Int. Res. Adv. 2018, 5(1), 5-19 Article

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Comprehensive Review on Enabling Reactivity Controlled Compression Ignition (RCCI) in Diesel Engines

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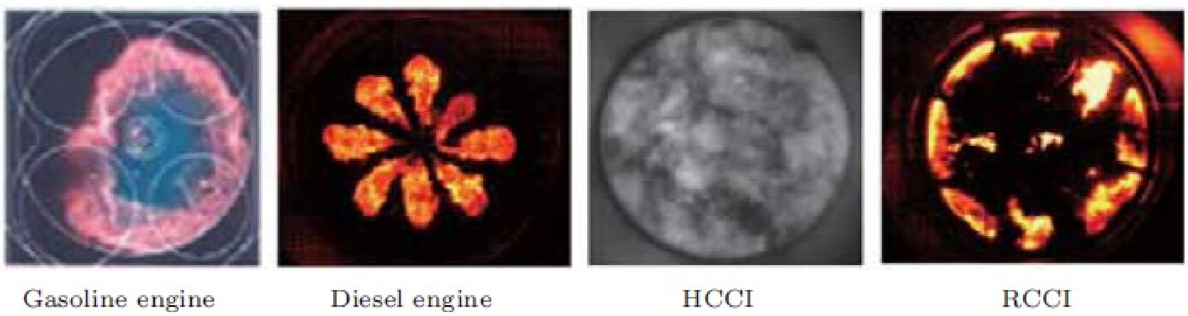
Received on: 02-Nov-2017, Revised on: 08-12-2017, Accepted and Published on: 22-Dec-2017

ABSTRACT



Schematic of gasoline, diesel, HCCI and RCCI engines

The demand for higher fuel economy and lower exhaust gas emissions of CI engine cannot be achieved by just using mechanically governed fuel injection systems. The electronically operated fuel injection system is the heart of a CI engine and has become one of the essential technologies to control critical in the recent years. High computing power of microcontrollers has made it possible for controlling injection of fuel electronically at any injection pressure, any crank angle and for any duration during the engine operation. A newer concept of RCCI operation further reduces the particulate matter and NOx emission to almost zero level with slightly higher BTE. Hence, there is a need to discuss the performance, combustion and emission of RCCI engine fuelled with alternative fuels and this paper presents an exhaustive review on these aspects.

Keywords: Compression ignition engine, Dual fuel engine, Homogeneous charge compression ignition engine, Reactivity controlled compression ignition engine

INTRODUCTION

Limited fossil fuel availability and air pollution have drawn a continuously increasing attention to the study of alternative fuels[1]. With the development of the world economy, the demand for crude oil is growing rapidly. The researchers are more interested in the CI engines for its better fuel economy with high compression ratio and no throttling losses. But for the conventional CI engine, it suffers from high NOx and PM emissions. By providing low emissions while maintaining the high efficiency, the diesel PCCI attracts increasing attentions in recent decades. In the PCCI combustion strategy, PM can be reduced by promoting mixing of fuel and air prior to combustion. Meanwhile, NOx is reduced by using lean fuel/air mixture as well as high EGR rate to cool the combustion temperature down. However, because of the low volatility and high flammability of diesel, there are still some problems yet to be solved for PCCI engine, including the formation of homogeneous mixture, ignition control, limited operating range, excessive wall impingement, and so on. To overcome these problems associated with diesel PCCI combustion, the RCCI concept has been proposed recently, which is an effective and clean combustion strategy by using two fuels with different properties and separate injection. The fuel with high octane number and low boiling point is injected into intake port, while diesel is injected into cylinder, so the stratified distribution

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Cite as: *Int. Res. Adv., 2018, 5(1), 5-19.*

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of fuel reactivity is formed which leads to stratified combustion when be ignited by compression[2]. RCCI is auto-ignition combustion strategy that operates through in-cylinder blending of diesel-like and gasoline-like fuels[3]. This combustion mode symbolizes an evolution from the PCCI concept, because fuel and air are mixed before combustion, but in this case fuel reactivity varies across the cylinder[4]. To generate the reactivity in cylinder, fuel with low CN, such as gasoline is injected from the port, whereas the fuel with high CN, such as diesel, is injected into the cylinder through an injector. In fact, the reactivity can be distributed into two types: one is called global reactivity; the other is called reactivity gradient. Global reactivity is only determined by the fuel types and their amount that is introduced into the cylinder. The reactivity gradient is varied with the injection strategy, because either early or late injection of low and high CN fuels could affect the mixing outcome among the two types of fuel and air. Hence, both the percentage and the injection strategy of each type of fuel could have an impact on RCCI combustion strategy[5]. Recent experimental and simulated studies confirm that RCCI concept allows an effective ignition control and a low maximum PRR while maintaining low engine out emissions levels and high fuel efficiency simultaneously, proving that the RCCI concept is a more promising LTC technique than HCCI[6]. In an RCCI engine fueled with diesel, the ignition timing can be controlled by adjusting the ratio of diesel and other fuel and varying the SOI[8].

RCCI PRINCIPLE

With the development of the world economy, the demand for crude oil is growing rapidly. The researchers are more interested in the CI engines for its better fuel economy with high compression ratio and no throttling losses. But for the conventional CI engine, it suffers from high NOx and PM emissions. By providing low emissions while maintaining the high efficiency, the diesel PCCI attracts increasing attentions in recent decades. In the PCCI combustion strategy, PM can be reduced by promoting mixing of fuel and air prior to combustion. Meanwhile, NOx is reduced by using lean fuel/air mixture as well as high EGR rate to cool the combustion temperature down. However, because of the low volatility and high flammability of diesel, there are still some problems yet to be solved for PCCI engine, including the formation of homogeneous mixture, ignition control, limited operating range, excessive wall impingement, and so on. To overcome these problems associated with diesel PCCI combustion, the RCCI concept has been proposed recently, which is an effective and clean combustion strategy by using two fuels with different properties and separate injection. The fuel with high octane number and low boiling point is injected into intake port, while diesel is injected into cylinder, so the stratified distribution of fuel reactivity is formed which leads to stratified combustion when be ignited by compression[2]. In fact, the reactivity can be distributed into two types: one is called global reactivity; the other is called reactivity gradient. Global reactivity is only determined by the fuel types and their amount that is introduced into the cylinder. The reactivity gradient is varied with the injection strategy, because either early or late injection of low and high CN fuels could affect the mixing outcome among the two types of fuel and air. Hence, both the percentage and the injection strategy of each type of fuel could have an impact on RCCI combustion strategy[5].

RCCI COMBUSTION USING FUEL ADDITIVES (SINGLE FUEL STRATEGY)

Splitter et al.[38] demonstrated a single fuel strategy (PFI fuel-gasoline, DI fuel-gasoline doped with a small quantity of DTBP at a mid-load condition on the HD diesel engine. They found that a very small percentage of an appropriate additive could be used to establish a sufficient large reactivity gradient to match the performance of a dual fuel strategy when operated in the RCCI regime. They used DTBP as a cetane improver for the pump gasoline. The study focused on investigating the feasibility of using a single fuel as both the high and low reactivity fuels. The strategy involved PFI of gasoline and DI of gasoline doped with 0.75%, 1.75% and 3.5% DTBP by volume, which accounts approximately 0.2% of the total fueling. DTBP was selected based on research by Tanaka et al.[40]. In that research, rapid compression machine experiments demonstrated that for gasoline-like fuels, 2% DTBP addition provided a greater effect in decreasing ignition delay than 2% addition of 2-EHN. The single fuel results with DTBP were compared to previous high-thermal efficiency, low-emissions results with port injection of gasoline and direct injections of diesel. The comparison between the fueling strategies found that the higher volatility of gasoline enabled a reduction in the direct injection pressure from 800 bar with diesel to 400 bar with gasoline. At the tested conditions, the peak gross indicated based thermal efficiency was over 57%.

PROGRESS AND CHALLENGES IN COMBUSTION RESEARCH

Unlike the conventional gasoline and diesel engines as shown in Fig.1, which mainly rely on, respectively, the propagation and transport of premixed and diffusion flames to produce heat release, advanced HCCI and RCCI engines use partially or fully premixed combustion processes with multi-pulse early fuel injection and EGR dilution. As such the combustion process in HCCI and RCCI engines is more dominated by volumetric ignition than flame front propagation. As a result, in advanced engines combustion processes involving auto-ignition and ignition to flame transition play an important role. Ignition process is highly governed by radical initiation and branching processes which depend strongly on the size and structure of fuel molecules Therefore, the heat release rate of advanced engines such as HCCI and RCCI is more affected by initial pressure, temperature, and fuel reactivity than conventional engines[51].

