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Residual Gas Fraction Measurement and Estimation on a Homogeneous Charge Compression Ignition Engine Utilizing the Negative Valve Overlap Strategy

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ABSTRACT

This paper is concerned with the Residual Gas Fraction measurement and estimation on a Homogeneous Charge Compression Ignition (HCCI) engine. A novel in-cylinder gas sampling technique was employed to obtain cyclic dynamic measurements of CO_2 concentration in the compression stroke and in combination with CO_2 concentration measurements in the exhaust stroke, cyclic Residual Gas Fraction was measured. The measurements were compared to estimations from a physical, 4-cylinder, single-zone model of the HCCI cycle and good agreement was found in steady engine running conditions. Some form of oscillating behaviour that HCCI exhibits because of exhaust gas coupling was studied and the model was modified to simulate this behaviour.

INTRODUCTION

Homogeneous Charge Compression Ignition (HCCI) holds a great promise in reducing NO_x emissions and fuel consumption in gasoline Internal Combustion engines. To avoid excessive rates of heat release, it requires high levels of charge dilution, which can be achieved by retaining a significant amount of residual gas in the combustion chamber via non-standard valve strategies. The amount of residual gas determines the temperature and constitution of the charge when the inlet valves close and it subsequently affects the combustion phasing and the rate of heat release. In general, the higher the level of the Residual Gas Fraction (RGF), the lower the peak combustion temperature is and hence the lower the NO_x emissions are. RGF in HCCI depends on valve settings, engine speed, fuel amount, intake and exhaust manifold pressures.

In view of the above, the need for an accurate RGF determination is evident. Direct continuous measurement involves comparing the content in HC or CO_2 between the exhaust gas and the fresh charge. The latter requires sampling gas from the cylinder during the compression stroke and analysing it, which has only become possible in the last couple of decades, with the aid of fast emissions measurement systems [8]. It has been possible to measure species' concentrations of in-cylinder gas in IC engines using slow emissions analysers and simple sampling techniques. This involved sampling at the same point in the cycle over many cycles and feeding the gas to an analyzer, on the test bench. Apart from being a laborious and time-consuming process, this method is only valid for steady engine running conditions. Some recent measurement methods involve sampling from pre-programmed motoring cycles occurring every 10-15 cycles [12, 13]. Apart from being constrained to non-dynamic measurements, these methods cannot be used in HCCI because one cycle's misfire would most probably lead to continuous misfiring. Cyclic measurements are only possible by coupling a fast sampling system to a fast response sensor. Most of the challenges associated with this set-up are related to the amount of gas that should be drawn. It has to be small compared to the total cylinder charge, but yet sufficient to give a fast measurement. The most challenging point in the cycle to extract gas is when the cylinder pressure is low.

Residual Gas Fraction can be estimated via thermodynamic gas exchange analysis based on the cylinder pressure signal from a piezoelectric transducer. To attain reasonably accurate estimates, the analysis has to be very detailed, taking into account the dependence of specific heat capacities on species fractions and temperature. Additionally, if engine measurements such as exhaust gas temperature and mass flow of air from a hot film sensor are required by the algorithm, errors are introduced in the computation because of inaccuracies in the measurements. Another estimation approach is to use single-zone or multi-zone models, where the cylinder pressure, temperature and mass are predicted by the model and the only requirement is that the model is calibrated with the use of data from the engine. The long computational time is the main drawback of multi-zone models, especially HCCI models, where detailed chemical kinetics codes are used to predict combustion initiation and duration. It has been demonstrated that at a specific operating point in HCCI with high RGF, the RGF prediction from a single-zone model is only 1% different from the multi-zone's model prediction [6].

EXPERIMENTAL MEASUREMENT OF THE RESIDUAL GAS FRACTION

ENGINE SET-UP

The engine used for the tests is a Ford 2.0L 4-cylinder GDI engine coupled to a dynamometer pneumatically. The engine speed is controlled by the hardware dynamometer controller and a torque output reading is given by the dynamometer load cell. The engine was run with gasoline of 95 Research Octane Number. 4 Kistler 6123 piezoelectric transducers in combination with AVL charge amplifiers facilitated individual cylinder pressure monitoring and recording. An optical encoder provided a pulse every degree of crank revolution, triggering ADC cards for high-frequency data recording. A Horiba Universal Exhaust Gas Oxygen (UEGO) sensor attached to the exhaust tailpipe was used to track AFR. Thermocouples attached to the intake and exhaust ports provided temperature readings. Precise measurement of the fuel flow rate was obtained from a coriolis flow meter. A fast Flame Ionisation Detector HFR400, with a time constant of 4ms, was used to measure uHC.

Homogenous Charge Compression Ignition is attained via Exhaust Gas Trapping [11]. The valve lifts are lower and the valve opening durations are shorter than in a conventional SI engine. The engine is also equipped with dual independent variable camshafts. At a typical HCCI operating point, the exhaust valves close early, well before TDC, to trap residual gas in the cylinder. The inlet valves then need to open well after TDC and for a short period, to avoid significant amount of backflow to the intake manifold. This valve strategy is usually referred to as Negative Valve Overlap (NVO).

THE FAST SAMPLING VALVE

The RGF measurements were obtained by comparing the CO_2 mole fraction in the compression stroke to that of the previous exhaust stroke. A novel in-cylinder sampling technique was employed, using a modified production Gasoline Direct Injection (GDI) injector as a very high speed and compact sampling valve, the Cambustion CSV [1]. The valve was coupled to a Cambustion fast Non-Dispersive Infrared (NDIR) sensor to measure CO_2 volume fractions.

The CSV was used to sample gas from the cylinder every cycle for a specified small crank angle window. Typically minimum valve opening times are less than 1 ms.

The in-cylinder gas was sampled from Cylinder 1 using a drilling in the cylinder head parallel to the head face. As the CSV sampling line is 3mm diameter, this drilling could be made entirely in metal, i.e. with no water jacket interaction. The sample for the second channel of the fast NDIR came from a probe in the exhaust port of cylinder 1, inserted so as to be close to its exhaust value. Note that the sensor measures wet gas concentrations, so there is no need for correction in the measurements.

