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The effect of operating conditions on HCCI exhaust gas temperature

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1. Abstract

To successfully use an exhaust aftertreatment system to overcome high HC and CO emissions in Homogeneous Charge Compression Ignition (HCCI) engines requires high exhaust gas temperatures for oxidation catalysts. Low exhaust gas temperature in certain HCCI conditions is a limiting factor to obtain a large desirable operating range in HCCI engines. This paper investigates the influence of combustion chamber charge conditions on the exhaust gas temperature in a single cylinder experimental engine at over 160 operating points. For the conditions tested, more than half of the collected data exhibits an exhaust gas temperature below 300°C which is below the light-off temperature of typical catalytic converters in the market. Location of ignition timing is found as a main factor to influence HCCI exhaust gas temperature. HCCI combustion occurring immediately after TDC indicates a lower exhaust temperature compared to the HCCI combustion that occurs late after TDC. HCCI ignition timing also influences HC and CO and NOx emissions and advancing ignition towards TDC causes lower HC and CO but higher NOx emissions. In addition, results at a constant load condition indicate longer burn duration and higher fueling rate in Spark Ignition (SI) mode lead to have higher exhaust gas temperature in SI mode comparing to that of HCCI mode.

2. Introduction

Ultra low NOx and negligible PM emissions combined with a modest fuel economy penalty make Homogeneous Charge Compression Ignition (HCCI) a promising alternative to conventional spark/diesel combustion [1]. However, higher HC and CO emission compared to conventional combustion is a major challenge of HCCI engines. These high HC and CO emissions in HCCI are predominantly due to low in-cylinder temperature caused by lean-burn or high-dilution combustion. This can result in incomplete combustion and decrease of post-combustion oxidation rates inside the cylinder [1, 2].

If exhaust gas temperature is high enough for the oxidation process in catalyst to occur, then high HC and CO engine-out emissions in HCCI can be significantly reduced. The oxidation catalysts can reach conversion efficiencies of up to 95% for HC and CO pollutants [3], but only when the catalytic converter is at operating temperature. The light-off temperature (the temperature at which the catalyst becomes more than %50 effective) is about 250 to 300 °C for most of the oxidation catalysts [4, 5]. Methods to reduce light-off time in catalysts are classified into active or passive systems. Active systems such as electrically heated catalysts or fuel burners require extra energy supply and have more complexity and cost [6]. In contrast, passive systems rely mainly on thermal management of the energy from exhaust gases; thus highly depend on engine operating conditions. Since HCCI has typically low exhaust temperature which can be as low as 120°C [7], the HC and CO abatement by oxidation catalysts is an important concern in HCCI engines. This could limit the desirable operating range of HCCI engines. In addition, the low HCCI exhaust temperature provides little energy for turbocharging which is often used to extend HCCI operating range for high loads [8]. For a better modulation of engine charge variables to extend desirable HCCI operating range it is essential to understand the factors influencing HCCI exhaust temperature.

To the authors' knowledge, there is not any major study in the literature dedicated to analyze the exhaust temperature variations in HCCI engines. The variations of HCCI exhaust temperature have been partially investigated [9] where

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cycle to cycle variations of exhaust temperature and its relation with combustion phasing are discussed for a camless gasoline HCCI engine which operates with an early exhaust valve closing strategy. They find that the exhaust temperature at the end of expansion stroke does not affect the HCCI combustion phasing for the next cycle; instead combustion phasing is influenced by the gas temperature at the beginning of the compression stroke. Results in [10] indicate an increase in the exhaust gas temperature when equivalence ratio is increased for a medium-duty Direct Injection (DI) HCCI engine.

The primary aim of this study is to provide discussion of several variables affecting the exhaust temperature in an HCCI engine by using experimental data collected during both steady-state and transient operating conditions.

