteristics.

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1. INTRODUCTION

Internal combustion (IC) engines are widely used in numerous applications throughout the world. A new mode of combustion is being sought in order to reduce the emissions levels from these engines: homogeneous charge compression ignition (HCCI) engine technology is a potential candidate. The HCCI technique is the process by which a homogeneous mixture of air and fuel is compressed until auto-ignition occurs near the end of the compression stroke, followed by a combustion process that is significantly faster than either Compression Ignition (CI) or Spark Ignition (SI) combustion [1]. The major disadvantage of SI engines is its low efficiency at partial loads. The compression ratio in SI engines is limited by knock and can normally be limited in the range from 8 to 12 contributing to the low efficiency.

Conventional diesel combustion, as a typical representation of CI combustion, operates at higher compression ratios (2–4) than SI engines. In this type of engine, the air–fuel mixture auto-ignites as a consequence of piston compression instead of ignition by a spark plug. The processes which occur between the two moments when the liquid fuel leaves the injector nozzles and when the fuel starts to burn are complex and include droplet formation, collisions, breakup, and evaporation and vapour diffusion. The rate of combustion is effectively limited by these processes. A part of the air and fuel will be premixed and burn fast, but for the larger fraction of the fuel, the time scale of evaporation, diffusion, etc. is larger than the chemical time scale.

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Therefore, the mixture can be divided into high fuel concentration regions and high temperature flame regions. In the high fuel concentration regions, a large amount of soot is formed because of the absence of O_2. Some soot can be oxidized with the increase of in-cylinder temperature.

The in-cylinder temperature in a Conventional diesel engine is about 2700 K, which leads to a great deal of NOx emissions. HCCI technology claimed to improve the engine thermal efficiency while maintaining low emissions and can be implemented by modifying either SI or CI engines using any fuel or combination of fuels [5,6]. The air/fuel mixture quality in HCCI engines is normally lean, it auto ignites in multiple locations and is then burned volumetrically without discernible flame propagation [7].Combustion takes place when the homogeneous fuel mixture has reached the chemical activation energy and is fully controlled by chemical kinetics [8] rather than spark or injection timing. Since the mixture is lean and it is fully controlled by chemical kinetics, there are new challenges in developing HCCI engines as it is difficult to control the auto-ignition of the mixture and the heat release rate at high load operation, achieve cold start ,meet emission standards and control knock [9,10]. The advantages of using HCCI technology in IC engines are: (1)high efficiency relative to SI engines approaching the efficiency of CI engines due to the ability of these engines to high compression ratio (CR) and fast combustion [8,9]; (2) the ability to operate on a wide range of fuels [9–11]; and (3) the ability to be used in any engine configuration: automobile engines, stationary engines, heavy duty engines or small sized engines [2,12,13]. On the other hand, HCCI engines have some disadvantages such as high level so fun burned hydrocarbons (UHC) and carbon monoxide (CO) [6, 14, 15] as well as knocking under certain operating conditions [6, 14, 16]. Emissions regulations are

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becoming more stringent and NOx and soot emissions levels in HCCI engines have been greatly reduced without sacrificing efficiency, which is close to that of CI engines [14]. However, knocking is still the major issue because of its sudden onset. Knocking is due to premature combustion where the ignition takes place before the piston reaches top dead centre (TDC) and it reduces engines reliability due to high vibration effects. The performance of an HCCI engine is strongly dependent on the fuel type, and this affects the emissions levels as well. Since the emissions levels become one of the factors driving engine technology today, HCCI development has moved to a next level. Due to the importance of HCCI technology, which potentially can replace the conventional SI and CI engines, there is a need to report the recent development of HCCI engines. This paper discusses the various parameters, behind the operation of HCCI mode engine

2. FUNDAMENTALS OF HCCI ENGINE

Homogeneous charge compression ignition (HCCI) is a combustion concept that constitutes a valid approach to achieve high efficiencies and low nitrogen oxides and particulate emissions in comparison with traditional compression ignition (CI) direct injection (DI) engines [2]. Although HCCI combustion was demonstrated about 20 years ago [3], only the recent advances made an intake air charge, varying the compression ratios and exhaust gas recirculation (EGR) have made it controlled combustion and less knocking. The schematic HCCI mode engine setup shows in Fig1. HCCI has been successfully applied both to spark ignition (SI) and compression ignition (CI) engines, and proved to be fuel flexible. It has been achieved using with gaseous fuels such as propane or natural gas and liquid fuels like traditional gasoline or diesel fuels. Fig2 shows the different operating principle of SI, CI and HCCI engine. The HCCI process operates on the principle of having a lean, premixed, homogeneous charge that reacts and burns volumetrically throughout the cylinder as it is compressed by the piston [4].

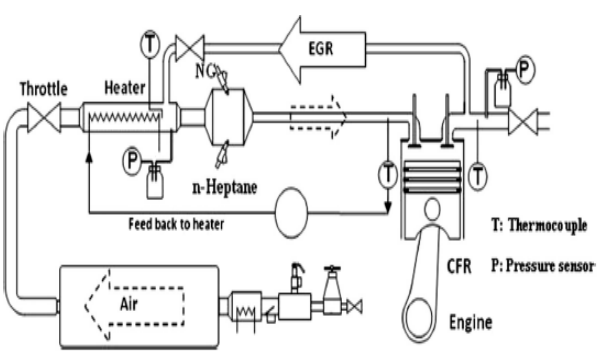

Fig.1 Schematic of experimental setup of HCCI mode

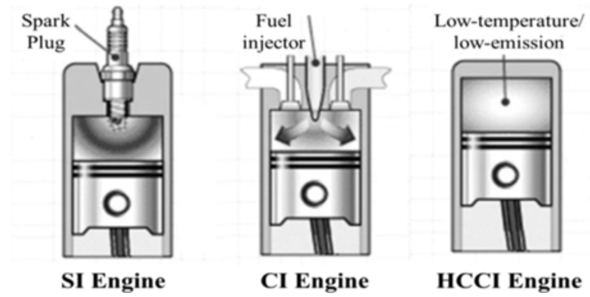

Fig. 2 Schematic diagram of SI, CI and HCCI

The mixture can be prepared either internally or externally. The internal preparation of homogeneous air-fuel charge in HCCI engine by injected the fuel at combustion chamber using in cylinder injector during the suction stroke. It can prepare the homogeneous mixture in HCCI mode engine. In externally preparation of homogeneous air-fuel mixture by provide the electronic controlled injector on engine intake manifold instead of DI method. In this method the fuel is injected on the air stream during the suction stroke. It gives enough time to prepare the homogeneous charge. The premixed charge minimizes particulate emissions because the combustion occurs simultaneously throughout the cylinder volume rather than in the form of a flame front. This avoids the well known drawbacks of mixing – controlled combustion that lead to soot formation in DI Diesel engines. Moreover, since the combustion occurs without flame propagation, it leads to lower gas temperatures, thus reducing NOx emissions [5].

3. PERFORMANCE AND EMISSION COMPARISON

3.1 Combustion

The heat release rate of HCCI mode engine is depends on the fuel, which is used in HCCI mode engine. Rate of heat release is based on combustion percentage and point of combustion starts in combustion chamber. If the combustion starts at end of the compression stroke, it leads to increase the heat release rate and also if the combustion starts at piston moves to BDC. It reduced the combustion efficiency and heat release rate (16).From the fig3 shows the rate of heat release rate of HCCI mode engine has higher than conventional CI engine. In HCCI mode of operation the fuel and air mixed before enter to the combustion chamber. This homogeneous charge leads to reduce the ignition delay and combustion duration. And the combustion will be stated at end of the compression stroke.

The addition of biogas in dual fuelled HCCI mode engine, increase the ignition delay and also reduced the heat release rate. This was because of the presence of CO_2 and also because of methane presents in biogas has a high self ignition temperature. The variation heat release rates at a charge temperature of 80ºC indicates that though the peak heat release rate increase with BMEF, combustion gets delayed. This was because at higher BMEFs larger biogas energy ratios to be used while working at the knock limit for good thermal efficiency. Even though the maximum heat release rate raise with BMEF the MRPR reduced (17).

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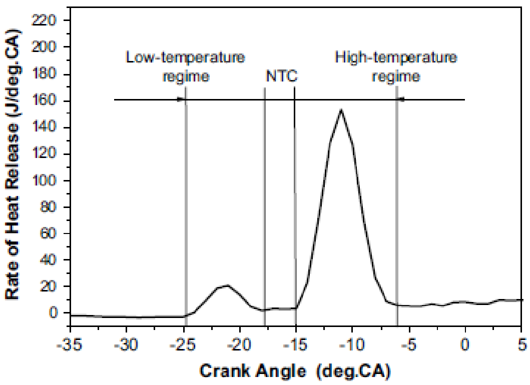

Fig3. Heat release curve of HCCI mode combustion

Canakci et al.(18) analysed the heat release rate based on cylinder gas pressure data. The differences in cylinder pressures and apparent heat release rates caused by the boost pressures at different engine speeds for the same global equivalence ratio and intake air temperature. The start of combustion was advanced with increasing boost pressure. So that the peak of cylinder pressure increased and the heat release rate advanced with increasing boost pressure. The reason for this is the combustion reaction velocity which increases with the increasing boost pressure. When the heat release rates are compared, it is apparent that only the cool flame (pre combustion which appears before the main reaction) occurred in 138 kPa boost pressure and 700 rpm engine speed case. In other engine speeds, the cool flame regions are very small for the highest boost pressure which may be neglected. The combustion durations at 1300 rpm case are slightly shorter at all boost pressures when compared to other engine speeds. Combustion duration decreases with increasing boost pressure and engine speeds. As seen in the figure, the maximum rates of pressure rises were increased with increasing boost pressure and they were obtained at 1300 rpm engine speed case. However, there is no significant difference on the points of the maximum in-cylinder pressures (Pmax) when the engine speed was changed.