RESIDUAL GAS FRACTION MEASUREMENT

The residual gas fraction is given by:

$$x_r = \frac{m_r}{m_c} \tag{1}$$

where m_r is mass of residual gas left over from the previous cycle and m_c is the total mass of trapped charge after inlet valves close. For this engine configuration, with no external EGR, the Burnt Gas Fraction (BGF) is equal to the RGF. To a good approximation, the cyclic RGF can be determined from measurements of CO_2 mole fractions in the compression and exhaust strokes using [5]:

$$x_{r,m} = \frac{y_{CO_2,c}}{y_{CO_2,x}} \tag{2}$$

where $y_{CO_2,c}$ is the CO_2 mole fraction in the compression stroke and $y_{CO_2,x}$ is the CO_2 mole fraction in the previous exhaust stroke. In fact Eq. (2) as given in [5] is not correct, the actual RGF as defined in Eq. (1) is equal to:

$$x_r = \frac{y_{CO_2,c}}{y_{CO_2,x}} \cdot \frac{m_{CO_2,r}}{m_{CO_2,c}} \cdot \frac{M_x}{M_c} \tag{3}$$

where $m_{CO_2,r}$ and $m_{CO_2,c}$ are the CO_2 masses in the residual gas and trapped gas after IVC respectively and M_x and M_c are their mean molecular weights.

The difference in molecular weights is not large [5], introducing a small percentage deviation in RGF from Eq. (2) that becomes greater for richer combustion and for lower RGF. The error has been accounted for in the RGF calculation for engine steady conditions but Eq. (2) was used during transient engine operation.

With regard to the CO_2 masses, though at first sight, $m_{CO_2,r} = m_{CO_2,c}$ for an engine without external EGR, in the HCCI situation here, any reaction in the NVO leading to CO_2 production, will lead to an increase in the $m_{CO_2,c}$ above $m_{CO_2,r}$.

CO_2 VOLUME FRACTION MEASUREMENTS

A sampling valve opening time of 2.5ms in the compression stroke, which corresponds to 30 CAD at 2000 RPM, was sufficient for most of the sampling experiments. A gas transport delay of about 30 ms, equivalent to half an engine cycle's duration at 2000 RPM, was observed for both the gas sampled from within the cylinder and the exhaust manifold. The CO_2 mole fraction figures that were used to calculate the RGF level from Eq. (3) are obtained by averaging over flat plateaus of the CO_2 signals after the transit delays. The most important consideration in the CO_2 measurements is that the time constant during transients is small compared to the engine cycle. This requirement is satisfied by the small size of capillaries used to draw the gas to the sensors and the high frequency of the NDIR sensor.

Figure 1 shows the two raw CO_2 volume fraction measurements in HCCI steady engine running conditions. The data were post-processed such that the height of each window represents an average of the CO_2 mole fractions over the period that the gas reaches the sensor. The width of the low window is equal to the sampling period and the width of the high window is equal to the exhaust valve opening duration.

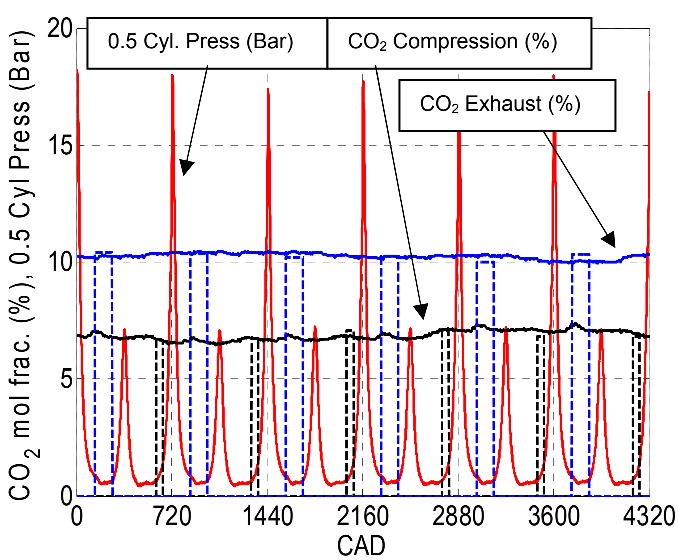

Figure 1: CO_2 volume fractions measurements and average levels in steady state HCCI.

RESIDUAL GAS FRACTION PREDICTED BY A SINGLE ZONE MODEL

Residual gas in SI engines is attributed to two factors:

1. Residual gas that fails to escape from the cylinder by the time the inlet valves open.

2. Exhaust gas that flows from the exhaust manifold to the cylinder during valve overlap.

Zero-dimensional models developed to estimate the RGF treat these two factors separately and ignore any backflow from the cylinder to the intake manifold [3] as the gas is expected to return to the cylinder either later in the cycle or in the next intake event.

In HCCI operation, utilising the Negative Valve Overlap strategy the estimation is simpler, since only the first factor has to be considered. The RGF estimate $x_{r,e}$ is then:

$$x_{r,e} = \frac{m_{IVO}}{m_{IVC}} \quad (4)$$

where m_{IVO} and m_{IVC} are the in-cylinder masses at Inlet Valve Opening (IVO) and Inlet Valve Closing (IVC) respectively. Note that this calculation assumes that the fuel injection is in the intake stroke. If the fuel is injected in the NVO, the fuel mass has to be subtracted from m_{IVO} in Eq. (4).

The RGF estimations came from a zero-dimensional physical dynamic model of the combustion chamber processes of an HCCI engine [2]. The model is based on simple thermodynamic concepts, a simplified chemistry equation and a semi-empirical model to predict the Start of Combustion. Good agreement between RGF measurements and model estimates was found in steady engine running conditions. Some instability, in the form of cyclic oscillations that the HCCI engine exhibits, was investigated and the model was modified slightly to replicate the behaviour and support the analysis.

The model was originally developed as it was felt that more insight would be gained about the cyclic processes especially during transition from SI to HCCI, than would be possible using a commercial code, where the physics of the problem are largely hidden. As the model is single-zone and it does not include any chemical kinetics, it is relatively fast. Its main advantage though is that the model persists between multiple cycles enabling the cycle-to-cycle coupling through the residual gas that exists in HCCI. The models for the 4 cylinders are identical and they are coupled together by a single-state intake manifold model. The VCT actuation is also unique for the 4 cylinders to represent the engine set-up used.

Details on the model assumptions, equations and parameters can be found in [2].

PREDICTION OF START OF COMBUSTION

The ignition timing in HCCI is solely determined by the chemical kinetics in the cylinder. Even low-order chemistry mechanisms may include up to 100 species and 500 reactions [6]. The computational times required for detailed chemical-kinetics models are long. Given that the detailed chemistry for real fuels is unknown, simple semi-empirical techniques are used extensively. The simplest possible prediction of HCCI combustion initiation is to assume that it occurs at a certain temperature level, since the exponential effect of temperature in the reaction rate is dominant. Here, we use a slightly more complex model, where the temperature/time history, fuel and air effects are modeled. Combustion is triggered when the Integral of an Arrhenius-type expression exceeds a constant threshold K_{th}. It occurs at the time $t = t_{soc}$, the time of Start Of Combustion (SOC), when

$$A\int_{t_{ivc}}^{t} T\exp\left(-\frac{E_a}{R_u T}\right)\left[C_nH_m\right]^a\left[O_2\right]^b \mathrm{d}t = K_{th} \qquad (5)$$

where t_{ivc} is the time of Inlet Valve Closing, $\frac{E_a}{R_u}$ is the fuel activation temperature. The constants a, b and $\frac{E_a}{R_u}$ for pure hydrocarbons are tabulated in [10]. The values for isooctane were chosen. An estimate for K_{th} was found using a least squares approach using 10 datasets with different fuel rates and speeds.

