# INFLUENCE OF INTAKE AIR TEMPERATURE AND EXHAUST GAS RECIRCULATION ON HCCI COMBUSTION PROCESS USING BIOETHANOL

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**Abstract.** The combustion process and exhaust gas emission inside an internal combustion engine are adamantly dependent on the composition of the mixture and method of preparation. Better energy use results are obtained when the air-fuel mixture ignites spontaneously by increasing the temperature inside the cylinder. But in HCCI (homogeneous charge compression ignition) engines, the mixture is prepared outside the cylinder, and it is hard to control the process. The paper presents the thermodynamic characteristics of homogeneous air-bioethanol mixtures on HCCI engines performance. The experiment was carried out by changing the intake air temperature, exhaust gas recirculation and mixture composition. When the intake air was preheated up to 100 ºC, and the relative air-fuel ratio (λ) was 1.9, the air-bioethanol mixture started burning earlier. When the relative air-fuel ratio was higher or lower, the combustion started later. Stable burning was fixed in the range of $\lambda$ = 2.18-4.28, when the intake air temperature was 120 °C. The heat release rate and cylinder pressure were proportional to the temperature of air-bioethanol mixture. The heat release rate was 107.3 and 201 $kJ\cdot(m^3\ CAD)^{-1}$ (CAD – crank angle degrees), and the cylinder pressure was 53.3 and 60.9 bar at a mixture temperature of 100 and 120 °C, respectively. The combustion temperature on HCCI engines was low due to small quantities of nitrogen oxides. During this experiment, the amount of $NO_x$ (nitrogen oxides) of many measurement points was not higher than 10 ppm. But with richer mixtures, the amount of $NO_x$ increased up to 74 ppm. The lowest quantity of CO (carbon monoxide) and HC (hydrocarbon) realized was 0.05 % and 0.06 %.

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

## Introduction

The main problem with the HCCI is that the ignition is completely controlled by chemical kinetics, and is therefore affected by the fuel composition, equivalence ratio, and thermodynamic state of the mixture. Homogeneous charge compression ignition (HCCI) is a new combustion mode, which has the advantages of higher thermal efficiency and lower emissions of $NO_x$ and particulate matter (PM). But on this same combustion process, the amount of CO and HC increases because the combustion temperatures decrease. As λ increases, the peak pressure in the cylinder and the maximum pressure rising rate decrease, also the crank angles corresponding to them both delay. This is because the larger λ is, the less energy in unit volume there is for mixture [1].

Indian scientists said that increasing of intake air temperature gives a chance to ignite leaner mixtures in HCCI mode. But relatively richer mixture tends to knock. It was found that at lower intake air temperature it is possible to ignite the richer mixture $\lambda > 2$ in HCCI combustion mode [2]. As the intake air temperature increases, the engine running on richer mixture tends to knock with very high rate of pressure rise. But at higher intake air temperature it is possible to ignite the leaner mixture $\lambda < 5.5$ in HCCI combustion mode [2].

Stable HCCI combustion for ethanol at the intake air temperature of 120-150 ºC and constant engine speed of 1500 rpm, is achieved in range $\lambda$ = 2-5. The maximum IMEP (indicated mean effective pressure) obtained during the experiment was 4.3 bars. The maximum indicated thermal efficiency was found 44.78 % at relative air–fuel ratio of 2.5 for intake air temperature of 120 ºC. Extremely low $NO_x$ (<10 ppm) was emitted from all stable HCCI operating conditions. The HC and CO emissions are however higher. HC emissions decrease and CO emissions increase with an increase in relative air–fuel ratio [3].

Increasing EGR (exhaust gas recirculation) dilutes decreases the combustion temperature and leads to incomplete HCCI combustion and, therefore, increases CO emission. Scientist Ghazikhani and his team said that increasing EGR rate dilutes the intake charge and reduces its oxygen. Dilution also decreases the combustion temperature, which results in a reduction of the amount of burnt fuel thus HC emission increases in comparison with no EGR [4].

Scientists from the United Kingdom investigated that the thermal effect of the trapped EGR is due to its high temperature. After mixing with cool air/fuel mixture, the hot EGR improves the temperature of the entire engine inlet charge, therefore, increases the cylinder temperature uprising during the compression process. Similar to the technique to generate HCCI combustion by increasing the inlet temperature, hot EGR can initiate HCCI combustion and advances ignition timing if more EGR is introduced. An assumed EGR at a fixed temperature of 800 K was used to calculate its thermal effect. The trend of the calculated results agrees with the experimental results [5].