Almost same Pmax values were obtained for each boost pressure at the different engine speeds. However, the points of Pmax closed to top dead center (TDC) with increasing boost pressure. If the combustion process takes place only after TDC, the combustion rate will be slower because of the volume increase during the combustion. The results of this study indicate that the SOI's were selected properly to achieve the maximum engine torque for each case. As expected, the Pmax values increased with increasing boost pressure. This increment occurred sharply at 1300 rpm case for all boost pressures because of the late SOI and short combustion duration, e.g.; less homogeneous mixture. At this operating condition, there was no any knocking in the engine. When the engine speed was increased at the 101 kPa and 117 kPa boost pressures, the SOI was retarded to achieve the maximum engine torques. However, for the 138 kPa boost pressure, the SOI changed slightly when the engine speed was varied from 700 to 1300 rpm. It is evident that, with the increase of engine speed, the combustion duration will be shortened with respect to time. The earlier SOI represents a more homogeneous mixture, while the later one represents a more fuel-stratified mixture.

Many studies have focused on investigating the effect of EGR on HCCI combustion engines. EGR dilution has been commercialized in both spark ignition (SI) and compression ignition (CI) engines mainly to decrease the combustion high temperature and to reduce the related NOx emission. The maximum amount of EGR in conventional engines is restricted by the in-cylinder charge dilution limit dictated by flammability limits. These limits are much wider in an HCCI combustion engine. Zhao et al. [19] investigated how each EGR effect changes the operating parameters. They identified five different effects of EGR: the charge heating, dilution, heat capacity, chemical, and stratification effects. They concluded that the charge heating effect advances the auto-ignition, increases the heat-release rate, and shortens the combustion duration. The dilution effect is not responsible for phasing the auto-ignition, but extends the combustion duration and slows down the heat-release rate. The heat capacity effect, like the dilution effect, extends the burn duration.

The chemical effect just slightly reduces the combustion duration at high EGR rates. HCCI combustion is facilitated by the presence of a high temperature mixing region between the burned gases and the fresh fuel/air mixture as the stratification effects. When EGR is used, no change in combustion efficiency has been reported [20], because the absolute combustion speed is fast enough to compensate the dilution effect. Applying EGR affects in-cylinder pressure. This reduction is due to the fact that recycled exhaust gases serve as a thermal sink, which in turn, as afore mentioned, reduces the charge temperature during combustion. As can be seen, the pressure rise rate reduces with EGR, too. This decrease is to some extent due to the lower peak temperature. By the way, in a fixed air/fuel ratio the delivered fuel mass decreases with EGR which results in a reduction in combustion pressure and temperature.

The reduction in pressure rise rate by EGR is a means to avoid incidence of knocking. Heat transfer rate is also calculated from MFLHR model. Heat transfer coefficient is proportional to pressure and inversely proportional to temperature. Heat transfer coefficient does not vary significantly with EGR, since by increasing EGR rate both temperature and pressure experience a reduction. However, as afore mentioned, temperature decreases with EGR, hence the heat flux of convective heat transfer reduces with EGR as a result of charge temperature reduction and thus temperature difference reduction. As can be seen, applying EGR has profound effect on HRR magnitude and phasing. Recycled exhaust gases substitute part of fresh intake, and as a result affect in-cylinder equivalence ratio, and consequently the combustion process. Also the species in the recycled exhaust gases by their chemical effects can be utilized to control the HRR. This can delay SOC and prolong burn duration. These trends increase operating range and removes maximum IMEP limit. EGR increases specific heat of the charge. Thus, by applying EGR, in-cylinder charge temperature decreases and as a result the ignition tends to be initiated later in the cycle. As mentioned before, EGR introduces non-reacting species

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such as tri-atomic molecules and also affects in-cylinder equivalence ratio. These trends tend to slow down the pre-ignition reactions leading to auto-ignition of the charge and prolong the combustion duration. The results indicated that increased EGR rate led to an increase in combustion duration.

3.2 Brake thermal efficiency

The brake thermal efficiency (BTE) of an engine is the ratio of brake output power to input power and describes the brake power produced by an engine with respect to the energy supplied by the fuel. A study has been performed [20] to determine the best BTE for biogas fuels with an energy ratio ranges from 40% to 57% of biogas and an intake temperature of 80 ºC, 100ºC and135ºC. The biogas is main component was methane (460%), which was produced by a naerobic fermentation of cellulose bio mass materials [21]. It was reported that the best energy ratio with diesel in HCCI mode was 51% biogas and the optimum efficiency occurred when the intake temperature was 1351C.High energy ratios lower the heat release rate and the efficiency [20]. However, even when operated at the optimum biogas energy ratio, the BTE was no better than diesel running in CI mode. Duc and Wattanavichien [21] reported that biogas-diesel running in dual-fuel non-HCCI engines has a lower efficiency than diesel single fuel in either HCCI or non-HCCI mode. Hydrogen,on the other hand,had a higher BTE than pure diesel in non-HCCI mode, increasing the BTE by13–16% [30]. Szwaja and Grab Rogalinski [22] reported that the BTE was increased from 30.3% to 32% with an addition of 5% hydrogen. The increase of BTE in hydrogen diesel mode might be due to the uniformity of mixing of hydrogen with air [23].

Hydrogen as a single fuel running in HCCI mode gives a BTE of upto 45% [24], showing that hydrogen is able to operate with extremely lean mixtures and still maintain a relatively high efficiency compared to diesel engines. Table5 shows that hydrogen in HCCI mode yields better results compared to the conventional mode (45% vs.42.8%)and hydrogen with diesel in non-HCCI mode produces a. Therefore, it is expected that the combination of hydrogen and diesel in HCCI mode will produce a higher BTE than diesel in CI mode. This conclusion was reached with partially premixed charge compression ignition (PPCCI) configurations when reformed exhaust gas recirculation (REGR) was used with hydrogen-rich gas (no morethan24%) added to the intake manifold [25]. Some researchers use PPCCI instead of HCCI as they differ in their injection methods and both of them have the same purpose: to achieve a homogeneous mixture.

The brake thermal efficiency is calculated by dividing the actual brake work to the amount of fuel energy input. The brake thermal efficiency showed similar trend with brake torque which increased with increasing boost pressure. However, when compared the brake thermal efficiency to combustion efficiency for the engine speed, an inversely relationship was found in general. This means that, with increasing boost pressure, the brake thermal efficiency increased while the combustion efficiency decreased. For each injection pressure down the homogeneous combustion rate, especially at high boosting pressure. MEP is a useful engine performance parameter to compare the engine output since the cylinder gas pressure in an engine continuously changes during the cycle. In general, the MEP's increase with increasing boost pressure and decreased with increasing engine speed, except for 138 kPa boost pressure case. When compared the changes between IMEP and BMEP at each case, it is seen that the change trend is not same because of the efficiencies which depend on the engine speed and boost pressure. The reasons for this may be explained with the change in volumetric efficiency and increased friction forces at the higher engine speeds. The maximum IMEP was obtained at 138 kPa boost pressure and 1300 rpm engine speed case.

Almost in all cases, it can be said that there are no direct relationships obtained between the peak values of the heat release or maximum cylinder pressure and the maximum IMEP value because of the different SOI timing, the rate of cylinder pressure rise and combustion duration. To keep the same equivalence ratio for three boost pressures, it is necessary to increase the fuel flow rate since higher amount of air will be introduced into the cylinder at higher boost condition. Therefore, this caused lower BSFC to be obtained at the high boost pressure due to more IMEP produced. At the same time, the reduction in the IMEP with increasing engine speed causes higher BSFC at higher engine speeds for each boost pressure [22]. The results indicate that an increase in the boost pressure causes the need of leaner mixture, and requires more advanced injection timing to achieve the maximum engine torque.Swami Nathan et al. (20) Investigated biogas-diesel fuelled HCCI mode engine with different intake charge temperature as 80 ºC, 100ºC and 135ºC. The variation of intake air temperature is used to control and improve the combustion process. The brake thermal efficiency of the engine lies of combustion efficiency. The experimental result showed that the brake thermal efficiency of HCCI mode engine was reduced compared to diesel fuelled CI mode engine.

3.3 Specific Fuel consumption

Indicated specific fuel consumption (ISFC) shows how effectively the charged fuel is utilized in the engine to do work on pistons. HCCI engine has consuming less fuel than conventional CI engine. The HCCI mode engine operates under lean air and fuel mixture at all loads conditions. The main factor contributing to the increasing SFC are as follows, the off- phasing of combustion process and the negative work due to spilt the combustion stages are the first reasons. Second, the significant increase in HC and CO emissions observed is indicative of fuel energy loss due to incomplete combustion (26). The SOC retardation tends to reduce specific fuel consumption; meanwhile increasing EGR percentages has opposite effect due to its influence on incomplete combustion. This trade-off led to a decrease in SFC with EGR which is the effect of SOC retardation.