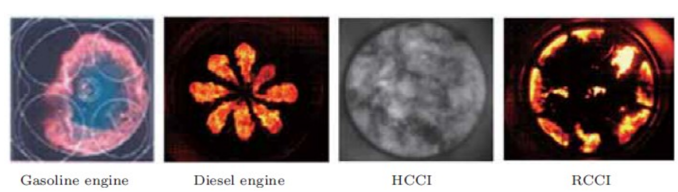

Fig.1. Schematic of gasoline, diesel, HCCI and RCCI engines[51]

PERFORMANCE, COMBUSTION AND EMISSION CHARACTERISTICS

Several literatures have studied the effect of load, speed, compression ratio, pilot fuel injection timing, inlet manifold condition, combustion chamber shapes, exhaust gas recirculation have been achieved. These investigations by the researches are conducted in CI engines using high octane fuel as primary fuel and diesel as high cetane fuel as injected fuel. RCCI mode operation can result in fairly better performance and extremely at high power outputs compared to diesel fuel operation. Researchers have studied performance, combustion and emission characteristics such as brake specific fuel consumption, equivalent indicated specific fuel consumption, indicated power, indicated thermal efficiency, gross indicated efficiency, gross thermal efficiency, fuel efficiency, exhaust gas temperature, heat release rate, in-cylinder pressure, peak pressure rise, ignition delay, ringing intensity, combustion duration, combustion phasing, indicated mean effective pressure, engine knocking identification, engine knocking mitigation, CH_4 mole fractions, HC, CO, NOx, Soot and CO_2 emissions. These parameters depend on physic-chemical properties of fuel used, design of engine and operating conditions used during the test. Based on literature a brief review on the utilization of high octane fuels in CI engine is presented here.

Brake specific fuel consumption (BSFC)

BSFC for RCCI combustion has been reported by Benajes et al.[16,18] and Molina et al.[22]. The first advance in diesel SOI resulted in a notable improvement in BSFC. That improvement was related with the better combustion phasing achieved which allowed enhancing the fuel to work efficiency. Once a desired combustion phasing was reached, the combined modification of the EGR rate and diesel SOI allowed modulating the combustion development in order to search slight improvements in BSFC[16]. At low loads the increase in GF from 65% to 75% resulted in more delayed combustion with almost equal duration, leading to better fuel consumption fuel consumption. Bathtub piston allowed an improvement compared to the stock geometry due to the slight delay in combustion development. The smooth and late burn provided by the stepped piston resulted in worsen BSFC than the stock piston. At medium loads the gain in RoHR with bathtub piston leads to remarkable lower BSFC. At high loads stepped piston allowed the minimum BSFC, while bathtub piston worsened it with respect to the stock geometry[18]. At low loads, there was a trade-off when using a single injection strategy: delayed SOI and increased fuel consumption. Lower local equivalence ratios from double injection strategies provide later combustion phasing that contributed to improve fuel consumption. An advanced double diesel injection strategy was not enough to attain the region of interest, because of the early phasing of combustion, which resulted in high fuel consumption[22].

Equivalent indicated specific fuel consumption (EISFC)

EISFC for RCCI combustion has been reported by Yaopeng Li et al.[2,7,25]. With retarded SOI, the EISFC decreased due to less negative work with delayed ignition timing. When the methanol mass fraction was increased from 20% to 45%, there was little change in EISFC with SOI variation. That may because the effect of negative work was reduced as the ignition timing was pushed towards TDC with methanol addition. When the methanol mass fraction was greater than 45%, the ignition timing was almost after TDC at earlier SOI the higher combustion temperature without concerning the negative work leads to higher combustion efficiency and consequently better EISFC[2]. By increasing the methanol fraction from 37.54% in Design-2 to 82.4% in Design-4 while remaining other operating parameters unchanged, EISFC increased substantially due to the relatively low combustion temperature. Since the auto-ignition timing in Design-5 was well before TDC, the EISFC of Design-5 further deteriorated than that of Design-3 due to the increased heat transfer loss and negative work in the compression stroke. There was an upper limit for EGR rate in order to remain EISFC at a low level. Once EGR rate exceeded the limit, EISFC rapidly increased. As EGR rate was reduced within the range of 20% variation from the base value, EISFC was not significantly affected. Once the initial pressure and temperature were reduced from the base value, EISFC increased sharply[7]. The CA50 in the EISFC boundaries nearly remain constant and were independent of the variations of initial temperature and EGR rate. The advanced CA50 resulted in overall higher combustion efficiency and more heat transfer loss. As a result, the final work increased with advanced CA50, leading to the improved EISFC[25].

Indicated power

Factor E was the most significant one for improving indicated power. Factors A and C had positive effects on increased indicated power. Factors B, D and F lead to deterioration in indicated power. The highest indicated power could reach 15.76 kW and the corresponding fuel conversion efficiency was 55.91%. The lowest indicated power was 8.78 kW, which was only 55.7% of the maximum value. The peak pressure of R11 was much higher than that of R15, which paves the higher indicated power of R11[5].

Indicated thermal efficiency (ITE)

ITE for RCCI combustion has been reported by Zhu et al.[20], Fang et al.[24] and Qian et al.[27]. With an advance in the direct injection timing, the ITE gradually increased for each tested fuel. Gasoline and n-butanol had the similar ITE value at any SOI condition. For ethanol, when the SOI was above 10° CA BTDC, its ITE was significantly higher than that of the other two fuels. At retarded direct injection timing, the in-cylinder incomplete combustion leads to a low ITE. The higher ITE for ethanol at advanced SOI was partly due to the suitable combustion phase[20]. At lower loads the ITE was decreased along the optimization path. The ITE was reduced when combustion phasing was more delayed from TDC in RCCI combustion due to lower effective expansion ratio. Along the optimization path, the CA50 was delayed, resulted in a reduction in ITE. At higher loads the ITE was decreased along the optimization path except at the last step, where it was increased. The reduction in the ITE at the first two steps was likely due to the delayed combustion phasing, similar to the lower load condition[24]. The ITE for both gasoline/n-heptane and ethanol/n-heptane first increased and then decreased. At the same RP, the ITE of gasoline/n-heptane was higher than that of ethanol/n-heptane. The trend of an initial increase followed by a decrease in ITE was partly due to the combustion efficiency and partly due to the combustion phase. With increased RP, the in-cylinder temperature gradually increased. The combustion became more complete, and the ITE increased[27].

Gross indicated efficiency (GIE)

GIE for RCCI combustion has been reported by Desantes et al.[6], Paykani et al.[8] and Benajes et al.[16]. An increase of 1% in GIE was achieved in -40 and -35 CAD ATDC compared with the extremes of the range tested. The better combustion phasing combined with shorter combustion duration contributed to enhance the GIE. The test with 38% EGR and 79.6% of gasoline reach the same maximum peak in the GIE i.e. 49%. The GIE was clearly correlated with the combustion efficiency, which showed a maximum value of 48.5% for the higher combustion efficiency case[6]. A slight increase of GIE occurred with increased natural gas mass fraction. Misfiring occurred at 89% natural gas fraction and thus, the GIE significantly dropped at that point. When SOI-2 timing closes to TDC, a larger amount of heat release occurred after TDC, resulted in reduction of GIE. A slight increase of GIE occurred with retarded SOI-2 timing. The better CA50 combined with shorter combustion duration in those cases contributed to the enhanced GIE. A slight decrease of GIE occurred with increased SOI-1 fuel fraction. The highest GIE occurred when the peak HRR was just after TDC[8]. At CR 14.4:1 the GIE increased as engine speed increased. The higher value found was 47.2% at 1800 rpm and 25% load, which corresponds to the test with the highest GF. The trend of GIE as a function of the engine load showed a parabolic behavior, with maximum values between 25% and 50% load, which also correspond with the higher GF levels in the map. At CR 11:1 the GIE increased as engine speed and load increase. The highest value found was 48.2% at 1200 rpm and 96% load[16].

Gross thermal efficiency (GTE)

GTE for RCCI combustion has been reported by Splitter et al.[3] and Gharehghani et al.[11]. The region of highest GTE roughly resides within the self-imposed pressure rise rate and combustion efficiency constraints. ULSD operation yielded a lower GTE relative to 3.5% EHN + Gas. In-cylinder fuel reactivity gradients that existed in RCCI directly affect the engine GTE and in operable range. Fuels with lower reactivity gradients tend to require more DI fuel and thus leaner global operation was needed to reduce PPRR. At those leaner and more homogeneous conditions the GTE was found to be higher. The difference in peak GTE between EHN + Gas and ULSD resulted from the reduced incomplete combustion losses that were seen at leaner global equivalence ratios[3]. GTE of biodiesel was slightly higher than that of diesel at any given load. Greater oxygen content and better combustion initiation of biodiesel leads to an improved combustion and as a result better GTE. The dual fuel RCCI mode demonstrated higher GTE for both pilot fueled diesel or biodiesel. Lower combustion temperature reduced the heat loss through the walls which was the main reason for that improvement. For all engine loads, the CNG/biodiesel dual fuel mode had about 1.6% more GTE as compared to the CNG/diesel mode. Higher adiabatic flame temperature of methane increased cylinder temperature in the natural gas/diesel cases, and as a result increased heat transfer losses to the walls occurred which affected the GTE. Combustion loss and pumping losses in dual fuel modes were about 18.85% for CNG/biodiesel and 20.88% for CNG/diesel mode. So that decrease of about 2% was directly converted to gross work because GTE of CNG/biodiesel case was about 2% higher than CNG/diesel case at engine full load. Combustion loss for biodiesel case was about 14.5% while that amount for diesel fueled case was about 16.72%[11].