During the re-compression stroke, the heat release curves indicate more heat being released from directly injected alcohol fuels. However, no chemiluminescence was detected for alcohol fuels. The effect of spark discharge is prominently shown in the gasoline fuel but not much effect in alcohol fuels. This is because the CAI (controlled autoignition) combustion of alcohol fuels started earlier due to increased charge temperature. With spark-assisted ignition, flame propagation dominates the combustion at the early period followed by spontaneous autoignition for CAI combustion. Because of the high dilution rate, the flame propagates at a slow speed and accounts for a very small portion of the total heat released. Methanol shows fastest and earliest auto-ignition combustion, followed by ethanol while gasoline is the slowest and least combustion. Charge temperature during compression is the key factor for this kind of engine. Due to limited molecule and reaction pathway, oxygenated fuels, particularly methanol, present lower image intensity for all of the emissions compared with gasoline [6].

The HCCI combustion process fueled by DME (dimethyl ether) and methanol can be controlled by adjusting the EGR rate. EGR cannot extend the maximum IMEP of HCCI operation range fueled by DME and methanol [7].

## Materials and methods

The tests have been conducted on naturally aspirated, four-cylinder, four-stroke, direct injection, 4.75 l displacements, compression rate 16 diesel engine D-243. One of the cylinders of the engine was modified to operate in HCCI mode. The other three cylinders operated in traditional diesel engine mode. Bioethanol was injected by using the electromagnetic fuel injector mounted in the intake manifold. By NI cRIO 9022 system with cRIO 9114 platform and DRIVVEN PFI Rev G module the fuel injector – the amount of bioethanol was controlled. The intake air was heated by using glow plugs and 7.7 kW air heaters. 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 piezo-electric pressure transducer GU24D (AVL) and connected to the MICROIFEM amplifier. To measure the crank angle position a precision AVL crank angle encoder 365C (±0.1$^{o}$) was coupled with the engine crankshaft. The in-cylinder pressure was recorded for 100 cycles at 0.1 crank angle degrees resolution by using the 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 were performed by using AVL IndiCom Mobil program. The exhaust gas recirculation was adjustable by using the mechanical valve installed on the exhaust pipe.

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

Experiments were conducted at the intake air temperature varying from 80 to 130 ºC at different air-fuel and exhaust gas recirculation ratios with ethanol – air mixtures.

## Results and discussion

The intake air temperature and the composition of the air-bioethanol mixture have influenced the self-ignition moment. When the intake air temperature was 100 ºC and $\lambda$ = 1.9, the ignition of air-bioethanol mixture was the earliest (Fig. 1). By increasing or decreasing of the relative air-fuel ratio, the combustion began later. The combustion started directly on the TDC (top dead center), when the $\lambda$ = 2.4 and the intake air temperature was 100 ºC. Further increases of $\lambda$ resulted, that the combustion began later, after the TDC. By increasing the amount on the fuel in the mixture, the combustion started before TDC and cylinder knocking was observed.

By increasing the intake air temperature up to 120 ºC the stable combustion process was possible with thinner air-bioethanol mixture. The mixture with relative air-fuel ratio of 2.2 began to burn at first. On the same intake air temperature, the combustion commenced on TDC with $\lambda$ = 2.8. But the knocking combustion for higher intake air temperature started rather at $\lambda$ = 2.1.

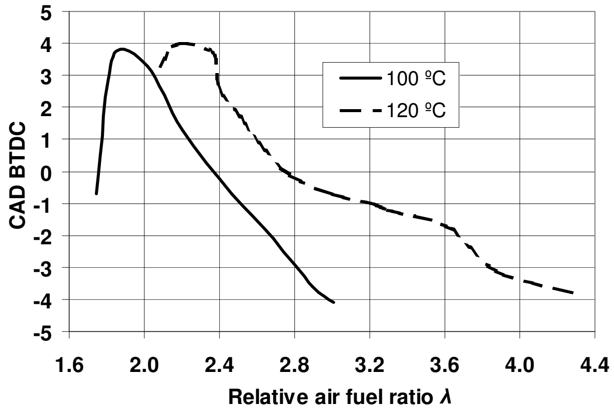



Fig. 1. **Dependencies of bioethanol auto ignition with regard to TDC on air-fuel equivalence ratio and mixture temperature**

The start of combustion was possible to be controlled by using the exhaust gas recirculation (EGR). With 20-30 % exhaust gases in the mixture, the combustion process began earlier than the combustion of air-bioethanol mixture without EGR (Fig. 2). The air-bioethanol mixture began to burn 6 CAD before TDC by using 20 % of EGR. For different EGR values, the combustion process starts later compared to TDC.

