IMPACT OF CYLINDER-TO-CYLINDER DISPERSION OF EXHAUST GAS RECIRCULATION ON THE THREE-WAY CATALYST PERFORMANCE AND TAILPIPE EMISSIONS OF SPARK-IGNITION ENGINES

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ABSTRACT

New generations of spark-ignition engines include exhaust gas recirculation (EGR) to improve the engine efficiency. Depending on the design of the EGR routing, some differences in the total amount of recirculated gases that reach each cylinder can be induced. This affects the air-to-fuel ratio on each cylinder due to the combination of the different temperature and composition of the gases at the intake valve closure. As a

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consequence, significant deviations in the combustion process and the subsequent composition upstream the three-way catalyst can be reached. This paper explores these effects on catalyst performance and tailpipe emissions, individualizing the behavior for each regulated species. The study was performed in a 4-cylinder naturally aspirated engine with Atkinson cycle and a close-coupled three-way catalyst. The most significant deterioration in conversion efficiency appeared for the nitrogen oxides, directly linked to the EGR dispersion level. In the case of CO emissions, no significant impact was observed except at high average EGR rates, where one or more of the cylinders exceeded the EGR tolerance for that speed and load. Based on these results, a strategy where the fuel injector command is adapted to correct the air-to-fuel ratio deviations induced by the EGR was developed and implemented.

1. INTRODUCTION

Hybrid-electric vehicle (HEV) platforms have been developed in the past few years as a means to rapidly reduce carbon dioxide emissions from road transportation [1]. However, this has substantially increased the cost and complexity of the complete powertrain system [2]. Therefore, efforts are being made to mitigate this trend both on the engine and electrical system sides [3]. For this reason, most HEV are based on spark-ignition (SI) engines, characterized by lower costs thanks to lower requirements on the fuel injection system, lower in-cylinder pressure levels [4] and easier exhaust emissions management through the three-way catalyst (TWC) [5]. However, the need to maintain stoichiometric conditions limited the engine efficiency due to increased pumping work losses at low load [4]. This has been partially compensated by introducing turbocharging systems and the subsequent engine downsizing trend [6], but at the expense of more risk for knock at boosted conditions [7]. Additionally, including the turbocharging system increases the engine’s overall complexity with respect to a naturally aspirated counterpart [8].

Cooled exhaust gas recirculation (EGR) is another tool that can be used to simultaneously mitigate pumping losses and increase knock resistance [9,10] in SI engines. Its implementation can lead to substantial improvements in fuel efficiency [11], especially at higher loads thanks to a more optimal combustion phasing [12,13]. Furthermore, the cost impact of adding the EGR routing is relatively low [14]. However, to effectively reach the expected fuel efficiency benefits it is critical to maintain a proper matching between the residuals level in the cylinder (combination of EGR and internal residuals) and the management of fuel injection and spark actuations [15]. Otherwise, combustion can become unstable, increasing the emitted levels of the total unburned hydrocarbons (THC) and carbon monoxide (CO) emissions [16] in a way that may not be manageable by the TWC.

The correct management of the residuals level and the cylinder control may be more critical depending on how the EGR routing is designed [17]. When using low-pressure systems (LP-EGR) in turbocharged engines, the external EGR is introduced upstream the compressor, so a completely homogeneous mixture of air and EGR gases are produced before reaching the cylinders [18]. Thus, the expected residuals gas fraction would be approximately the same on each cylinder [19]. Instead, in turbocharged engines with high-pressure EGR circuit (HP-EGR) as well as in naturally aspirated engines the exhaust gases are introduced directly in the intake manifold (mono-point injection) [20] or in the intake runners (multi-point injection) [19]. Consequently, non-negligible cylinder to cylinder variations of the residuals fraction can appear [21], especially under highly transient operation [22].