Fuel efficiency

Fuel efficiency for RCCI combustion has been reported by Yaopeng et al.[7,25]. As the methanol fraction was very small, the lean methanol/air mixture in the low temperature region near the cylinder wall and the injected diesel in the fuel-rich region near the piston bowl cannot be completely oxidized, which deteriorate the fuel efficiency. The high reactivity of diesel fuel leads to the advanced ignition timing before TDC, which increased heat transfer loss and negative work during the compression stork, further resulted in the relatively poor fuel efficiency. Increased EGR rate leads to the reduction of combustion temperature and the extension of heat release duration, which deteriorate the fuel efficiency. The combined effect of high EGR rate and high methanol fraction leads to excessive ignition delay and incomplete combustion, resulted in low fuel economy[7]. When CA50 was kept constant, the increased methanol fraction and initial temperature promote fuel efficiency. The higher initial temperature is one of the major reasons for the improvement of fuel efficiency because of the more complete combustion. Due to the increase of methanol fraction, the in-cylinder direct injected diesel mass decreased, which could eliminate the rich fuel region and reduce wall wetting and consequently leads to the improvement of fuel economy[25].

Exhaust gas temperature (EGT)

EGT in dual fuel CNG/biodiesel case was the least amount throughout different engine loading. The EGT for that case was about 4.8% lower than that of dual fuel CNG/diesel case. Higher oxygen content of WFO biodiesel leads to earlier combustion and together with lower heating value of biodiesel resulted in lower EGT. The EGT of CNG/diesel case was about 12% lower than that of the conventional diesel mode[11].

Heat release rate (HRR)

HRR for RCCI combustion has been reported by Yaopeng et al.[2,7,25], Benajes et al.[4,16,17,18,19], Jing et al.[5], Desantes et al.[6], Paykani et al.[8], Gharehghani et al.[11], Zhou et al.[12,13], Li et al.[14,15], Zhu et al.[20], Kokjohn et al.[23], Fang et al.[24] and Qian et al.[26,27]. As methanol mass fraction was increased from 40% to 60%, both the CN value and the area of high CN decreased. So the ignition timing delayed with increased methanol due to the lower CN i.e. the decreased fuel reactivity. Methanol was injected into the intake port, so the specific heat of the intake mixture became larger during compression when methanol fraction was increased and consequently the rise rate of in-cylinder temperature slowed down before ignition[2]. As the diesel/gasoline in-cylinder blending ratio was reduced from 100/0% to 50/50%, the ignition delay increased due to the global lowering of fuel reactivity. The two staged combustion showed a squared shape with lower maximum peaks of heat release and a proper combustion phasing. If the diesel/gasoline ratio was further reduced, from 25/75% up to 10/90% the combustion process was highly worsened. At the most delayed injection timing, the initial stage of HTHR was a sharp peak in which takes place the premixed burn of most diesel fuel and the entrained air-gasoline[4]. The HRR of R11 was much sharper and shorter than that of R15. At 0° CA ATDC, the temperature of R11 reached around 1700 K, which was higher than that of R15[5]. Due to the increase in the in-cylinder temperature as a consequence of the first low temperature reactions, the thermal ignition of the n-heptane occurred. Those differences between the simulated and experimental HRR are associated to the slow thermal ignition of the n-heptane in that case, resulted in low in-cylinder temperatures which inhibit to burn the iso-octane[6]. With the increased methanol fraction from 33.25% to 66.5%, the peak of HRR increased. When the methanol fraction was large, CA50 was substantially retarded due to the reactivity dilution effect of methanol. The long ignition delay and the nearly homogeneous fuel distribution due to the high methanol fraction lead to a HCCI like combustion, which resulted in high single peak of HRR. When the CA50 was between -3.69° CA ATDC and 4.3° CA ATDC, for the cases with low methanol fraction, the low peak of HRR counteracts the high combustion temperature and pressure resulted from the early CA50. As the CA50 was earlier than -3.69° CA ATDC, the increased premixed combustion with more injected diesel leads to higher peak of HRR[7]. The HRR decreased as natural gas mass fraction increased. That may be due to the fact that the fuel distribution was more homogeneous with increased premixed natural gas, thus the local rich fuel region could be avoided, resulted in a larger area with high temperatures. By increasing diesel fuel percentage, in-cylinder reactivity increases, and combustion phasing was advanced. As the SOI-1 timing was retarded, combustion phasing advanced, resulted in increased peak cylinder temperatures. As the SOI-1 timing was retarded the reduced mixing time between SOI-1 and SOI-2, resulted in a higher concentration of n-heptane, which increased the local fuel reactivity, and therefore leads to reduced ignition delay. As SOI-2 was retarded from -60 to -30 ATDC, the cylinder pressure and HRR were increased, and combustion phasing was advanced, resulted in higher cylinder temperature[8]. The highest in-cylinder peak pressure and most heat release rate were achieved when the engine was dual fueled with CNG/biodiesel at 50% load. When biodiesel was used as a pure fuel in conventional mode or as a pilot in dual fuel CNG/biodiesel, the premixed combustion initiates much earlier. The difference between peak pressures reduced in dual fuel modes by increasing engine load while the value for CNG/biodiesel was higher than the CNG/diesel[11]. At 10% load while increased the methanol induction rate, the low reactivity of methanol dominated the combustion process, contributed to more unburned fuel and decreased the peak HRR. For 50% load, the peak HRR increased with the increased methanol mass fraction from 0% to 60%, and decreased from 60% to 80%. The highest peak HRR at 100% load was found to be at 80% methanol case. Considering the longer ignition delay caused by increased methanol induction and the higher in-cylinder temperature at 100% load conditions resulted in better fuel evaporation, showed highest peak HRR at 80% methanol mass fraction at 100% load was acceptable. High HRR could lead to serious engine knocking especially for the cases of 60%, 80% methanol mass fraction which were over premixed and more similar to PCCI[12]. The phenomenon of LTC around -23° CA ATDC could be found from the HRR curve for Case-2, but not for the base case. Case-3 with the SOI timing of -35° CA ATDC showed low peak HRR thereby low combustion efficiency. Case-2 was without fuel reactivity gradient, thereby advancing the start of ignition timing and generating high HRR during the working process[14]. At the LTC stage, more gasoline lowered the peak HRR and it will increase the peak HRR and shorten the duration of heat release of HTC stage. The HRR profile of LTC stage depends on the biodiesel since the lower peak HRR was found with less biodiesel. The shorter and faster heat release process with more gasoline was due to a relatively more homogeneous mixture formed in the cylinder[15]. A notable increase in the maximum HRR peak was appreciated in the second stage, while in the first one was progressively reduced. Combustion phasing remained almost unchanged with the EGR modification. Focusing on the HRR shape of SOI -32 CAD, it was clear that almost the whole combustion development takes place after TDC. The improved in-cylinder reactivity, due to the EGR reduction, resulted in a higher HRR peak[16]. At lower loads in the case of B7+E85 an appreciable change in the HRR slope during the combustion development was found. B7+E85 with PER = 24%, a strong HRR peak followed by a late soft burn was appreciated. At medium loads B7+E85 exhibited the same behavior found at low load, with a first premixed HRR peak followed by a softer combustion development. The strong improvement in the second combustion stage that B7+E10-95 provided compared to B7+E20-95 fuel blend for the highest PER tested. The higher reactivity of E10-95 resulted in more than double HRR peak during the second combustion stage, when the majority of E10-95 was consumed, leading to combustion phasing 7.4 CAD advanced[17]. At low loads lower first HRR peak and delayed combustion progression was found for the most advanced strategy. The stock geometry enhanced the charge motion, which resulted in a faster mixing and earlier autoignition. The higher HRR was related to differences in the equivalence ratio stratification at SOC. Bathtub piston allowed an improvement in fuel consumption as compared to the stock geometry due to the slight delay in combustion development together with the similar maximum HRR peak. At medium loads greater differences in the first HRR peak were found[18]. B7+E85 exhibited earlier HTHR growth, followed by B7+E10-98, B7+E20-95 and finally B7+E10-95. The higher oxygen content of E85 compared to other fuels also contributed to the more evident advance in the HTHR onset. The diesel fuel mass distribution became leaner as PER was increased, whatever the fuel blend. The low reactivity fuels with higher reactivity enhanced the autoignition process, which resulted in higher maximum HRR peaks during the HTHR stage. The maximum LTHR peak became reduced as PER was increased, whatever the low reactivity fuel used. For the same PER, the maximum LTHR peak depends on the reactivity of the low reactivity fuel. The maximum LTHR peak was well related to the octane number of the low reactivity fuels, with higher LTHR peaks found as the RON and MON were decreased[19]. With a retard in the direct injection timing, the peak values of the in-cylinder HRR decreased. That was due to earlier combustion phase occurring near the top dead center and leading to a higher in-cylinder gas pressure. The highest peak value of HRR for the less retarded direct injection timing may be attributable to the better mixed charge of n-heptane and air compared to other direct injection timing, which could accelerate combustion initially. The peak value of the HRR of the RCCI combustion mode with n-butanol as port fuel was the highest, followed by that of ethanol[20]. Around -19° ATDC, low temperature reactions pushed the apparent HRR above zero. The energy release from low temperature reactions reached a peak near -13° ATDC and by -11° ATDC enough energy released from low temperature reactions to raise the cylinder pressure above the motored pressure. The high temperature heat release begins near -6° ATDC and peaks near 2° ATDC[23]. At lower loads the HRR curves exhibited a two stage heat release event, represented the combustion of the high reactivity diesel fuel in piston bowl followed by the premixed combustion of the remained well mixed fuel-air charge. At higher loads one significant difference was the absence of the first peak of diesel fuel combustion at the high load condition. All the operating points exhibited a single peak HRR curve with a remarkably longer combustion duration compared to the low load condition. Despite the higher fuel input quantity and overall equivalence ratio at the high load condition, the peak HRR was not significantly higher than at the low load condition[24]. The in-cylinder combustion process was more intensive with elevated initial temperature and higher methanol fraction. The peak combustion temperature and peak HRR significantly increase despite of the constant CA50[25]. The peak values of HRR curves sharply decreased and the combustion phase was delayed when the premixed ratio increased from 0.47 to 0.85 with port injection of ethanol. When fueled with n-butanol and n-amyl alcohol, the peak values of the HRR slightly decreased and the combustion phases were slightly delayed. That was mainly because the amount of directly injected n-heptane decreased and the reaction activities of in-cylinder mixtures decreased with the increase of premixed ratio. That would lead to the reduction in the premixed combustion zone of n-heptane during compression ignition[26]. With increased RP, the peak values of the heat release rates first increased and then decreased. With increased RP, the concentration of the homogeneous premixed ethanol was higher because more ethanol mixed with n-heptane was ignited by compression in the premixed combustion flame, the heat release will be higher[27].