3.4 Exhaust gas emissions

Emissions levels have become a major focus in new engine developments now a day because regulatory bodies, such as those in Europe, the United States (US) and Japan are imposing stringent vehicle emissions quality standards [27–29]. Emissions in HCCI engines consist of UHC, CO, NOx, and particulates. HCCI engines claimed to have low emissions

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levels on NOx and particulate matters [14] and high levels of unburned hydrocarbons (HC) and carbon monoxide (CO) [30].Fig 3 shows the region of HCCI combustion well above the UHC and CO oxidation limit and escapes the formation of both NOx and soot. However, the emissions levels of an engine is vary from one engine to another and is dependent on the operating conditions of the engine, fuel properties and the engine design [31,32]. Thus, the emissions levels from different engines cannot be directly compared with another engine because of those factors

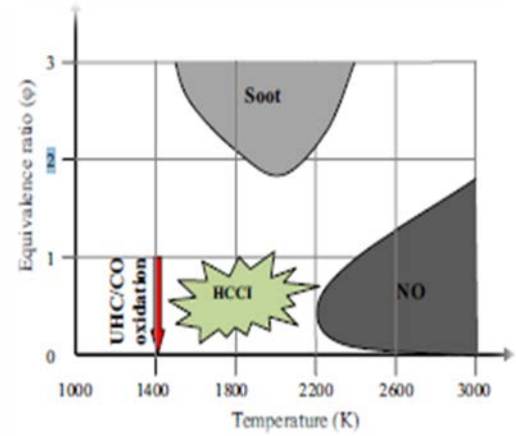

Fig.4 Reduction of NOx and Soot in HCCI combustion (33)

3.4.1. UHC, CO and CO_2

UHC and CO emissions in HCCI engines are higher than the conventional diesel engines as reported in [34, 35] and have two major challenges that need to be resolved. Unburned hydrocarbon is the consequence of incomplete combustion caused by low combustion temperatures [36, 37] which cause deposition of fuel in boundary layers and crevices. The level of UHC is generally specified in total hydrocarbon concentration, which is expressed in parts per million carbon atoms [38]. The source of UHC is reported from the crevice region, cylinder wall with thin layer of oil left when the piston moves down and any combustion wall which has a cold area [39, 40].The amount of CO_2 and CO is dependent on the combustion efficiency, where the combustion efficiency can be defined as the ratio of CO_2 to the total of fuel carbon present in the exhaust including CO, CO_2 and UHC [41]. CO emissions are controlled primarily by the fuel– air equivalence ratio [39, 42]. The principal reaction mechanism of CO formation is RH-R-RO_2-RCHO-RCO-CO, where R is the hydrocarbon radical [39].

The CO oxidation is dominated by the reactions of CO with OH radicals as COþOH) CO_2 pH, which also forms hydrogen radicals. The conversion of CO to CO_2 occurs when the concentration of OH radicals increases during combustion [43]. For the HCCI engine with a low combustion temperature, the OH level is reduced resulting in an incomplete oxidation of CO [44]. The low combustion temperature will reduce the combustion efficiency as a result of the lower oxidation activity of the hydrocarbons and the lower conversion rate of CO to CO_2 [45]. Results of simulations confirm that the piston ring crevice needs to be resolved in order to accurately predict UHC and CO emissions. UHC and CO emissions originate in the crevices and boundary layer, which are too cold for complete consumption.

3.4.2 NOx and particulate matter (PM)

The NOx formation is explained by several formation mechanisms: Zeldovich mechanism, Fenimore mechanism, fuel bound NOx, NO_2 mechanism and N_2O mechanism. In the Zeldovich mechanism, also known as thermal NOx, the NOx is not formed from the fuel because the reisno nitrogen component in the fuel. NOx is for medina high temperature reaction, where the nitrogen in air dissociates in to nitrogen radicals to form NO when reacting with oxygen. Some NO is converted to NO_2 when further reactions occur in the chamber. The thermal NOx is not significant when the combustion temperature is below 1800K [46]. Under the Fenimore mechanism, also known as prompt NOx, the NOx is promptly formedin laminar premixed flames long before the NOx is formed by the thermal mechanism. The Fenimore mechanism explains the additional NOx produced over the Zeldovich mechanism in hydrocarbon flames. Prompt NOx is important for hydrocarbon fuels in fuel-rich conditions, where NOx is formed by rapid reactions of hydrocarbon radicals (CH, CH_2, C_2, C_2H and C) with molecular nitrogen.

Miller and Bowman [46] reported that NOx formed by thermal mechanism is the dominant source of NOx only in the equivalence ratio range of Ø ¼ 0:8_1:0. For fuel–air equivalence ratio, Øo 0:8, the temperature is sufficiently low and NOx was formed by Fenimore mechanism. The fuel-bound NOx mechanism is used for coal and coal- derived fuels, where nitrogen exists as chemically bound to the major fuel. The NOx formation is dependent on the local combustion temperature, stoichiometric conditions and the level of the nitrogen compounds in the fuel–air mixture. The NO_2 mechanism on the other hand, is based on the chemical kinetic calculations near the flame zone, where NO_2 is formed due to the reaction between NO and HO_2. The NO_2 then reacts with Hand Oradicals to form NOx. The N_2O mechanism is also based on the chemical kinetic calculations, where the N_2O is formed due to the reactions of various nitrogen radicals with NO. The N_2O will finally react with oxygen radicals to form NOx. In short, the NOx formation is still under investigation and one cannot claim that all routes have been found [47]. Generally, most parts of the NOx formation are determined by the peak temperature during combustion, where the peak temperature is dependent on other parameters as well such as equivalence ratio, fuel composition and the initial temperature of the fuel–air mixture [46]. The region of formation of NOx and Soot emissions in diesel combustion depend on equivalence ratio as shown in fig5.

Junnian et al.(48) Investigated effect of operating parameter on NOx emission and NOx formation mechanism of natural gas fuelled HCCI engine To using the single zone simulations, Figs. 6–8 show the contributions of the different NOx mechanisms to the total NOx emissions at a few typical operating conditions. Using a stacked area chart, Fig. 6 shows the total NOx concentration and the NOx from thermal, prompt, nitrous oxides, and NO2 mechanisms as a function of engine speed. As engine speed increases, the ignition timing is retarded (a result of the kinetics) and the IMEP values at

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different engine speeds are different. In Figs. 7 and 8, the effects of equivalence ratio and EGR level on the different mechanisms have also been examined. The increase in equivalence ratio restrains the importance of N_2O intermediate mechanism, while enhancing the thermal and prompt mechanisms as the mixture becomes richer, and the gas temperature becomes higher. Higher EGR percentage greatly reduces the absolute value of NOx from each mechanism due to the dilution effect and the lowered temperature. However, the increase in EGR level slightly increases the contribution of N_2O mechanism which is less dependent on temperature than the other mechanisms.

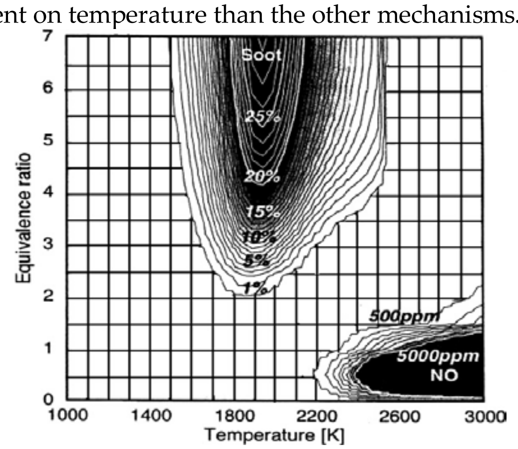

Fig.5 Formation of NOx and Soot emissions (47)

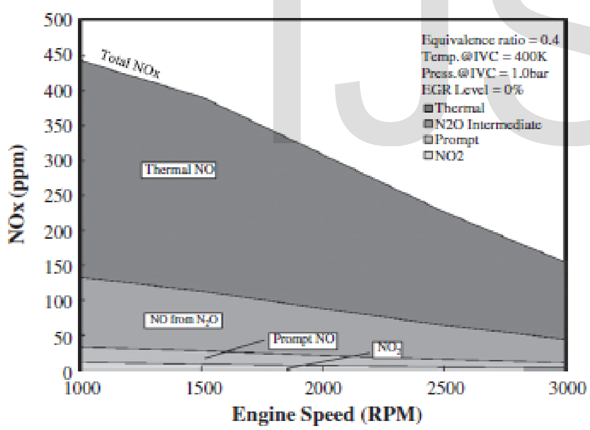

Fig.6 Formation of NOx with engine Speed

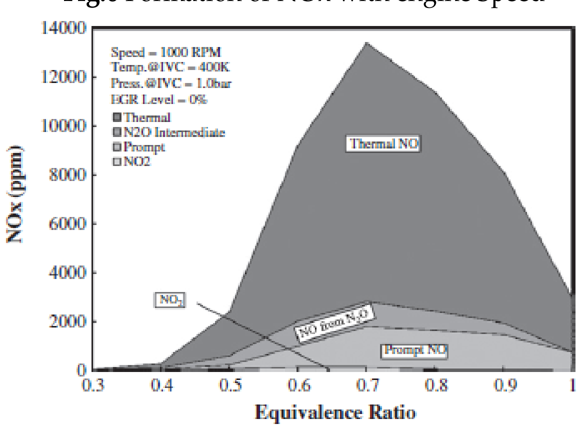

Fig.7 Effect of equivalence ratio on NOx emissions

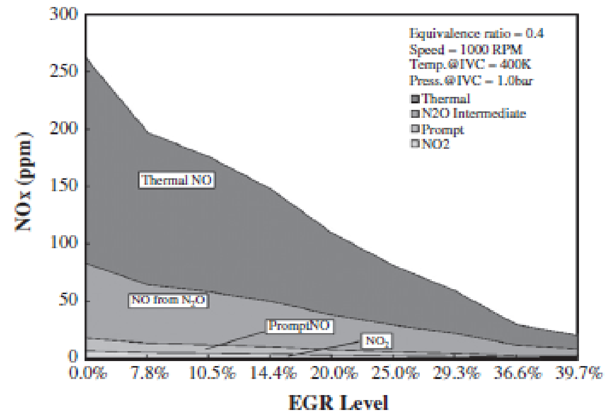

Fig. 8 Effect of EGR percentage on NOx emissions

Tanaka et al. [49] also reported that the NOx levels are increased substantially when the equivalence ratio about 0.7 and get reduced at higher equivalence ratio. It causes the maximum in-cylinder temperature to exceed 1800K and produce more NOx. In HCCI engines, the NOx and particulate matters (PM) are reported to be very low [49, 50] by implementing high CR engines. NOx can also be reduced using high EGR rates by reducing the local temperature and decreasing the oxygen amount in the cylinder [42]. However, unburned HC and CO emissions will increase due to insufficient oxygen. Bression et al. [37] implemented a high pressure loop EGR without cooler and Variable Valve Timing (VVT) to reduce the NOx, unburned HC and CO by increasing the combustion temperature. They reported that internal EGR by using VVT is an effective way to reduce the NOx instead of using high pressure EGR.

