

An Experimental Study on the Effect of Premixed and Equivalence Ratios on CO and HC Emissions of Dual Fuel HCCI Engine

M. Ghazikhani, M. R. Kalateh, Y. K. Toroghi, and M. Dehnavi

Abstract—In this study, effects of premixed and equivalence ratios on CO and HC emissions of a dual fuel HCCI engine are investigated. Tests were conducted on a single-cylinder engine with compression ratio of 17.5. Premixed gasoline is provided by a carburetor connected to intake manifold and equipped with a screw to adjust premixed air-fuel ratio, and diesel fuel is injected directly into the cylinder through an injector at pressure of 250 bars. A heater placed at inlet manifold is used to control the intake charge temperature. Optimal intake charge temperature results in better HCCI combustion due to formation of a homogeneous mixture, therefore, all tests were carried out over the optimum intake temperature of 110-115 °C. Timing of diesel fuel injection has a great effect on stratification of in-cylinder charge and plays an important role in HCCI combustion phasing. Experiments indicated 35 BTDC as the optimum injection timing. Varying the coolant temperature in a range of 40 to 70 °C, better HCCI combustion was achieved at 50 °C. Therefore, coolant temperature was maintained 50 °C during all tests. Simultaneous investigation of effective parameters on HCCI combustion was conducted to determine optimum parameters resulting in fast transition to HCCI combustion. One of the advantages of the method studied in this study is feasibility of easy and fast transition of typical diesel engine to a dual fuel HCCI engine. Results show that increasing premixed ratio, while keeping EGR rate constant, increases unburned hydrocarbon (UHC) emissions due to quenching phenomena and trapping of premixed fuel in crevices, but CO emission decreases due to increase in CO to CO₂ reactions.

Keywords—Dual fuel HCCI engine, premixed ratio, equivalence ratio, CO and UHC emissions.

I. INTRODUCTION

OVER the past decade, numerous studies have been conducted to improve engine performance and reduce exhaust emissions. Although there have been advances in both areas, still more research needs to be done in order to meet the strict upcoming emission regulations such as EUR-V [1]. There are many alternatives being researched to improve the engine-out emission further. An alternative combustion

process that has received considerable interest in recent times due to lower emissions is the homogeneous charge compression ignition (HCCI) [2]. HCCI, also referred to as Controlled Auto-Ignition (CAI), has been researched for some thirty years. It was first identified by Noguchi et al. and Onishi et al. as a method to reduce emissions and fuel consumption of two-stroke engines at part-load conditions [3]. In 1983, Najt and Foster performed HCCI experiment with a four-stroke engine, they are the first to apply HCCI combustion concept in a four-stroke gasoline engine [4].

In HCCI engine, lean homogeneous fuel/air mixture is essentially inducted into the cylinder without throttling losses and then compressed to auto-ignition which occurs simultaneously through the cylinder without flame propagation. These features lead to very low NO_x and PM emissions while maintaining high thermal efficiency [5-8]. The HCCI engine combines features from both spark ignition (SI) and compression ignition (CI) engines, and incorporates the advantages of both spark ignition (SI) engines and compression ignition direct injection (CIDI) engines [9,10].

The Results from experimental and numerical studies show that the HCCI combustion has a low temperature heat release and a high temperature heat release, and both heat release occur within certain temperature ranges. The low temperature heat release is one of the most important phenomena for HCCI engine operation and the occurrence of it depends chemically on the fuel type [2, 11-13].

In HCCI engines, auto-ignition and combustion rate are dominated by the fuel chemical kinetics, which is extremely sensitive to the charge composition and to the pressure and temperature changes during the compression stroke, therefore HCCI combustion is widely assumed to be kinetically controlled [7, 9, 14]. Thus, it is challenging to maintain the onset of HCCI combustion close to the top dead center (TDC) at a wide range of loads and speeds in commercial engines. The occurrences of misfiring at low load and knocking at high load are usually noted which result in a limited operation range of HCCI engine [1,9, 14].

