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In-cylinder Burned Gas Rate Estimation and Control on VVA Diesel Engines

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ABSTRACT

In this paper, we propose a strategy to control the in-cylinder burned gas rate inside the combustion chambers of turbocharged Diesel engines equipped with low pressure EGR loop and VVA actuator. We first design a mean-value model of the in-cylinder composition and validate it on a high frequency reference simulator. Then, we develop the control strategy. It is based on the coordination of existing low-level controllers acting separately on the EGR valve and on the VVA actuator. The objective is to improve transient response, taking into account the responses of the low-level controllers. Supportive simulation results show the relevance of this approach.

INTRODUCTION AND MOTIVATIONS

In this paper, we propose a method to improve the control of premixed combustion modes (LTC: Low Temperature Combustion) for an automotive engine. Using a high Exhaust Gas Recirculation (EGR) rate along with advanced combustion timing allows to reduce the combustion temperature and the air/fuel ratio, and, in turn, decreases Nitrogen Oxides (NO_x) emissions [7]. Following [3], in view of real-life reliability requirements and considering the maximum cooling potential of most engines at LTC operating limit, the engine under consideration in this study is equipped with Low Pressure (LP) EGR loop. The reduced temperature, resulting from the LP EGR, tends to deactivate the oxidation catalyst and then to produce a sharp increase in Hydrocarbons (HC) and Carbon Monoxides (CO) emissions.

To compensate the exhaust temperature reduction, an Internal Exhaust Gas Recirculation (IEGR) can be considered. It can be achieved using a Variable Valve Actuation (VVA) device [3]. The actuator under consideration produces a variable exhaust valve re-opening (see Figure 1). Hot exhaust gases are re-admitted into the cylinder, increasing the mixture temperature, and then allowing activation of the oxidation catalyst in the post-treatment system (which permits to decrease HC and CO emissions).

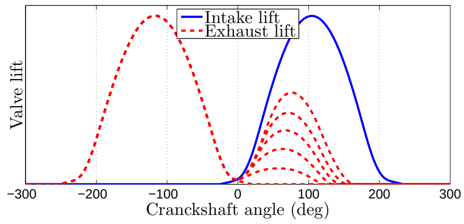

Figure 1: VVA actuator controls the exhaust valve lift re-opening.

On classical diesel engines which do not incorporate any VVA actuator, the in-cylinder Burned Gas Rate (BGR) is defined by the intake manifold BGR (only intake manifold mixture is aspirated into the cylinder).

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Then, controlling the in-cylinder BGR is equivalent to controlling the intake manifold BGR (see [4] for example). By contrast, on engines equipped with VVA actuators, IEGR affects the in-cylinder filling by allowing burned gases to be re-admitted into the cylinder during the intake stroke. Then, controlling intake manifold BGR is not sufficient to control the in-cylinder BGR.

We consider a VVA Diesel engine equipped with LP EGR capability, whose control system includes a low-level intake manifold BGR controller and a low-level VVA actuator controller. These controllers are not detailed in this paper. In closed-loop, the intake manifold BGR subsystem and the VVA subsystem have varying performance (mainly speed of convergence). Their responses can be considered as moderately fast to slow, and, at occasions, they can outperform each other. The intake manifold BGR response is very slow (its time constant being of several seconds) because of the size of the EGR loop. Indeed, contrary to the EGR valve, VVA actuators have an immediate effect on the cylinder filling process. Therefore, the IEGR dynamics is directly impacted by the VVA actuator dynamics (whose time constant is approximately 100 ms).

We propose to coordinate the two low-level control subsystems to improve the overall performance. Because the intake manifold BGR subsystem is slow, it can lead to unacceptable NO$_x$ emissions peaks during transients. It is then possible to adjust the VVA actuator set point to speed-up the in-cylinder BGR dynamics and to limit NO$_x$ emissions peaks. In detail, the controller is based on an in-cylinder BGR mean-value model. The model uses intake and exhaust thermodynamics quantities and is used to reconstruct the in-cylinder composition at the Intake Valve Closing (ivc).

The paper is organized as follows. After presenting the reference simulator and engine setup, we propose a mean-value model of the in-cylinder composition. Then, we expose the control strategy and report simulation results. Finally, we conclude and give future directions.

REFERENCE SIMULATOR

The engine system simulation tool used for this study is the IFP-ENGINE library. It has been developed under the IMAGINE numerical platform AMESim, which is an environment for modeling and simulating dynamical systems (using a Bond Graph approach).