3. Engine Setup

The experimental single cylinder Ricardo Hydra Mark III engine with a Rover K7 head is depicted in Figure 1 and the configuration of the engine is given in Table 1. Two separate fuel systems with 3-bar fuel pressure are used to ensure injection on closed intake valves. One fuel system is used to inject n-Heptane and the other is used to inject Isooctane. The separate flow rate control of each of these two fuels allows any desired octane number to be obtained. Both n-Heptane and Isooctane injectors are aimed directly at the back of the intake valves. The fresh intake air entering the engine is first passed through a laminar air-flow meter for flow rate measurement. Then, a supercharger driven by a variable speed electric motor adjusts the intake manifold pressure and a 600W electrical band-type heater sets the mixture temperature to a desired value using a closed-loop controller. Finally the exhaust gases exiting the cylinder are sampled for emission analysis. A 5-gas emissions test bench is used to collect emissions data – NO_x is measured with 1 ppm resolution using Horriba CLA-510SS emission analyzer and Horiba FIZ-510 emission analyzer is used to measure HC with 10 ppm resolution. CO is measured with 0.01% resolution using Siemens ULTRAMAT6 emission analyzer.

The engine out Air Fuel Ratio (AFR) value is measured by ECM AFRecorder 1200 UEGO and intake temperature is measured with 2°C resolution using a K-type thermocouple positioned in the intake manifold before the charge entering into the cylinder. A 1/32″ sheathed J-Type thermocouple is used for fast exhaust temperature measurement. The thermocouple is placed in the exhaust as close as possible to the exhaust valve. The thermocouple voltage is amplified before sending to the data logging system and is sampled every crank angle degree. A polynominal fit outlined in [11] is used to calculate temperature from the amplified voltage signal. Measurement of the cylinder pressure is done using a Kistler water-cooled ThermoCOMP (model 6043A60) piezoelectric pressure sensor that is flush mounted in the cylinder head.

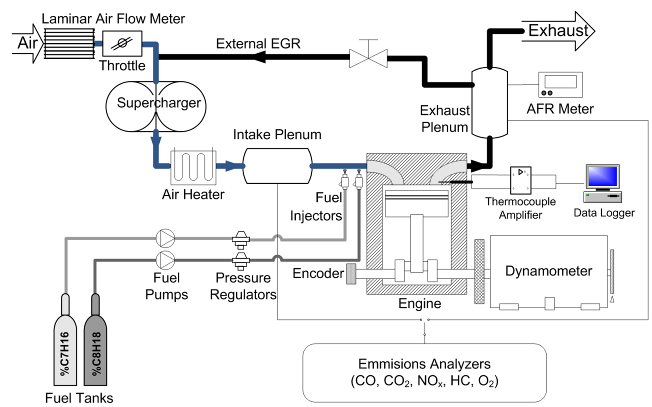

Figure 1: Schematic of the testbench used to obtain experimental data

The Ricardo engine is run to collect two types of experiment measurements: steady-state points, and transient points. Table 2 details the experimental conditions of 165 steady-state points used in this study. Indicated Mean Effective Pressure (IMEP) of the engine ranges from 3.8 bar to 7.3 bar. For each steady-state test point, pressure traces from 200 consecutive engine cycles with 0.1 CAD resolution are recorded. The start of combustion in HCCI is determined using the third derivative of the pressure trace criteria [12]. The Rassweiler method [13] is used to calculate CA_{50} which is the crank angles for 50% burnt fuel. The net heat release rate is determined using the usual heat release method [4] that applies the first law analysis on the engine charge assuming ideal gas properties. Burn duration is defined as the difference between the crank angles of 10% and 90% heat released. For transient tests, fuel injection pulse width is open loop scheduled by a dSpace MicroAutobox 1401/1501 ECU to achieve desired Φ and octane number. SI-HCCI transitions is enabled by programming the ECU with an open loop look-up table for fueling, in both SI and HCCI modes. A software switch is used to command the engine from stoichiometric SI mode to HCCI mode [14]. Engine variables are measured at a constant sample rate of 100Hz during each transient test.