Despite the many advantageous features of HCCI combustion, the use of HCCI engines is not widespread due to

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NOMENCLATURE

HCCI	Homogeneous Charge Compression Ignition
EGR	Exhaust Gas Recirculation
BTDC	Before Top Dead Center
DI	Direct injection
r_p	Premixed ratio
h_{up}	Heating value of premixed fuel (kJ/kg)
\dot{m}_p	Mass flow rate of premixed gasoline (kg/s)
\dot{m}_d	Mass flow rate of injected fuel (kg/s)
h_{ud}	Heating value of diesel fuel (kJ/kg)
\dot{m}_{aEGR}	Mass flow rate of intake air with EGR (kg/s)
\dot{m}_a	Mass flow rate of intake air without EGR (kg/s)
\dot{m}_{EGR}	Mass flow rate of EGR gasses (kg/s)
ϕ	Equivalence ratio
ΔH	Water height difference in manometer (m)
D	Orifice diameter (m)
p	Ambient pressure (kPa)
T_{exh}	Temperature of EGR gasses (K)
T_{in}	Intake air temperature (K)
MON	Measured octane number
$(A/F)_s$	Stoichiometric air-fuel ratio
UHC	Unburned hydrocarbon
CO	Carbon monoxide
CO ₂	Carbon dioxide

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some unresolved issues such as: controlling ignition timing and burn rate over a range of engine speed and loads, cold starts, transient response of the HCCI engines, minimizing UHC and CO emissions particularly at low loads, high heat release rates, extending the operating range of HCCI to high loads, Nox emission increase at high loads and formation of a completely homogeneous mixture [1, 4, 9, 15].

HCCI engine has low emissions of particulate matter and NOx, while high emissions of CO and UHC are its major hurdles. HCCI combustion has higher emissions of CO and UHC in comparison with typical conventional engines. CO emissions from HCCI combustion are controlled by chemical kinetics. CO emissions from internal combustion engines are controlled primary by the fuel/air equivalence ratio. For fuel-rich mixtures, CO concentrations increase steadily with increase of equivalence ratio. For lean fuel mixtures, equivalence ratio has a slight effect on CO emissions. CO in HCCI engine is a result of incomplete combustion in intermediate temperature regions where the OH radical concentration becomes significantly diminished, resulting in less conversion of CO to CO₂, or low temperature regions such as boundary layers, cylinder wall (quenching phenomenon) and crevices.[4,9,14,16].

The lack of a well-defined ignition timing control has led a range of control strategies to be explored. Numerous studies have been conducted to investigate HCCI combustion control methods such as intake air preheating [1, 17, 18], Variable

Valve Actuation (VVA) [19], Variable Valve Timing (VVT) [5], Variable Compression Ratio (VCR) [20] and EGR rate [2]. Moreover many studies also focused on the effects of different fuel physical and chemical properties, for instance the octane number and the cetane number, using the primary reference fuels and fuel additives. HCCI researchers have investigated different fuels such as isooctane, ethanol, natural gas, hydrogen, gasoline, diesel fuel, methanol, propane and n-pentane [9, 21-23].

Another concept to overcome the disadvantages of HCCI is to use HCCI / SI dual mode combustion. Equipped with the VVA and spark ignition system, the HCCI/SI dual mode engine is able to operate in HCCI mode at low to medium loads and it can switch into SI mode to meet the large power output requirements. However the mode transition, especially from HCCI to SI, is not very stable and smooth so that more improvements would be needed on the control strategies to diminish the cycle-to-cycle variation [9, 18, 19]. Generally, this dual mode engine combines the HCCI and SI together to achieve the best performance. Many studies have focused on the application a partial HCCI engine as a control mechanism for HCCI combustion. The HCCI-DI compound combustion mode could be considered as a compromise between premixed HCCI and conventional CIDI [1,9]. Several studies have revealed the advantages of similar combined combustion mode.

Experimental studies have shown that since the timing of

fuel injection in direct injection can affect the stratification of in-cylinder charge, it plays an important role in HCCI combustion phasing. More advanced ignition contributes to formation of a more homogeneous mixture, but causes knock at high loads and thus reduce the operating range of HCCI engine. However, less advanced ignition increases the upper load limit and leads to increase of the stratification of in-cylinder charge [21, 24, 25].