With EGR values ranging between 40 – 50 %, combustion was started later. When $\lambda = 3$ and EGR = 0 %, combustion began at 0.7 CAD before TDC. By EGR = 10 % combustion began 1 CAD before TDC, by EGR = 20 % 2 CAD before TDC, by EGR = 30 % 1.9 CAD before TDC, by EGR = 40 % 0.8 CAD before TDC and by EGR = 50 % combustion started 1 CAD after TDC by the same relative air-fuel ratio.

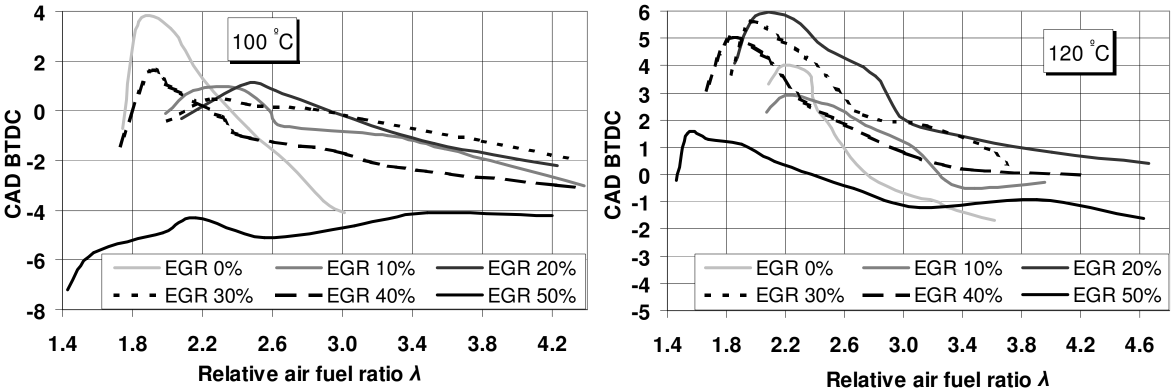



Fig. 2. **EGR influence on auto ignition**

By increasing the amount of exhaust gases in the air-fuel mixture, the knock combustion went for richer air-fuel mixtures. The knocking combustion was heard when the relative air-fuel ratio was less than 2.1, intake air temperature of 120 ºC without EGR. By EGR = 30 % the knock combustion was heard at $\lambda < 1.8$ and EGR = 50 % – $\lambda < 1.5$ at the same intake air temperature of 120 ºC.

The intake air temperature affected the heat release rate (HRR) and gas pressure on the cylinder. The higher air intake temperature resulted in higher heat release ratio and cylinder pressure at the same relative air-fuel ratio (Fig. 3). By the intake air temperature of 100 and 120 ºC the heat release ratio was 107.3 and 201 $kJ\cdot(m^3\,CAD)^{-1}$. The maximum in cylinder pressure was 53.3 and 60.9 bar respectively.

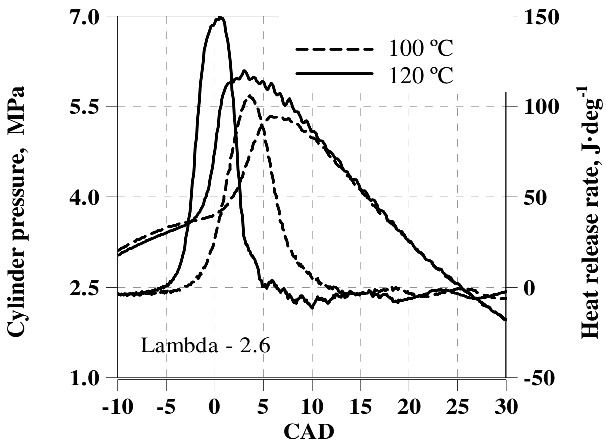



Fig. 3. **Dependencies of the heat release rates and cylinder pressure with regard to TDC on mixture temperature**

By higher intake air temperature combustion began previously and resulting that the maximum of the heat release rate and cylinder pressure occurred rather versus the crank angle. By increasing the intake air temperature of 20 ºC led that the heat release rate has advanced 4 CAD and cylinder pressure has advanced 3 CAD.