Table 1: Configurations of the Ricardo single cylinder engine.(IVO: Intake Valve Opening, IVC: Intake Valve Closing, EVO: Exhaust Valve Opening, EVC: Exhaust Valve Closing, aBDC: after Bottom Dead Center)

Parameters	**Values**
Bore $\times$ Stroke [mm]	80×88.9
Compression Ratio	10
Displacement [L]	0.447
Number of Valves	4
IVO, IVC [aBDC]	$-175°$, $55°$
EVO, EVC [aBDC]	$-70°$, $-175°$

Table 2: Operating conditions of 165 steady-state data points used in this study

Variables	**Values**
Fuel (PRF)	0 - 10
Engine speed [rpm]	800 - 1200
Intake manifold temperature [^oC]	41 - 120
Equivalence ratio	0.3 - 0.6
Intake manifold pressure [kPa]	88 - 134
EGR [%]	0
$T_{coolant}$ [^oC]	31 - 83
CA50 [CAD aTDC]	1 - 22

4. Results

Values of measured emission and calculated IMEP versus exhaust gas temperature (T_{exh}) are shown in Figure 2 for all 165 HCCI data points used in this study. A significant portion of the data points in Figure 2 have T_{exh} lower than typical catalyst light-off temperatures with 54% of the HCCI data having T_{exh} lower than 300°C, almost 40% of the data has T_{exh} lower than 200°C and T_{exh} can be as low as 147°C. The majority of the data points with low exhaust temperature ($T_{exh} < 300$°C) have negligible NO_x emissions ($NO_x < 10$ ppm) and IMEP is as high as 7.3 bar. These are desirable HCCI operating points although low T_{exh} in these points could be a limiting factor for