In-cylinder pressure

In-cylinder pressure for RCCI combustion has been reported by Zhou et al.[12], Li et al.[15] and Qian et al.[26]. At 10% load condition, in-cylinder decreased with the increased methanol mass fraction, which could be attributed to more unburned fuel and lower thermal efficiency because of the low reactivity and low temperature in the chamber. With more fuel injected and consumed in the combustion chamber at medium load and high load, the peak pressure appeared at 60% methanol induction case. When the methanol percentage reached 80%, the poor methanol ignition ability started to restrict the combustion. For those reasons, the highest peak pressure exhibited at 60% methanol mass fraction both at medium and high engine loads[12]. There was not clear trend in C-SOI case and in A-SOI case the peak pressure was higher with more premixed gasoline. There was no combustion found in the case with 100% of gasoline. That may be due to two reasons. The first was that for the gasoline fueled HCCI combustion, a fuel-lean mixture was formed in the cylinder when the same boundary condition was used as the diesel combustion. Secondly, the gasoline was also difficult to be ignited due to its high octane number[15]. The peak values of in-cylinder pressure curves sharply decreased and the combustion phase was delayed when the premixed ratio increased from 0.47 to 0.85 with port injection of ethanol. When fueled with n-butanol and n-amyl alcohol, the peak values of the in-cylinder pressure slightly decreased and the combustion phases were slightly delayed. That was mainly because the amounts of directly injected n-heptane decreased and the reaction activities of in-cylinder mixtures decreased with the increase of premixed ratio. That would lead to the reduction in the premixed combustion zone of n-heptane during compression ignition. In that condition, the initial in-cylinder temperature and pressure was decreased, causing decreased concentrations of active radicals. Then, the combustion of n-butanol and n-amyl alcohol outside the pre-mixed flame would be delayed to some degree[26].

Peak pressure rise (PPR)

PPR for RCCI combustion has been reported by Splitter et al.[3], Li et al.[14] and Benajes et al.[16]. The reduced direct injection fuel reactivity of the EHN + Gas strategy resulted in a more abrupt combustion process. The reduced gradient in reactivity shortened the combustion event, even though the mass of EHN + Gas was greater than that of the equivalent ULSD operation. Mixing distributions were likely less influential on the combustion rate than the local reactivity. The pressure traces showed that the faster burn with EHN + Gas increased the PPRR relative to ULSD[3]. Without the presence of fuel reactivity gradient, the peak PPR of case-2 could reached about 17 bar/degree which tends to produced knocking and was quite damaging to the engine. The PPRR of the base case could be well controlled within 10 bar/degree which was acceptable during the operation. The peak PRR for case-4 exceeded 10 bar/degree which was not favorable. Case-3 with the SOI timing of -35° CA ATDC showed a low peak PRR, thereby a low combustion efficiency[14]. At CR 11:1 the higher PPR as BMEP was increased and engine speed was reduced. At 50% load, slightly higher maximum PPR values than in the case of CR 14.4:1 were obtained. The maximum PPR registered was 25.1 bar/CAD at 1200 rpm and 96% load[16].

Ignition delay

Ignition delay for RCCI combustion has been reported by Benajes et al.[4], Li et al.[15], Zhu et al.[20], Kokjohn et al.[23] and Qian et al.[26,27]. As the diesel/gasoline in-cylinder blending ratio was reduced from 100/0% to 50/50%, the ignition delay increased due to the global lowering of fuel reactivity. If the diesel/gasoline ratio was further reduced, from 25/75% up to 10/90% the combustion process was highly worsened. The diesel injection event was too short and the energy given by the diesel injection was not enough to onset multiple propagation flames. As the diesel SOI was advanced, the local equivalence ratio stratification was reduced because of the increase of the ignition delay. Increasing the ignition delay and shortened the combustion duration mainly by enhanced the second combustion stage, as the injection timing was advanced[4]. With C-SOI, as gasoline ratio increased from 0.0 to 0.8, the ignition delays were almost the same but with a slightly downward trend. When SOI was advanced to -35° CA ATDC with more premixed gasoline the ignition delay was significantly lengthened. The gasoline had low reactivity; consequently more gasoline was believed to lead to longer ignition delay[15]. With retard in the direct injection timing, the ignition was also delayed. That was mainly due to that the well mixed low reactivity fuel from the port injection could hardly be ignited by compression and was ignited by directly injected fuel with high reactivity and the combustion of the directly injected fuel was controlled by its injected timing. The ignition delay of the RCCI combustion mode with ethanol as port fuel was longest, followed by that of n-butanol while maintained the same direct injection timing. The ignition delay decreased with an increase in the premixed ratio with port injection of gasoline. When fueled with n-heptane/ethanol, the ignition delay increased[20]. Reduced the difference in ignition delay between the most reactive and least reactive regions in the combustion chamber may improve the engine efficiency. The temperature stratification had only a minor effect on ignition delay. For primary reference fuel blends from 0 to 75, the NTC behavior resulted in a nearly constant ignition delay. The ignition delay was shortest in the cooler regions at PRF blends of 50 and less. As the PRF was increased past 50, the ignition delay was shortest in the higher temperature regions[23]. The ignition delay of ethanol was longer than that of n-butanol and n-amyl alcohol. That was mainly due to the higher latent heat and lower cetane number of ethanol compared to both n-butanol and n-amyl alcohol[26]. The ignition delay of gasoline/n-heptane was smaller than that of ethanol/n-heptane due to the lower latent heat of vaporization and higher cetane number of gasoline and the higher peak values of ethanol heat release rate curves were mainly due to the greater ignition delay of ethanol/n-heptane. The longer ignition delay, the mixture of ethanol, n-heptane and air became more homogeneous. Once ignited, it burned rapidly and the peak values of the heat release rate curves became higher[27].

Ringing intensity (RI)