Experimental results indicate that the combustion stability limit of the HCCI engine is extremely sensitive to coolant temperature. Lowering the wall temperature leads to greater reduction in the bulk burn rate which is a result of ignition delay, thus indicating the significance of thermal stratification in the near-wall boundary layer. Combustion of gasoline HCCI has great sensitivity to thermal conditions. The range of coolant temperature variations for gasoline HCCI engine is limited at the low end. Decreasing coolant temperature leads to cyclic variation and prevents stable movement of the engine. The wall temperature effect on HCCI combustion is stronger than the intake charge temperature effect [17].

Although considerable improvements have been reported, there is no direct method to control auto-ignition process. None of these methods perform satisfactorily over a wide range of engine speeds and loads. Currently, EGR has been considered to be an effective method in controlling HCCI combustion. The effect of EGR on HCCI combustion can be divided into three parts: a dilution effect (inert gasses present in the EGR), a thermal effect (heat exchange, thermal loss to the wall, EGR ratio mixture quality, EGR temperature, heat capacity), and a chemical effect. The chemical effect influences not only the overall kinetics, but it also can change a specific reaction path, which makes this effect particularly interesting for the investigation of the auto-ignition process [26, 27]. EGR provides the appropriate temperature to enhance auto-ignition, while maintaining the combustion temperature sufficiently low. In this project, external EGR was used in gasoline-diesel dual fuel HCCI engine.

The objective of this study is to investigate CO and HC emissions of dual fuel HCCI engine, using premixed gasoline. The effect of premixed and equivalence ratios on CO and HC emissions of HCCI at different EGR rates has also been investigated.

II. EXPERIMENTAL APPARATUS AND TESTING PROCEDURE

A. Description of the Engine

The engine used for the experiment was a four stroke VCR single-cylinder naturally-aspired DI diesel engine with a displacement volume of 582 cc. Specifications of test engine are presented in Table I. Premixed gasoline is provided by a carburetor connected to intake manifold and equipped with a fuel adjustment screw and diesel fuel is injected into the cylinder by an injector. The engine is also equipped with torque measurement system, coolant system that adjusts the

coolant temperature, dynamometer, gas analyzer, electric heater and dimmer (that are used to control intake charge temperature), fuel consumption measurement system, EGR system, electrical control board and air mass flow-meter (surge tank and orifice system). Fig. 1 presents a schematic of the test engine.

Dynamometer is a DC electrical motor that can be used to start the engine or to measure braking power, as well as inserting load on engine. Coolant system comprises water pump, coolant flow meter, gate valve, and coolant radiator. Coolant temperature can be adjusted to the desirable temperature by altering coolant flow rate and turning off/on the radiator fan. Fuel flow was measured by using constant volume fuel consumption system and time recorder. Tubes that return exhaust gasses and a throttle valve are the components of EGR system. CO and HC emissions are measured by a Plint RE 205 Gas Analyzer.

TABLE I
ENGINE SPECIFICATIONS

Number of cylinders	1
Displacement volume (cm ³)	582
Test compression ratio	17.5:1
Bore×stroke (mm)	95×82
Number of injection holes	4
Injection pressure (bar)	250

B. Testing Procedure

To start the engine, dynamometer is set to start mode and is used to rotate the engine. After the engine starts, dynamometer is switched to generator mode and is used to insert load on the engine and control its speed under carburetor full-throttle condition. Power is controlled by altering fuel rate of carburetor. Load can be altered by varying the resistance in electric resistant of dynamometer. Dynamometer is mounted on bearings and can rotate freely. Torque is measured by a load cell and a torque arm which is attached to dynamometer. A 2 kW electric heater placed at the intake manifold entrance controls the intake charge temperature. By using an orifice and manometer system, mass flow rate of intake air can be measured in terms of (kg/s) as follows:

$$\dot{m} = 3.65 \times 10^{-6} \times D^2 \times \sqrt{\frac{P \times \Delta H}{T}} \quad (1)$$

Where ΔH is water height difference in manometer, D is orifice diameter, P is ambient pressure, and T is ambient temperature.