Papagiannakis and Hountalas [51] investigated the combustion and exhaust emission characteristics of a CI engine fuelled with a blend of diesel (direct injected) and natural gas (port injected). They found that the NOx level for dual fuel engine operation is lower than diesel CI engine. The NOx can also be reduced by using hydrogen addition. The concentration of NOx between natural gas and hydrogen in diesel HCCI mode is different due to different combustion temperatures, because hydrogen has a higher temperature and flame speed compared to natural gas and diesel [52]. A survey of research papers by Akansu et al. [53] shows that the NOx level is increased when the hydrogen content in natural gas hydrogen mixtures is increased. It shows that the combustion temperature and the flame speed of hydrogen contribute to a higher level of NOx emissions. In biogas–diesel HCCI engines, the NOx level was low when the biogas energy was increased (54). This might be due to a higher homogeneity level achieved between air and fuels. Van Blarigan [55] in his study reported that the mixture must be homogeneous and lean in order to eliminate the production of NOx.

Olsson et al. (56) Stated that the NOx level is low in natural gas HCCI engines and when combined with exhaust gas recirculation (EGR), it drops further. Even in natural gas–diesel non-HCCI mode, the NOx level is lower than diesel conventional CI engines [57]. Hydrogen on the other hand produces zero UHC, CO and CO2, due to the absence of carbon in the fuel, but still produces NOx [44]. Hydrogen operated as a single fuel in CI mode yielded lower NOx levels than diesel [51]. Saravanan and Nagarajan (58) in their study of hydrogen addition to diesel in non-HCCI mode showed that lower NOx emissions were obtained for all load ranges compared to diesel in conventional mode. They reported that the formation of NOx depends on temperature more than the availability of oxygen. Therefore, hydrogen diesel in HCCI mode results in extremely low NOx emissions levels with no significant amount of PM [59]. Kayes and Hochgreb [60, 61] developed a mechanism model for PM and they reported that PM is produced due to liquid fuel nucleation from port fuel injection. The PM can also be produced from gas-phase nucleation in fuel rich conditions. The air-to-fuel ratio strongly affects PM and burning liquid fuel will increase the amount of PM in the exhaust gas emissions [62], hence the use of a high pressure injection is advisable so that the fuel is in an atomized condition. PM can be reduced significantly when hydro- gen is added to a diesel fuelled engine [49] and also by heating the inlet [59]. When the engine runs in a dual fuel mode between diesel and natural gas, no significant reduction in PM was reported [63]. Hydrogen addition, on the other hand, shows a good advantage in reducing the PM emission levels.

3.5 Knocking

Knocking in SI engines is a phenomenon where the unburned mixture in the compressed gas ignites before it is reached by the propagating flame front [64]. Knock is physically detected when the engine vibrates excessively and a pinging sound can be heard outside as a result of the combustion activity. It causes loss of power and if not controlled, knocking could lead to severe engine damage and shorten its life. Knocking can occur in any reciprocating engine. HCCI engines are prone to knock since they are controlled by chemical kinetics and there is no fixed mechanism to control knock in them. Knocking phenomena limits the load range of an HCCI engine: high load operations can easily initiate knock, so upper load limits have to be applied.

In all engines, knocking occurs when the combustion starts before the piston reaches TDC, while misfire is when combustion commences after TDC. Knocking and misfire are two different behaviours which must be avoided in engine operation as both of them can contribute to deterioration of engine performance. In natural gas–diesel HCCI engines, a misfire occurs when a high natural gas flow rate is combined with a low diesel flow rate while knocking starts with the opposite configuration (65). If HCCI engines operate on hydrogen–diesel fuels, knocking is expected to occur when high amount of hydrogen is added. Knocking will take place if the hydrogen content is more than 16% of the energy ratio [56] which is confirmed by Guo et al. [66], who reported that it was necessary to use hydrogen with mass fraction less than 15% to achieve stable combustion. If an HCCI engine runs on natural gas with Di methyl ether (DME), the knock limit is at an in-cylinder pressure of 9MP and a very limited load range is obtained while getting unstable operation for high natural gas concentrations.

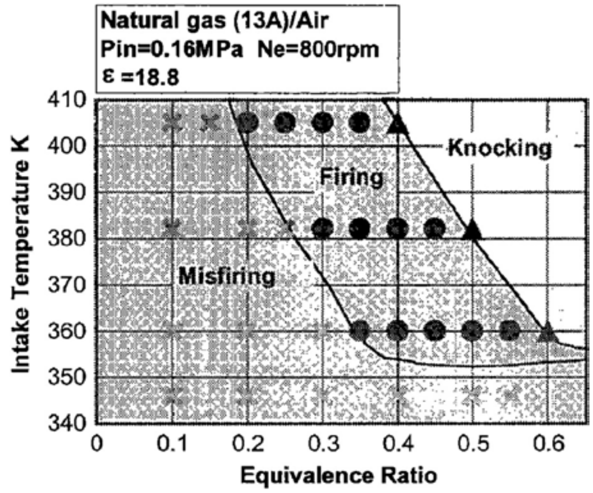

Fig.9 Knocking and misfire region of natural gas fuelled HCCI engine (16)

An investigation of auto-ignition and combustion of natural gas HCCI engines showed that knocking starts to happen when the equivalence ratio is less than 0.45 with an intake temperature of 380K [62] as shown in fig10.Thus, natural gas fuelled HCCI engines will not be able to operate at high load conditions due to this limitation. Knocking occurs when a rapid release of energy in the remaining unburned mixture causes a rapid increase in local pressure. In hydrogen–diesel engines, the engine will not work in a stable condition if 100% hydrogen is used. A combination of hydrogen and di ethyl ether (DEE) produces knock on 100% load [58]. As such, higher hydrogen content leads to a higher probability of knock on set.

Knocking phenomena can be detected in the in cylinder pressure variations, observing a rapid instantaneous local pressure rise. The graph formed depends on the knocking frequency: the higher the frequency, the more severe the knock [22].Fig11 shows the in cylinder pressure generate during the combustion. The maximum increase in pressure for knocking is unpredictable and this abnormality usually occurs when there is an incorrect operating condition.

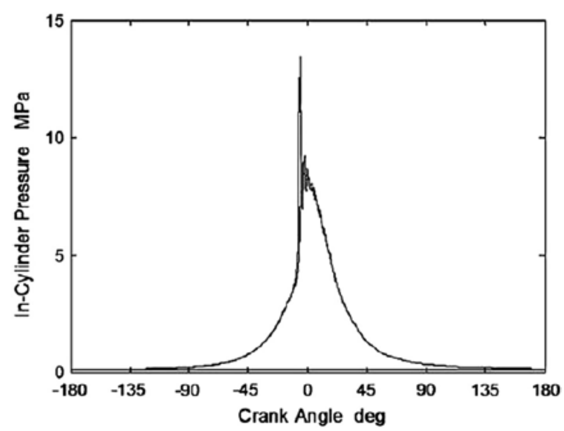

Fig.10 In-cylinder pressure formation in a Hydrogen HCCI engine (22)

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Flowers et al. [67] have described the HCCI ignition by H_2O_2 decomposition. It accumulates due to low temperature reactions in the compression stroke. Once the in-cylinder temperature reaches 1050–1100 K, H_2O_2 decomposes rapidly into two OH radicals forming an enormous amount of OH radicals, which will rapidly consume the fuels. Reduction in OH concentration in low temperature regions leads to incomplete combustion, which delays the high-temperature oxidation [68]. However, there is no concrete mechanism in the literature describing the knocking phenomenon in HCCI engines and the methods to control it. In SI engines, it is quite easy to control knock. For example, car manufacturers usually install knock and oxygen sensors in the engine. If the mixture is prone to generate knocking, the sensors will automatically adjust the spark timing accordingly. However, in HCCI engines, there is no spark to control such situations and there is a strong dependence on the right mixtures and conditions.

4. IMPLEMENTATION OF HCCI COMBUSTION IN DI DIESEL ENGINES

Considering that homogeneity of charge is the key feature of HCCI combustion, the fuel injection process, particularly the injection timing is crucial to the development of HCCI combustion. The following discusses how HCCI combustion is implemented in both commercial and research DI diesel engines using various injection timing strategies. Other governing factors in the development of stable HCCI operation such as injector characteristics, injection pressure, and piston bowl geometry and compression ratio are also reviewed.

4.1. Injection timing

Since charge homogeneity is necessary for HCCI combustion, port fuel injection would appear initially to be the obvious way to generate a homogeneous mixture by allowing sufficient mixing time. However, due to the high boiling point range of diesel fuel which leads to poor vaporisation at typical diesel intake manifold temperatures, it is not feasible to deliver diesel fuel via port injection [69]. Smoke and HC emissions increase as the non-evaporated liquid fuel adheres to the walls of the intake system and combustion chamber [70]. Although increasing the intake gas temperature can alleviate to a certain extent the problem of poor vaporisation, engine efficiency is lowered [71]. Thus, direct injection into the combustion chamber is used for current diesel engines to eliminate the need to raise intake temperature to counter the vaporisation problem encountered in port injected systems. To achieve the required degree of air fuel mixture homogeneity necessary for HCCI combustion, injection timing is of paramount importance since the length of mixing time of air and fuel is dependent on when the fuel is injected into the combustion chamber. Here, HCCI operations can be broadly classified into three categories: early injection, multiple injections and late injection.