COMBUSTION CHEMISTRY AND GAS PROPERTIES

A simple combustion chemistry equation was included in the model, considering only main products and reactants, ignoring CO and NO_x formation. The resulting mole fractions were used to determine the specific heat capacities of the cylinder gas. The temperature dependence was taken into account using look-up tables built from JANAF data [9] and a weighted average based on the mole fractions was calculated.

CYLINDER MASS AND CO_2 PREDICTIONS

The in-cylinder mass and the CO_2 mole fraction are computed continuously within and between engine cycles.

Figure 2 shows the mass of cylinder contents during an engine cycle under typical steady engine running conditions. As it might be expected, the model predicts some backflow at the beginning and the end of the short intake event. Even though the total duration of backflow is quite significant, the total mass drawn into the intake manifold is small because the valve lift is low during the backflow periods. The model assumes that only air moves during backflow. This is a reasonable assumption because the gas drawn into the manifold will return to the cylinder later in the cycle or in the next intake event besides which, there is no mixing model included for the intake manifold. This assumption makes the model simpler and lowers its CPU time. However it may result in a slight overestimation of the RGF during transient operation. Fuel injection is assumed to occur at a constant rate and it has a small contribution to the total mass. Finally, the model indicates the occurrence of choked flow at the beginning of the exhaust event, when the cylinder pressure is high. The RGF is computed using Eq. (4) accounting for the fuel mass, which is all injected in the NVO.

The CO_2 mole fraction prediction under some different steady engine running conditions is shown in Figure 3. It rises unrealistically at the beginning and the end of the intake event because of the assumption of air flowing during backflow. A small drop is apparent during fuel injection. The model assumption of homogeneous gas expulsion during the exhaust event forces the CO_2 mole fraction to remain constant in that period.

COMPARISON OF RGF MEASUREMENTS TO MODEL PREDICTIONS

The model was calibrated at one operating point, independently of the RGF measurements, based on cylinder pressure data, AFR measurements and IMEP and a broad model validation followed. Accuracy within 5-10% in the main engine variables was observed, which is reasonable given the level of complexity of the model and the approximations involved in the prediction of SOC and the heat transfer model. RGF predictions were then compared to measurements in steady engine running conditions.

EXHAUST VCT SWEEP

The engine was run at 2000 RPM, 2.10 Bar BMEP in steady state with the settings shown in Table 1. The EVC angle was then changed in steps of 2.5 CAD with the rest of the settings unchanged. The CO_2 concentrations in the compression stroke and in the exhaust runner, upstream of the common point, of Cylinder 1 were measured to yield RGF measurements (Fig. 4). As expected, the Residual Gas Fraction drops as the exhaust VCT is retarded and the HCCI operation becomes leaner. The allowable range of VCT sweep, with no other engine setting change, is limited to 15 CAD because of the sensitivity of the HCCI operation to the residual gas trapped. More retarded VCT would lead to large cycle to cycle variations and eventually misfiring. More advanced VCT would result in unacceptably high pressure gradients and engine knocking.

Table 1: Engine and sensor settings

Parameter	Value	Units
Speed	2000	RPM
Throttle opening	100	%
EVC angle	-95	CAD ATDC
IVO angle	95	CAD ATDC
Fuel Flow Rate	2.06	Kgh^{-1}
Spark Angle	50	CAD BTDC
Start of Inj. Angle	325	CAD ATDC
CSV window	660-690	CAD ATDC
NDIR Press. Compr.	55	kPa (Absolute)
NDIR Press. Exh.	80	kPa (Absolute)
CSV Spill Vac. Press.	90	kPa (Absolute)

The engine speed, valve settings, throttle opening and fuel amount were used as model inputs and the combustion duration was fixed to 25 CAD. The RGF predictions agree reasonably well with the experimental measurements, especially for the most retarded EVC angles. The model slightly overestimates the change in RGF across the feasible VCT sweep. This is thought to be due to the simplifying assumption of a constant exhaust manifold pressure.

LOAD SWEEP

In another set of experiments, the RGF was measured for a span in the load from 3.0 to 5.7 Bar Net IMEP, at 2000 RPM (Fig. 5). An increase in the load in HCCI involves increasing the fuel amount and reducing the amount of trapped residuals, which is achieved by retarding the EVC in this engine configuration. The IVO angle was set symmetric to the EVC angle with respect to TDC, for each operating point. Note here that even though the Net IMEP is an engine output quantity measured from cylinder pressure data, it is used here to represent a set of engine inputs.

The RGF decreases monotonically as the load rises in an almost linear fashion. The model inputs were matched to the engine settings and the combustion duration was set to 25 CAD. The model over-predicts the RGF by about 2.5%, absolute level. This level of accuracy is reasonable, given that moving towards higher IMEP requires changing three model inputs. Increasing the fuel flow rate and retarding the exhaust VCT both tend to decrease the residual gas fraction. Intake VCT has a non-monotonic effect on the RGF for fixed fuel amount and exhaust VCT. This is because for the fixed short valve duration, a too advanced intake VCT results in a significant backflow at the beginning of the intake event and a too retarded intake VCT results in a large backflow towards the end of the intake event. Both tend to reduce the air inducted and increase the RGF slightly.

CYCLE BY CYCLE ALTERNATING BEHAVIOUR

In HCCI, the high level of residual gas trapped couples one engine cycle to its previous one. In a conventional SI engine, such coupling is sometimes observed at idle, especially in pre-VVT engines. The exhaust gas coupling in HCCI can have self-sustaining stabilising effects [14] but at high loads, combustion can be gradually driven unstable [15]. Combustion instability can be in the form of cylinder misfiring or cyclic oscillations in ignition timing, work output, AFR and RGF. This form of instability has been explained via the cyclic thermal effects and via the chemistry involved [4, 16].

In this study, cyclic oscillatory engine behaviour, where some pre-oxidation occurs in the NVO every other cycle, is investigated using cylinder pressure data, fast CO_2 measurements and fast uHC measurements.

