An Experimental Study on the Effect of EGR and Engine Speed on CO and HC Emissions of Dual Fuel HCCI Engine

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Abstract—In this study, effects of EGR on CO and HC emissions of a dual fuel HCCI-DI engine are investigated. Tests were conducted on a single-cylinder variable compression ratio (VCR) diesel 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 was 110-115ºC due to better formation of a homogeneous mixture causing HCCI combustion. Timing of diesel fuel injection has a great effect on stratification of in-cylinder charge in HCCI combustion. Experiments indicated 35 BTDC as the optimum injection timing. Coolant temperature was maintained 50ºC during the tests. Results show that increasing engine speed at a constant EGR rate leads to increase in CO and UHC emissions due to the incomplete combustion caused by shorter combustion duration and less homogeneous mixture. Results also show that increasing EGR reduces the amount of oxygen and leads to incomplete combustion and therefore increases CO emission due to lower combustion temperature. HC emission also increases as a result of lower combustion temperatures.

Keywords—Dual fuel HCCI engine, EGR, engine speed, CO and UHC emissions.

I. INTRODUCTION

DUE to the world-wide air pollution and strict upcoming emission regulations, a new concept of combustion technology that can reduce the NO_X and PM is urgently required. Therefore, technologically advanced countries are exerting efforts to develop environmentally friendly engines for vehicles through improvement of combustion [1]. The homogeneous charge compression ignition (HCCI) engine is a promising concept for future automobile engines and stationary power plants for its potential efficient energy conversion with low environmental impact [2, 3]. It was first identified by Noguchi et al. and Onishi et al. as a method to

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reduce emissions and fuel consumption of two-stroke engines at part-load conditions [4].

In HCCI engines, the fuel and air are premixed to form a homogeneous mixture before the compression stroke. As a result, the mixture ignites throughout the bulk without discernable flame propagation due to occurrence of auto-ignition at various locations in the combustion chamber (multi-point ignition), which may cause extremely high rates of heat release, and consequently, high rates of pressurization [5-7]. In HCCI engines, auto-ignition and combustion rate are mainly controlled by the fuel chemical kinetics, which is extremely sensitive to the charge composition and to the pressure and temperature evolution during the compression stroke, therefore HCCI combustion is widely assumed to be kinetically controlled [3, 8, 9]. The main objective of HCCI combustion is to reduce soot and NO_X emissions while maintaining high fuel efficiency at part load conditions [3, 8]. In some regards, HCCI combustion combines the advantages of both spark ignition (SI) engines and compression ignition (CI) engines [10, 11]. The results from experiment and simulation show that the HCCI combustion has a low temperature heat release and a high temperature heat release, and both heat releases 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 [12-14]. However there are certain number of obstacles and problems in its application that have not been resolved. These problems are the control of ignition and combustion, difficulty in operation at higher loads, higher rate of heat release, higher CO and HC emissions particularly at light loads, difficulty with cold start, increased NO_X emissions at high loads and formation of a completely homogeneous mixture [15-17]. 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 [16, 18, 19], Variable Valve Actuation (VVA) [6], Variable Valve Timing (VVT) [2], Variable Compression Ratio (VCR) [20] and EGR rate [12]. Moreover many studies also focused on the effects of different fuel physical and chemical properties to gain control of HCCI combustion [11, 21-23].

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)
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

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 [18].

Experimental studies have shown that fuel injection timing has a significant effect on homogeneous charge formation and on controlling the combustion mechanism of HCCI. As the time between injection and ignition increases, the variability in combustion stability improves. Start of injection (SOI) would be an effective and useful needed parameter to control ignition timing for transient engine applications. More advanced injection 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 [1, 21, 24].

The objective of this study is to investigate the effects of EGR and engine speed variation on CO and HC emissions of dual fuel HCCI engine using premixed gasoline.

II. FORMATION OF CO AND UHC IN HCCI COMBUSTION

HCCI combustion has higher emissions of CO and UHC in comparison with typical conventional engines. CO emissions from HCCI are controlled by chemical kinetics [11]. CO formation is one of the basic reaction steps in the hydrocarbon oxidation mechanism, which may be summarized as follows [25]:

$$RH \rightarrow R \rightarrow RO_2 \rightarrow RCHO \rightarrow RCO \rightarrow CO$$

Where, R stands for the hydrocarbon radical. CO 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 CO_2, or low temperature regions such as boundary layers, cylinder wall (quenching phenomenon) crevices and so on. For high cetane number fuels, the ignition timing and combustion events occur at zone before the TDC. The incomplete oxidation products in boundary layers and crevice layer were squished into the center of combustion chamber by the combustion turbulence. As a result, the incomplete oxidation productions were further oxidized. This is the reason why EGR rate has a slight effect on CO emissions [9, 11, 17, 26]. But for high octane number fuels the ignition and combustion events occur near the TDC, and then the incomplete reaction products in the boundary layer and crevice layer are discharged into the exhaust pipe during the expansion stroke. Furthermore, the gas temperature

decreases with the increase of EGR rate, the further oxidation in exhaust pipe also reduces. These reasons lead to higher CO levels with the increase of EGR rate [9].