To show the effect of fuels in dual fuel HCCI engine, the premixed ratio (r_p) is defined as a ratio of premixed fuel energy (Q_p) to total energy (Q_t). It can be obtained from the following equation [9]:

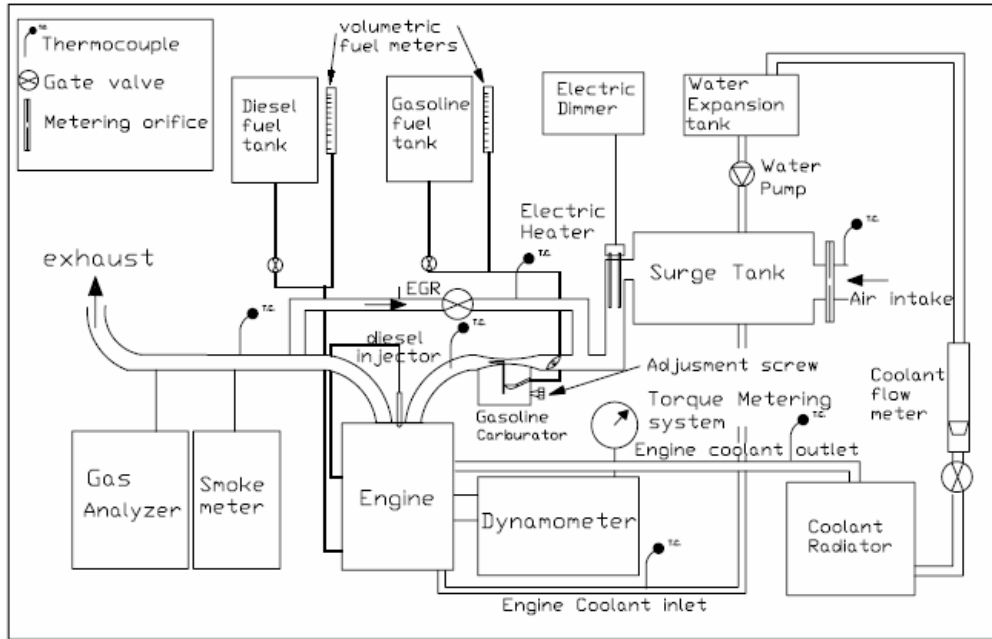


Fig. 1 Schematic of test engine

$$r_p = \frac{\dot{Q}_p}{\dot{Q}_t} = \frac{\dot{m}_p h_{up}}{\dot{m}_p h_{up} + \dot{m}_d h_{ud}} \quad (2)$$

Where, \dot{m}_p represents mass flow rate of premixed gasoline, \dot{m}_d is mass flow rate of injected fuel, h_{up} is heating value of premixed fuel and h_{ud} is heating value of diesel fuel. Therefore, $r_p=1$ corresponds to single fuel HCCI combustion and $r_p=0$ corresponds to typical CIDI combustion.

EGR rate also is calculated as follow [15]:

$$EGR (\%) = \frac{\dot{m}_{EGR}}{\dot{m}_{EGR} + \dot{m}_{aEGR}} \quad (3)$$

In the above equation, \dot{m}_{aEGR} is mass flow rate of intake air with EGR, and \dot{m}_{EGR} is mass flow rate of EGR gasses.

To be able to determine the specified EGR rate at different operating loads and speeds, a code was written in FORTRAN which computes the corresponding orifice ΔH at the specified EGR rate. To provide the code, the engine was tested without EGR at different speeds with a coolant temperature of 50 °C and the values of ΔH and exhaust gas temperature were measured. Mass flow rate of intake air was calculated using eq. 1. Then a polynomial was obtained by extrapolating the calculated values of ΔH , T_{exh} and mass rate of intake air thus giving the function related to each data. Using these functions, values of ΔH , T_{exh} and mass rate of intake air without EGR can be calculated at different speeds which were employed to determine EGR rate in the computational code, based on the data including initial speed of engine, ambient temperature and pressure, air density, air temperature at orifice intake, orifice diameter, assumption of the value of mass flow rate of EGR gasses, error and the specified EGR. Then, mass flow rate of intake air with EGR at a specific speed can be

calculated using eq. 4:

$$\dot{m}_{aEGR} \approx \dot{m}_a - \dot{m}_{EGR} \times \frac{T_{exh}}{T_{in}} \quad (4)$$

In the above equation, \dot{m}_a is mass flow rate of intake air without EGR, T_{exh} is temperature of EGR gasses, and T_{in} is intake air temperature at the specified engine speed.