EXPERIMENTAL RESULTS

The oscillating behaviour was examined at two operating points. It was initially observed at 1500 RPM, 2.62 Bar BMEP, λ=1.1. It was then examined further at the lower load point of 1500 RPM, 1.0 Bar BMEP, λ=1.21 at which oscillations are more intense and frequent. The engine was operated at WOT with a SOI = 40 CAD BTDC in the NVO. The oscillations would last for 6 to 20 cycles, followed by stable engine running for tens of cycles. It is not entirely clear what triggers the oscillations but presumably they are initiated by the normal cyclic variations in AFR and RGF.

In summary, the following effects were observed:

- Cyclic oscillations in Net IMEP and Gross IMEP.
- Fuel oxidation in the NVO taking place every other cycle, as evidenced by the work in the NVO.
- Alternating advanced and retarded main combustion events.
- Higher levels of uHC in the exhaust stroke of a late combustion.
- Strong correlation of high IMEP with high residual gas fractions.

The IMEP calculation was chosen to start at the BDC after the main combustion event, so that one cyclic calculation can capture the NVO oxidation and the main combustion event uninterrupted (Figs. 6, 7). In cycles with high IMEP, the Net IMEP is slightly higher than the

Gross IMEP. This is an indication of some energy release in the NVO, which is greater than the small pumping work of unthrottled HCCI. Higher peak cylinder pressures in the recompressions preceding the main combustion events of these cycles were also observed.

The alternating oxidation in the NVO was confirmed by logP – logV diagrams (Figs 8-11). They are set to start at the BDC, end of the expansion stroke. When oxidation occurs, the pressure during expansion exceeds the pressure during compression at the symmetric positions about TDC. Typically, the NVO oxidation starts at 10 CAD BTDC and results in advancing the main SOC from 1 to 8 CAD BTDC.

During the oscillating periods, fast measurements of uHCs consistently revealed higher levels in the exhaust events of late-igniting cycles compared to early-igniting ones (Fig. 7). This suggests that cycles with high IMEP most probably burn more fuel. Approximately, a Net IMEP difference of 0.5 Bar between consecutive cycles is associated with a difference of 350 ppm in HC emissions. Exhaust gas temperature measurements from a 0.2 mm-diameter thermocouple fitted close to the exhaust valve of the same cylinder showed a slight tendency of high IMEP cycles to be correlated with high exhaust gas temperatures.

Figure 12 shows the RGF measurements during cyclic oscillations at 1500 RPM 2.62 Bar BMEP. Cycles with high IMEP are also cycles with high RGF. The raw CO_2 mole fraction measurements and the cyclically average levels are shown in Figure 13. High IMEP cycles have higher CO_2 levels in the compression and exhaust events compared to low IMEP cycles. The higher exhaust CO_2 level can be partially attributed to the fact that more fuel burns in that cycle and due to the fact that the high IMEP cycles are less lean. The oxidation in the NVO produces some CO_2, some of which flows back to the intake manifold at the beginning of the intake event and it re-enters the cylinder later on. The more fuel that oxidizes in the NVO, the higher the CO_2 volume fraction in the compression stroke is. Since the fuel oxidised in the NVO is relatively small, the high level of compression CO_2 mole fraction is attributed to the smaller air mass inducted because of the lower density of the hotter residual gas.

ANALYSIS AND DISCUSSION

Koopmans et al [4] have observed a very similar phenomenon on a Port Fuel Injected engine set-up utilizing the NVO. They examined it with the aid of fast uHC and fast exhaust gas temperature measurements. In this work, the setup is a GDI engine and the whole fuel amount is injected before TDC in the NVO. Further to that, the fast uHC measurements during oscillations were taken at a lower load point than in [4] and the cycle-by-cycle differences in uHC are not so prominent. Otherwise, part of the analysis here looks similar to [4].

The oscillating behaviour can mostly be explained via the cyclic thermal effects associated with HCCI operation and the completeness of combustion in each cycle. A slight oxidation in the NVO yields the following subsequent effects compared to the case of no oxidation: Hotter gas when the inlet valves open, hotter charge when the inlet valves close, more advanced and sometimes shorter main combustion. The reduced amount of air inducted due to the higher residual temperature during the intake event has also some contribution to the higher temperature in the compression stroke.

A late-igniting cycle is associated with a higher level of uHC, an indication that the combustion is less complete than in an early-igniting cycle. The combustion in a late-igniting cycle takes place mostly in the expansion phase where the gas temperatures are lower and the heat delivered by the fuel is probably quenched. The high levels of dilution imply that early-igniting cycles burn more fuel than late igniting cycles and this is the main reason for the high Net IMEP values observed.

The most challenging part of the analysis is to determine how the gas condition in NVO is different enough between consecutive cycles to trigger some fuel oxidation in one NVO and not in the next one. The whole fuel amount is injected before any NVO oxidation is observed but the time is limited for the fuel to evaporate and prepare for combustion. The temperature and type of hydrocarbons left in the cylinder from the previous cycle are considered to be key factors in initiating oxidation. Late-igniting cycles result in a few hundred ppm higher uHC than early igniting ones which are probably enough to initiate oxidation in the following NVO. Late–igniting cycles are also leaner cycles. The residual gas that remains in the cylinder before combustion of a late-igniting cycle is cooler and denser allowing more air to enter. In the following recompression, the oxygen content is higher and this might have some contribution to combustion but no safe conclusions can be driven from the data gathered.

The residual gas fraction measurements, taken at the medium load point, confirm that a high IMEP cycle has a higher RGF than a low IMEP cycle. This is due to the high temperature low density residuals that follow the NVO oxidation, constraining the amount of air drawn in the intake event. An early-igniting cycle is followed by no oxidation in the NVO and therefore lower residual gas fraction in the following cycle. There is no apparent difference in the cylinder pressures during the exhaust events of consecutive cycles indicating that the absolute residual mass in the NVO and compression strokes of consecutive cycles is very similar.

It is likely that the amplitude of oscillation in the RGF is in practice slightly lower than measured. The RGF is estimated as the ratio of CO_2 mole fractions in the compression stroke and the previous exhaust stroke. The exhaust gas and therefore the residual gas retained in the cylinder after a low IMEP poorly burning cycle has a lower CO_2 mass content than normally and this would result in an overestimation of the RGF of the next cycle. The following fuel oxidation increases the CO_2 mass content in the charge after the inlet valves close and adds to the apparent RGF. In overall, it is thought that the RGF of the early-igniting cycle is slightly overestimated.

To gain further insight to this phenomenon, the model developed for SI to HCCI transition control was deliberately set to simulate the oscillating behaviour. At a first stage, provision for a second combustion in the NVO was added. As the model includes no chemical kinetics and the prediction of combustion is mainly based on temperature evolution the oscillating behaviour cannot be predicted. Instead combustion in the NVO was deliberately set to happen every other cycle burning around 15% of the fuel to match logP - logV diagrams from the low load point examined.