RI for RCCI combustion has been reported by Yaopeng et al.[2], Desantes et al.[6], Paykani et al.[8], Kakaee et al.[9], Zhou et al.[12] and Benajes et al.[17,19]. The RI decreased with the increased methanol. With the increase of methanol, the decrease of RI was found. Since the peak of HRR was closely related to the ringing intensity, that was the reason for the fact that the highest RI was found at highest peak of HRR with mid-SOI[2]. Higher levels of ringing intensity appeared when the combustion phasing was closer to TDC. The slightly delayed CA50 allowed the ringing intensity to remain below the target level[6]. Once the CA50 was beyond 4.3° CA ATDC, the RI decreased rapidly with the retarded CA50 for all the parameters. Retarding CA50 later than TDC was very effective to reduce RI. Both methanol fraction and initial pressure demonstrate the similar characteristics on RI as CA50 was later than -3.69° CA ATDC. RI remained negligible change and then reduced substantially with the increased methanol fraction or the decreased initial pressure. When the CA50 was further retarded later than 4.3° CA ATDC, the overall RI was slowed down due to the cooling effect of expansion from the retarded CA50. The RI quickly decreased with the increased methanol fraction[7]. The monotonic drop of RI occurred with increased natural gas mass fraction. The slightly delayed CA50 allowed the RI to remain below the target level. A slight increase of RI occurred with retarded SOI-2 timing. By increasing initial temperature, reaction rates increase, which increased RI and advanced combustion phasing. At engine speed of 800 rpm, the required time for initiation of combustion reactions was higher and combustion occurred at lower CA50 and higher RI values[8]. For gas with lower WN, increased initial temperature rises reaction rates, and thus reduces the available time for complete combustion which increased RI. At engine speed of 800 rpm, the required time for initiation of combustion reactions was higher and combustion occurred at lower CA50 and higher RI values[9]. At 10% load operating condition, the ringing intensity dropped monotonically with increased methanol mass fraction. For the 50% load, with larger percentage of methanol being introduced from the intake port, the increase of IP per cylinder and drop of RI were found. If a small amount of direct injected was burnt, higher premixed fuel with low cetane number could suppress autoignition because of its lower reactivity. Hence, drop tendency of ringing intensity was found at 10% and 50% load. If a large amount of fuel was burnt in that stage, higher temperature and active radical pool could enhance end-gas autoignition, which could explain the increased ringing intensity at 100% load[12]. At medium loads ringing intensity, its values were clearly related with the blend reactivity. Values under 5 MW/m^2 were achieved using B7+E10-98 for all conditions tested. B7+E10-95 combustion leads to RI values greater than 5 MW/m^2 for the most advanced SOI_{main} whatever the PER. B7+E20-95 exhibited an intermediate behavior, with low RI when using the highest PER and greater one when using advanced SOI_{main} combined with the lowest PER. At higher loads no clear trend was found in terms of ringing intensity. All the values are confined between 12 and 15 MW/m^2 for B7+E20-95 and B7+E10-98 and a unacceptable RI value around 25 MW/m^2 was registered for B7+E10-95 at -9 CAD ATDC with the highest PER (64%)[17]. Ringing intensity trends between the different blends were the same regardless the PER. The higher RON and MON, the lower RI registered. RI values for each fuel became lower as PER was increased. Slightly higher RI values were found with the lower PER due to the enhancement in the blend reactivity through the higher diesel fuel mass[19]. As the CA50 was retarded from 8.5 to 10.0° CA ATDC, lower RI was found due to the retarded ignition timing and prolonged combustion duration. When initial temperature was less than 380 K, the methanol strategy without EGR can achieve satisfactory RI compared to the EGR strategy. When initial temperature was higher than 380 K, a certain amount of EGR rate was necessarily introduced to avoid the dramatic increase of RI with elevated initial temperature[25].

Combustion duration

Combustion duration for RCCI combustion has been reported by Li et al.[15] and Benajes et al.[19]. Combustion duration was shortened with the increase in gasoline ratio regardless of SOI timing. Varying gasoline ratio from 0.0 to 0.8, combustion duration decreased from 20° to 10° CA with C-SOI and from 80° to 5° CA with A-SOI. In the case with gasoline involved, once combustion started, the gasoline which was homogeneous would deplete simultaneously; thus, more gasoline leads to shorter combustion duration. The long combustion duration with A-SOI under low gasoline ratio were caused by poor mixing process in squish region[15]. The low reactivity fuels with higher reactivity enhanced the autoignition process, which resulted in shorter combustion duration. The late combustion phase, from +10 to +20 CAD ATDC, was almost equal whatever the PER and blend. The EOC was almost the same between fuels for the same PER[19].

Combustion phasing

The combustion phasing fluctuated over the swept gasoline ratios with C-SOI; however, it significantly shifted backwards to top dead center with A-SOI. The less variation of CA50 with C-SOI may be due to the low sensitivities of ignition delay and combustion duration to gasoline ratios. The significant shift with A-SOI might be resulted from the large variations of ignition delay and combustion duration[15].

Indicated mean effective pressure (IMEP)

IMEP for RCCI combustion has been reported by Benajes et al.[17] and Qian et al.[27]. At medium loads IMEP values were very similar except for delayed diesel SOI_{main} timings, in which a drop was found for B7+E20-95 and B7+E10-98. The improved reactivity of B7+E10-95 allowed maintaining almost constant values at that point. At higher loads IMEP, values almost collide for all the fuel blends, with slight differences associated to the combustion duration[17]. The CA10 and CA50 of gasoline/n-heptane were more advanced than those of ethanol/n-heptane. Under high loads, the CA90 of gasoline/n-heptane occurred later than with ethanol/n-heptane. With increased IMEP, more fuel was injected into the cylinder by both port injection and direct injection. In that condition, more latent heat of vaporization was generated. That would lead to lower initial in-cylinder temperatures. With increased load, more fuel should be consumed when CA10, CA50 and CA90 were reached[27].

Engine knocking identification

The pressure analysis could play a significant role in predicting engine knock. It was ideal to apply mean pressure to identify engine knocking because engine knocking especially end-gas autoignition phenomenal should be captured at a local region. The regions near the cylinder wall were prone to engine knocking than other regions. That was mainly due to the autoignition of methanol in those regions. The value of filtered local pressure could be feasible for identifying engine knocking, especially in RCCI dual-fuel engine where local parameters were crucial to determine the engine knocking tendency. Engine load, methanol mass fraction, SOI and EGR rate could comprehensively affect the engine knocking occurrence. Engine knocking of case-4 was more intense than that of case-3 simply because of higher load in case-4, which resulted in higher temperature rise, promoting the end-gas autoignition. The presence of HCO indicated that end gas autoignition was about to take place and the concentration of HCO could in some respect represented engine knocking intensity. In the base cases, the concentrations of HCO for case-4 were the highest while the smallest for case-1, which was consistent with their engine knocking intensity[13].

Engine knocking mitigation

With 10-20% induction of EGR, the knocking intensity index did not show obvious drop compared to that of 30-50% EGR. Engine knocking mitigation effects of EGR was resulted from reduced reaction rate due to the dilution of the fresh charge. Engine operation at 50% load showed very light engine knocking, whereas the engine operating at 100% load need high EGR rate to suppress the engine knocking. At 10° ATDC, the biodiesel started ignition, leading to high temperature in piston bowl. At 20° ATDC, the flame propagated towards cylinder wall in which the temperature rises. The EGR was able to suppress the temperature and flame propagations, mitigating engine knocking. With delayed SOI, the knocking intensity showed a general decreasing trend. Engine knocking tendency increased with the augment of premixed methanol from port-injection, which increased the possibility of end-gas autoignition because of higher methanol concentration at Region-7. That may be due to the high concentration of methanol at Region-7, indicated high reactivity of the end gas and easy autoignition. Higher premixed methanol ratio with advanced injection timing could remarkably increase engine knocking tendency[13].

CH_4 emissions

The mole fractions of CH_4 were increased at the beginning of the combustion process. That may due to the oxidation of DME can produced some CH_4. With the addition of hydrogen, the consumption rate of CH_4 was increased, especially in the initial 10 to 15 crank angles of the combustion period[1].