III. ROLE OF EGR IN HCCI ENGINE

According to different methods of the exhaust gas recirculation, EGR technique can be divided into internal EGR and external EGR. The internal EGR rate can be obtained by changing valve overlap period to negative valve overlap (NVO) and the external EGR rate can be adjusted by using an EGR valve. For high-octane fuel (such as: gasoline) HCCI, negative valve overlap (NVO) is recognized as one of the possible implementation strategies of HCCI closet to production. The effect of inlet air temperature through NVO is insignificant when the engine runs well inside of the HCCI operating range [2, 12]. 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 [27]. The effects of EGR that have been investigated are: increase in intake charge temperature (heating effect), reduction of oxygen concentration (dilution effect), increase in specific heat of the mixture (heat-capacity effect), chemical interactions involving the CO_2 and H_2O species of the recycled burned gases (chemical effect), and stratification of the recycled burned gases (stratification effect) [5, 28]. The dilution and heat capacity effects are responsible for reducing the heat-release rates and delaying HCCI combustion. The heating effect is mainly responsible for the advance timing of auto-ignition, and the residuals-stratification effect facilitates HCCI combustion. Reactive species, present in the residuals, facilitate auto-ignition, and the inert species slow down the combustion rate (dilution effects) [5, 27, 28]. 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.

IV. 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 (cm³)	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:

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

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 [11]:

$$r_p = \frac{\dot{Q}_p}{\dot{Q}_t} = \frac{\dot{m}_p h_{up}}{\dot{m}_p h_{up} + \dot{m}_d h_{ud}} \qquad (2)$$
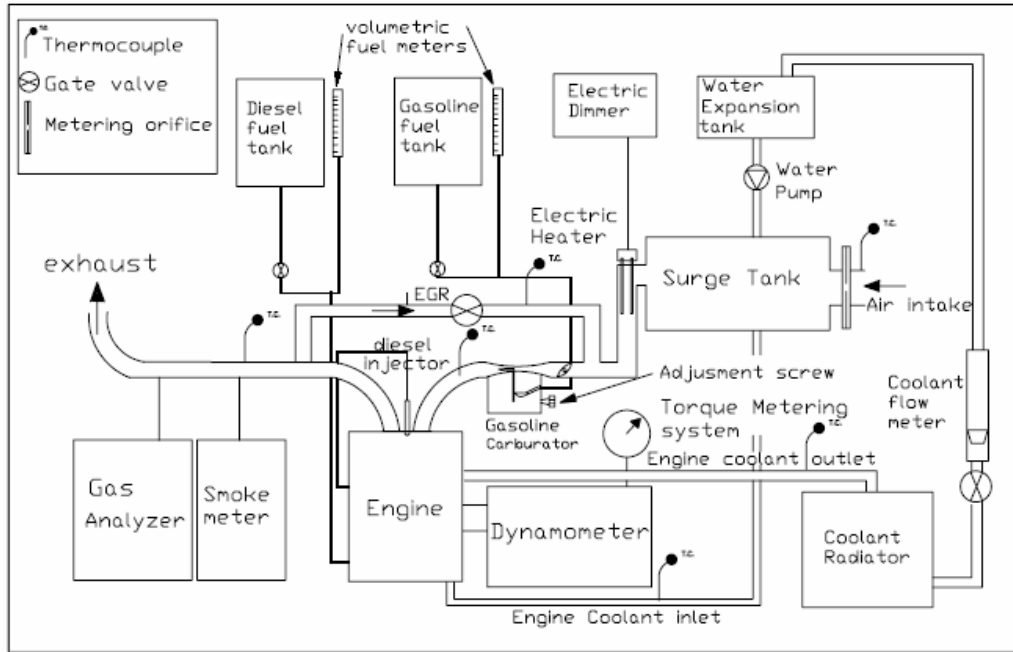
Fig. 1 Schematic of test engine

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]:

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

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 ºC 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:

$$\dot{m}_{aEGR} \approx \dot{m}_a - \dot{m}_{EGR} \times \frac{T_{exh}}{T_{in.}} \tag{4}$$

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 115ºC for the test without EGR, and up to 80 ºC 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 ºC, HCCI combustion showed better results at 50 ºC. Therefore, coolant temperature was maintained 50 ºC 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, engine speed 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 (ºC)	110-115
Coolant temperature (ºC)	50
EGR rate (based on mass flow rate of intake air)	0-15%
Injection timing	35 BTDC
Premixed ratio (r_p)	0-1

139

TABLE III
FUEL SPECIFICATIONS [25]

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 ºC)	305	270
Density (kg/m^3)	720	780
(A/F)$_s$	14.6	14.5

V. RESULTS AND DISCUSSIONS

A. Effect of Engine Speed Variations on CO Emissions

Fig. 2 shows the effect of engine speed variations on CO emissions of dual fuel HCCI engine.
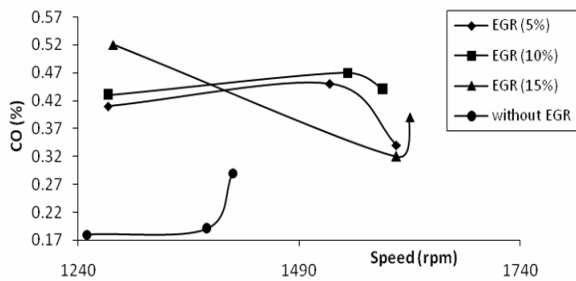

Figure 2. Effect of engine speed variations on CO emission of dual fuel HCCI-DI engine

As shown in Fig. 2, at a constant EGR rate, engine speed variation has a slight effect on CO emission.