The model does not allow for quenching and the combustion is complete in every cycle, as long as fuel and oxygen are available. No significant IMEP oscillations were indicated by the model, confirming that the actual IMEP oscillations are due to the different amounts of fuel burnt. The model predicts that oxidation in the NVO results in earlier main combustion and higher peak pressure. Finally, it is confirmed that early igniting cycles are less lean with a higher RGF (Fig. 14).

CONCLUSION

Residual Gas Fraction is a critical parameter for performance and emissions in HCCI engines. In this paper, a fast sampling technique was used on a Gasoline HCCI engine to facilitate cyclic dynamic measurements of Residual Gas Fraction. The RGF measurements were compared to the predictions of a single-zone physical model of HCCI combustion and gas exchange processes and reasonably good agreement was found with respect to Exhaust VCT and load changes.

Some form of cycle to cycle oscillations exhibited in HCCI engines at medium and low load points was examined by the means of cylinder pressure data, fast CO_2 and HC measurements. The oscillations arise as fuel oxidation occurs in the recompression every other cycle due to the higher level of unburnt hydrocarbons following a late-igniting cycle with less complete combustion. The secondary oxidation results in a hotter charge in the main compression and therefore an earlier and more complete combustion than the previous one. Early igniting cycles are associated with a higher Residual Gas Fraction due to the smaller amount of air drawn. The model was used to support the analysis of the results.

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ABBREVIATIONS

ADC Analogue to Digital Converter

AFR Air to Fuel Ratio

ATDC After Top Dead Center

CSV Cambustion Sampling Valve

HCCI Homogeneous Charge Compression Ignition

RGF Residual Gas Fraction

NDIR Non-Dispersive Infrared

NVO Negative Valve Overlap

BTDC Before Top Dead Center

SOC Start Of Combustion

EVC Exhaust Valve Closing

GDI Gasoline Direct Injection

IVC Inlet Valve Closing

IVO Inlet Valve Opening

SOI Start of Injection angle

BDC Bottom Dead Center

TDC Top Dead Center

IMEP Indicated Mean Effective Pressure

BMEP Brake Mean Effective Pressure

CAD Crank Angle Degrees

VCT Variable Cam Timing

RPM Revolutions per Minute

uHC Unburnt hydrocarbons

HC Hydrocarbons

VVT Variable Valve Timing

FID Flame Ionisation Detector

WOT Wide Open Throttle

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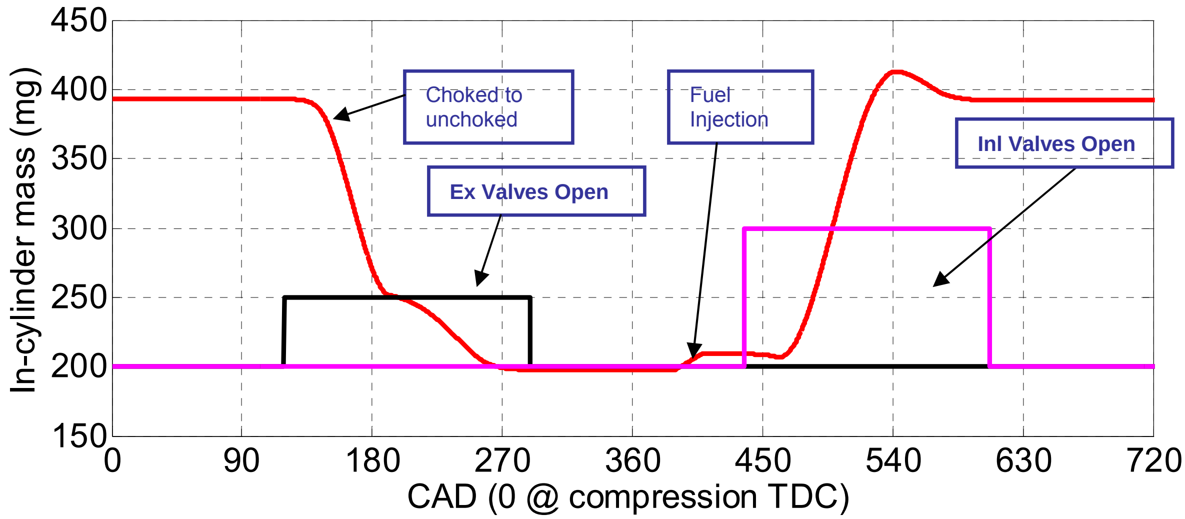


Figure 2: Model prediction of in-cylinder mass at 2000 RPM, 3.0 Bar Net IMEP in steady state

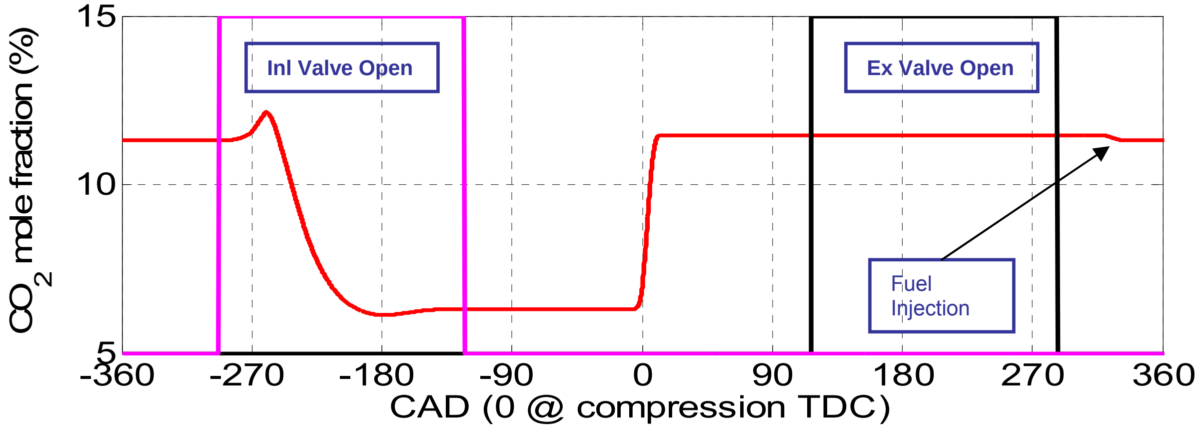


Figure 3: Model prediction of CO_2 volume fraction at 1500 RPM, 3.0 Bar Net IMEP in steady state