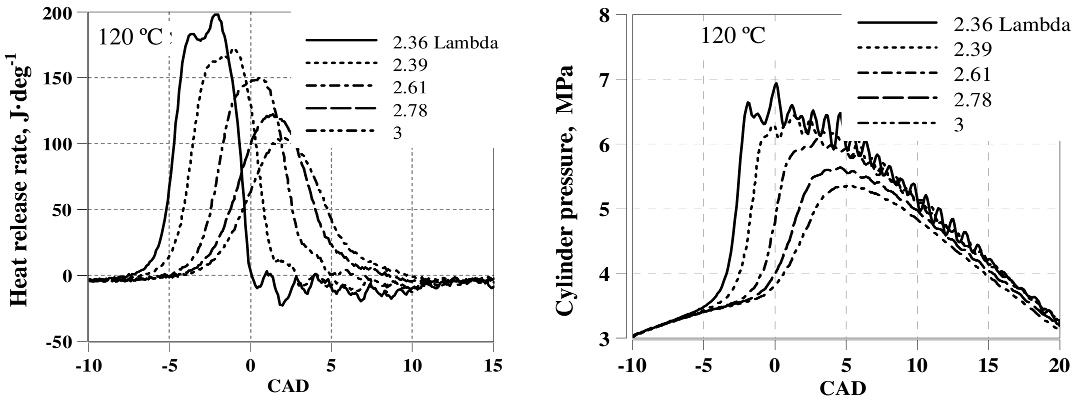



Fig. 4. **Influence of equivalence air-bioethanol ratio on the heat release ratio and cylinder pressure**

The heat release ratio and cylinder pressure were dependent on the amount of bioethanol in the mixture (Fig. 4). Whit the slimmer mixture the maximum of heat release ratio and cylinder pressure were less.

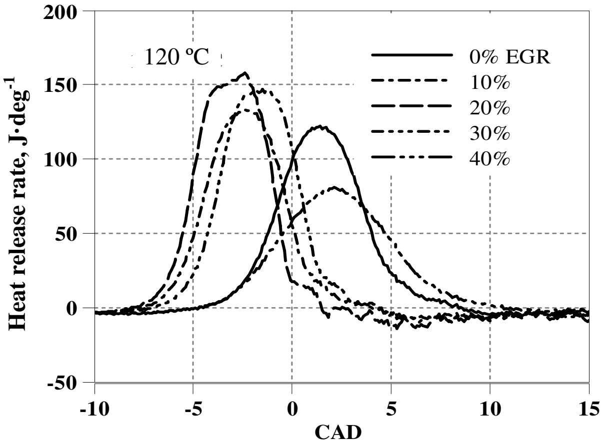



Fig. 5. **Influence of EGR on the heat release rate**

The heat release ratio was proportional to the amount of the exhaust gases in air-fuel mixtures (Fig. 5). HRR increase of 5 %, when 10 % of EGR was supplied. By increasing EGR up to 40 %, the HRR increase of 32 %. By EGR = 50 %, the HRR was 35 % less.

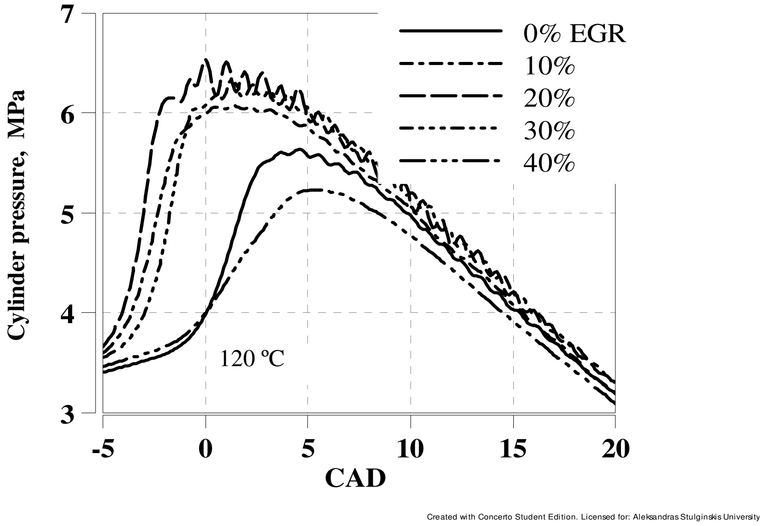



Fig. 6. **Influence of EGR on the cylinder pressure**

The cylinder pressure was at the highest level by using 20-40 % of EGR when the relative air-fuel ratio was constant (Fig. 6).