These variations can affect the operation of the TWC, which is highly sensitive to the air-fuel ratio (AFR) [23]. During transient lean AFR conditions, the high oxygen concentration affects TWC performance causing a significant drop in nitrogen oxides (NOx) conversion efficiency [24]. Otherwise, transient-rich AFR conditions in SI engines can negatively affect the TWC's THC and CO conversion efficiency due to the excess fuel in the exhaust gas [25].

In this regard, the effect of exhaust gas composition changes caused by EGR dispersion, and its effect on aftertreatment, have been studied on compression-ignition engines due to its negative impact on NOx emissions management [26], but the best of the authors' knowledge, this impact has not been explored yet in a modern SI engine platform. Therefore, the current study focuses on the analysis of the EGR dispersion impact on combustion development and regulated exhaust emissions management. For this purpose, a 4-cylinder spark-ignition Atkinson cycle engine was selected. In this engine, a multi-point EGR system takes the gases from the outlet of the close-coupled TWC and injects them approximately 120 mm upstream of the inlet of the intake ports. An intake CO_2 probe is installed 50 mm downstream the injection point to evaluate the rate of recirculated gases in each cylinder. The system incorporates a set of fine-step gate valves to control the amount of recirculated exhaust gases on each cylinder and the subsequent EGR dispersion. Finally, a Horiba-Mexa 7100DEGR exhaust gas analyzer coupled with a line selector was used to measure the regulated emissions either upstream or downstream of the TWC.

With this setup, the combustion characteristics and the operation of the TWC is analyzed for a set of steady-state engine operating points, defined in terms of speed and load, as well as different levels of cylinder-to-cylinder dispersion of recirculated exhaust gases. In the first step, the engine will be run at stoichiometric conditions according to the lambda sensor placed upstream of the catalyst. Based on these results, a strategy where the fuel injector command is adapted for each cylinder will be proposed and evaluated.

The paper is divided into 5 sections. Section 2 details the engine configuration and instrumentation used for the study and describes the testing procedure and schedule. Section 3 focuses on the effect of the dispersion on combustion duration and tailpipe emissions when AFR is set as stoichiometric according to the lambda sensor upstream the aftertreatment. The methodology to correct the AFR management is described and validated in Section 4. Finally, the main conclusions of the study are summarized in Section 5.

2. MATERIALS AND METHODS

In order to accomplish the objectives of the current work, a fully instrumented multi-cylinder spark-ignition engine was set up. The current section describes the characteristics of the experimental arrangement, the calculations made from the in-cylinder pressure signal to evaluate the combustion performance,the testing procedure, and the methodology used to analyze the TWC performance.

2.1. Multi-cylinder engine

For this study, a four-stroke 1.3-liter four-cylinder, naturally-aspirated, direct-injection spark-ignition engine has been used. The main engine characteristics are summarized in Table 1. The engine counts with a Variable Valve Timing (VVT) system, capable to change the phasing of both intake and exhaust valve events in a range of 40 crank angle degrees (CAD). The lift profile of the intake valve was optimized to work with an Atkinson cycle by inducing a late intake valve closure strategy. The EGR circuit extracts the exhaust gases from the TWC outlet section and directs them to the intake runners, approximately 120 mm upstream of the connection to the cylinder head, representing 200 mm from the intake valve section. The EGR rate can be controlled in two different ways. First, an electronically controlled valve is located downstream of the EGR cooler, setting the pressure of the EGR gases upstream of the connection to the runners. Then, one fine-step manual valve is placed in the connection to each of the individual cylinders, so that different levels of cylinder-to-cylinder dispersion of the external EGR rate can be set.