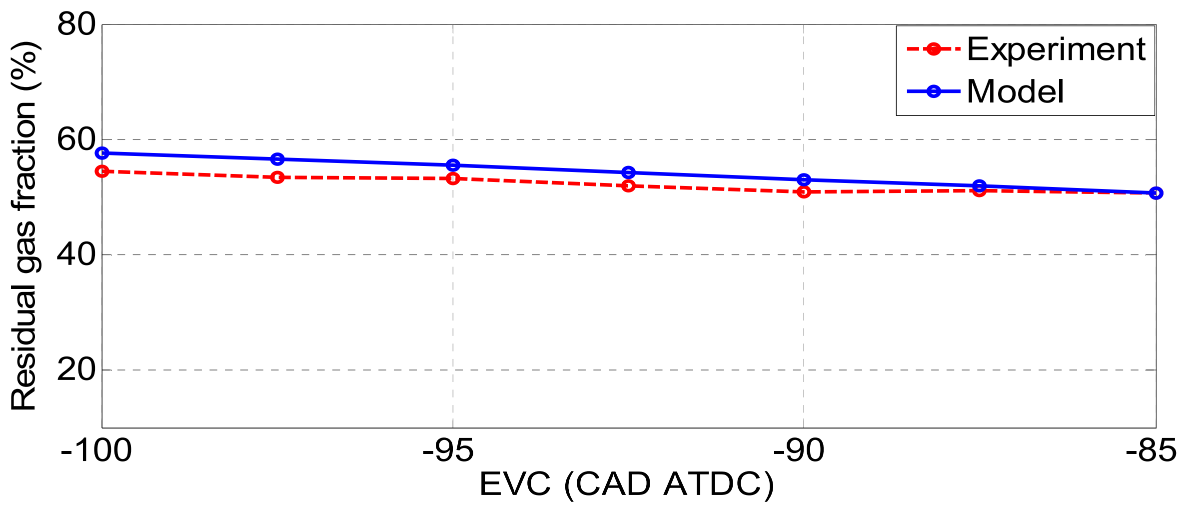


Figure 4: Residual gas fraction comparison with respect to changes in EVC

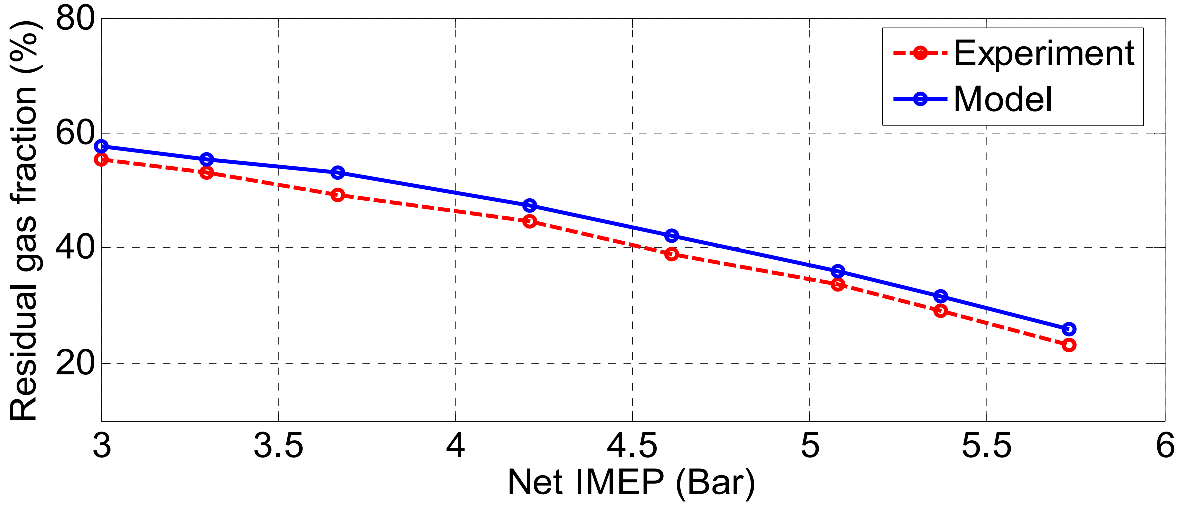


Figure 5: Residual gas fraction comparison across a span in load

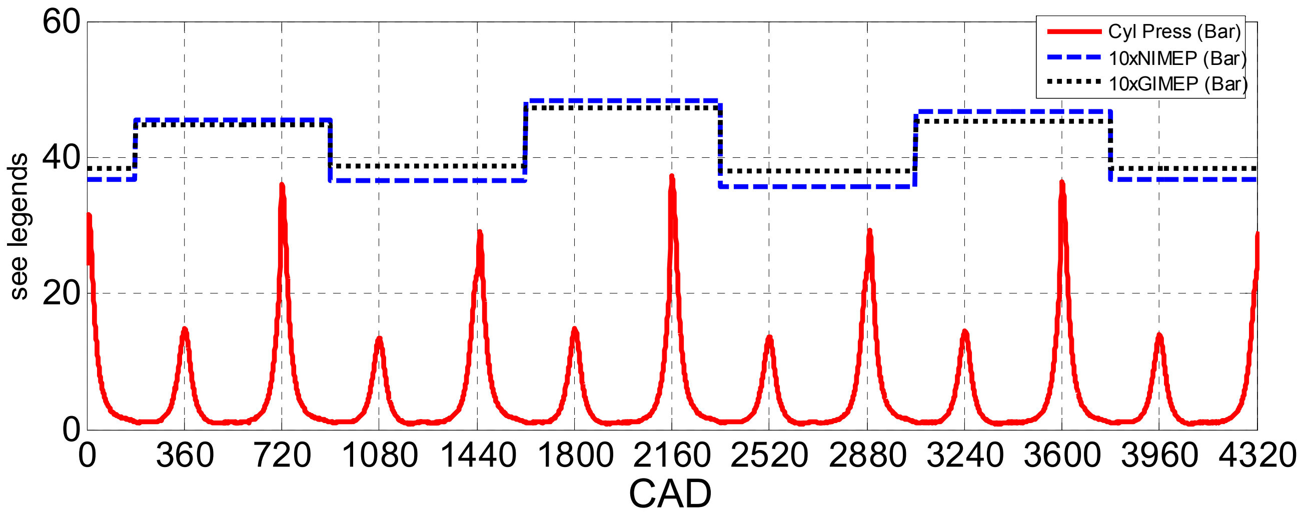


Figure 6: Cyclic oscillations in the Net and Gross IMEP at 1500 RPM 2.62 Bar BMEP