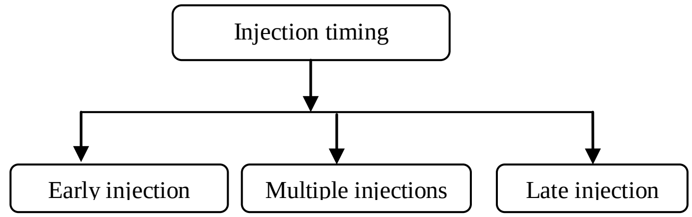

Fig.11 Strategies of Injection timing

4.1.1. Early injection

In conventional HCCI combustion, fuel is injected early in the compression stroke, which allows sufficient mixing time for the formation of a homogeneous mixture. Takeda et al. [72] described one of the earliest implementation of this strategy in the premixed lean Diesel Combustion (PREDIC) mode operated on a DI four-cycle naturally aspirated single-cylinder engine. Using two side injectors and one centre injector, different injector configurations were tested in combination with varying injection timings and excess air ratios, k. For fixed excess air ratios, the ranges of operational injection timings were limited to misfiring (too early) and knocking (too late). For instance, at k = 2.7, injection timing had to be between approximately 80º Before Top Dead Centre (BTDC) to 60ºBTDC. NOx emissions were significantly lower under PREDIC operation compared to conventional diesel combustion, with values reaching as low as 1/10 of the minimum concentrations emitted by standard diesel operation.

However, this was accompanied by increased HC and CO concentrations due to over-leaning of the air fuel mixture. The shortcomings of the PREDIC combustion system included limited partial load operation and lack of ignition timing control [73]. The Premixed Compression-Ignited (PCI) combustion system developed by Mitsubishi Motors Corporation is another example of early injection strategy to form a premixed lean mixture prior to self-ignition [74]. With early injection in the range of 75– 40º BTDC and an equivalence ratio of 0.38, comparison tests between a conventional hole nozzle and an impinged-spray nozzle were carried out on a DI, four-stroke, single-cylinder diesel engine. With the former, spray penetration was found to be significant with fuel adhering to the combustion chamber wall. Compared to conventional diesel combustion, NOx emissions reduced to a very low level but both HC and smoke emissions increased. Using an impinged-spray nozzle allowed high fuel dispersion and low spray penetration which resulted in low emissions of both NOx and smoke. The PCI system still produced relatively high HC emissions and inferior fuel consumption in contrast to conventional combustion. Suggested counter-measures include the use of an oxidation catalyst and optimising the compression ratio. Likewise to PREDIC, PCI combustion was limited to partial load conditions. Under high load conditions, the formation of a lean, premixed air fuel mixture was not possible due to the larger amount of injected fuel leading to insufficient mixing with air and knocking.

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4.1.2. Multiple injections

Multiple injections strategies have been developed subsequently to single early injection strategy for diesel combustion in HCCI mode. The systems featuring multiple injections include the multiple stage Diesel Combustion (MULDIC) [75] and Homogeneous charge intelligent Multiple Injection Combustion System (HiMICS) [76, 77]. In the former, following an early injection at 150ºBTDC, a second injection occurs within the range of 2ºBTDC to 30º After Top Dead Centre (ATDC) as can be seen in Fig. 12. The first stage of combustion is premixed lean combustion which lowers NOx emissions, while the second stage is diffusion combustion which occurs under high temperature and low oxygen conditions.

This enabled the MULDIC operating range to be extended to higher load conditions, with substantial reduction in NOx emission at higher load conditions, compared to the PREDIC system. The HiMICS, a combination of a very advanced preliminary injection followed by a main injection around Top Dead Centre (TDC) and a late stage injection at approximately 30ºATDC to reduce smoke is carried out. Compared to standard injection and pilot injection cases, trade-off relations worsen between NOx and fuel consumption as well as NOx and smoke in the region of ordinary injection timing.

In the region of excessively retarded timing, these trade-off relations can be improved for HiMICS due to the ultra low NOx emission. Nevertheless, drawbacks of this combustion system included high levels of HC and CO emissions, premature ignition and inadequate homogenisation of the pre mixture. An early injection is implemented for the creation of a homogeneous mixture through fuel diffusion and fuel conversion to lower hydrocarbons (LTR). A late injection is used to trigger combustion of all fuel, including the partially combusted fuel from the first injection and the premixed fuel of the second injection. Optimisation experiments with first injection timing and quantity showed that the early injection needs to be between 54 ºBTDC and 36ºBTDC with 5 mm3/st of fuel. The late injection consisting of 25 mm3/st of fuel is fixed at approximately 3ºATDC. Low NOx as well as smoke emissions were obtained for this combustion system.

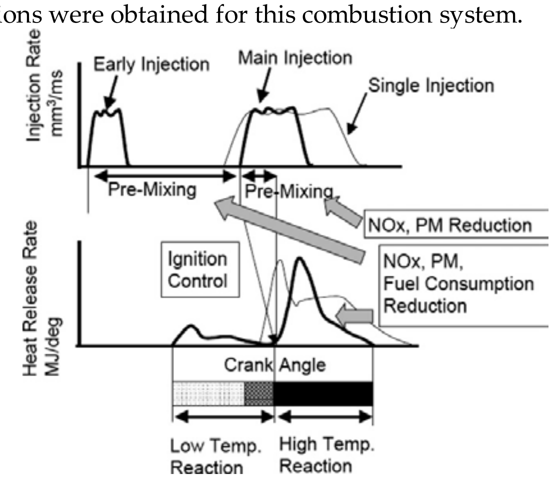

Fig.12 UNIBUS Combustion Strategy (150)

4.1.3. Late injection

With late injection timing starting just before TDC till later crank angles, gas temperature and density decrease because of piston expansion which leads to longer ignition delay (ID) and an improved mixture formation. The in-cylinder conditions become favourable for HCCI combustion. Additionally, if high levels of EGR are utilised, the peak heat release rate decreases leading to lower NOx formation. ID is also prolonged, allowing adequate time for improved air fuel mixing, which reduces net soot production. The Modulated Kinetics (MK) combustion system developed by Nissan Motor Corporation falls under this category of late injection timing. The first generation of MK combustion system was successfully implemented in a four-valve-per-cylinder DI diesel engine (YD25) that was introduced into the Japanese market in June 1998 [78, 79].

The three fundamental operational features of this first generation system are large amount of EGR, retarded injection timing and high swirl ratio. Large amount of EGR which reduces oxygen concentration from 21% to 15% is utilised as the main approach to suppressing NOx formation through low temperature combustion. Start of injection (SOI), meanwhile, is retarded from 7ºBTDC to 3ºATDC to allow thorough mixing of fuel and air prior to ignition to reduce smoke emission. Under this combustion mode, both NOx and smoke levels reduced dramatically with a penalty on HC emissions. A high swirl ratio of 3–5 is introduced to suppress smoke and counter HC emissions by further improving mixing of in-cylinder charge. Another key advantage of the MK combustion system is the reduction of diesel combustion noise [80].the region of application of the first generation MK combustion system is limited to low load. As the operation enters higher load region, fuel input and temperature of the EGR gas are increased leading to longer injection duration and shortened ID. The combination of these two effects prohibits the complete delivery of the desired amount of fuel before combustion commences.

Hence, in the second generation of MK engines, injection duration is shortened by using a larger nozzle hole with high injection pressure and ID is extended by cooling the EGR gas and reducing the compression ratio. The Homogeneous Charge Late Injection (HCLI) and the Highly Premixed Late Injection (HPLI) are the latest HCCI diesel combustion concepts featuring late injection timing [81]. In HCLI, the injection is carried out at approximately 40ºBTDC and a rapid homogenisation occurs. For HPLI, SOI is after TDC to ensure that the ID is sufficient for mixture formation and homogenisation before combustion.

4.2. Injector characteristics

In order to create the homogeneous mixture necessary for HCCI combustion, the fuel injection system should ideally generate sprays which have low penetration, uniform and wide dispersion, and a high injection rate. Experimental observations in PCI engines have shown that injectors with impinging spray nozzles are more suited for this than conventional injectors [69, 74]. Larger impingement angle increases the spray angle and decreases the penetration resulting in a more uniform fuel concentration within the spray. The angle is limited by the nozzle geometry, with 60º being typically employed in PCI combustion systems. Harada et al. [11] demonstrated the use of a swirling-flow pintle-type injection nozzle in creating a more uniform mixture and avoiding spray collision with the in-cylinder liner. Comparisons between a typical pintle-type, a pintle-type with two swirl grooves above the nozzle seat and a pintle-type with a throttle below the nozzle seat showed that the one with swirl grooves can reduce fuel adhesion to the cylinder liner and dilution of lubricating oil [81].

Nozzles with narrow spray cone angles are design features of some HCCI diesel combustion systems. A narrow spray cone angle of less than 100ºwas utilised in conjunction with an early injection strategy in the Narrow Angle Direct Injection (NADI™) engine to promote mixing of air and fuel and to limit fuel wall impingement [82]. Similarly, fuel injection angle was reduced from 156º, typical of conventional diesel engine, to 60º on a single-cylinder DI diesel engine to reduce wall wetting and avoid an out of bowl injection at early injection timing [83]. Additionally, to promote mixing of air and fuel, the nozzle hole diameter can be reduced. It has been demonstrated that the required mixing period, defined as the time necessary for mixture of equivalence ratios of 2.0 or greater to disappear, decreased by approximately 25–50% when the hole diameter was reduced to 25% and 40% of the conventional nozzle hole diameter at injection pressures ranging from 40 MPa to 140 MPa [84].

