INFLUENCE OF FUEL TYPE AND INTAKE AIR PROPERTIES ON COMBUSTION CHARACTERISTICS OF HCCI ENGINE

Kastytis Laurinaitis, Stasys Slavinskas
Aleksandras Stulginskis University, Lithuania
kastytis.laurinaitis@asu.lt, stasys.slavinskas@asu.lt

Abstract. The paper presents the analysis of the combustion characteristics of homogeneous charge compression ignition (HCCI) combustion using diesel fuel, jet fuel, gasoline and ethanol. The experiments were conducted on a modified four-stroke, four-cylinder engine at constant engine speed of 1400 rpm. The in-cylinder pressure, heat release analysis and exhaust emission measurements were employed for combustion diagnostics. The effect of the intake air temperature, relative air/fuel ratio and exhaust gas recirculation (EGR) rate on the combustion parameters and emission were analyzed. The experimental results indicate that the intake-air temperature, air-fuel ratio and EGR rate have significant effect on the maximum in-cylinder pressure, heat release rate, start of combustion and emissions. The engine in HCCI mode operates more stable when fuels with a lower cetane number (gasoline and ethanol) were used. When working at lower intake air temperature homogeneities of the air fuel mixture decreased especially when heavier fuels were used. The combustion of ethanol-air mixtures at $\lambda \leq 3.33$ started 3 CAD (Crank Angle Degree) before TDC when the air temperature was 120 °C. When the intake air temperature was increased to 130 ºC the engine ran stably until $\lambda = 4.4$. The exhaust gas recirculation allows the extension of air-fuel mixture composition limits towards higher fat mixtures without knocking combustion. The results show that for all stable operation points NOx emissions were lower than 12 ppm, however HC and CO emissions are higher.

Keywords: HCCI, emission, intake air temperature, relative air/fuel ratio, exhaust gas recirculation.

Introduction

Year by year the regulation of emissions on current vehicles is becoming more and more stringent. Traditional compression (CI) and spark (SI) ignition engines are no longer meeting the strict emission regulations. Therefore, a new engine combustion concept is very attractive now, i.e., homogeneous charge compression ignition (HCCI).

HCCI is a process, when the air-fuel mixture auto-ignites as a result of the temperature increase in the compression stroke. The air and fuel are mixed outside the cylinder. The HCCI and SI engines are similar in the sense that both of them use the premixed charge. HCCI and CI engines are similar because in both cases combustion starts after the auto-ignition process. But in practice it is very difficult to use this HCCI process. To use the HCCI engine the following questions have to be solved: how to control the ignition timing and combustion velocity by changing the speed and load of the engine; how to control the engine running under high loads; starting of cold engine; minimize hydrocarbon and carbon monoxide emissions.

It is widely accepted that HCCI combustion was dominated by chemical kinetics [1]. HCCI engines have been shown to operate with a range of fuels, i.e., gasoline, natural gas, diesel fuel, ethanol, dimethyl ether, propane and butane.

In HCCI combustion, most hydrocarbon fuels exhibit two stage ignitions: low temperature reaction (LRT) and high temperature reaction (HTR). The timing and heat release during the LTR significantly influence the HTR [2].

There are several possibilities to control the HCCI combustion: mixture control, temperature control, fuel modification (blending and additive) and others. The experimental studies showed that for the same partial equivalence ratio of n-heptane, both the low temperature reaction and high temperature reaction are delayed due to the methyl tert-butyl ether addition [2].

The test results with neat n-heptane and 10 – 50 % ethanol/n-heptane blend fuels show that by the introduction of ethanol in n-heptane, the maximum indicated mean effective pressure can be expanded from 3.38 bar of neat n-heptane to 5.1 bar [3]. The indicated thermal efficiency can also be increased up to 50 % at large engine loads. But the thermal efficiency deteriorated at light engine load. Due to the much higher octane number of ethanol, the cool-flame reaction delays, the initial temperature corresponding to the cool-flame reactions increases. The peak value of the low-temperature heat release decreases with increase of the ethanol addition in the blend fuels. The combustion duration whit 10 % ethanol/n-heptane blends is very short due to the early ignition timing. For 20-50 %

ethanol/n-heptane blend fuels, the ignition timing is gradually delayed to the top dead center (TDC) by ethanol addition.