The test bench was controlled with AVL-PUMA software, allowing instantaneous speed and engine torque by means of a dynamometric brake AVL AFA 200/4-8EU. Besides, any modification of the intake and exhaust VVT system, the throttle valve position, the fuel injection command, and the spark timing was done via a partially opened ECU, which was equipped with an air flow meter to ensure a proper lambda control when operating with EGR. The main ECU variables were registered through INCA v7.1. Instead, the EGR main valve was controlled in an open loop configuration independently from the ECU based on a National Instruments PXI system, previously described in [21]. The same system was used for high-frequency acquisition. For this purpose, instrumented spark plugs (AVL ZI33) were mounted in all four cylinders, in addition to the instantaneous intake and exhaust manifold pressures, measured in their respective manifolds (Kistler 4007). An AVL optical encoder was installed to provide a crank angle reference for instantaneous pressure measurements with a sampling of 0.2 crank angle degrees.

The emissions were sampled with a HORIBA MEXA-7100D-EGR gas analyzer, which was placed both before and after the aftertreament module. Additionally, intake CO_2 probes were installed upstream of each cylinder, at a 50 mm distance with respect to the EGR injection point, to measure the EGR rate on each cylinder according to the ratio of CO_2 between intake and exhaust. The instrumentation was completed with mean pressure and temperature acquisition in the most critical sections of the exhaust, intake, and cooling systems. The air and fuel mass flows were also acquired through AVL FLOWSONIX and 733S systems, respectively. The complete engine and instrumentation layout are summarized in Figure 1. For all tests, a commercial 98 octane number gasoline fuel was used. Its main properties are summarized in Table 2.

2.2. In-cylinder pressure analysis

Combustion characteristics are evaluated by analyzing the evolution of the in-cylinder pressure. For this purpose, the signal from each cylinder is processed following these steps:

1. The signals from all cylinders are phased so that crank angle zero represents firing TDC position. This position was previously attained based on an engine characterization in motoring conditions.
2. The in-cylinder pressure is pegged to match the instantaneous pressure trace in the intake manifold during a crank angle window represented by an intake valve lift higher than 0.7 mm, using a least squared error criterion.
3. For combustion analysis, a low-pass filter with a cutting frequency of 4.5 kHz is applied to remove high-frequency content associated with resonance effects and/or electrical noise. Instead, a passband filter between 4 and 20 kHz is applied to a window between -20 and 70 CAD aTDC for knock evaluation.

After these steps, the following parameters are calculated for each cylinder and cycle:

- The indicated mean effective pressure (IMEP) on each cylinder and cycle. From this information, the coefficient of variance (COV) is computed as the ratio between the standard deviation and the average values.
- The apparent heat release rate assuming a single-zone model, according to equation (1):

$$HRR = \frac{\mathrm{k}}{\mathrm{k}-1} P_{cyl} \cdot dV + \frac{1}{\mathrm{k}-1} V \cdot dP_{cyl} \qquad (1)$$

Where HRR is the instantaneous apparent heat release rate at a certain crank angle position, Pcyl and V the corresponding in-cylinder pressure and volume (according to a kinematic slider crank mechanism without mechanical deformations) and k the polytropic coefficient, for which values of 1.35 for the intake stroke and 1.3 for the exhaust stroke are imposed, in order to take into account the heat transfer losses associated to the temperature increase after combustion. The HRR is calculated in a window from the spark activation up to the exhaust valve opening so that the constant-mass approach implicit in equation (1) is valid [17].

- The maximum amplitude of the pressure oscillations (MAPO), calculated as the maximum absolute value of the pressure signal after the passband filter, is used as a knock estimator. An acceptable threshold of 0.3 to 1.1 bar is set as a function of the engine speed after analyzing in detail the resonances induced by the main combustion in the in-cylinder pressure signal [27].

2.3. Test matrix and methodology

The impact of EGR dispersion is assessed in a total of 7 partial load steady-state keypoints, ranging from 1500 to 3000 rpm and from 6 to 10 bar BMEP. In all tests, the engine is operating at fully warmed-up conditions, represented by approximately 90ºC at the outlet of both coolant and oil circuits. For each keypoint, the following steps are applied:

I. An EGR sweep was performed by acting on the main EGR valve (electronically commanded) while maintaining the manual EGR valves upstream each cylinder at fully open condition. This sweep was used to find the optimal calibration (EGR rate, VVT and spark) from fuel efficiency perspective for each operating condition.

