AN EXPERIMENTAL STUDY TO LOWER EMISSION IN CNG HCCI ENGINE USING VARIABLE VALVE TIMINGS WITH NON-LINEAR QUASI-STATIC COMPENSATION SYSTEM FOR TIMING CONTROL

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Abstract

The homogeneous charge compression ignition engine, HCCI, has the potential to combine the best of the spark ignition and compression ignition engines with high octane number fuel. The engine operates with high compression ratio and lean mixtures giving CI engine equivalent fuel consumption or better. Due to premixed charge without rich or stoichiometric zones the production of soot and NOx can be avoided but HC and CO emission are high due to low temperature in combustion zones. HCCI combustion is achieved by controlling the temperature, pressure and composition of the air/fuel mixture so that it auto ignites near top dead centre (TDC) as it is compressed by the piston. Expanding the controlled operation of an HCCI engine over a wide range of speeds and loads is probably the most difficult hurdle facing HCCI engines. Out of this follows that timing/phasing of the combustion is one of the main difficulties with HCCI combustion concepts. This paper deals with control of HCCI by aid of variable valve timing with specialization in usage of a non-linear quasi-static compensation to lower HC and CO emission in CNG HCCI engine.

Keyworks: Valve tining, Non-linear Quasi-static compensation system.

I. INTRODUCTION

During recent years an increased environmental awareness has lead to stricter emission rules for vehicles and thereby vehicular engines.

Table 1. Emission levels for Heavy Duty truck engines

Legislation	From Year	NOx [g/kWh]	Particulates [g/kWh]
Euro I	1992	8.0	0.36
Euro II	1996	7.0	0.25
Euro III	2000	5.0	0.10
Euro IV	2005	3.5	0.02
Euro V	2008	2.0	0.02
EPA 07-10, US	2007 - 2010	0.27	0.013

In combination with higher fuel prices and the growing concern for the large scale burning of fossil fuel causing a global warming, improvement of the Diesel engine has this far, EU I to EU IV Table 1, largely been achieved by engine internal combustion system optimization. In order to meet the upcoming EPA 07-10 legislation and keep the fuel consumption down after treatment to reduce NOx and particulates is expensive and still not generally available on the market. The basis of Homogeneous Charge Compression Ignition (HCCI) engines is their fast and flameless combustion after an auto ignition process of a homogeneous mixture. HCCI combustion achieves high fuel efficiency with low NOx emission due to limited cylinder peak temperature (below 1700 K). The timing of HCCI is determined by mixture conditions, rather than the spark timing for the fuel injection timing controlled auto ignition requires regulation of the mixture properties (temperature, pressure and composition) at the intake valve closing. The variable valve timing (VVT) flexibility can provide control over the mixture conditions at IVC.

II. CONTROL

Timing of Start of Combustion (SOC) of HCCI combustion is for a number of reasons crucial for an HCCI engine. If combustion occurs too early, high pressure rise speed, which is related to combustion noise and high maximum pressure will result. If on the other hand the combustion occurs too late the charge will in the limit misfire and cause large HC - emission and no work output. Figure 1 shows an example of a measured cylinder pressure trace and an estimated cylinder temperature trace for a typical engine cycle.