HC and CO emissions

HC and CO emissions for RCCI combustion has been reported by Liu et al.[1], Yaopeng et al.[2,7,25], Splitter et al.[3], Benajes et al.[4,16,17,18,19], Li et al.[5], Desantes et al.[6], Paykani et al.[8], Kakaee et al.[9], Gharehghani et al.[11], Zhou et al.[12], Zhu et al.[20], Fang et al.[24] and Qian et al.[26,27]. The CO emissions of the natural gas dual fuel engines were significantly higher than that of the normal diesel engines, especially at part load conditions. CO species were produced at the initial combustion stage and then part of them was consumed by oxidation reactions. With the addition of hydrogen, CO emission was reduced. That may be due to the addition of H_2 advanced the ignition time and increased the combustion temperature, which promoted the oxidation of CO [1]. The addition of methanol was beneficial for emission reduction, especially for HC and CO emissions when methanol mass fraction was less than 60%. There was slight increment in HC emissions as methanol mass fraction was increased from 0% to 10%. When methanol was more than 60%, part of the fuel burned incompletely, leading to less formation of CO. With increased methanol fraction, the CO emission decreased monotonically. There was substantial decline in HC emissions with advanced SOI. The diesel fuel was more homogeneous with advanced SOI, resulted in larger region with high combustion temperature. The high-temperature region leads to a low level of HC emissions. When SOI was 37° CA BTDC, the entire bowl stays in high-temperature environment, especially the temperature of squish region was very high at 10° CA ATDC thus, the HC emission was extremely low[2]. The local mixture characteristics dominate CO formation in RCCI. The operating parameters were able to strongly influence the CO emissions, and simultaneously lean premixed and global equivalence ratios increased CO emissions[3]. If the diesel/gasoline ratio was reduced from 25/75% up to 10/90% the combustion process gets highly worsened. HC and CO were reduced, as the diesel SOI was advanced. That may be due to the fact that the peak of premixed burn increased and also does the maximum adiabatic flame temperature. The shorter ignition delay of delayed injection timings implied the richer local equivalence ratios. Those high temperatures contributed to improve oxidation processes and to reduce CO and HC emissions[4]. Factor E was the most significant one for reducing CO emissions. Factors A and C had positive effects on reducing CO emissions. A negative correlation between indicated power and ISCO for the interaction of Factors B and D which implied low CO could be achieved by controlling ignition timing near top dead center. The oxidization process of CO could rapidly happen if ignition happened near top dead center. Low CO was achieved when ignition happened near top dead center. Most of CO of R16 was formed near the center of cylinder with a long 2nd ID. Long 2nd ID could create a short penetration for diesel and retard the ignition timing. The ignition happened long time after the top dead center, when the in-cylinder pressure was decreased, as well as temperature, which was not favorable for oxidation of CO[5]. At maximum combustion efficiency the minimum values of 2.9 g/kW-hr and 2.8 g/kW-hr were found for unburned HC and CO emissions, respectively. Compared the tests with the maximum and minimum combustion efficiency it was possible to note a reduction of 6.47 g/kW-hr and 1.61 g/kW-hr in CO and unburned HC, respectively[6]. When SOI was retarded later than 14.6° CA BTDC, the HC emission begins to increase. As SOI was advanced than the base value, the reduced combustion temperature resulted from the retarded CA50 accounted for the increase of both HC and CO emissions. SOI timing determinate the target of the injected diesel, which further affected the utilization of the in-cylinder oxygen, and consequently had an impact on CO emission[7]. The CO emission increased with increased natural gas mass fraction. With more natural gas induction, lower combustion temperature, as well as lower introduced carbon, was beneficial for less CO generation. CO oxidation deteriorates by the poor fuel combustion, which enhanced CO emission. Higher temperature resulted in more rapid oxidation of CO; therefore, low CO was achieved when ignition happened near TDC[8]. Higher temperature resulted in more rapid oxidation process of CO which could lead to lower CO. For gas with lower WN, increased initial temperature rises reaction rates, and thus reduced the available time for complete combustion which could decrease UHC. At engine speed of 800 rpm, the required time for initiation of combustion reactions was higher, and combustion occurred at lower CA50. The temperature was also higher which resulted in lower CO and UHC emissions. Increased the engine speed, the available time for chemical reactions was reduced, which resulted in more incomplete combustion and thus CO and UHC emissions were enhanced[9]. For dual fuel mode either pilot fueled with diesel or biodiesel, the CO concentration was significantly higher than for conventional combustion mode at all engine loads. In dual fuel RCCI mode, CNG was induced into intake manifold and form a premixed mixture at intake stroke and that leads to lower local fuel-rich region. The reduced in-cylinder temperature in this mode together with lower oxygen concentration caused the incomplete combustion which resulted in more carbon monoxide and unburned hydrocarbon. The fuel trapped in crevices during combustion process leads to higher CO and UHC emissions[11]. At 10% load, the CO emission increased with increased methanol mass fraction except for the 80% methanol case. At the early stage of combustion more CO was generated for less methanol induction cases. With the continuation of combustion, more CO was oxidized, which resulted in less CO emissions. With more methanol induction, lower combustion temperature, as well as lower introduced carbon, was beneficial for less CO generation. The poor fuel combustion would deteriorate the CO oxidation, which gives rise to more CO emission finally[12]. At CR 14.4:1 the HC and CO emissions were found to decrease as BMEP increased. The lower HC and CO emissions are located in the region of the map with great combustion stability and also high PRR. The engine operating condition with the best balance in terms of HC and CO emissions achieved was found at 1500 rpm and 50% load, with 3.9 g/kW-hr and 4.4 g/kW-hr respectively. At CR 11:1 as found with CR 14.4:1, HC and CO emissions levels were notably reduced as BMEP increased. The lower HC and CO emissions were located in the zone of the map with the greater combustion stability and PRR. The values of HC = 0.17 g/kW-hr and CO = 1.75 g/kW-hr were attained at 1500 rpm and 85% load. That represents a 95% improvement in HC and 60.2% in CO versus the best balanced operating point in terms of these emissions at CR 14.4:1[16]. At lower loads tests with PER = 54% resulted in a combustion development with a $COV_{IMEP} > 4.5\%$ and CO and HC emission levels greater than 32 and 14 g/kW-hr, respectively. At medium loads the worst test with PER = 49% resulted in a $COV_{IMEP} = 3.5\%$ and CO and HC emission levels greater than 32 and 8 g/kW h, respectively[17]. At low loads improved HC and CO emissions due to the enhanced HRR peak and temperature was found. The advanced CA50 resulted in higher combustion temperatures leading to lower CO and HC emissions[18]. Unburned HC emissions correlate with the fuel blend reactivity, which was also related with the maximum energy released during the combustion. As the fuel blend reactivity was increased lower unburned HC were registered. Reduction in CO emission levels were achieved as PER was increased from 49% to 69% for all the blends[19]. With an advance in the direct injection timing, the CO emissions gradually decreased. With an advance in the direct injection timing, more CO converted to CO_2 because of the higher in-cylinder temperature. At retarded direct injection timing, the direct injection fuel does not have enough time to blend and begins to burn. A large proportion of diffusive combustion leads to high CO emission[20]. At lower loads the HC emissions monotonically increased along the optimization path, while the CO emissions had an opposite trend. The increase of HC emissions at the first two steps along the optimization path was likely caused by increased FEF. A greater fraction of ethanol in total fuel input was likely to cause higher HC emissions since more ethanol fuel stayed in the squish and crevice volumes and was left unburned. At the last step along the optimization path, although FEF was decreased to maintain stable combustion, the HC emissions still increased[24]. At constant CA50, HC emissions decreased with higher EGR rate and initial temperature, even though the average in-cylinder combustion temperatures were almost identical for the cases with the same CA50. HC emissions mainly form at the low temperature region near the cylinder wall, thus the elevated initial temperature could directly promote HC oxidation by increased the temperature in the near wall region. HC emissions decreased accordingly with the increase of initial temperature and methanol fraction. CO emissions reduced more obviously with the increased initial temperature and methanol fraction. The combustion temperature in the region adjacent to the cylinder wall and piston rim became higher with increased initial temperature and methanol fraction, which reduce both HC and CO emissions[25]. CO emissions had a tendency of first decreased and then increased with port injection of ethanol. The CO emissions were mainly determined by the degree of uniformity of the combustible mixture and the in-cylinder temperature. With RP increased, the mixture in the cylinder became more uniform and when the premixed ratio increased from 0.45 to 0.7, a more homogeneous mixture helped to suppress CO emissions[26]. When the RP was approximately 0.47 and the system was fueled with gasoline/n-heptane, CO emissions reached a maximum value. With further increase in RP, CO emissions decreased significantly. With increased premixed ratio, CO emissions decreased slightly, reached a minimum at an RP of approximately 0.67[27].