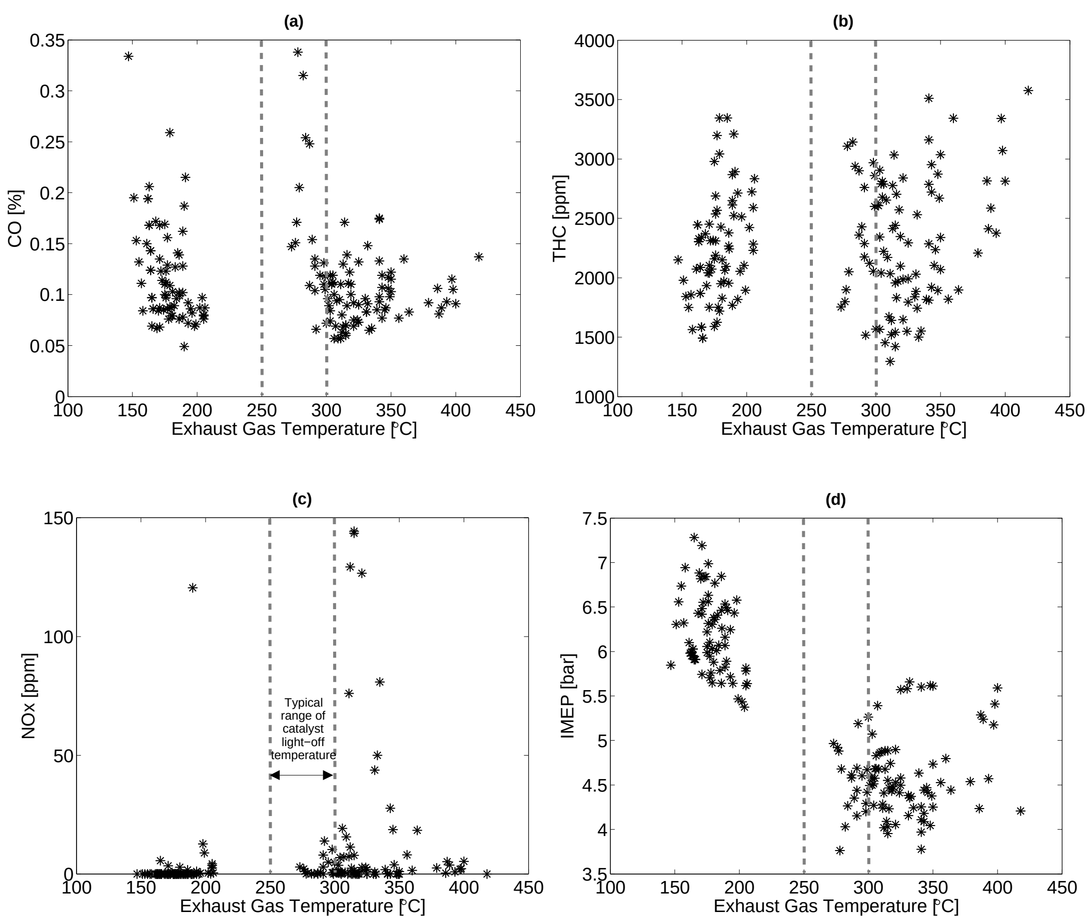

Figure 2: Trend of variation in exhaust gas emissions and IMEP as a function of the exhaust gas temperature.

oxidizing HC and CO pollutants in the catalytic converters. HCCI operation is possible only within a certain range of operating conditions to avoid knock/misfiring. So the data points in Figure 2 are divided into different zones of exhaust temperatures where T_{exh} is either below 206°C or above 273°C and none of the points appeared between these two limits. This zoning is the reflection of the operating conditions of the data points used in this study.

The influence of changing intake pressure on T_{exh} is shown in Figure 3. T_{exh} decreases by 33°C when the intake pressure is increased from 90 kPa to 101 kPa ($\Delta T_{exh}/\Delta P_m = -3°C/kPa$). The advancing the HCCI combustion when the intake pressure is increased is the predominant cause. CA_{50} for the data points of Figure 3 advances from 14.7 CAD aTDC to 4.0 CAD aTDC when the intake pressure increases by 11 kPa. With delayed ignition, more of the energy is released partway down the expansion stroke and this increases the exhaust temperature. As expected the IMEP increases by boosting the intake pressure in Figure 3. Thus for a fixed equivalence ratio fueling condition in Figure 3, HCCI exhaust gas temperature decreases with increasing the engine load (IMEP). Although the location of ignition timing in Figure 3 acts as the main factor affecting T_{exh} when intake pressure varies, other factors can become important when other engine variables such as intake temperature are changed as indicated in Figure 4.

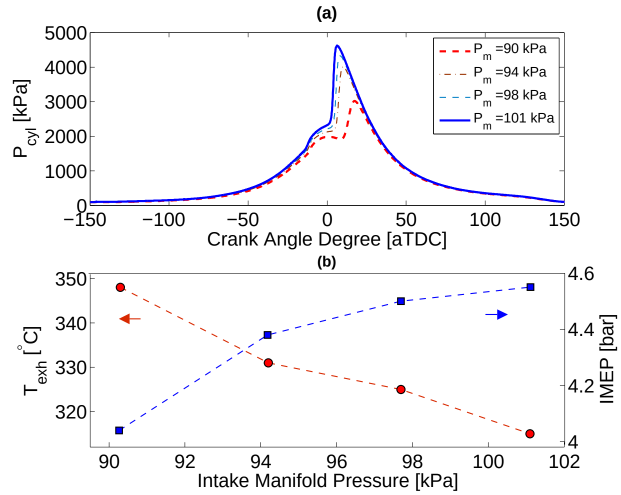

Figure 3: HCCI exhaust temperature variations as a function of the intake pressure. (Fuel: PRF0, 900 rpm, Φ = 0.49, T_{man} = 100°C)

Results in Figure 4 indicate 8.5°C decrease in T_{exh} by increasing the intake temperature from 79°C to 117°C ($\Delta T_{exh}/\Delta T_m = -0.2°C/°C$). This does not exhibit a large change in T_{exh} despite 38°C change in the intake temperature. However, CA_{50} advances significantly from 13.5 CAD aTDC to 7.6 CAD aTDC when the intake temperature increases. This suggests that there is another factor beside CA_{50} variation that causes T_{exh} variations. The second main factor is that higher initial temperature at higher intake temperature conditions which leads to higher in-cylinder gas temperature for most of the engine cycle. This factor acts against the ignition timing factor for changing T_{exh} when the intake temperature is varied. Thus variation in T_{exh} by changing intake temperature in Figure 4 is not as significant as those seen in Figure 3.