The experimental investigation on the effect of the intake air temperature and air-ethanol ratio on HCCI combustion shows that at 120 ºC the engine could be operated in HCCI mode with air-fuel mixture as λ = 2-4 [4]. Mixture leaner than λ = 4 could not be used for successful combustion due to appearing misfiring cycles. The operating window for air-fuel mixtures for HCCI was λ = 2.5-5.0 when the temperature of the intake air was 140 ºC. When the intake air temperature is raised to 160 ºC, the engine can run at λ = 3.0-5.5. Increasing the intake air temperature it becomes possible to ignite leaner mixtures. The richer mixture tends to knock.

The experimental results at different engine operating conditions at constant engine speed of 1500 rpm with ethanol as fuel are presented in [5]. The intake air temperature was changed in the range 120-150 ºC. Stable ethanol-air mixture combustion at intake air temperature 120 ºC was in the range λ = 2.0-3.5. A higher intake air temperature, richer fuel-air mixture has advanced ignition timing and the rate of pressure rise was also very high, which led to knocking combustion.

Low temperature combustion suggests higher emissions of unburned hydrocarbons. Lower combustion temperatures prevent NO_x formation. But at times this becomes too low to fully oxidize the fuel. The trend shows that unburned hydrocarbon emissions increase when the mixture becomes leaner [5].

The purpose of the research was to investigate the effect of the intake air temperature, exhaust gas recirculation and the relative air/fuel ratio of a homogeneous air-fuel mixture on HCCI combustion parameters.

Materials and methods

The tests have been conducted on a four-stroke, four-cylinder, direct-injection, naturally aspirated diesel engine D-243. One of the cylinders of the engine was modified to operate in HCCI mode. The other cylinders operated in the traditional diesel engine mode. Fuel was injected by using the electromagnetic fuel injector mounted in the intake manifold. The fuel injector was controlled by NI cRIO 9022 system with cRIO 9114 platform and DRIVVEN PFI Rev G module. The intake air was heated by using 7.7 kW heater and glow plugs. The coolant outlet temperature was kept at the level of 85 ºC. The engine speed was fixed at 1400 rpm. The engine torque was measured with AC stand dynamometer KS-56-4 with an accuracy of ± 1 Nm.

The in-cylinder pressure was measured by using a piezo-electric pressure transducer GU24D (AVL) mounted into the first cylinder head and connected to the MICROIFEM amplifier. To measure the crank angle position a precision AVL crank angle encoder 365C (± 0.1º) was coupled with the engine crankshaft. The in-cylinder pressure was recorded for 100 cycles at 0.1 crank angle degrees resolution by using AVL indication and data acquisition system and averaged to calculate the rate of heat release and other combustion parameters. The cylinder pressure history data acquisition and combustion analysis was performed by using AVL IndiCom Mobil program.

The amount of nitrogen oxides NO_x, carbon monoxide CO and total unburned hydrocarbons HC in the exhaust were measured with the Testo 350 XL gas analyzer. The exhaust opacity was measured with the Bosch RTT 100 RTT opacity-meter with an accuracy of ± 0.1 %.

The experiments were conducted at constant engine speed of 1400 min^{-1} and the intake air temperature varying from 80 to 13 ºC at different air-fuel and exhaust gas recirculation ratios with gasoline-air, diesel-air, jet-air and ethanol-air mixtures.

Results and discussion

Figure 1 shows the in-cylinder pressure and heat release rate curves for diesel and JET fuels HCCI combustion at the inlet air temperature of 80 ºC. As it can be seen, both fuels have the two-stage auto ignition and combustion process. The combustion of the air-diesel and air-jet fuel mixtures starts too early – at 20-18 CAD (Crank Angle Degree) before TDC because the cetane number is high. The heat release rate reaches its peak at 11-13 CAD before TDC. This can be changed by reducing the intake air temperature, but it would be difficult to prepare homogeneous mixture at lower temperature because of low volatility of both fuels. In this case, the fuel does not evaporate completely and a part of the fuel flows on the manifold walls in the form of fuel film into the cylinder. The JET fuel has higher volatility compared to that of diesel, therefore the combustion process of the air-jet fuel mixture goes faster.