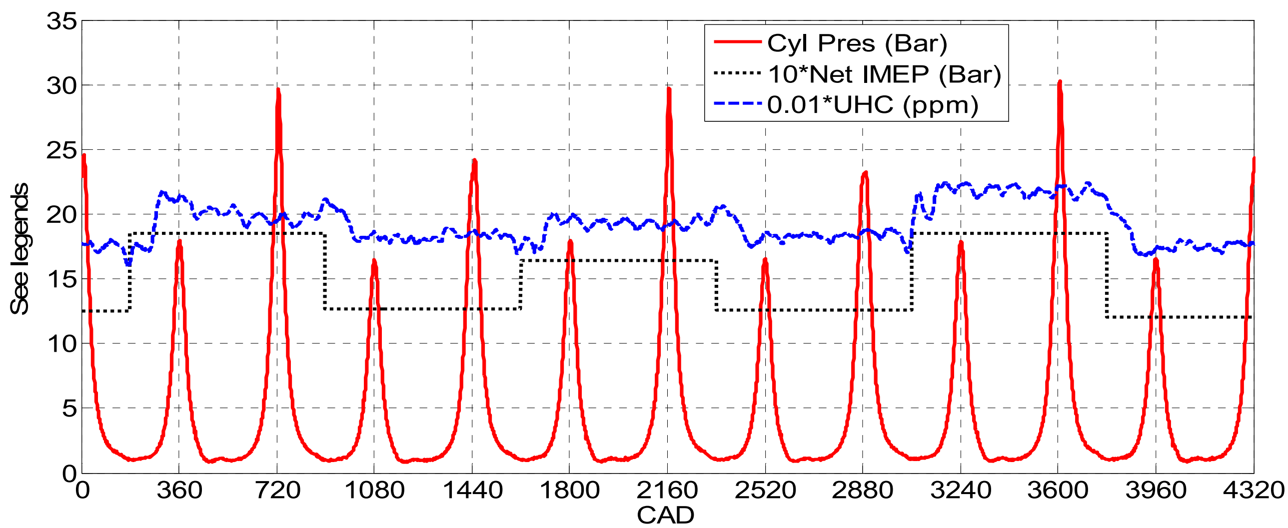
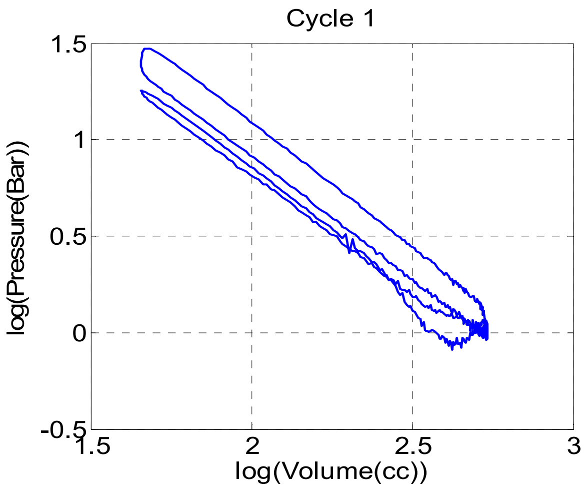


Figure 7: Cyclic oscillations in the Net IMEP and uHC at 1500 RPM and 1.0 Bar BMEP



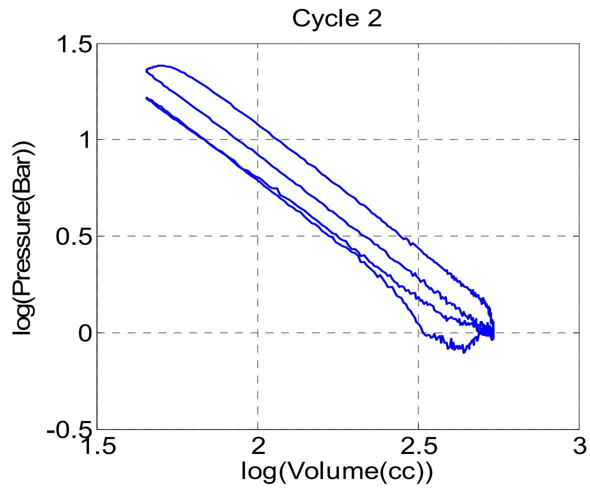
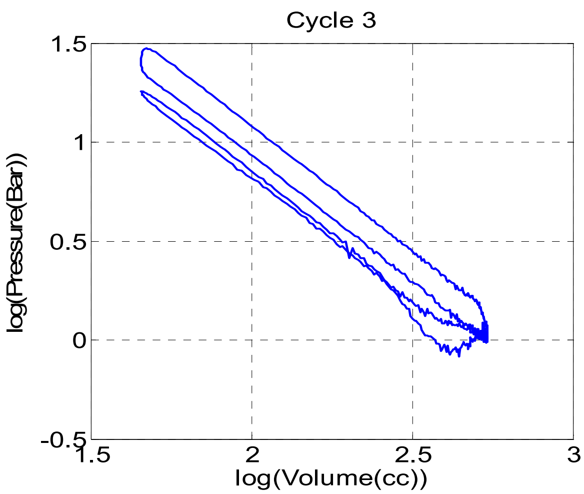


Figure 8: Log P vs log V at 1500 RPM 1.0 Bar BMEP. Significant Oxidation in the NVO is followed by early main combustion.

Figure 9: Log P vs log V at 1500 RPM 1.0 Bar BMEP. Very Slight Oxidation in the NVO is followed by late main combustion.



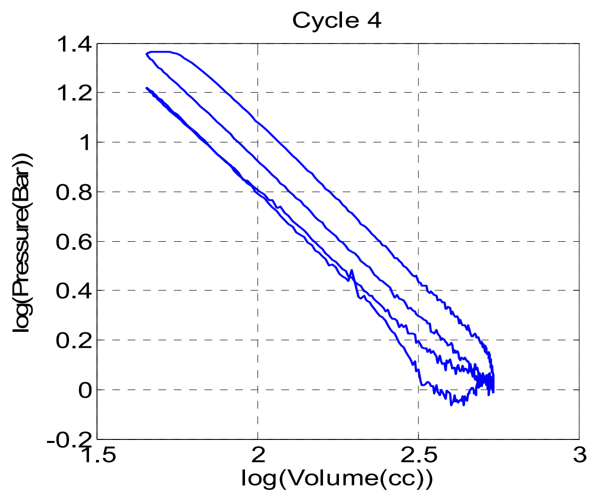


Figure 10: Log P vs log V at 1500 RPM 1.0 Bar BMEP. Significant Oxidation in the NVO is followed by early main combustion.

Figure 11: Log P vs log V at 1500 RPM 1.0 Bar BMEP. Very Slight Oxidation in the NVO is followed by late main combustion.