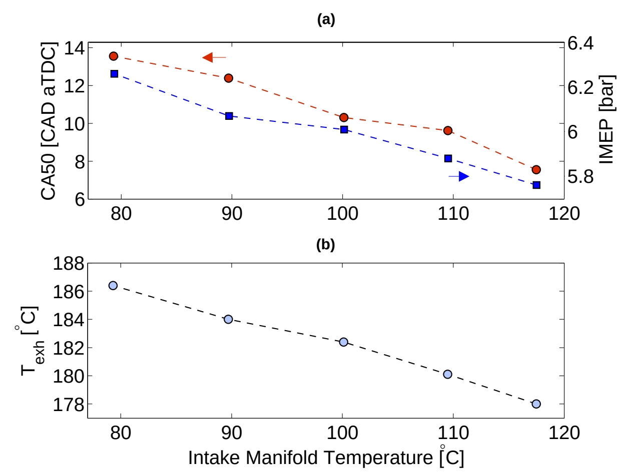

Figure 4: HCCI exhaust temperature variations as a function of the intake temperature (Fuel: PRF10, 1000 rpm, Φ = 0.46, P_{man} = 110 kPa)

Different relation of T_{exh} variations versus engine load (IMEP) is observed in Figure 3 compared to that of Figure 4. This comparison indicates that T_{exh} is not necessarily a function of engine load in HCCI. In Figure 3 T_{exh} decreases by increasing engine load, but T_{exh} increases by increasing engine load in Figure 4. This is because T_{exh} is mainly function of other engine charge variables that may also influence engine load. But IMEP in HCCI mode mainly depends on the fueling rate rather than the ignition timing; however, IMEP becomes strongly dependant on the ignition timing for the conditions that have a constant fueling rate [12]. This explains why IMEP increases in Figure 3 and Figure 4 by either increasing intake pressure or decreasing intake temperature. Since Φ is kept constant in Figures 3 and 4, higher intake pressure and lower intake temperature require more fuel mass flow rates.

Emissions characteristics for the data points from P_m sweep (Figure 3), T_m sweep (Figure 4) are shown in Figure 5. A strong relation is observed between CA_{50} and amounts of CO and HC emissions. This relation is independent of the type of the engine input variation in Figure 5 and mainly depends on the resulting CA_{50} from changing the engine variable. Both CO and HC emissions increase with increasing CA_{50} and this can be explained by higher in-cylinder temperature during HCCI combustion at earlier ignitions. High combustion temperature which is well characterized by the peak cylinder cycle temperature is the main requirement for CO and HC oxidation in HCCI engines [1, 2, 15]. However, higher peak combustion temperature at earlier ignitions causes higher NO_x emissions. This is evident in NO_x values in P_m sweep data points where NO_x increase sharply as CA_{50} advances. However NO_x generation is not influenced by changing CA_{50} in T_m sweep data in Figure 5-c since engine condition (combustion temperature) is far from the conditions to generate NO_x.

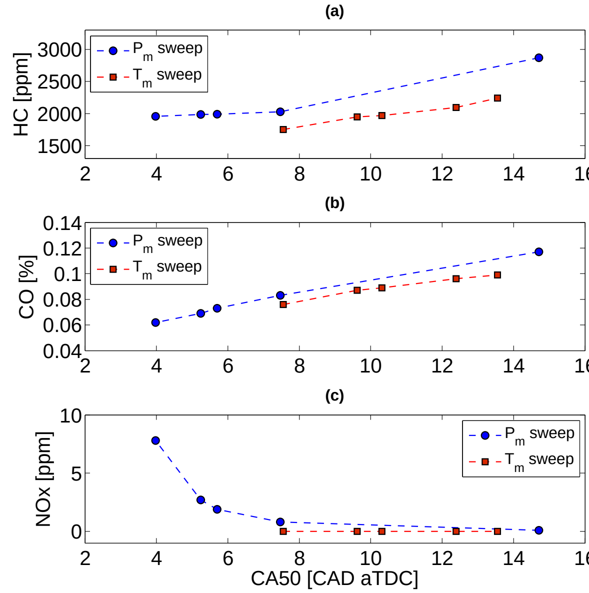

Figure 5: Variations in HC and CO emissions with changing two engine variables: (1) P_m sweep from Figure 3, (2) T_m sweep from Figure 4.