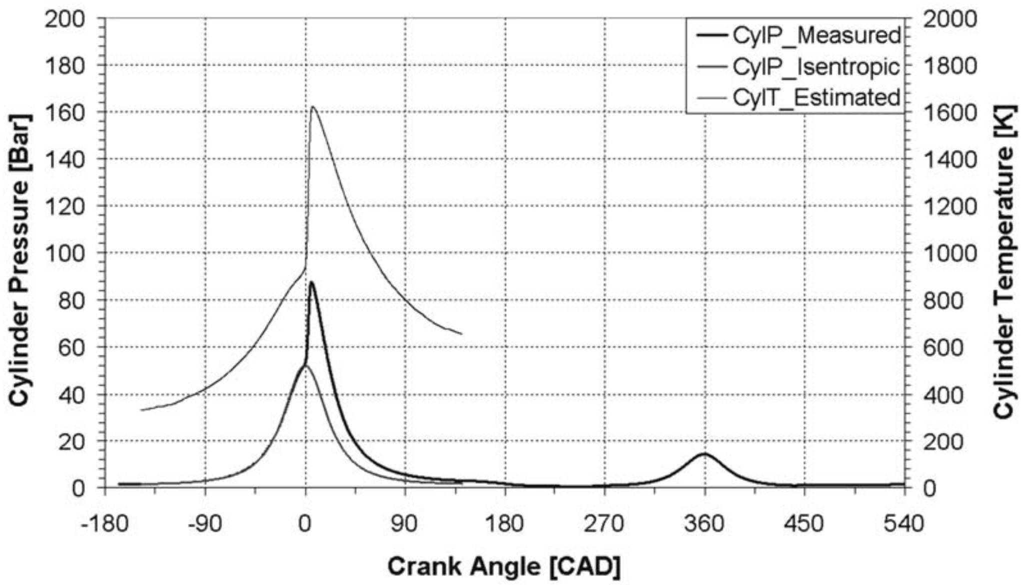

Fig. 1. Measured cylinder pressure and estimated cylinder temperature for the HCCI Engine according to Table operated at 1000 rpm, an inlet pressure of 1.5 bar absolute, inlet temperature of 60°C and a net IMEP of 6.0 bar. Also shown is the estimated isentropic pressure that would have occurred during a motored cycle. The estimated parameters are calculated during the part of the cycle when all engine valves are closed.

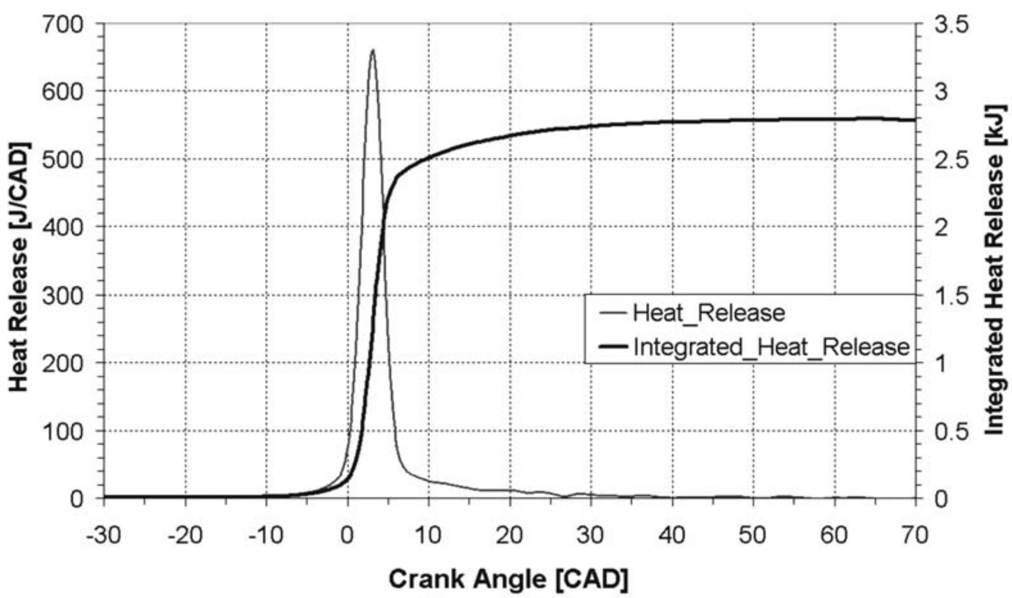

Fig. 2. Calculated heat release based on the measured pressure trace

The figure also shows the estimated isentropic pressure that would have resulted in the absence of combustion. Figure 2 shows the heat release that is occurring in Figure 1 and that cause the rise in pressure above the isentropic pressure. The engine is operated at 1000 rpm resulting in that the whole cycle, two crank shaft revolutions, last for 120 ms. In this case the combustion is occurring close to CTDC.

III. ROUGH SOC MODEL

In some situation a course approach is desirable to explain the basic mechanisms that governs SOC as a chemical kinetics simulation is very complex and computer intensive. In order to get a rough understanding of the ignition mechanism the knock - integral-method described in [1] can be used. The use for HCCI combustion has been proposed in [2], [3] and [4].