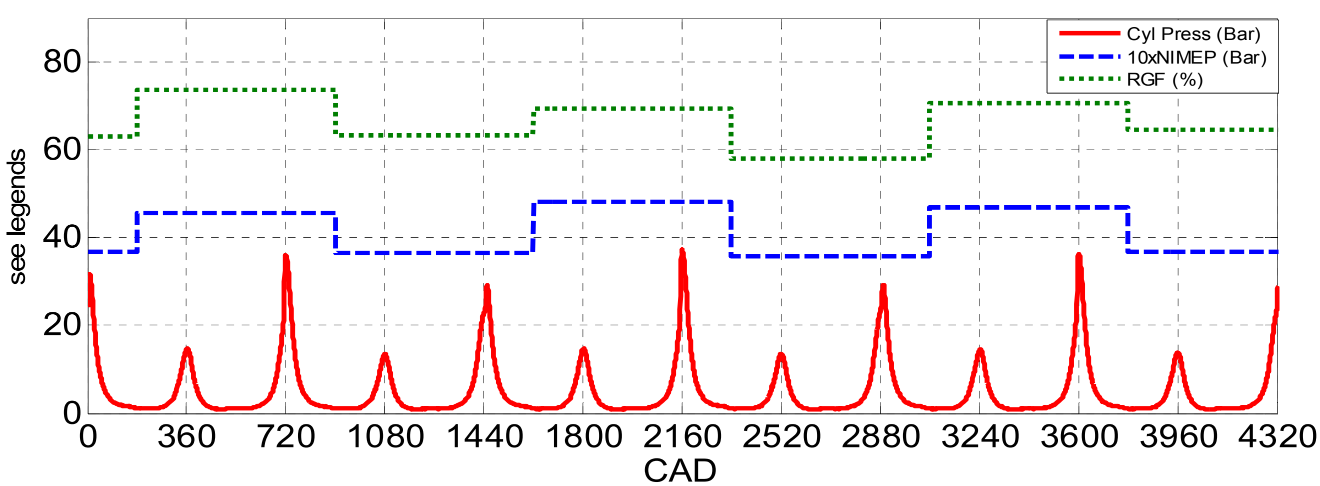


Figure 12: Cyclic oscillations in the Residual Gas Fraction at 1500 RPM 2.62 Bar BMEP

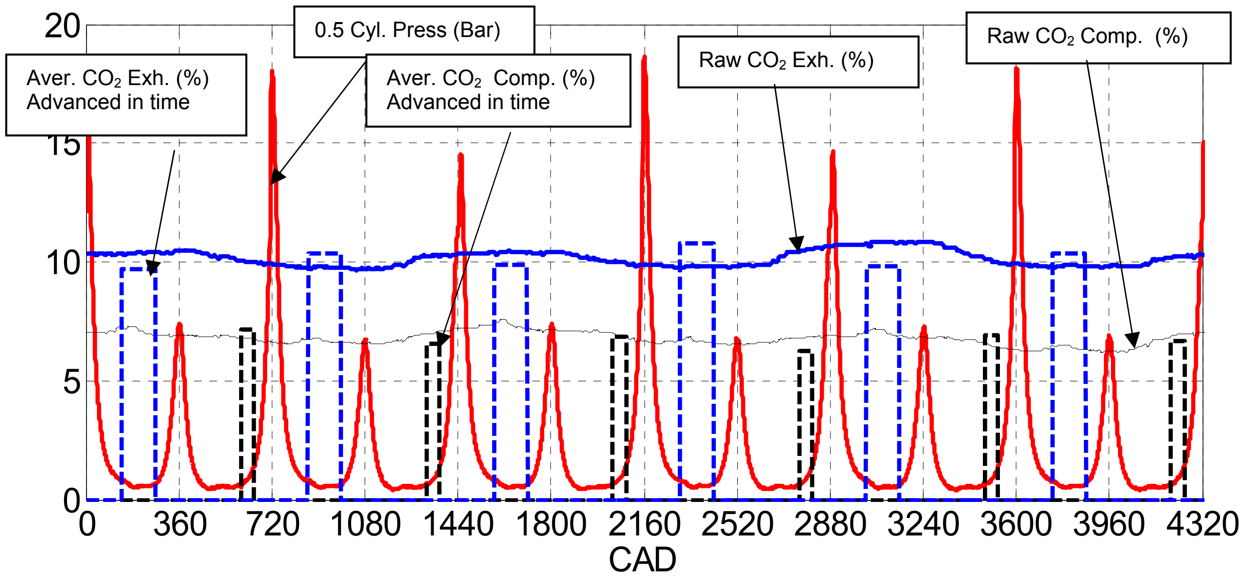


Figure 13: Cyclic CO_2 volume fraction measurements and average levels at 1500 RPM 2.62 Bar BMEP (See legends for units)

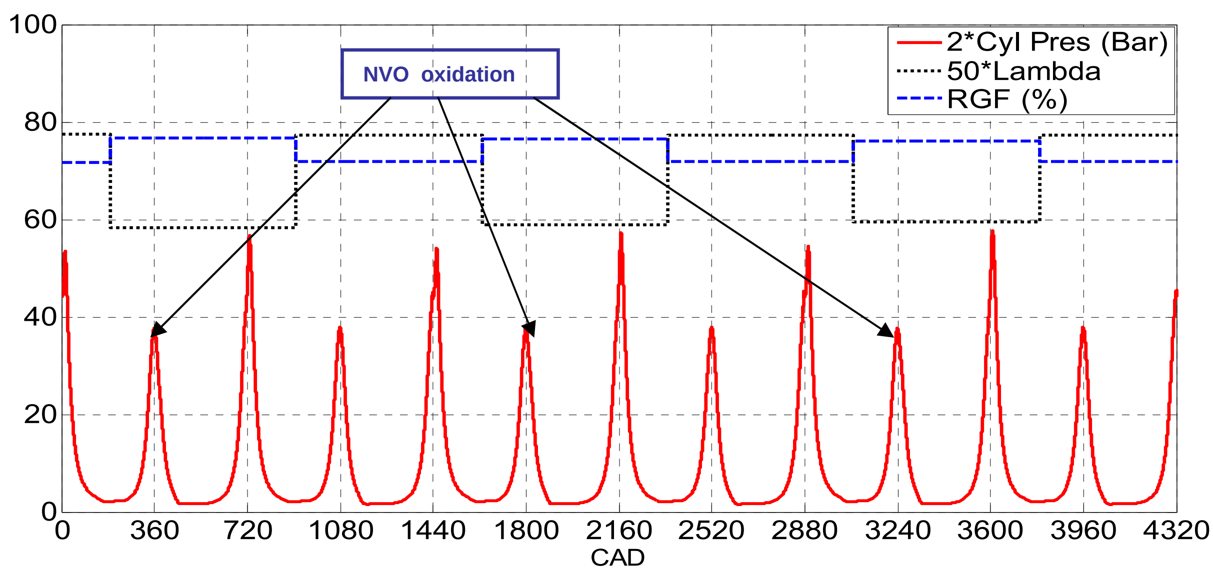


Figure 14: Modelled oscillating behaviour in RGF and AFR at 1500 RPM 1.6 Bar Net IMEP (See legends for units)