Table 3 summarizes the optimal EGR rate as well as the limit of EGR rate acceptable assuming a maximum COV of 4% on any cylinder.

II. The main EGR valve was fixed at the value previously set, and dispersion was induced by closing the manual valve individually on each cylinder, until overcoming the COV limit or closing completely the EGR. The spark is set constant and the AFR is managed in closed-loop by the ECU to maintain stoichiometric operation according to the lambda sensor upstream the TWC.

III. If the optimal EGR rate is significantly lower than the maximum corresponding to combustion stability, a second sweep is performed by setting the same EGR rate with a higher opening of the main EGR valve, an intermediate position of the manual valves (equal for all cylinders), and applying further openings of the manual valve on each cylinder so that the EGR rate increases with respect to the optimal value.

After these measurements, the lambda at which each individual cylinder operates is estimated as follows:

- The engine volumetric efficiency is calculated considering the air mass flow and the average EGR rate, pressure and temperature conditions.
- The air mass flow through each cylinder is computed assuming that the volumetric efficiency is constant for all cylinders and independent from the position of the manual valve, but considering the individual EGR rate and intake temperature values.

- The lambda is computed by dividing each individual air mass flow by 1/4 of the measured fuel mass flow and the stoichiometric AFR.

3. ORIGINAL CONTROL STRATEGY

Figure 2 shows the result of the EGR dispersion sweep induced when acting on the manual valve for cylinder #4 for the operating condition of 1750 rpm and 6 bar BMEP. In this figure, the x-axis represents the relative position of this manual valve, considering 0% the value for fully closed and 100% the fully open. The figure includes, from top to bottom, the sensed EGR rate on each cylinder together with the mean value, the combustion phasing characterized by the CA50, the combustion duration (TOC) as the difference between CA10 and CA90 and the combustion stability computed as the coefficient of variation of the IMEP.

As it can be seen, as the manual EGR valve on cyl#4 is closed, the pressure dynamics in the EGR circuit and intake manifold produce that cyl#3 maintaining approximately the original EGR level, while the EGR rate in the other cylinders tends to increase. As a result, the mean EGR value is approximately constant along the whole sweep. Since the spark advance is set constant for all cylinders, the combustion centering and combustion duration show a direct correlation with the in-cylinder EGR rate, while combustion stability shows no clear trend since the optimal EGR rate in this case (23%) was still far enough from the maximum EGR tolerance from combustion stability perspective (32%).

Figure 3 shows the analysis of the tailpipe regulated emissions for the same working conditions. Additionally, different values of lambda are represented. First, the lambda measured from both the lambda sensor upstream the TWC used for the fuel management (labelled ECU) is compared with the lambda computed from the oxygen balance in the emissions bench (labelled Emi). Both values show a good agreement around a value of 1, confirming that on average the air-to-fuel ratio is set at stoichiometric condition. Then, the estimated variation of lambda from each cylinder, calculated according to the methodology described in Section 2.3, is depicted. The results confirm that cyl#4 operates at leaner conditions than the rest of the cylinders, producing an excess of oxygen that leads to a substantial increase in the tailpipe NOx emissions. Instead, the available oxygen results in a complete abatement of the CO emissions, which was not possible in the original scenario. Finally, the unburned hydrocarbon emissions are kept at a low level since the unbalance of lambda does not result in rich operation in any of the cylinders. This is due to the fact that most hydrocarbons have a higher selectivity compared to CO for their oxidation at the exhaust temperature and composition typical of EGR operation.

Figure 4 shows the instantaneous evolution of the NOx and CO emissions, together with the oxygen concentration, for the manual valves positions of 100% (A) and 50% (B). The color of the line is used to distinguish between raw emissions (at engine-out) in red and at the tailpipe in black, while the kind of line is used to distinguish each species in the exhaust line. The recorded data represents 30 seconds of operation with an acquisition rate of 1 Hz.