A K-type thermocouple was used for measuring intake charge and exhaust gas temperatures by using an interface and computer data logging system. Intake air temperature was increased up to 115°C for the test without EGR, and up to 80 °C for the test with EGR. The amount of exhaust gasses was recirculated up to 15 % mass flow rate of intake air. Altering the coolant temperature at the 40 to 70 °C, HCCI combustion showed better results at 50 °C. Therefore, coolant temperature was maintained 50 °C throughout all the tests. For all tests, premixed fuel adjustment screw was initially closed and the engine was started with diesel fuel. Then, the values of premixed fuel were increased gradually as well as load to approach HCCI combustion. Having reached HCCI combustion, emissions, value of premixed fuel and other data were recorded. Test conditions are reported in Table II. Table III presents the specifications of the fuels used in dual fuel HCCI test engine.

TABLE II
 TEST CONDITIONS

Speed (rpm)	1200-1700
Intake charge temperature (°C)	110-115
Coolant temperature (°C)	50
EGR rate (based on mass flow rate of intake air)	0-15%
Injection timing	35 BTDC
Premixed ratio (r_p)	0-1

TABLE III
 FUEL SPECIFICATIONS [28]

Fuel type	Gasoline	Diesel fuel
MON	87	-
Cetane number	-	54
Higher Heating Value (kJ/kg)	47300	46100
Lower Heating Value (kJ/kg)	44000	43200
Heat of vaporization (kJ/kg), at 1 atm, 25 °C)	305	270
Density (kg/m ³)	720	780
(A/F) _s	14.6	14.5

III. RESULTS AND DISCUSSIONS

A. Effect of Premixed Ratio on CO Emissions

Fig. 2 shows the effect of premixed ratio on CO emissions of dual fuel HCCI-DI engine. As shown in Fig. 2, CO emission decreases as premixed ratio increases. By increasing premixed ratio, more energy is released; therefore CO emission decreases due to more conversion of CO to CO₂ at higher in-cylinder temperature. It can be noted that increasing EGR, which results in dilution of in-cylinder mixture and reduction of combustion temperature, increases CO emission. Since CIDI combustion has low emissions of CO and UHC, formation of CO and UHC emissions in premixed gasoline of HCCI-DI engine is due to HCCI combustion.

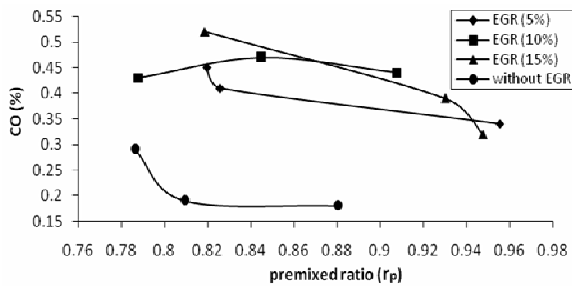


Fig. 2 Effect of premixed ratio on CO emission of dual fuel HCCI-DI engine

B. Effect of Equivalence Ratio on CO Emissions

Effect of equivalence ratio on CO emission of HCCI engine is shown in Fig. 3. CO emission decreases as equivalence ratio of the premixed fuel increases which is similar to the effect of premixed ratio. Increasing equivalence ratio of the premixed fuel moves CO to CO₂ reactions towards completion thus CO emission decreases.

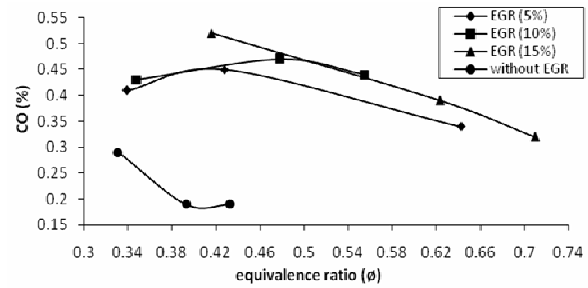


Fig. 3 Effect of equivalence ratio on CO emission of dual fuel HCCI-DI engine

C. Effect of Premixed Ratio on HC Emission

Fig. 4 demonstrates the effect of premixed ratio on HC emission of dual fuel HCCI-DI engine.