Extending HCCI operating range can be accomplished by switching from HCCI to SI mode for higher loads [16]. This causes a large change in T_{exh} during transient HCCI-SI mode switch as shown in Figure 6. The exhaust gas temperature is substantially higher in SI mode compared to that of HCCI mode for similar load condition in Figure 6. The value of Φ changes from 0.45 to 0.95 when the Ricardo engine switches from HCCI to SI mode in Figure 6. Total heat released in Figure 6 increases by about 40J when the Ricardo engine switches from HCCI mode to SI mode. Higher heat released (455J) in SI mode in Figure 6 contributes to have a higher T_{exh} in SI mode than that of HCCI mode. In addition, burn duration in SI mode is significantly longer than that of HCCI mode. For instance in the test condition in Figure 6, burn duration in SI mode is twice that of HCCI mode. Longer combustion in SI results in releasing more energy at later crank angles compared to that of HCCI. This contributes to higher T_{exh} in SI mode.

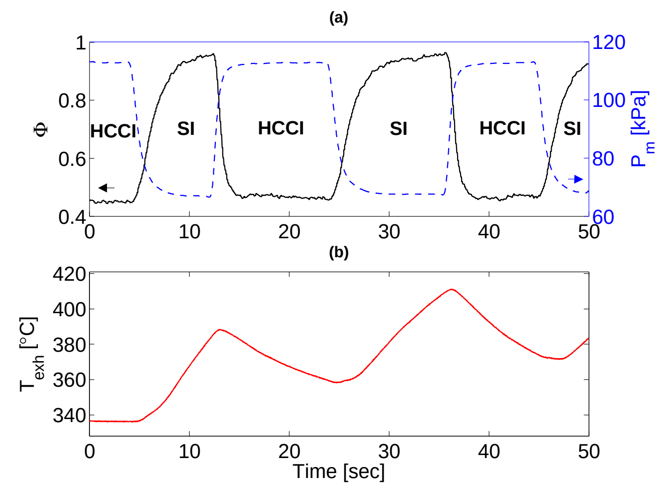

Figure 6: Variation in the exhaust gas temperature in HCCI-SI mode switch with similar steady state loads (IMEP= 4.6bar±0.2bar, CA50 = 7.7±3.2 CAD aTDC, N= 1000 rpm, T_m= 94±1°C.)

5. Conclusions

An experimental study of an HCCI engine at over 160 operating conditions indicate 54% of collected HCCI points have an exhaust temperature below 300°C while having desirable NO_x and IMEP characteristics. Since HC and CO emissions are high in HCCI and conversion of HC and CO by catalysts requires a light-off temperature of about 300°C, low T_{exh} could limit the practical HCCI operating range. It is found location of ignition timing (CA_{50}) significantly influences both exhaust temperature and engine-out emissions. Higher T_{exh} and higher CO and HC emissions but lower NO_x emission are observed when shifting CA_{50} from early ignitions to late ignitions after TDC. Other factors like initial charge temperature reduce the influence of the effect that location of ignition timing causes on T_{exh}. Opposite to SI mode, T_{exh} in HCCI mode is not a strong function of engine load. For the same engine load, a significantly higher T_{exh} is oberved in SI mode compared to that of HCCI mode caused by longer burn duration and higher fuel energy content in SI mode compared to HCCI mode.

Since CA_{50} influences T_{exh}, engine-out emissions and output power the optimal control of CA_{50} to adjust exhaust temperature and engine output power would be beneficial for providing a larger HCCI operating range for which the catalyst would have sufficient exhaust temperature.

Acknowledgments

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