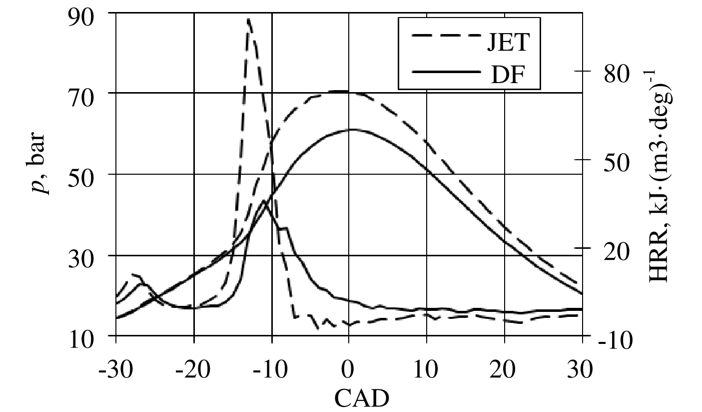

Fig. 1. **In-cylinder pressure and the heat release rate (HRR) for diesel and jet fuels**

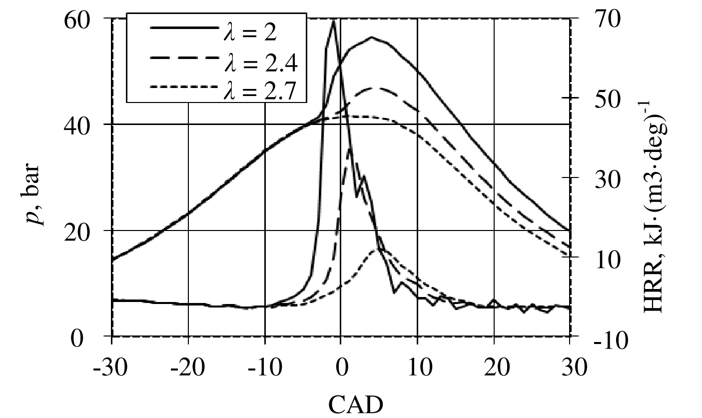

Fig. 2. **In-cylinder pressure and the heat release rate (HRR) for gasoline at different relative air/fuel ratio λ**

A more stable combustion process was obtained by using gasoline-air mixture (Fig. 2). The intake air temperature has to be increased up to 100 ºC because the cetane number is low. This alleviated preparation of homogeneous fuel – air mixture. It can be seen that the gasoline-air mixture has one stage auto-ignition (high temperature reaction). At λ = 2.7 combustion of the gasoline-air mixture starts 2.4 CADs before TDC and the peak of the heat release is 12.0 kJ·(m^3·deg)$^{-1}$. The gasoline-air mixture combustion starts earlier and the ratio of the heat release (ROHR) is higher when λ decreases. At λ = 2.4 and 2.0 the combustion starts 3.8 and 6.2 CAD before TDC and ROHR peak increases to 37.1 and 69.5 kJ·(m^3·deg)$^{-1}$, correspondently.

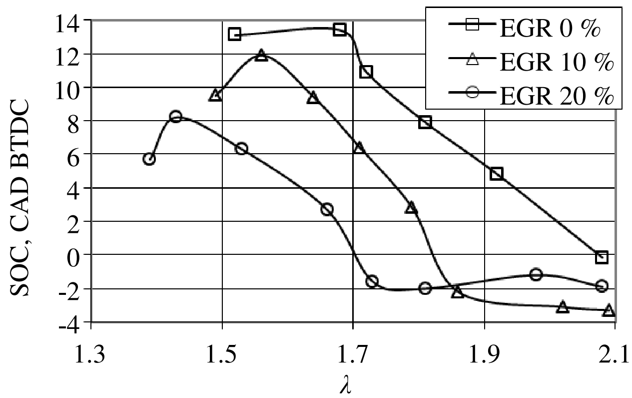

Fig. 3. **Influence of the EGR rate on the start of combustion (SOC) for gasoline**