The building of nitrogen oxides was very dependent on the burning temperature. It formed larger quantities of oxides by higher burning temperature in the cylinder.

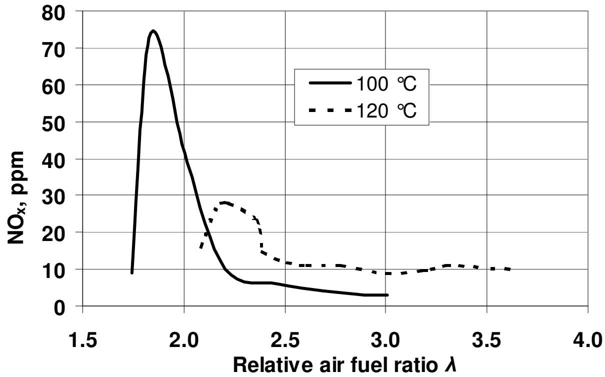
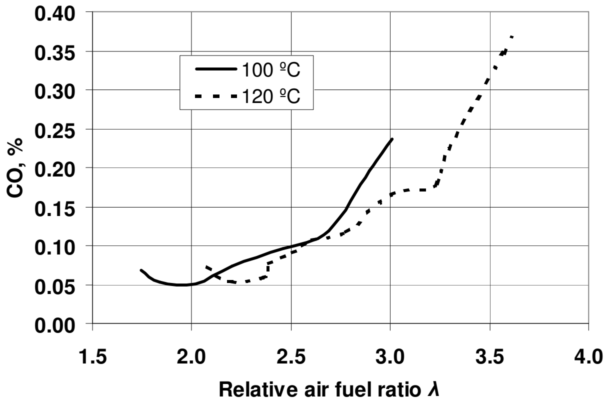



Fig. 7. **Dependencies of $NO_x$ emitted on air-fuel equivalence ratio and mixture temperature**

Of a lot of measurement points the emission of nitrogen oxides was not more than 10 ppm. Only by richer air-fuel mixtures the amount of $NO_x$ was much higher 28 and 74 ppm at the intake air temperature of 120 and 100 ºC (Fig. 7). But it is important to remember, that on these measurement points knock burning took place. And that the gas cylinder temperature was high. On other measurements points, the amount of $NO_x$ was at a low level.

The amount of CO and HC was at a higher level because the combustion temperature was low (Fig. 8). The heat was not enough to burn CO and HC to the end. By higher relative air-fuel ratio the heat on the cylinder was low. It is the point why the CO and HC cannot burn to the end. The intake air temperature had a less important role to the CO and HC emissions. The minimum of CO was 0.05 % and a minimum of HC – 0.06 %.



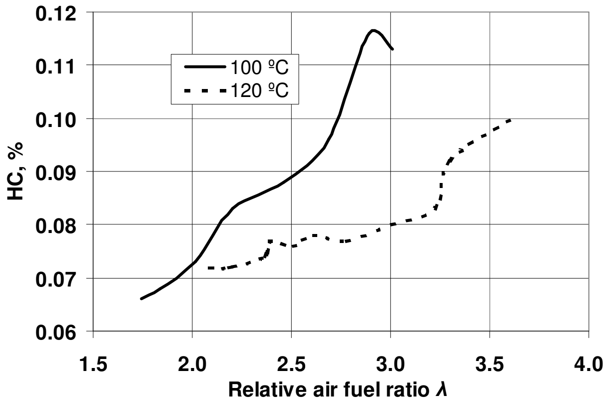




Fig. 8. **Dependencies of CO and HC emitted on air-fuel equivalence ratio and mixture temperature**

The quantity of CO and HC was higher by using of 50 % of EGR. But at the same time, the amount of $NO_x$ strongly dropped. The smoke opacity of the exhaust is not higher than 3.5 % in all test modes.

## Conclusions

1. The higher intake air temperature leads to working with thinner air bioethanol mixture, but the knocking combustion starts previously. The air-bioethanol mixture began to burn 4 CAD before TDC and 3.8 CAD after TDC, when the relative air-fuel ratio was 2.18-4.28 at the intake air temperature at 120 ºC.
2. The engine worked stable with richer air-bioethanol mixtures by using EGR.
3. The highest amount of $NO_x$ was 74 ppm, when the engine worked on HCCI mode fueled with air-bioethanol mixture, and the lowest quantity of CO and HC was 0.05 and 0.06 %.

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