4.3. Injection pressure

Increased fuel injection pressure can promote better mixing of the in-cylinder charge especially when used in combination with smaller nozzle orifice [85]. At high fuel injection pressure, injection speed increases leading to a high rate of air entrainment and mixing which results in favourable spray structure and better combustion [77]. Furthermore, investigations on a pressurised vessel with optical access and common rail (CR) fuel injection system indicated that leaner sprays which are important for PCI combustion are produced when fuel injection pressure is increased [69].

4.4. Piston bowl geometry

In the case of early injection, piston bowl geometry is often adapted to enable the development of HCCI combustion. Flat and shallow dish types combustion chambers have been developed specifically for HCCI diesel engines [74, 86]. A shallow dish type can reduce the formation of fuel wall-film on the surface of the piston bowl wall leading to lesser soot, HC and CO emissions compared to a standard re-entrant bowl as well as improve Indicated Mean Effective Pressure (IMEP) and combustion stability [87]. The BUMP combustion chamber combined with multi-pulse fuel injection has been demonstrated to work successfully for HCCI diesel combustion [88, 89]. In recent works, computer modelling and simulation is utilised to generate the optimum piston bowl geometry for better combustion and emissions characteristics [90, 91].

4.5. Compression ratio

By lowering the compression ratio, ID is prolonged which allows complete injection of all fuel prior to ignition, a prerequisite for premixed combustion and a reduction in the gas temperature of the combustion field at TDC of the compression stroke. Reducing compression ratio from 18:1 to 16:1 was part of the strategy used in the second generation of MK diesel engines to extend low temperature, premixed combustion to higher load conditions [79]. It has been demonstrated by Ryan and Callahan [92] that knock in HCCI diesel combustion can be prevented by reducing the compression ratio. When compression ratio is reduced, the accompanying reduction in temperature rise of the end gas prevents explosive self ignition from occurring.

Peng et al. [19] conducted experiments on a four-stroke, single cylinder, variable compression ratio engine using n-heptane which verified the effects of compression ratio on HCCI combustion. The results showed that the possibility of knocking decreased, with the knock limit pulled to lower air fuel ratio (AFR) region (richer mixture) when compression ratio was reduced from 18:1 to 12:1. The maximum IMEP attainable on this engine thus increased from 2.7 bar to 3.5 bar.

5. CHALLENGES OF HCCI COMBUSTION

Before implementing the benefits of the HCCI combustion engines, it has to overcome some of the barriers for mass production. The challenges of the HCCI combustion are reviewed below. Among these challenges, the homogeneous mixture preparation and combustion phase control play vital role in determining the efficiency and emissions.

5.1. Combustion phase control

The main challenge of the HCCI engine is to control ignition timing, which influences the power and efficiency. The conventional engines have a direct mechanism to control the start of combustion. Unlike, spark timing in SI engines and fuel injection timing in CI engines, the HCCI engine lacks start of combustion controlled by auto-ignition. The fuel–air mixture is premixed homogeneously, before the start of combustion initiated by the auto-ignition of time temperature history. This phenomenon of auto-ignition leads to the main combustion control which is affected by the few factors [93-95]: fuel auto-ignition chemistry and thermodynamic properties, combustion duration, wall temperatures, concentration of reacting species, residual rate, degree of mixture homogeneity, intake temperature, compression ratio, amount of EGR, engine speed, engine temperature, convective heat transfer to the engine, and other engine parameters. Hence, the HCCI combustion control over a wide range of speeds and loads is the most difficult task. Controlling combustion is the most important parameter, because it affects the power output and the engine efficiency. If combustion occurs too early, power drop in terms of efficiency and serious damage to the engine occurs, and if combustion occurs too late, the chance of misfire increases. Most of the researchers believe on the fact that HCCI combustion is governed by chemical kinetics [96, 97].

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5.2. Abnormal pressure rise with noise

The instantaneous heat release which is caused by auto-ignition of the whole homogeneous charge simultaneously during compression stroke. The instantaneous heat release results in abrupt rise in temperature followed by abrupt pressure rise, and then high levels of noise. Controlling this sudden heat release is extremely important, because it is the main source of pressure rise, which may cause a severe damage to the engine. The acceptable pressure is limit is E8 bar/CA for noise [98].

5.6. Homogeneous charge preparation

The mixture preparation is the key to achieve high fuel economy and low exhaust emissions from the engine. The thermodynamic cycle time of internal combustion engine takes a very short span and within that, the homogeneous mixture preparation time for combustion is much lower. The degree of homogeneity of the fuel–air mixture is greatly improved only by increasing the time for mixture preparation.

6. HOMOGENEOUS CHARGE PREPARATION STRATEGIES

The preparation of the homogeneous mixture is the main factor in reducing the particulate matter (soot) emissions, and local fuel rich regions to minimise oxides of nitrogen. The local fuel-rich regions can be decreased by an effective mixture preparation. However, the preparation of the homogeneous mixture for the cycle-to-cycle variation of speed and load is a difficult task due to less time availability of mixture preparation. The effective mixture preparation for the HCCI combustion includes both the fuel–air homogenisation and temperature control over in combustion. The strategies for mixture preparation are either in cylinder direct injection, or external mixture. Both the preparation methods have their own disadvantages that the external mixture has a low volumetric efficiency and in-cylinder mixture is prone to an oil dilution. This session describes the strategies and implementations of mixture preparation. Table 2 gives the various method of homogeneous charge mixture prepared by researchers.

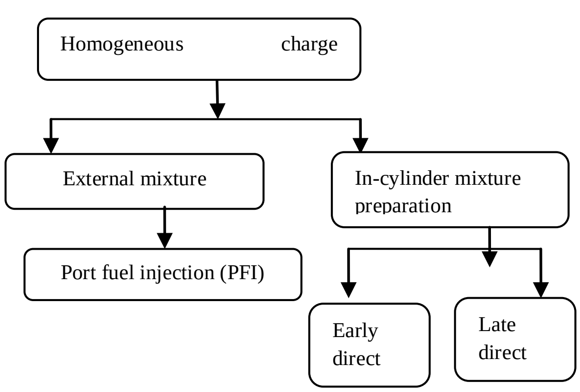

Fig.13 Methods of Homogeneous Charge Preparation in HCCI engine

Table.1

Overview on method of homogeneous charge preparation from literature

Reference	Used fuel	Homogeneous charge preparation method	Advantages	Disadvantages
Swami Nathen et al.	Biogas / Diesel	Manifold	Good atomisation leads to improve efficiency, reduced fuel consumption	Due to early injection it may misfire.
Ramesh et al.	Acetylene	Manifold	Lower NOx and soot emissions	Higher unburned hydrocarbon and CO emissions
Ganesh et al.	Diesel	Fuel vaporiser with port injection	High fuel efficiency and low emissions	Control of combustion is problematic and presents of knock
Morteza fathi et al.	n-heptane/ natural gas	Port injection	Reduced NOx emissions and increased fuel consumption	Increased UHC and CO
Avinash kumar agarval et al	Diesel	Port injection	Low NOx emission was noticed	Limited to part load operation
Qiang et al.	Diesel	Late direct injection	Reduced NOx	Smoke increased

6.1. External mixture preparation

The homogeneous mixture which is prepared external to the engine cylinder is the most effective and more mixing time availability, before the start of combustion. This method of preparation is more suitable for high volatile fuels like gasoline and alcohols. The mixture preparation strategies are port fuel injection (PFI), manifold induction, fumigation, wide open throttle (WOT) carburetion etc. However, the low volatile fuel like diesel can also be used by using fuel vaporiser. The gaseous fuels are ready to mix with the air and preparation of homogeneous mixture externally is pretty simple, but the engine may suffer with the volumetric efficiency, if the calorific value of the gas is low.

The gaseous fuels are mixed mostly in the intake manifold and some early implementations are acetylene [99], biogas

[100-102], hydrogen [103–106] etc. A first study on HCCI combustion process has been performed on two stroke engines by Onishi et al.in 1979 [107]. There is no flame propagation, as in a conventional SI engine instead, the whole mixture burns slowly at the same time. They called it active-thermo atmosphere combustion (ATAC) [108–110]. The same combustion was demonstrated at the Toyata Motor Co.Ltd.and named as "Toyota-Soken (TS) combustion" [16]. Noguchi et al. [16] demonstrated the same combustion process in an opposed-piston two stroke engine. Later, Honda R & D Co.,Ltd. Investigated on activated radical combustion(ARC)on two stroke gasoline engines [111-115] by winning the fifth place in Granada-Dakar rally competition.

During radical combustion, the exhaust port throttling has been tested at arrange of 2 to 16mm (exhaust port reduction area is 1–8%) by Saqaff et.al. [116] on a two stroke engine. It is reported that, the exhaust gas temperature decreased by about 16.7–22.5% at all engine speeds and loads, while the fuel consumption reduced by about 11.1–49.8%.The PCCI (premixed charge compression ignition) engine developed by the Toyota Central Research [117] in which combustion of premixed lean mixture arises from a multi-point ignition is very promising and necessary for achieving both higher efficiency and lower nitrogen oxide (NOx) emission. The PCCI engine operates stably in the air–fuel ratio range of 33–44 and ignition occurs spontaneously at unspecified points as it does in diesel engines. Some researchers introduced an electronically controlled fuel vaporiser for low volatile and high boiling point fuel such as diesel [118-121]. The diesel vaporiser formed a very light and dispersed aerosol with a very fast evaporation due to a very high surface to volume ratio.