NOx emissions

NOx emissions for RCCI combustion has been reported by Liu et al.[1], Yaopeng et al.[2,7,25], Benajes et al.[4,16,17,18,19], Li et al.[5], Desantes et al.6, Paykani et al.[8], Kakaee et al.[9], Gharehghani et al.[11], Zhou et al.[12], Li et al.[14,15], Zhu et al.[20], Molina et al.[22], Fang et al.[24] and Qian et al.[26,27]. In dual-fuel engines, the flame temperatures were less than that in diesel engines, since most of the fuel burned under lean premixed conditions. So the NOx emissions of the dual fuel engines tend to be lower than that of the diesel engines. The combustion of the natural gas-air mixture usually begins in the expansion stroke, which suppresses the temperature increase and freezes NOx production chemistry. With the addition of hydrogen, NO emission was increased. That may be due to the addition of H_2 advances the ignition time and increases the combustion temperature, which promotes the production of NO[1]. NOx decreased with increased methanol. The in-cylinder temperature rises with increased methanol fraction, and large area with high temperature existed in the cylinder. So more methanol adding promoted NOx formation due to the high combustion temperature. The retarded ignition timing was helpful for decreasing the residence time of mixture in the high ambient temperature. As a result, there was less time left for NOx formation. Increased methanol fraction was beneficial for the reduction of NOx emission. SOI makes insignificant differences in NOx reaction duration. There was larger area with high-temperature zone existed in the cylinder with advanced SOI, where NOx was mainly formed. Thus the NOx emission increased with advanced SOI[2]. Longer ignition delay helped in slight reduction of NOx emissions. The maximum adiabatic flame temperatures were higher and NOx emissions increased. The lean local equivalence ratios and the low adiabatic flame temperatures agreed with the low levels of NOx emissions. NOx increased as the diesel SOI was advanced[4]. Factor E was the most significant one for reducing NOx emissions. Factors A and C had negative impacts on NOx emission. Factors B, D and F had positive impacts on NOx emission. Early 2nd SOI timing had a negative impact on NOx[5]. As the main direct injection timing was delayed, combustion phasing was closer to TDC which elevated the combustion temperature. That high temperature achieved during the combustion development enhanced the NO formation reactions promoted an increase in the NOx emissions[6]. The retarded injection timing in Design-2 also contributed to the reduced resident time in high temperature by delaying the time of CA50, which could inhibit the NOx formation[7]. NOx decreased with increased natural gas content. The in-cylinder temperature dropped with increased natural gas fraction. More natural gas reduced NOx formation due to the low combustion temperature. Along with more natural gas, the retarded ignition timing is favorable for reducing the residence time of the mixture with high ambient temperature. There was less time for NOx formation[8]. As CA50 moved towards to TDC, the in-cylinder temperature increased and therefore NOx enhanced. The gas with lower WN had lower peak pressure and temperature either in high or low initial temperatures. For both gases, by increasing the initial temperature, peak temperature was increased and NOx increased as well. At engine speed of 800 rpm, the required time for initiation of combustion reactions was higher, and combustion occurred at lower CA50. The temperature was also higher which resulted in higher NOx emissions[9]. Dual fuel mode of operation for either pilot fuels showed much lower NOx emission than conventional mode. Premixed nature of dual fuel mode which was followed by CNG injection into intake manifold and also earlier pilot fuel injection at 45° BTDC led to a low temperature HCCI-like combustion which reduced NOx emission relative to conventional mode. When engine fueled with CNG-biodiesel, much higher NOx emission was produced as opposed to CNG-diesel which was about 36% on average through all engine loadings. NOx formation for CNG-biodiesel mode was much lower than conventional combustion modes for either diesel or biodiesel[11]. More methanol induction leading to poor combustion at 10% load resulted in low temperature. As a result NOx under 10% load showed decreased trend with more methanol. For both 50% and 100% load there were no tangible variations of the NOx emission from 20% to 60% methanol mass fraction but reduction at 80%[12]. The NOx emission which was about 230 ppm for case-2 was a little bit higher than that of the base case which was about 170 ppm. That may because the longer critical time in high temperature region for case-2 where ignition delay was shorter compared to the base case. The fuel reactivity gradient could help retard the ignition and consequently cut the critical time period in local high temperature region, subsequently reduced the NOx emissions[14]. Generally increased in gasoline ratio could reduce NOx emissions. When gasoline ratio exceeded 0.2, A-SOI could reduce NOx emissions compared with C-SOI. That might be due to the extended ignition delay of A-SOI which provide enough time for air-fuel mixing. Final value of NOx emissions were at the same or even higher level when gasoline ratio equals to 0.0 and 0.1 compared with that adopting A-SOI[15]. NOx emissions remained below the limit of 0.4 g/kW-hr as EGR rate was decreased. Further reduction in the EGR rate will cause the non-compliance in terms of NOx emissions. At CR 14.4:1 all the NOx values were below EURO VI limitation (0.4 g/kW-hr)[16]. At lower loads, a delay in diesel SOI_{main} as well as a reduction in PER resulted in a NOx emissions increase whatever the fuel blend. At lower loads, a reduction in PER enhanced the combustion process leading to higher maximum HRR peaks. CA50 was shifted to high temperature instants in the cycle as diesel SOI_{main} was delayed. Both effects directly contributed to increase the NOx emissions[17]. At low loads the advanced CA50 resulted in higher combustion temperatures leading to remarkable higher NOx. The increase in GF from 65% to 75% resulted in more delayed combustion with almost equal duration, leading to better fuel consumption while decreasing NOx emissions. Double injection strategy allowed to meet EURO VI NOx limitations for the three piston geometries. Only the stepped piston reached EURO VI limits for NOx emissions[18]. E85 leads to significantly higher NOx emission levels than the other low reactivity fuels since much lower EGR rate was required to maintain the proper combustion phasing. The EGR rate reduction promoted an increase in combustion temperature. That higher temperature achieved during the combustion development enhanced the NO formation reactions promoted an increase in the NOx levels. As PER was increased, NOx emissions increase whatever the blend. E20-95, E10-98 and E10-95 were valid to fulfill EURO VI NOx limits independently on the PER[19]. With an advance in the direct injection timing, the NOx emissions quickly increased. That may be due to the rise of the in-cylinder temperature. The NOx emission with gasoline as port fuel was higher compared to that of alcohol fuels regardless of the direct injection timing. With the advance of direct injection timing, the ignition delay would be longer. The diffusion combustion of n-heptane would reduce and the NOx emissions were inhibited[20]. The higher level of NOx emissions found from the single injection strategy was justified by the existence of wider regions of high temperature, for longer in the cycle. At medium loads, the use of double diesel injection strategy was able to attain the NOx reduction. An advanced double diesel injection strategy was not enough to attain the region of interest, because of the early phasing of combustion, which resulted in high NOx emissions. At high loads, Regarding NOx emissions, mainly the SOI-2 delay was able to reduce them, but remained far away from the region of interest[22]. At lower loads the NOx emissions were simultaneously reduced along the optimization path. With more advanced diesel injection timing, higher fraction of diesel fuel in the first injection event and higher diesel injection pressure, the mixing process of fuel and air was improved, leading to more premixed combustion and lower NOx emissions. The reduction in intake air pressure also had a statistically significant effect on increased NOx emissions[24]. NOx emissions decreased with higher initial temperature and larger EGR rate when CA50 was fixed and NOx emissions were controlled less than 0.13 g/kW-hr for the cases along the RI boundary with the highest NOx emissions. The levels of NOx and soot emissions under different methanol fractions were very low. When initial temperature was less than 380 K, the methanol strategy without EGR can achieve satisfactory NOx emissions compared to the EGR strategy[25]. With higher premixed ratio and less n-heptane, the amount of heat released from the first period decreased, and the premixed combustion of n-heptane decreased; therefore, NOx emissions decreased [26]. As RP increased NOx emissions initially increased to a high level and then decreased to a low level for both gasoline/n-heptane and ethanol/n-heptane. At increased RP, the concentration of port injection fuels increased. After the injection of n-heptane and while the piston was moving upward, the premixed n-heptane mixed with premixed port injected fuel was ignited. With the increase in RP, more of the port-injected fuel would be ignited in the n-heptane premixed combustion zone. The higher concentration of port-injected fuel would also increase its own combustion. All those factors produced a higher in-cylinder pressure and higher heat release rate, leading to higher in-cylinder temperatures. The higher temperature resulted in an increase in NOx emissions[27].

Soot emissions

Soot emissions for RCCI combustion has been reported by Yaopeng et al.[2,7], Benajes et al.[4,16,17,18,19], Li et al.[5], Paykani et al.[8], Zhou et al.[12], Li et al.[14,15], Zhu et al.[20], Molina et al.[22], Fang et al.[24] and Qian et al.[26,27]. Soot emission reduced with the increment of methanol until methanol fraction was more than 30%, and then soot emission increased. Soot emission decreased with advanced SOI. The soot emission reduced with advanced SOI for the longer ignition delay and the less fuel-rich region where soot was mainly formed. The high in-cylinder temperature with advanced SOI was also favorable for soot oxidation[2]. As the diesel/gasoline in-cylinder blending ratio was reduced from 100/0% to 50/50%, soot emissions were strongly reduced due to longer ignition delay and the increase of the low reactivity fuel amount. The lean local equivalence ratios and the low adiabatic flame temperatures agreed with the low levels of soot emissions. Soot increased as the diesel SOI was advanced [4]. Factors A and C had positive effects on reducing soot emissions. Early 2nd SOI timing had a positive effect on soot. At the 5° CA ATDC, soot was formed mainly near the cylinder liner and the center for both cases. The soot formed near the center of cylinder was mainly due to the 2nd injection[5]. The soot emission was nearly zero at relatively high methanol fraction more than 46.55% for all the tested operating conditions. For the RCCI combustion fueled with methanol and diesel, the chemical properties of methanol and the improvement of fuel distribution through separate fuel supplement were very important for the soot reduction[7]. The soot emission reduced with increase of natural gas until natural gas fraction reached 89%, and then soot increased due to the decreased combustion efficiency resulting from incomplete combustion. As the SOI-1 timing was retarded, soot emissions were slightly increased. Soot emissions had significant increase with additional second injection fuel. Soot reduced which could be attributed to more complete combustion with higher initial temperature[8]. The Soot emission showed obvious drop with increased methanol under all engine loads. With more methanol being introduced, RCCI combustion was achieved under more homogenous fuel reactivity gradient and hence more premixed combustion mode i.e. more premixed combustion mode reduced Soot emission[12]. The base case emitted the lowest level of Soot emissions, followed by cases-2 to case-4. Increased in Soot emissions were found for cases-2 to case-4 compared with the base case. The level of the generated Soot emissions from BFM (cases-3 and case-4) combustion was higher than that from DFM (base case and case-2) combustion[14]. With A-SOI higher levels of soot emissions were generated compared with C-SOI. High ER was the major contributor to the generation of soot emissions. When $\gamma = 0.8$, ER in cylinder was out of the range of generating soot emissions for both SOI timings. Hence, negligible soot emissions were found. When $\gamma = 0.0$, final soot emissions of A-SOI were much higher than that of C-SOI. The fuel spray targeted area might be responsible for such an increase in soot emissions with A-SOI[15]. The Soot emissions were decreased as EGR was reduced due to the enhancement of its oxidation process. At CR 14.4:1 the Soot emission levels with limit of 0.01 g/kW-hr could be achieved across the whole engine map. At 50% load soot levels were very close to the limit (mainly at 1200 rpm). Further increase in load pushed the Soot levels over the maximum allowed value, even with GF greater than 90%. At CR 11:1 the lower CR allowed to advance the diesel SOI to minimize Soot formation[16]. At lower loads, soot levels registered were low in all tests for B7+E20-95, B7+E10-98 and B7+E10-95. Advanced enough injection strategy for the direct injected fuel provided sufficient mixing time prior to the start of combustion to inhibited soot formation. B7+E85 fuel blend exhibited a different behavior. The higher PER tested not allowed to inhibited soot formation even using an advanced injection strategy for the high reactivity fuel. As PER was decreased, soot emissions were increased due to the higher diesel amount in the blend[17]. At low loads the low amount of diesel fuel used (25-35%) resulted in soot levels below the minimum detection limit of the smoke meter, even in the case with the shortest mixing time available. double injection strategy allowed to meet EURO VI soot limitations for the three piston geometries. At medium loads the advance in the main injection from -12 to -21 CAD ATDC resulted in advanced combustion development with higher in-cylinder temperatures, which leads to lower soot emissions. At high loads both modifications (advance in SOI and increase GF) resulted in lower soot emissions. The significant improvement in soot emissions for bathtub piston for the most delayed case. Soot emissions were minimized for the stock and stepped geometries, but not for bathtub piston[18]. Engine out soot emissions from RCCI operation were zero whatever the low reactivity fuel used. An advanced enough injection strategy for the direct injected fuel was required to provide sufficient mixing time prior to the start of combustion and inhibit soot formation. The -30 CAD ATDC was to allow an adequate mixing time for that second fuel mass too, achieving soot levels below the minimum detection limit of the AVL 415S Smoke Meter in all tests[19]. The emissions initially decreased and then remain steady with an advance in the direct injection timing. The formation of soot was mainly based on the diffusive combustion of n-heptane. With the advance of the direct injection timing, the ignition delay increased correspondingly. A well-mixed charge reduced the proportion of diffusive combustion and improved the soot emission. The soot emission was reduced with an increase in the premixed ratio[20]. At low loads using single injection strategies there appear rich regions which promoted soot formation; however the high temperatures existed help to their oxidation resulted in low soot emissions. Using double injection strategies, the high equivalence ratio regions were strongly reduced, and therefore, the low level of soot emissions was ruled by avoiding its formation[22]. At lower loads the Soot emissions were simultaneously reduced along the optimization path. Higher diesel fuel rail pressure was well known to reduce soot by improving the mixing of diesel fuel and air. With more retarded CA50, the soot emissions were lower. At higher loads the Soot emissions were reduced along the optimization path except at the last step, where an increased occurred[24]. Soot emissions were decreased with RP increased. When the premixed ratio was higher than 0.5, soot emissions for ethanol, n-butanol and n-amyl alcohol as port injected fuels sharply decreased. With RP increased, the mixture would be more homogenous. That would lead to a decrease in soot emissions[26]. As RP was increased, the soot emissions produced by the burning of ethanol/n-heptane decreased faster than those of gasoline/n-heptane. That may be due to the fact that gasoline contained many kinds of cycloalkanes and aromatic substances. During the combustion process, cycloalkanes and aromatic substances readily generated C_3H_3 and C_4H_4, which were soot precursors. That would benefit soot formation[27].