The model under consideration is presented in [1]. Airpath includes a compressor, pipes, heat exchangers, etc. All these elements are represented by dedicated submodels. The combustion chamber is connected to the airpath through the cylinder head which acts thanks to valve lift laws and permeability behavior model derived from experimental characterization. The fuel injection system allows to perform up to three injections per cycle which are controlled with the common rail pressure by means of injection starting time and duration. Cylinder wall heat losses are modeled using a Woschni approach with three independent temperature variables for the cylinder head, the piston and the liner. The combustion heat release model is based on a conventional 0D Diesel combustion model approaches ([6], [2]) extended to multi-pulse injection, auto-ignition delay and EGR effect correction in order to get good combustion behavior over the whole range of operating set points, especially in both Highly Premixed Combustion (HPC) and conventional combustion modes. The airpath of the engine is equipped with an air throttle, an EGR valve, a Variable Geometry Turbocharger and a VVA actuator (see Figure 2).

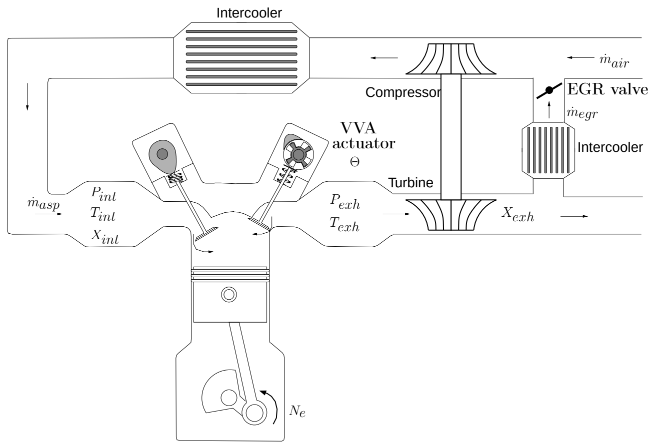
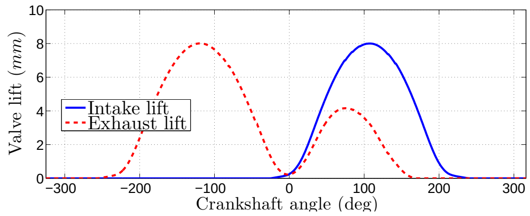


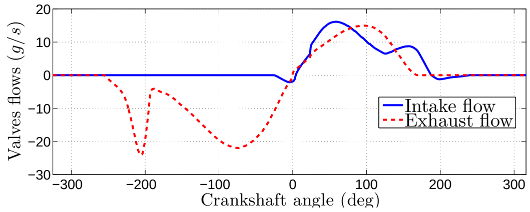
Figure 2: Engine scheme and notation of the considered variables. P, T, X and $\dot m$ stand respectively for pressure, temperature, BGR and mass flow. int and exh mean intake and exhaust manifold.

MODELING

HIGH FREQUENCY ANALYSIS - Figure 3 represents the instantaneous flows going in and out of the cylinder, during exhaust and intake phases. Intake and exhaust valves lifts profiles are given in Figure 3a. VVA actuator position permits a large re-opening of the exhaust valve. Figure 3b shows that, before TDC (Top Dead Center - $0\,\mathrm{deg}$), in-cylinder residual gases are expelled towards the exhaust runner (negative flow). Then, the intake valve opens and the exhaust valve re-opens while the piston starts to go down. Fresh mixture (air + burned gas coming from external EGR) is aspirated into the cylinder through the intake valve. Meanwhile, exhaust gases are re-aspirated into the cylinder, through the exhaust valve.



(a) Valves lifts



(b) Instantaneous cylinder flows

Figure 3: Instantaneous in and out cylinder flows. Intake boost pressure and exhaust back pressure are equal to 1.02 bar and 1.05 bar respectively.

Now, we present a mean-value model for the in-cylinder composition at ivc.