The start of combustion can be controlled by using the exhaust gas recirculation (EGR). The auto-ignition of air-fuel mixtures retards when using EGR (Fig. 3). For example, without EGR the start of the combustion of $\lambda = 1.9$ air-gasoline mixture occurs 5.2 CAD before TDC. When running at 10 % EGR, the air- fuel mixture of $\lambda = 1.78$ autoignites at the same 5.2 CAD. Furthermore, when using higher EGR of 20 % the much richer air and fuel mixture of 1.58 autoignites at the same crank angle. This indicates that the exhaust gas recirculation allows the extension of air-fuel mixtures composition limits towards higher fat mixtures without knocking combustion.

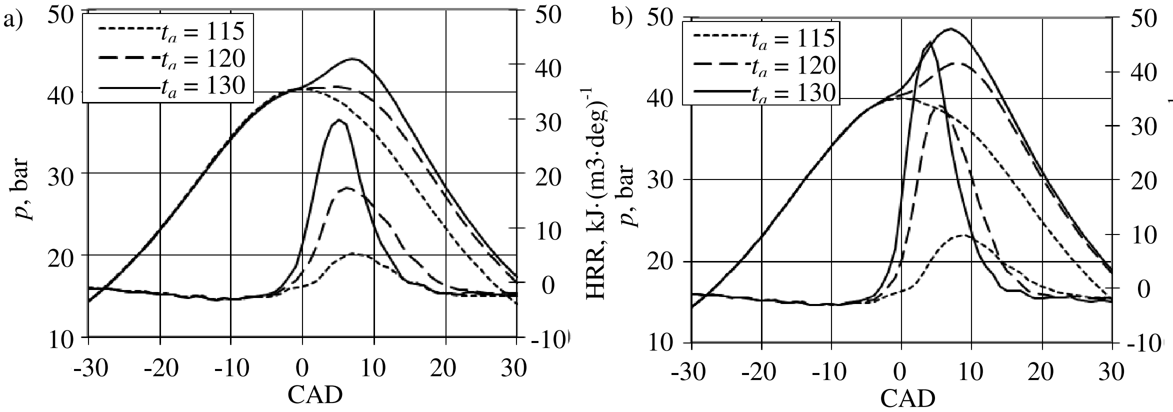

Fig. 4. **Influence of the intake air temperature (t_a) on the in-cylinder pressure and heat release rate of air - ethanol mixture:** a – $\lambda = 2.6$, b – $\lambda = 1.9$

Ethanol has high resistance to auto-ignition and very big latent heat of vaporization. When using this type of fuel the intake air must be preheated to temperature of 115-130 ºC. The combustion process starts after TDC when the intake air temperature is 115 ºC for all relative air/fuel rations tested (Fig. 4 and 5). In this case, the combustion takes place during the expansion stroke and the in-cylinder pressure increases smoothly.

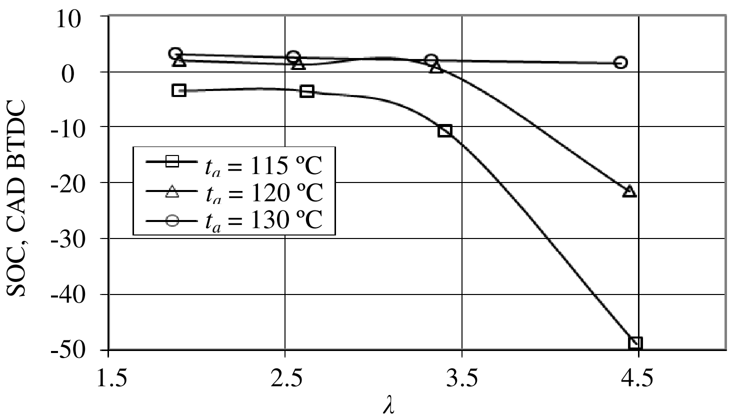

Fig. 5. **Influence of the relative air/fuel ratio and the intake air temperature on the start of combustion (SOC) for air-ethanol mixture**

When the intake air temperature was 120 ºC and λ less than 3.36, the auto-ignition time was about 3 CAD before TDC. At relative air/fuel ratio $\lambda > 3.36$ the combustions starts late and occurs after TDC. At 130 ºC intake air temperature the combustion of the ethanol-air mixture was stable up to $\lambda = 4.4$. The temperature increase does not have significant influence on the auto-ignition moment, but the heat release rate increases. When the intake air temperature was increased from 120 ºC to 130 ºC the peak of the heat release rate increased by about 70 %. At $\lambda < 1.9$ and at all intake air temperatures tested the knock combustion occurred.