Comparing both charts, the following aspects can be highlighted:

- In both cases, the raw CO and O_2 emissions show high-frequency variations as a consequence of the lambda management by the ECU. This is performed to ensure a sufficiently high proportion of oxygen arriving to the catalyst and stored so that CO and THC can be oxidized later on. On average, the O_2 concentration upstream the catalyst is similar, while the CO emissions show a significant increase for the 50% opening as a consequence of the higher EGR rate seen on cylinders 1 and 2, as previously analyzed.
- In the case of 100% opening, the tailpipe emissions traces are practically constant despite this high-frequency variation. Particularly, the tailpipe NOx are at a near zero value due to the combined effect of the EGR on the raw emission and the high catalyst conversion efficiency. This results in a lower amount of CO involved in the NOx reduction, so CO conversion efficiency depends mostly on its oxidation with the available oxygen. However, the relatively low concentration of oxygen linked to stoichiometric operation and the higher selectivity for the unburned hydrocarbons explains the remaining CO at the tailpipe [18].
- For the 50% opening of the manual EGR valve the situation is slightly different. Despite the higher raw CO characteristic of this condition, the extra available O_2 coming from the cylinder operating in lean condition helps to produce the complete oxidation of the CO, but inhibits the reduction of NOx. This inhibition is coupled with the dynamics of oxygen storage and consumption in the TWC, producing a low frequency oscillation in both the tailpipe NOx and O_2.

Figure 5 shows again the combustion related data for a manual EGR valve sweep performed on the 2500x8 operating point. In this case, the manual valves for all cylinders were set at 50% opening, and the target average EGR rate (14.5%) was reached with a further opening of the main EGR valve. Starting from this condition, the manual valve of cyl#3 was opened to investigate the effects of dispersion induced by EGR values higher than the optimal one. Increasing the manual valve opening up to 100% the EGR rate in cyl#3 increases up to 25%, while it reduces slightly in the other cylinders, with a higher sensitivity for cyl#1. As a result, maintaining constant spark actuation, the CA50 in the actuated cylinder retards significantly due to the lower reactivity induced by the EGR gases, and combustion stability deteriorates significantly, exceeding the 4% limit in the last point. It has to be noted that this situation is consistent with the data reported in Table 3, where it had been shown that the limit of EGR from combustion stability perspective was slightly over 20%.

The effect of the EGR increase in the exhaust gas composition can be seen in Figure 6. Again, the average lambda sensed both by the sensor connected to the ECU and by the emissions bench is constant at 1. Instead, the estimations from the volumetric efficiency and the EGR rate on each cylinder show a slight level of enrichment in cyl#3 as a consequence of the higher amount of EGR. This enrichment, together with the effect of the higher combustion instability, produces a significant increase of tailpipe CO emissions, which are multiplied by a factor of almost 4. Instead, the THC remains at low values, which is linked to a higher selectivity of oxygen towards hydrocarbons compared to CO in these working conditions. Finally, since the rest of the cylinders work around lambda=1, the NOx emissions are also under control.

Figure 7 represents the tailpipe CO emissions from the complete testing campaign. First, in Figure 7.a, these data are depicted against the ratio between the maximum EGR rate among all cylinders and the limit EGR rate reported in Table 3. In this way, a value greater than 1 represents that the limit EGR rate is overcome in at least one cylinder, while a value lower than 1 indicates that all cylinders work within an acceptable EGR level. As it can be seen, almost every condition that produce a tailpipe CO magnitude over 100 ppm coincides with a ratio higher than one, confirming the correlation between combustion instability and the capability to abate this emission.

The same CO values are depicted against the minimum lambda value among all cylinders in Figure 7.b. In this case, the dispersion of the data is much higher, and it can be seen how conditions with minimum lambda below 0.95 are still successful in oxidizing the CO induced. This is due to the fact that the combustion control is set at lambda=1 overall, so the cylinder working in rich conditions is compensated by one or more cylinders running lean, producing enough oxygen to complete the CO oxidation. This can be seen as a confirmation that tailpipe CO are controlled by the maximum EGR level reached more than by the working lambda on each individual cylinder.