The basic in the methods is that the ignition delay time, τ can be estimated through an Arrhenius correlation.

$$\tau = A \times p^{-n} \times e^{B/T} \quad \text{... [1]}$$

A, n and B - positive coefficients

p - Pressure

T - Temperature

If the pressure and temperature varies during the ignition delay, as is the case during the compression event, the time to ignite to can be estimated in accordance with Livengood and *Wu* as

$$t_{IVC} = t_{SOC}$$

$$\int_{t_{IVC}} 1/\tau\,(s)\,ds = 1 \quad \text{.... [2]}$$

τ - Ignition delay

t_{IVC} - Time at inlet valve closure

$t_{IVC} + t_{SOC}$ - corresponds to the time when enough radicals are present to ignite the charge.

As can be seen in equations in 1 and 2 SOC is affected by the pressure and temperature during the compression event. As a result SOC is affected by the boundary condition of the combustion chamber. For instance the pressure and temperature during the compression event is governed by the inlet pressure, inlet temperature, exhaust pressure, exhaust temperature and combustion chamber wall temperature. The engine speed and injection timing in a DI case governs the time during the compression event. Also, the coefficients in equation 1 are not fixed but rather functions of the charge state. The result is that apart from pressure and temperature it can be argued that the fuel load, fuel concentration and the trapped residual mass is affecting the ignition. The above dependencies result in a narrow combustion timing window. This makes combustion timing control over a wide range of engine conditions.

A. Control of Four-Stroke

In [5] a method called thermal management idea was analyzed through simulation. The idea was to use exhaust energy to heat the intake charge. In figure 3 the principle concept is shown.

B. Control through valve timings in detail

In principle there are two ways to affect the combustion timing by variable valve timing. The first way is to increase temperature of the charge here exemplified by what in the following is called the overlap method, and the second way is to affect the effective Compression ration (CR), here called IVC method. The overlap-method is also known under other names such as recompression or negative overlap.

The overlap method has been reported in [6], [7], [8] and [9]. The cause for this behavior is that the residual gas i.e., internal EGR that is trapped from the