MEAN-VALUE MODEL -

Table 1: Nomenclature

Symbol	Description	Unit
m_{asp}	Aspirated mass from intake	kg
m_{iegr}	Internal gas recirculation mass	kg
m_{ivc}	In-cylinder mass at ivc	kg
$\dot{m}_{air}$	Fresh air mass flow	kg/s
$\dot{m}_{asp}$	Aspirated mass flow from intake	kg/s
$\dot{m}_{egr}$	EGR mass flow	kg/s
N_e	Number of crankshaft revolutions	rpm
P_{int}	Intake manifold pressure	Pa
P_{ivc}	In-cylinder pressure at ivc	Pa
P_{exh}	Intake manifold pressure	Pa
R	Ideal gas constant	$J/kg/K$
V_{evc}	Cylinder volume at evc	m^3
T_{int}	Intake manifold temperature	K
T_{ivc}	In-cylinder temperature at ivc	K
T_{exh}	Intake manifold temperature	K
Θ	VVA actuator position	deg
X_{int}	Intake manifold burned gas rate	-
X_{ivc}	In-cylinder burned gas rate	-
X_{exh}	Fuel/air ratio	-

A nomenclature is given in Table 1. To model the in-cylinder BGR at ivc, X_{ivc}, we need to estimate the in-cylinder air and burned gas masses. Let m_{ivc} be the in-cylinder total mass at ivc. Let m_{asp} be the aspirated mass from the intake, consisting of fresh air and burned gases coming from external EGR (see Figure 2). Finally, let m_{iegr} be the recirculated gas mass coming from exhaust valve re-opening, consisting of burned gases and unburned air from the previous combustion. Neglecting residual gases trapped in the cylinder, the in-cylinder BGR writes

$$X_{ivc} = \frac{X_{int} m_{asp} + X_{exh} m_{iegr}}{m_{ivc}} \quad (1)$$

where X_{int} and X_{exh} are the BGR in the intake and exhaust manifolds. X_{int} can be estimated using an observer (see, e.g. [5]). We will not give any detail of such an observer in this paper and, in the sequel, we consider X_{int} as a known variable. X_{exh} is measured by an oxygen sensor situated at the exhaust of the engine (fuel/air ratio sensor).

Now, let us focus on the model of m_{asp}, m_{iegr} and m_{ivc}. Experimentally, the composition of the cylinder at ivc is very difficult to measure. However, one can obtain the aspirated mass under steady-states conditions. Indeed, a mass air flow meter situated at the intake of the engine measures the aspirated fresh air, $\dot{m}_{air}$. It is also possible, at a test bench, to measure the mass flow of the EGR loop, $\dot{m}_{egr}$. Then, the reference aspirated mass flow from the intake is obtained by

$$\dot{m}_{asp} = \dot{m}_{air} + \dot{m}_{egr}$$

We integrate the signal over one period $\Delta t = \frac{2\times 60}{N_e n_{cyl}}$ (n_{cyl} is the number of cylinders). The factor 2 is due to the fact that air is aspirated only every other cycle of a four stroke process and the factor 60 is due to the fact that N_e expresses in rpm (see [8]). Because $\dot{m}_{asp}$ is constant over one period, integration leads to

$$m_{asp} = (\dot{m}_{air} + \dot{m}_{egr})\Delta t \quad (2)$$

Model presentation - Following [9], m_{asp} is the reference variable for our in-cylinder composition model. Let $\hat{m}_{ivc}$, $\hat{m}_{asp}$ and $\hat{m}_{iegr}$ be respectively the estimates of m_{ivc}, m_{asp} and m_{iegr}. Then, from (1), an estimate of the in-cylinder BGR is

$$\hat{X}_{ivc} = \frac{X_{int} \hat{m}_{asp} + X_{exh} \hat{m}_{iegr}}{\hat{m}_{ivc}} \quad (3)$$

The aspirated mass equals the difference between the in-cylinder total mass and the recirculated gases mass

$$\hat{m}_{asp} = \hat{m}_{ivc} - \hat{m}_{iegr} \quad (4)$$

Figure 4 illustrates the mean-value model.