The relationship between tailpipe NOx and the maximum lambda reached on any cylinder is depicted in Figure 8. As it can be seen, for most of the conditions, the maximum lambda is between 1 and 1.04, and the NOx tailpipe emissions are under control. Instead, when this value is overcome there is a clear trend to increase the NOx values reached.

4. CORRECTION OF FUEL MANAGEMENT

From the previous studies it can be concluded that the EGR dispersion can significantly affect the capability of the TWC to abate regulated emissions by two different phenomena: exceeding the EGR tolerance when a cylinder works with a higher EGR rate, affecting mostly the CO emissions, and overleaning when a certain cylinder operates with a lower EGR rate than the average, affecting the NOx. In the case of the excessive EGR rate, a possible mitigation strategy would be to calibrate the engine to operate a lower EGR rate, increasing the margin with respect to the EGR limit, although it would imply some penalty in fuel efficiency. However, the overleaning effect can be more difficult to avoid, since it is not linked to the average level of EGR but to the dispersion itself.

For this reason, a potential solution to the increase of NOx emissions associated to this effect would be to apply a correction factor in the fuel injection management when EGR dispersion appears. This was implemented in the engine test bench by acting on the fuel control inside the ECU in open loop for the same working conditions reported in Figure 3. The results of this actuation on the NOx conversion efficiency can be seen in Table 4. With the standard (closed loop) fuel management, the excess of oxygen produced by the cylinder or cylinders running in lean condition produce a drop of conversion efficiency, which reaches values between 70 and 77%, linked to the level of dispersion induced. Instead, when the fuel was controlled in open loop imposing on each cylinder the fuel quantity estimated from the volumetric efficiency and the individual EGR rate, the NOx conversion efficiency increases to levels around 97%, comparable to the operation of the catalyst without any EGR dispersion induced. This confirms that the deviation in the in-cylinder lambda previously discussed is the mechanism behind the deterioration of the NOx emissions, and that the correction of the fuel management is the correct path to proceed.

The main difficulty for this strategy is the estimation of the EGR rate on each cylinder. The effect of dispersion on TWC performance could be identified based on a NOx sensor downstream, but in the current work the estimation of the individual lambda depended on direct measurement of the intake CO_2 on each cylinder, which is normally not available for the ECU. One possibility would be to estimate the individual EGR based on the measurement of the intake temperature upstream of each intake port, but the sensitivity of this temperature with respect to the EGR rate is relatively low. Another alternative represents the estimation of combustion-related characteristics from the analysis of accelerometer sensors installed close to each cylinder [28]. In this sense, Scocozza et al. [29] showed the capability of a virtual sensor based on the accelerometer signal to accurately predict the peak firing pressure. Posch et al. [30] applied a similar methodology and analyzed also the variability of the peak firing pressure, which was linked to the in-cylinder lambda. This possibility is subject of further investigation in future works by the authors.

5. CONCLUSIONS

In the current paper, an investigation of EGR cylinder-to-cylinder dispersion on exhaust emissions management is performed. For this purpose, a naturally-aspirated spark-ignition engine was set-up with a multi-point injection EGR system. A manual valve was placed upstream the EGR injection point for each cylinder so that different levels of dispersion could be reached. A total of 7 partial-load steady-state keypoints were analyzed by inducing different levels of dispersion to the optimal EGR rate from a previous engine calibration. From this study, the following conclusions can be drawn:

- When the dispersion is a consequence of the reduction of the EGR in a certain cylinder, an increase of the tailpipe NOx can be noticed.
- This increase can be directly correlated to a higher lambda in that individual cylinder, as a consequence of the lower intake temperature and lower level of residuals. This produces an increase in raw NOx emissions and O_2 during the exhaust of that particular cylinder, which cannot be reduced with the available CO.
- The negative effect of dispersion on NOx emissions management can be compensated by correcting the fuel mass command on each cylinder, confirming that the tailpipe NOx are linked to the individual in-cylinder lambda.
- When dispersion is induced by increasing the EGR level on one cylinder, the main risk is that EGR tolerance (defined as the maximum EGR rate that allows to operate with stable combustion) is overcome. In that case, tailpipe CO emissions are substantially increased.
- Unburned hydrocarbon emissions are under control despite the EGR variations thanks to the higher selectivity of the TWC for hydrocarbons when the engine works with high EGR level.

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NOMENCLATURE

aTDC	After top dead center
AFR	Air-fuel ratio
BMEP	Brake mean effective pressure
CA10	Crank angle at 10% cumulative heat release
CA50	Crank angle at 50% cumulative heat release
CA90	Crank angle at 90% cumulative heat release
CAD	Crank angle degree
CO	Carbon monoxide
COV	Coefficient of variance
ECU	Engine control unit
EGR	Exhaust gas recirculation
HEV	Hybrid-electric vehicle
HP-EGR	High-pressure exhaust gas recirculation
HRR	Heat release rate
IMEP	Indicated mean effective pressure
k	Polytropic coefficient
LP-EGR	Low-pressure exhaust gas recirculation
NOx	Nitrogen oxides
O_2	Oxygen molecule
P_{cyl}	In-cylinder pressure
RON	Research octane number
SI	Spark ignition
TDC	Top dead center
THC	Total unburned hydrocarbons
V	Volume
VVT	Variable valve timing

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TABLES

Table 1: Main engine characteristics.

Type	Spark-ignition
Displacement [cc]	1300
Compression ratio	10.6:1
Number of cylinders	4
Stroke [mm]	81.2
Bore [mm]	72
Type of injection	Direct injection (spray-guided)
Camshaft system	Variable Valve Timing (intake and exhaust) with Atkinson intake profile
Aftertreatment	Close-coupled three-way catalyst

Table 2: Fuel properties.

Magnitude	**Unit**	**Value**
Research octane number (RON)	[-]	98
Carbon content	[% mass]	85.98
Hydrogen content	[% mass]	12.09
Oxygen content	[% mass]	1.92
Sulfur content	[mg/kg]	7.83
Lower heating value	[MJ/kg]	42.82

Table 3: Optimal and maximum EGR rate.

Engine operating point [rpm x bar BMEP]	**Optimal EGR [%]**	**EGR limit [%]**
1500x6	23.1	31.5
2500x6	15.3	19.5
1500x8	16.3	17.3
2000x8	19.5	19.9
2500x8	15.5	20.5
3000x8	7.4	11.4
2500x10	5.5	5.5

Table 4: Comparison of NOx conversion efficiency for closed loop and open loop control at 1750X6.

Manual EGR valve	**Closed loop (original)**	**Open loop (original)**
50%	77.2%	98.1%
75%	74.5%	97.7%
100%	70.3%	97.3%

FIGURE CAPTIONS

Figure 1: engine layout and instrumentation.

Figure 2: combustion characteristics for 1750X6 engine point when closing the EGR valve for cyl#4.

Figure 3: analysis of tailpipe emissions for 1750X6 engine point when closing the EGR valve for cyl#4.

Figure 4: time evolution of tailpipe NOx and co emissions and O2 concentration upstream the catalyst for 100% (a) and 50% (b) manual valve position.

Figure 5: combustion characteristics for 2500X8 engine point when opening the EGR valve for cyl#3.

Figure 6: analysis of tailpipe emissions for 2500X8 engine point when opening the EGR valve for cyl#3.

Figure 7: relationship between tailpipe co and the maximum EGR rate (a) and the minimum lambda (b) on each cylinder.

Figure 8: relationship between tailpipe NOx and the maximum lambda on each cylinder.