The dependencies of the exhaust emissions NO_x, CO and HC produced from ethanol fuel are shown in Fig. 6. The NO_x formation is highly dependent on the intake air temperature. According to the Zeldovic mechanism the production of NO_x increases very fast when the combustion temperature is higher than 1800 K. Because the HCCI combustion process occurs at low temperature the amount of NO_x emissions decline. When the combustion process is stable the NO_x emission increases only up to 12 ppm.

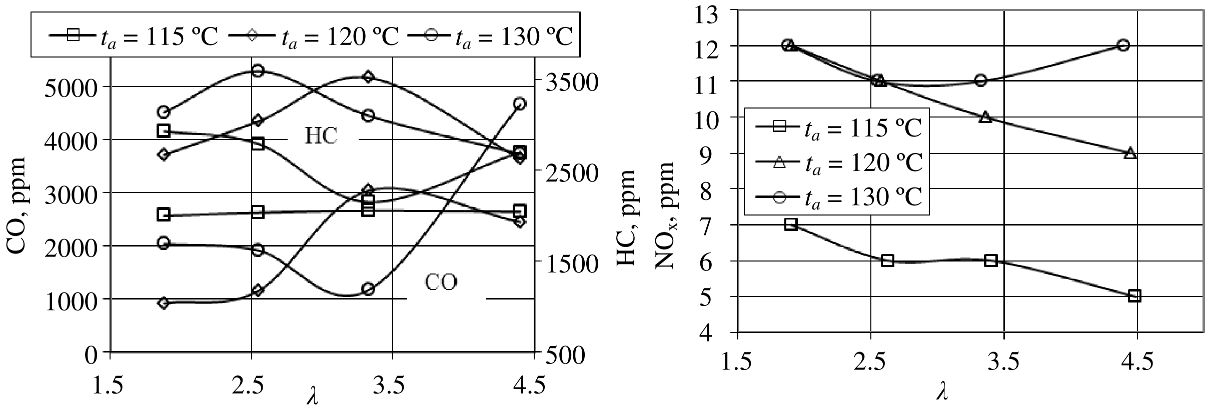

Fig. 6. **Relative air/fuel ratio impact on NO_x, CO and HC emissions**

The CO emission increases when λ increases to 3.36 at the intake air temperature of 120 ºC. After this λ point the CO emission starts to decrease. The auto-ignition of ethanol fuel at λ = 3.36 starts late, i.e., after TDC (Fig. 6). The HC emission changes in the same way. When the intake air temperature is 130 ºC and the relative air/fuel ratio increases from λ = 1.9 to λ = 3.36 the CO emission goes down; when λ > 3.36 the emission increases. The CO and HC emission change is opposite to that of NO_x. The lowest CO concentration reached during the study was 1000 ppm, and the lowest HC emission was 1200 ppm. Smoke opacity of the exhaust was lower than 3.5 % in all test modes.

Conclusions

1. The engine in HCCI mode operates more stable when using fuels with a lower cetane number: gasoline and ethanol. When working at lower intake air temperature the homogeneities of the air fuel mixture decrease especially when heavier fuels are used.
2. The combustion of ethanol – air mixtures at $\lambda \leq 3.33$ starts 3 CAD before TDC when the air temperature is 120 ºC. When the intake air temperature was increased to 130 ºC the engine ran stably until $\lambda = 4.4$.
3. The exhaust gas recirculation allows the extension of air-fuel mixtures composition limits towards more rich mixtures without knocking combustion.
4. When using air-ethanol mixtures during the HCCI operating mode the NO_x emission did not exceed 12 ppm. The smoke opacity was <3.5 %. Both, CO and HC emissions exceeded 1200 ppm level.

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