The smoke emissions were reported to be negligible and the EGR was used for combustion control and the NOx emissions. The operation temperature of vaporiser is above the boiling point of fuel for successful external mixture preparation [122]. Some researchers used a high intake air temperature [123-127] to vaporise the fuel in the intake manifold. The common disadvantage reported by them is the electric power consumption for vaporisation of diesel. Another study was reported by the researchers on the effect of premixed ratio in diesel engine with the partially pre mixed charge compression ignition (PPCI) combustion using diesel fuel [128–130].

An investigation [131] of diesel fuel with a port fuel injection with variable compression ratios reported that, the compression ratios need to be reduced in order to avoid knock in the HCCI combustion. The cool-combustion chemistry of diesel fuel leads to auto-ignition at approximately 800K during compression stroke [132]. The port fuel injection (PFI) is the simplest method of external mixture preparation, in which injector is mounted in the intake manifold, very close to the intake valve. This system improves the volumetric efficiency and fuel distribution over carburetion. The mixture enters into the cylinder during engine suction and the turbulence created by intake flow improves further homogenisation. This method of mixture formation has been reported to be successful with gasoline and alcoholic fuels [133–140]. The main drawback of this strategy is injection timing cannot influence the start of ignition. Furthermore, heavy fuels with lower volatility of PFI results in poor vaporisation with increased wall impingements.

6.2. In-cylinder mixture preparation

The demerits associated with diesel-fueled by the port fuel injection with an internal mixture formation has been investigated. Two strategies:(i) early direct injection and (ii) late direct injection for in-cylinder mixture formation have been adopted in the study. The injection timing for early direct injection was set during compression stroke, for late direct injection it was set after TDC. High injection pressures with a large number of small nozzle holes adopted to increase the spray disintegration which forms homogeneous mixture.

6.2.1. Early direct injection

The fuel injection process in the HCCI combustion is charge homogeneity, which is influenced by injection timing. Early injection method is mostly used method of achieving HCCI diesel combustion. The early injection allows a longer ignition delay along with the low temperatures to homogenise the diesel–air mixture. Unlike conventional diesel, direct injection in diesel engines, pulsed injection strategy is used. The early direct fuel injection during compression stroke results in wall wetting due to over-penetration of diesel as a result of poor volatility and low air density during early CAD compression stroke. A piezoelectric controlled common rail injector is capable to control injection with high injection pressures for performing variable pulsed injections. The area below the curve represents the fuel mass belonging to each pulse. The low gas density at the beginning of injection requires short pulses with the reduced injection velocities, and the time interval between the pulses is relatively large. As the piston moves up, density and temperature in the cylinder increase and penetration is reduced. The pulse durations can be prolonged, while the time intervals between sub sequent pulses are decreased. At the end of the pulsed injection the distance between nozzle and piston reduces significantly, and the mass injected per pulse must be reduced again in order to prevent fuel deposition on the piston [141]. The in cylinder mixture preparation strategies used in HCCI engines are listed in Table4. The early in-cylinder implementations used in diesel-fueled vehicles are PREDIC, MULDIC, HiMICS, UNIBUS and MULINBUMP. The fuel is injected during early compression stroke, which becomes partly homogeneous mixture and combustion starts closer to the TDC. This concept is called premixed lean diesel combustion (PREDIC) [142,143]. Takeda et al. [144] from New ACE institute, Japan reported low NOx and soot emissions, while the UHC emission is more due to poor combustion efficiency. The injection strategy was modified such that,two side injectors whose nozzle diameter was reduced from 0.17 mm to 0.08mm and the number of holes was also increased from 6 to 16. The operating region is limited to part load only. Nishijima et al. [145] used a nearly injection timing and found a problem that the fuel spray reaches the cylinder wall, which causes a higher HC emission and fuel consumption.

Iwabuchi et al. [146] from the Mitsubishi Motors corporation used early injection strategy, where premixed compression ignited (PCI) combustion system adopted in a fourstroke, single cylinder, diesel engine. The PCI combustion limited to part load and found more HC emissions with low NOx emissions. The strategy of multiple injections in the HCCI mode is used instead of a single injection strategy, in order to run at high load. The multiple stage diesel combustion (MULDIC) and homogeneous charge intelligent multiple injection combustion system(HiMICS) uses the first injection during early compression stroke and the second injection is just before the TDC. Hashizume et al. [147] studied the MULDIC in which the first stage combustion corresponded to the premixed lean combustion and the second stage combustion corresponds to diffusion combustion under the high temperature and low oxygen conditions. The HiMICS concept based on pre-mixed compression ignition combustion combined with a multiple injection developed by the Hino Motors,Ltd. [148]. The pre-mixture was formed by a preliminary injection performed during a period from the early stage of the induction stroke to the middle stage of the compression stroke and later injection after TDC to oxidise soot.

The emissions of NOx and soot are reported less, but high levels of HC and CO. The uniform bulky combustion system (UNIBUS) developed by the Toyota Motor Corporation in the Japanese market (1KD-FTV, 3 l-4cylinder) in August2000 [149]. A double injection technique was used. The first injection was used as an early injection for fuel diffusion and to advance the changing of fuel to lower hydrocarbons (i.e.low temperature reaction). The second injection was used as an ignition trigger for all the fuel. It is reported, that the ignition of the premixed gas could be controlled by the second injection, when the early injection was maintaining a low temperature reaction. The low NOx and smoke emissions are possible both in the first injection and in the second injection by this combustion. This system is limited to low loads and at high loads conventional diesel combustion is used.

The MULINBUMP is a compound combustion technology of premixed combustion and "lean diffusion combustion" in a DI diesel engine [150]. The premixed combustion is achieved by the technology of multi-pulse fuel injection. The start of pulse injection, injection-pulse number, injection period of each pulse and the dwell time between the injection pulses are controlled. The objective of controlling the pulse injection is to limit the spray penetration of the pulse injection, so that the fuel will not impinge on the cylinder liner, and to enhance the mixing rate of each fuel parcel by promoting the disturbance to the fuel parcels. The last or main injection pulse was set around the TDC. A flash mixing technology was developed from the design of a so-called BUMP combustion chamber, which was designed with some special bump rings. The combustion of fuel injected in the main injection proceeds under the effect of the BUMP combustion chamber at a much higher air/fuel mixing rate than it does in a conventional DI diesel engine, which leads to "lean diffusion combustion". The pulse injection mode modulation was investigated by variation of control signals, a series of injection modes were realized based on the pre judgment of combustion requirement.

The designed injection modes included, so called even mode, staggered mode, hump mode and progressive increase mode with four, five and six pulses respectively. An engine test was conducted with the designed injection modes. The experimental results showed that the HCCI diesel combustion was extremely sensitive to the injection mode. There were many ways to reach near zero NOx and smoke emissions, but the injection mode must be carefully designed for higher power output [151].

The development of diesel-fueled late DI HCCI system is the modulated kinetics (MK) combustion system developed by the Nissan Motor Co.,Ltd. [152,153]. This system combines two mutually independent intake ports, one of which is a helical port for generating an ultra-high swirl ratio and the other is a tangential port for generating a low swirl ratio. The tangential port incorporates a swirl control valve that controls the swirl ratio (3.5–10) by varying the flow rate. To achieve the premixed combustion, the fuel–air mixture homogeneity before ignition is required in MK combustion that can be achieved by increasing the ignition delay longer and rapid mixing with a high swirl. In the MK system, there are three features; (i) late fuel injection timing starts from 71 BTDC to 31 ATDC,(ii) high levels of EGR and (iii) high swirl ratio. The formation of NOx emissions can be suppressed by high EGR rates (reduces oxygen concentration from 21% to 15%) and low temperature combustion. The ignition delay was increased by decreasing the compression ratio to 16:1.

Kawamoto et al. [154] found a low compression ratio was effective in expanding the MK combustion region on the high-load side. Kimura et al. [155] examined the effects of combustion chamber insulation on the heat rejection and the thermal efficiency. The combustion chamber was insulated by using a silicon nitride piston cavity that was shrink-fitted in to a titanium alloy crown. The application of heat insulation reduced the angular velocity of the flame in the combustion chamber by about 10-20%. This reduction in the angular velocity of the flame was found to be one cause of combustion deterioration when the heat insulation was applied to the combustion chamber. The main advantage of late direct injection system is the combustion control by the injection timing over the port fuel injection and the early direct injection systems.

6.3. Narrow angle direct injection NADITM

In order to prevent fuel deposition on the cold cylinder liner, the angle between the spray must be reduced. The concept of narrow angle direct-injection(NADI)was suggested by Walter and Gatellier [156–157] to keep the fuel target within the piston bowl and avoid the interaction of the spray with the liner at advanced injection timing. The results indicated that the liquid fuel impingement on the bowl wall leads to fuel film combustion which is called "pool fire". Because of the rich air–fuel mixture and low temperature on the wall surface, the pool fire results in incomplete combustion and high soot formation for all early injection cases. A narrow fuel spray angle and a dual injection by Kim et al. examined [158] the fuel injection angle was narrowed from 1561 of conventional diesel engine to 601, while the compression ratio was reduced from 17.8:1 to 15:1 to prevent the early ignition of the mixture. The result showed that the NOx emissions were greatly reduced as the injection timing was advanced beyond 301 BTDC and the IMEP indicated a modest decree seal though the injection timing advanced to 50–601 BTDC in the case of the narrow spray angle configuration. Fig. 10 shows the narrow spray adoption in early in-cylinder direct injection. In early in-cylinder fuel injection, the spray direction adaption is important, because the volume between the injector nozzle and piston is larger.