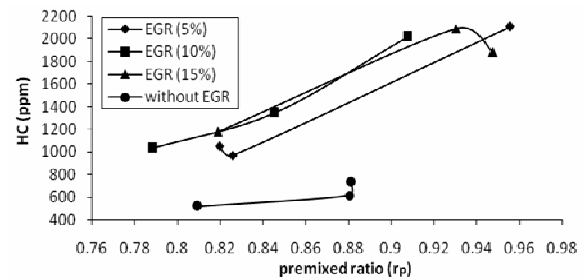


Fig. 4 Effect of premixed ratio on HC emissions of dual fuel HCCI-DI engine

According to Fig. 4, at all EGR rates, HC emissions increase by increasing premixed ratio. The main cause is the trapping of fuel premixed fuel in crevices and boundary layers that results in low oxidation at low temperatures. Heat release caused by direct injection during CIDI combustion can enhance the oxidation of HC produced during lower premixed fuel HCCI combustion and reduce HC emissions.

D. Effect of Equivalence Ratio on HC Emission

Effect of equivalence ratio on HC emission of HCCI engine is shown in Fig. 5.

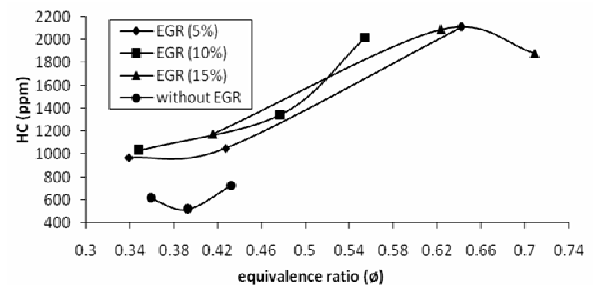


Fig. 5 Effect of equivalence ratio on HC emissions of dual fuel HCCI-DI engine

As shown, at all EGR rates, HC emission increases by increasing equivalence ratio of premixed fuel. This is due to increase in fuels trapped in crevices which do not oxidize well during low temperature HCCI combustion.

E. Effect of EGR on Extending the Operating Range of Dual Fuel HCCI Engine

According to the above figures, EGR can extend the operating range of HCCI combustion to higher equivalence ratios. This can be explained as follows (as mentioned by Machrafi et al and Chen et al [16, 26, 29]):

EGR consists of many gaseous chemical species, which includes the main components of burned gases, CO₂, H₂O, N₂ and O₂, partial burned gases such as CO, and particular matters and unburned HCs.

Generally, the effect of the chemical species in EGR played a very important role in influencing the amount of OH radicals that are present in the cylinder.

In an engine, a higher overall reactivity would be expressed by a decreasing ignition delay. Since, at higher equivalence ratios more OH radicals are formed, the effect of chemical species of EGR in the producing of OH radicals is negligible at higher equivalence ratios. Accordingly, the main effect of EGR at high equivalence ratios is dilution of in-cylinder mixture and reduction of maximum combustion temperature. Therefore, EGR can extend the operating range of HCCI combustion to higher equivalence ratios. This can be noticed in all the figures above. It can explain why the engine operating range is limited with no EGR. However, at lower equivalence ratios less OH radicals are formed, therefore the effect of chemical species of EGR in producing OH radicals is stronger at lower equivalence ratios and increases the overall reactivity. Thus, HC emission is lower at low equivalence ratios, which can be seen in Fig. 5.

IV. CONCLUSION

In this study, the effects of premixed and equivalence ratios on CO and HC emissions of gasoline-diesel dual fuel HCCI engine was investigated. Results can be summarized as:

- 1) An advantage of dual fuel engine, was fast and easy transition to HCCI mode.
- 2) By increasing premixed ratio, more energy is released; therefore CO emission decreases due to more conversion of CO to CO₂ at higher in-cylinder temperature.
- 3) Increasing EGR increases CO emission due to dilution of in-cylinder mixture and reduction of combustion temperature.
- 4) HC emission increases by increasing equivalence and premixed ratios which is a result of the crevices and quenching phenomenon near the cylinder wall, which prevent oxidation of mixture during low temperature HCCI combustion.
- 5) EGR extends the operating range of dual fuel HCCI engine to high equivalence ratios due to reduction of maximum combustion temperature.
- 6) At low equivalence ratios, EGR increases overall reactivity,

therefore HC emission is lower for low equivalence ratios in comparison with high equivalence ratios.

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