Fig. 2 also indicates that CO emission increases with increasing EGR. Increasing EGR dilutes the intake charge and reduces the amount of oxygen. Dilution also decreases combustion temperature and leads to incomplete combustion and therefore increases CO emission.

B. Effect of EGR on CO Emission at Different Engine Speeds

Fig. 3 shows the effect of EGR on CO emission at different engine speeds. As it can be seen, increasing EGR, which causes dilution of intake charge, and insufficient oxygen in intake charge, leads to lower combustion temperature and therefore increases CO emission. It can be noted that increasing engine speed with high EGR rates leads to reduction of CO emission, while with no EGR rates this trend is reverse. This is due to effect of more radicals in higher EGR rates which create different combustion chemical equilibriums at various engine speeds.
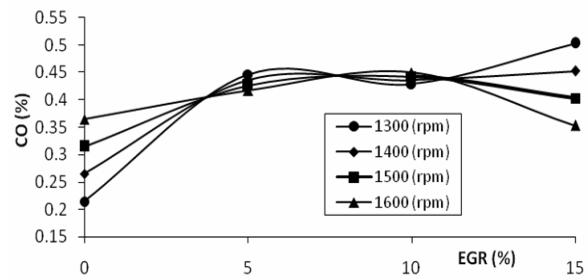

Fig. 3 Effect of EGR on CO emission of dual fuel HCCI-DI engine at different engine speeds

C. Effect of Engine Speed Variations on HC Emission

Fig. 4 demonstrates the effect of engine speed on HC emission of dual fuel HCCI engine. At higher engine speeds the insufficient time for formation of homogeneous mixture leads to incomplete HCCI combustion causes more HC emission.
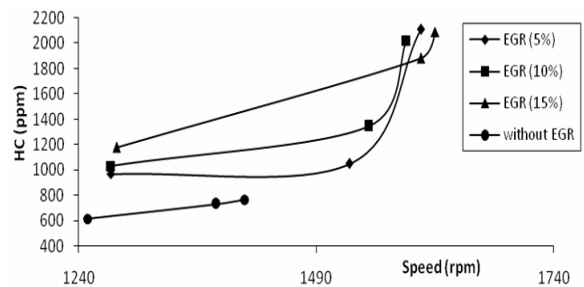

Fig. 4 Effect of engine speed variations on HC emissions of dual fuel HCCI-DI engine

D. Effect of EGR on HC Emission at Different Engine Speeds

Fig. 5 shows the effect of EGR on HC emission of HCCI engine at different engine speeds
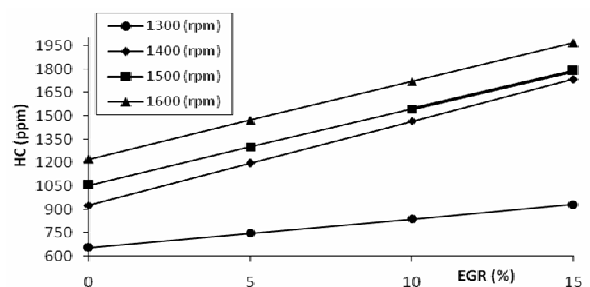

Fig. 5 Effect of EGR on HC emissions of dual fuel HCCI-DI engine at different engine speeds

Increasing EGR rate dilutes the intake charge and reduces its oxygen. Dilution also decreases combustion temperature, which results in reduction of the amount of burnt fuel thus HC emission increases. As mentioned earlier, with increasing

engine speed due to insufficient time for formation of homogeneous mixture HC emission increases.

VI. CONCLUSION

In this study, the effects of EGR rate and engine speed variation 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, which was noticed in all tests, was fast and easy transition to HCCI mode.
2) Increasing EGR dilutes the intake charge and reduces the amount of oxygen. Dilution also decreases combustion temperature and leads to incomplete HCCI combustion and therefore increases CO emission.
3) The insufficient time for formation of homogeneous charge mixture, caused by increasing engine speed, results in increase of CO emissions.
4) High engine speeds in HCCI mode results in insufficient time for formation of homogeneous mixture cause more HC emission due to incomplete HCCI combustion.
5) Increasing EGR rate dilutes the intake charge and reduces its oxygen. Dilution also decreases combustion temperature, which results in reduction of the amount of burnt fuel thus HC emission increases in comparison with no EGR.

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