CO_2 emissions

At lower loads the CO_2 emissions were constant at the first two steps but slightly increased at the last step. That increase was caused by decreased indicated thermal efficiency. CO_2 emissions were higher at the optimal point than at the starting point. At higher loads the CO_2 emissions were higher at the optimal point than at the starting point[24].

CONCLUSIONS

The CI engines are commonly used for applications such as automotive, power generation etc., because of their stability, fuel flexibility and higher BTE. They have undergone tremendous changes in their design concepts to comply with the increased emission regulations and for improved performance. However higher PM and NOx emissions from these engines require continued and sustainable research before they can be suitably modified to address the emission norms laid down by regulation authorities. From the exhaustive literature survey carried out to study the performance of the diesel engines fuelled with different fuel combinations in CI engines operated in conventional CI and RCCI modes, the following conclusions are made:

- With the addition of hydrogen, the peak cylinder pressure, NO emission is increased and CH_4, CO emission is reduced.

- RCCI combustion is able to produce an important reduction in soot and NOx with respect to neat diesel combustion and high levels of CO and HC were measured.
- As the diesel injection timing is advanced, the fuel mixture gets better stratified and less zones of low local reactivity exists across the cylinder.
- For RCCI engines, the premixed fuel plays a dominant role in improving the performance of engines, achieving high indicated power as well as fuel conversion efficiency and also premixed fuel could provide positive effects on the reductions of CO, NOx and soot emissions.
- The RCCI combustion with high energy fraction of methanol and advanced SOI exhibits higher fuel efficiency and lower emissions.
- By increasing initial temperature reaction rate increases which increases NOx and RI and advances combustion phasing.
- The use of biodiesel instead of diesel as a pilot fuel reduced cycle-to-cycle variations due to the combustion improvement resulted from high cetane number and oxygen content.
- An advanced injection strategy for the high reactivity fuel allows inhibiting soot formation.
- The reduced reactivity gradient between high and low reactivity fuels enhanced the combustion propagation which allowed a considerable reduction in HC and CO emissions.
- As the direct injection timing advances, CO and soot emissions gradually decrease while NOx emission increases.
- SCC is the best piston design for RCCI combustion among the different geometries.

SCOPE FOR FUTURE WORK

From the presented literature it is inferred that there is scope for further improvements in RCCI, as listed below[21].

- Further optimization of engine parameters is needed to fully realize the potential of dual-fuel RCCI operation. Of paramount interest is the feasibility of cycle-to-cycle control of dual fuel RCCI operation over a wide range of loads and during transient operation. The next steps in operating parameter optimizations should include other load points.
- The need for increased understanding of the performance of turbo-machinery for low temperature combustion is also apparent. The relatively low exhaust temperatures imply that high turbocharger efficiencies will be needed.
- Future work will help define the level of after-treatment required for meeting LD federal-emission standards and will need to address other design elements of interest, including the relative sizes of the direct-injected diesel fuel tank and the PFI fuel tank needed in a vehicle.
- The lower exhaust temperatures with RCCI also offer challenges for after treatment systems. Developments in oxidation catalyst after-treatment systems are required to treat HC and CO emissions at the available exhaust temperature. More research is required in the direction of low temperature catalysts.
- Choosing an inferior low reactivity fuel compared to gasoline may also be advantageous. Experiments conducted with ethanol as the low reactivity fuel show that it requires more high reactivity fuel at high load operation. This was also utilized to extend the load range in the LD multi-cylinder RCCI engine. Other potential of low reactivity fuels could also be investigated. Similarly, instead of diesel, biofuels (including neat biodiesel) could also be used as the DI fuel.
- More experiments are needed at higher speeds and loads to transfer the RCCI concept toward other applications (e.g., automotive/stationary).

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NOMENCLATURE

1D = one dimension; AHRR = apparent heat release rate; A-SOI = advanced start of injection; ATDC = after top dead center;

BFM = blend fuel mode; BMEP = brake mean effective pressure; BSFC = brake specific fuel consumption; BTDC = before top dead center; BTE = brake thermal efficiency; CA = crank angle; CA10 = crank angle at 10% mass fraction burnt; CA50 = crank angle at 50% mass fraction burnt; CA90 = crank angle at 90% mass fraction burnt; CAD = crank angle degree; CFD = computational fluid dynamics; CH_4 = methane; CI = compression ignition; CN = cetane number; CNG = compressed natural gas; CO = carbon monoxide; CO_2 = carbon di-oxide; COV_{IMEP} = coefficient of variation in indicated mean effective pressure; CR = compression ratio; C-SOI = conventional start of injection; DFM = dual fuel mode; DI = direct injection; DTBP = di tertiary butyl peroxide; EGR = exhaust gas recirculation; EGT = exhaust gas recirculation; EHN = ethyl hexyl nitrate; EISFC = equivalent indicated specific fuel consumption; EOC = end of combustion; EOI = end of injection; EPA = environmental protection agency; ER = energy ratio; FEF = fumigant energy fraction; GF = gasoline fraction; GIE = gross indicated efficiency; GTE = gross thermal efficiency; H_2 = hydrogen; HC = hydrocarbon; HCO = ; HD = heavy duty; HRR = heat release rate; HTHR = high temperature heat release; ICFB = in-cylinder fuel blend; ID = injection duration; IMEP = indicated mean effective pressure; IP = injection pressure; IVC = inlet valve closing; IVO = inlet valve opening; LD = light duty; LHV = lower heating value; LTC = Low temperature combustion; LTHR = lower temperature heat release; MON = motor octane number; NOx = nitrogen oxide; PCCI = premixed charge compression ignition; PER = premixed energy ratio; PFI = port fuel injection; PM = particulate matter; PPRR = peak pressure rise rate; PRF = primary reference fuel; PRR = pressure rise rate; PPR = peak pressure rise; R11 = run number 11; R15 = run number 15; RCCI = reactivity controlled compression ignition; RI = ringing intensity; RoHR = rate of heat release; RON = research octane number; rpm = revolutions per minute; SOC = start of combustion; SOI = start of injection; TDC = top dead center; UHC = unburnt hydrocarbon; ULSD = ultra-low sulphur diesel; WFO = waste fish oil.