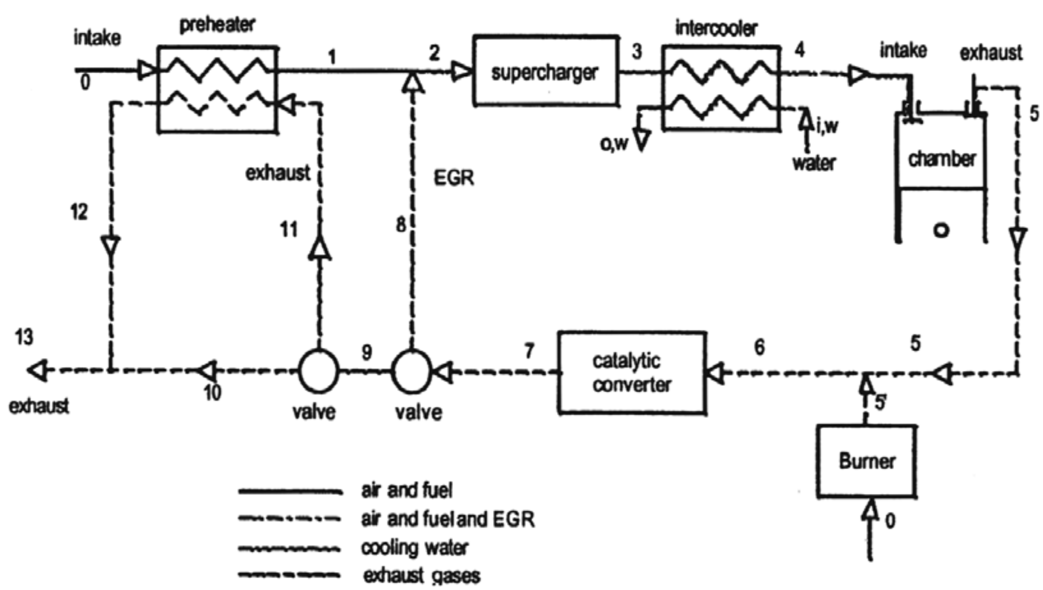

Fig. 3. Schematic of thermal management

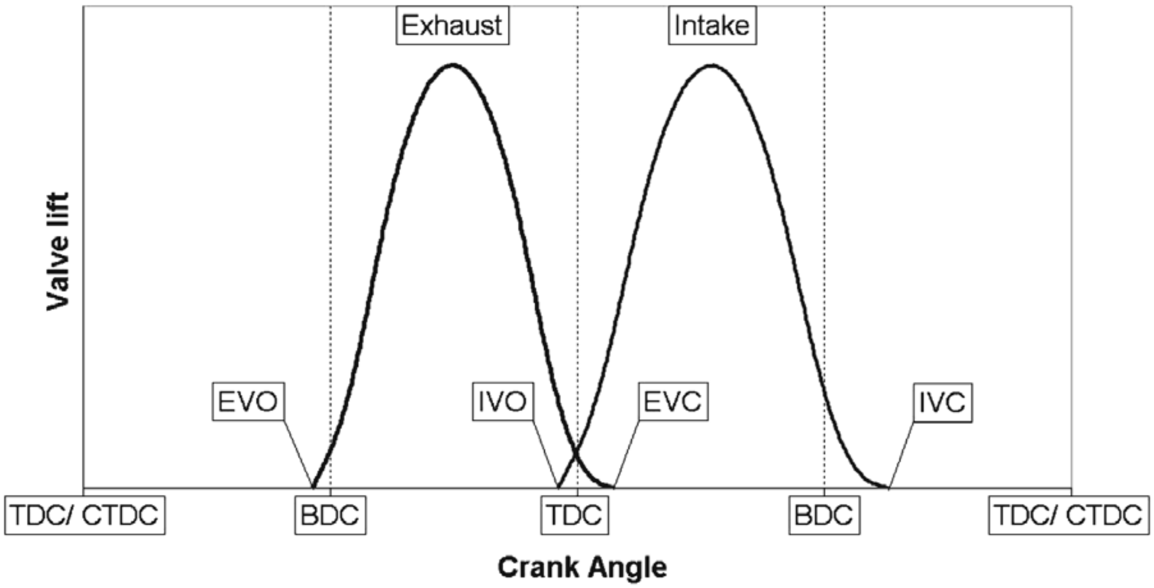

Fig. 4. Principle valve lifts as a function of crank angle

previous cycle is hot and thereby promoting reactions as described by equations 1 and 2.

In [6], [10] and [12] and alternative method to the overlap-method has been investigated. The idea is to use valve timing similar to those exemplified in Figure 4 and rise the temperature of the combustion engine.

By reopening or delayed closing of the exhaust valve. By doing so that exhaust valve is open during a part of the intake stroke and a mixture of fresh charge and hot burnt gases is prepared for the on coming combustion cycle. The main benefit is said to be lower heat transfer losses compared to the overlap-method.

C. Feedback signal

The task to time the combustion correctly requires feedback from the combustion chamber in some way or other. The most direct way of achieving this is by in-cylinder pressure measurements. This method is expensive and not practical to realize in series production at the moment. A drive to make the whole engine control system cylinder pressure based [13], [14] and [15] makes small/cheap in-cylinder pressure transducers forescable in the future. Recently, flow plug based cylinder pressure sensors aimed for series have been made public [16]. The ion-current has been evaluated for HCCI in [17] and [18] and used as feedback for control in [19].

The combustion timing signal displays in general an cycle-to-cycle variation as shown for example in [20] where an analyze of the cycle-to-cycle variation has been conducted for the engine used in this work with data according to Table 2 in auto correlation and spectrum.

Table 2. Engine Data

Swept volume	1.95 dm^3
Compression ratio	14.0 or 18.0
Bore	127 mm
Stroke	154 mm
Connecting rod length	255 mm
Number of valves	4

Analyze of the signal were done showing that the variation could be regarded as white noise. The statistical properties of the white noise were found to vary some with engine operating point.