Tiegang et al. [159], investigated the effects of two spray injection angles(i.e.,1501 and 701) on the combustion process in an HSDI optical diesel engine employing multiple injection strategies with high injection pressures (600 and 1000bar).The premixed combustion was observed for the150o tip with the high injection pressure, while other cases show diffusion flame combustion features. Anon-luminous flame was seen for the first injection of the 1501 tip, while two types of flames are seen for the first injection of the 701 tip including an on-luminous flame and a luminous film combustion flame. The flame was observed more homogeneous for the 1501 tip with the higher injection pressure, namely a combustion process close to the PCCI-like combustion, with a little soot formation. More soot luminosity is observed for the 701 tip due to fuel-wall impingement.

The fuel film combustion leads to the lower NOx emissions due to its rich mixture nature. For both the injection angles and higher injection pressures results in higher NOx emissions, because of the leaner air– fuel mixture and higher in-cylinder temperatures for the increased injection pressures. The French Institute of Petroleum,IFP [82] has developed a combustion system that was able to reach near zero particulate and NOx emissions, while maintaining the performance standards of the DI diesel engines. A Narrow Angle Direct Injection(NADI™) was applied to this dual fuel mode engine which applies HCCI at part load,and switches to conventional diesel combustion to reach full load requirements. At part load(including Motor Vehicle Emissions Group-MVEG-and Federal Test Procedure-FTP-cycles), the HCCI combustion mode allows near zero particulate and the NOx emissions and maintains very good fuel efficiency. At 1500 and 2500rpm,NADI™ reaches 0.6 and 0.9MPa (6 and 9bar) of indicated mean effective pressure(IMEP)with the emissions of NOx and particulate under 0.05g/kWh, which are lower by100 and 10 times respectively than a conventional diesel engine. At full load, NADI™ system is consistent with future diesel engine power density standard. Lately, IFP [160] has developed a near-zero NOx and particulate combustion process, the NADI™ concept, a dual- mode engine application switching from an ovel lean combustion process at part load to conventional diesel combustion at full load. The narrow spark cone angle injection can reduce liner wetting problem, when the fuel is injected at early CAD for the HCCI combustion.

7. HCCI COMBUSTION CONTROL STRATEGIES

Ignition control in HCCI engines is one of the challenges in developing HCCI engines. The challenges include (29,161): (1) auto-ignition control; (2) limiting the heat release rate at high load operations;(3) meeting emissions standards;(4) providing smooth engine operation by achieving cold starts(‘startability’ of the engine); and (5) limited load range.

Table.2 Overview of Combustion control techniques from literature

Reference	**Method of Combustion Control**	**Range**	**Advantage**
Flowers et al.	Intake charge temperature	80-110ºC	UHC and CO decreased
Christensent et al.	Variable compression ratio (VCR)		Increased BTE and reduced NOx emissions
Christensent et al.	High Boost injection pressure		Low NOx and UHC
Qiang Fang et al.	Exhaust gas Recirculation (EGR)	15%-25%	NOx reduced
Ganesh et al.	Cooled EGR	10%-30%	Low NOx emissions and delayed auto ignition temperature
Mortaza et al.	EGR	8%-42%	Reduced SFC,NOx
Sona visakhamoorthy et al.	Intake air heating	139-159ºC	Increased combustion efficiency and emissions of CO and HC tends to decrease with increased intake air temperature
Avinash kumar agarval et al.	Intake air heating	90-120ºC	
Ganesh et al.	Cooled EGR	10%	Reduced oxides of nitrogen
Ramesh et al.	EGR / intake air heating	40-110ºC	Reduced NOx and Soot
Swami Nathan et al.	intake air heating	80-135ºC	Increased BTE and reduced HC and CO

Various studies have suggested that ignition can be controlled by using promoters or additives, blending of low cetane number fuels with high cetane number fuels [161], pre-heating of the intake air [162], pressurizing the intake air [163], hydrogen addition and varying the amount of exhaust gas recirculation (EGR) [65] through early closure of the exhaust valve. Managing the ignition delay is one of the effective ways to control the start of combustion. Ignition delay is the time lag between the start of injection and the start of combustion. This is difficult to control because the combustion in HCCI engines is fully controlled by chemical kinetics. Too short an ignition delay leads to knocking while too long an ignition delay leads to misfiring. The ignition delay is strongly dependent on the gas temperature: an increase in temperature accelerates the chemical reactions, thereby helping to reduce the ignition delay [164]. Tanaka et al. [165] studied two-stage ignition in HCCI combustion, suggesting that the ignition delay can be controlled by the fuel air ratio, initial temperature and additive dosages.

7.1. Pre heat intake air

As discussed in natural gas and hydrogen fuelled in HCCI engine required high intake temperature. By preheating the intake air, ignition delay is reduced and thus the ignition timing can be controlled, methane and natural gas require high temperatures to auto ignite and methane did not ignite for an intake temperature of 400K at low CR [100]. Therefore, it is imperative to pre-heat the intake air to make the fuels combust smoothly. An tunes, Mikalsen and Roskilly [24] stated that heating the inlet air is the most useful method to control the ignition timing. Questions may arise regarding the practicality of including electric heaters [16] in an engine bay just for this purpose: the heater causes the operation and maintenance costs to increase and contributes to extra engine weight. Installing a heater is an option used by most researchers as it is the easiest way to get intake air heated to some specific temperature. Exhaust gas recirculation (EGR) could be another option to reduce the need for a high intake temperature [15,41,67].

7.2. Pressurized intake air

Turbochargers and superchargers are commonly used in real engine applications because they can be applied to any internal combustion engine. The operational concepts of these two devices are the same: to provide a high in take pressure into the combustion chamber, increase the charge density and there by increase the engine performance. Some studies [99,102,103] show that the start of combustion(SOC) is advanced if the intake pressure is increased by 0.1MPa.This indicates that pressurized intake air is able to improve the auto-ignition of the fuel. However, these situations also depend on the type of fuel used, and in this case they used primary reference fuels and gasoline. By increasing the intake pressure, it was possible to get the auto-ignition to start at 151 before top dead center (BTDC).On the other hand, super-charging (pressurizing intake air)is able to increase engine efficiency [7]. A super charged hydrogen–diesel engine, but in non-HCCI mode, was able to maintain high thermal efficiencies and it was possible to use more than 90% hydrogen energy substitution for the diesel [60]. Another study by Guo et al. [66] shows that hydrogen–diesel in HCCI mode with pressurized intake air (150kPa) is able to improve the atomization process, and therefore improve the combustion efficiency. CO, HC and NOx emissions were also decreased in this case.

7.3. Exhaust Gas Recirculation (EGR)

For early injection HCCI diesel combustion, EGR is used as a means of diluting the gas mixture in HCCI diesel engine thereby retarding the ignition timing. In a Premixed Charge Compression Ignition (PCCI) combustion system described by Kanda et al. [86], a large amount of EGR (54%) is introduced to retard the ignition timing toward TDC and improve the IMEP. In another PCI combustion system, it has been shown that high EGR rates up to 68% are used to effectively control the start of combustion (SOC) [44]. High EGR rates can also be used to counter combustion noise by controlling the start of combustion [48]. However, the drawbacks of such high rates of EGR include problems with transient response and temperature-stability characteristics [20]. Therefore, for early injection HCCI combustion, EGR should be combined with some other combustion control technology such as modification of fuel properties or adoption of some other chemical approach. In the case of late injection such as the MK combustion system, EGR is typically utilised as a NOx reduction measure with typical levels of approximately 40% [36]. NOx is reduced because of the lowering of flame temperature due to charge dilution and higher heat capacity of the cylinder charge when EGR is introduced [49].

Conclusion

The conclusions of the review on conversion of direct injection (DI) mode engine to HCCI mode engine and compare the combustion, performance and emission of both DI and HCCI mode engines. And also review on homogeneous mixture preparation and methods of control strategies of HCCI combustion and its impact on performance and emissions are as follows.

- The HCCI mode engines have the potential to improve the brake thermal efficiency with consuming less fuels, the HCCI engine has low emissions level of NOx and particulate. However, it has higher level of unburned hydrocarbon and carbon monoxides emissions.
- The conversion of DI mode engine into HCCI mode engine is a one of the main target of HCCI research. From the literature, HCCI mode engine is achieved by fuel injection process such as injection timing, injection pressure and injector characteristics. Particularly injection timing is crucial to develop the HCCI mode engine. And reviewed other governing factors in the development of HCCI engine were piston bowl geometry and compression ratios.
- Brief reviewed on challenges over to achieve the combustion process in HCCI mode engine. The main challenge of HCCI engine was to control ignition timing, which directly influences the efficiency and emissions. In HCCI mode engine the combustion has been achieved by chemical kinetics of fuel, intake charge temperature and method of exhaust gas recirculation.
- Homogeneous charge preparation in HCCI mode engine has a difficult task in the HCCI research. In this review briefly discussed about the method of homogeneous charge prepared in HCCI mode engine. The Homogeneous charge was prepared by external and in cylinder mixture preparation. There was many external methods were used to prepare the homogeneous mixture such as port fuel injection, manifold induction, fumigation and carburetion. Most of the researches used external mixture preparation by port fuel injection (PFI), manifold induction methods. In- cylinder mixture preparation has been divided into three strategies as early direct injection, late direct injection and narrow angle direct injection. Each injection methods have some advantages and disadvantages were discussed.
- The intake charge temperature, pressurized intake air and exhaust gas recirculation were used to control the combustion process of HCCI engine. In HCCI engine, the

heating coil was adopted in the intake manifold before the fuel injection. The higher intake air temperatures have been reduced ignition delay and ignition duration of the mixture. The combustion parameter would reduce the knocking and increased the thermal efficiency. The inlet air has been pressurised by turbocharger and supercharger, which have commonly used in real engine applications.

- Exhaust gas recirculation (EGR) was used in HCCI mode engine for control the combustion process. It was one of the effective methods of combustion control. The exhaust gas recirculation has diluted the fresh charge and it lengthens the ignition timing.

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