IV. RESULT AND DISCUSSION

Imaging conducted at increasing crank angles, although from different cycles, permits a combustion development sequence to be built up. Such results are presented in Figure 5, where the engine was again operated at stoichiometric with natural gas as fuel. Intake temperature was held constant at 200°C and the in-cylinder pressure indicated load (IMEP) of 2.4

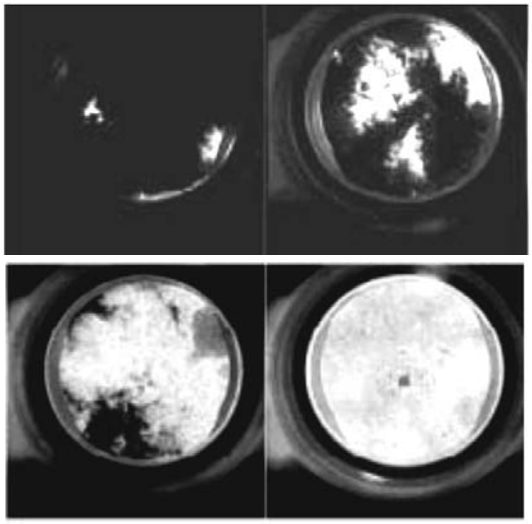

Fig. 5. Combustion development

bar. Intensifier integration period was 167 μs corresponding to 1 °CA.

A clear combustion development is obvious starting with multiple ignition sites and progressing to complete chamber engulfment. In this regard, various methods of image processing were performed; possibly the most informative were calculations of the area of the projected cylinder volume engulfed by combustion. This was performed by a simple threshold technique whereby pixel intensities above a threshold of 1.5% of the maximum grey level were assumed to be representing visible combustion, below that threshold they were not. Such results are presented in Figure 6 (crosses) where they are compared to mass fraction burned (lines with no markers).

In this case the engine was operated at stoichiometric with compressed natural gas or compressed natural gas with hydrogen H_2 addition (hydrogen replaced some of the natural gas to maintain lambda $=1$ in volumetric ratio of 1:10) fuels [21,22] for purpose of H_2 addition. Intake temperature was held constant at 200°C and the in-cylinder pressure indicated load (IMEP) varied from 2.5 bar to 2.9 bar depending on combustion phasing. Intensifier integration period was 167 μs corresponding to 1°CA.

Such an approach provides an opportunity to investigate how individual variables affect the combustion phasing and the rate of combustion development. In this context, Figure 6 shows the effect of H_2 addition on combustion development. Comparing the crank timing corresponding to the 5% and 95%

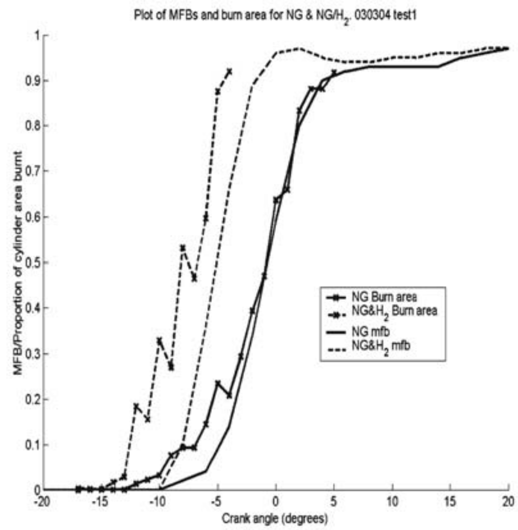

Fig. 6. Comparison of burn area and MFB for CNG HCCI, with and without H_2 addition (Combustion areas are average from 10 cycles, mtb average of 100 cycles)

chamber area engulfed indicates that while the H_2 addition advances the early combustion development (5% area) by only 3°, it significantly accelerates the rate of combustion development as the 95% chamber engulfed can be seen to be approximately 8° advanced. This was not as evident in the fuel mass fraction burned plots where the H_2 addition advanced the timing of 5% mtb by 3° CA and the 95° burn by only 5° CA. Although it should be noted that the part of the combustion chamber near the cylinder wall is not visible due to the limitations of the window size, the uncertainty in estimating the total combustion area is not expected as significant as for spark-ignition engines because of the nature of the multi-point ignition of HCCI.

V. CONCLUSION

HCCI engine cannot work without flexible controls, which is provided by the variable valve timing with specialization in usage of a non-linear quasi-static compensation. The variable valve timing control not only helps engines to meet stringent emission limits but also to achieve better drivability.

Advanced technology in variable valve timing system helps to reduce particulate NOx, HC and Co emissions.

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