- The in-cylinder mass at ivc is given by the perfect gas law

$$m_{ivc} = \frac{P_{ivc} V_{ivc}}{R T_{ivc}} \quad (5)$$

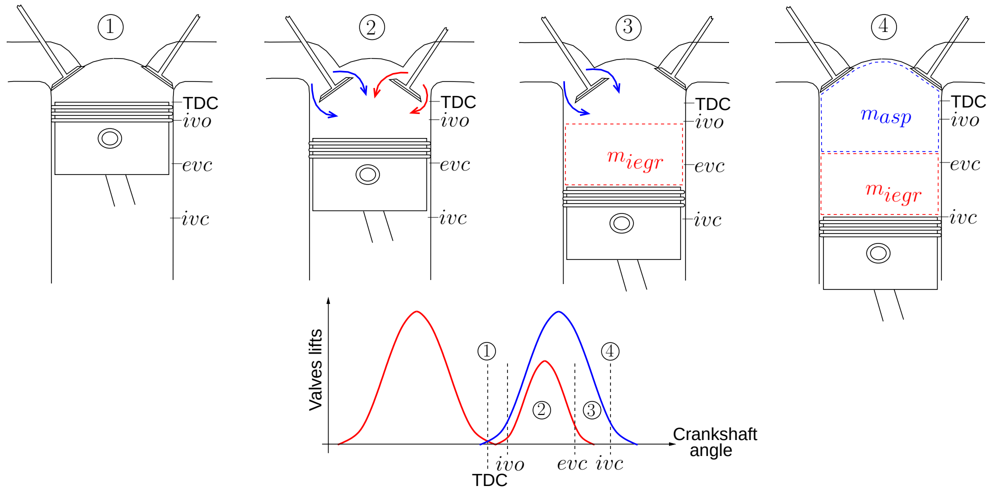


Figure 4: In-cylinder filling process. 1: piston is at TDC, end of the exhaust stroke. 2: piston starts to go down, both intake and exhaust valve open, gases are admitted from both sides. 3: exhaust valve closes, the rest of the cylinder volume is filled with intake mixture (fresh air and burned gases from EGR). 4: end of the intake stroke, total in-cylinder mass of gas at ivc, m_{ivc}, is equal to the aspirated mass from the intake, m_{asp}, plus the mass of gas coming from IEGR, m_{iegr}.

where P_{ivc}, V_{ivc}, T_{ivc} are respectively the in-cylinder pressure, volume and temperature at ivc. R is the ideal gas constant. One can easily assume that the pressure remains at equilibrium between intake manifold and cylinder at the end of the intake process ($P_{int} \simeq P_{ivc}$), and, further, consider a single constant value for R. Then, (5) becomes

$$\hat{m}_{ivc} = \alpha_1 \frac{P_{int} V_{ivc}}{R\hat{T}_{ivc}} \tag{6}$$

where α_1 is a correcting term which is a function of the intake manifold pressure and the engine speed. The estimate for the temperature at ivc, $\hat{T}_{ivc}$, is presented later on.

- A quasi-static flow from the exhaust towards the cylinder is assumed from TDC to evc (Exhaust Valve Closing). Then, the recirculated gases mass writes

$$\hat{m}_{iegr} = \alpha_2 \frac{P_{exh} V_{evc}(\Theta)}{RT_{exh}} \tag{7}$$

where α_2 is a correcting term function of the intake manifold pressure and the engine speed (it corresponds to the part of the intake mixture admitted during the exhaust re-opening phase). P_{exh} and T_{exh} are respectively the exhaust pressure and temperature. Finally, V_{evc} is the in-cylinder volume at evc, it is a function of VVA actuator position, Θ. Reader can refer to [7] for the computation of in-cylinder volume V.

Replacing (6) and (7) in (4) leads to

$$\hat{m}_{asp} = \alpha_1 \frac{P_{int} V_{ivc}}{R\hat{T}_{ivc}} - \alpha_2 \frac{P_{exh} V_{evc}(\Theta)}{RT_{exh}} \tag{8}$$

Neglecting the difference in heat capacities, the in-cylinder temperature at ivc is approximated as the weighted temperature of the mixing aspirated masses from the intake and from the exhaust [11],

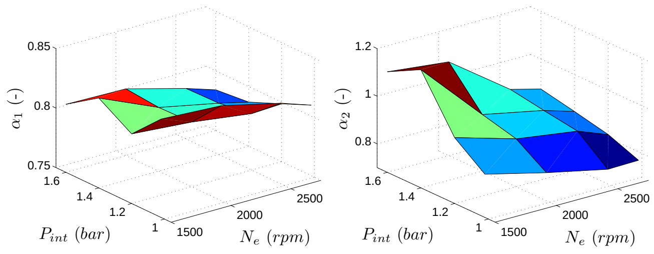
$$\hat{T}_{ivc} = \frac{T_{int}\hat{m}_{asp} + T_{exh}\hat{m}_{iegr}}{\hat{m}_{ivc}}$$

Replacing $\hat{m}_{ivc}$ and $\hat{m}_{iegr}$ by their expressions (6) and (7), one obtains

$$\hat{m}_{asp} = \alpha_1 \frac{P_{int} V_{ivc}}{RT_{int}} - \alpha_2 \frac{P_{exh} V_{evc}(\Theta)}{RT_{int}} \tag{9}$$

<u>Model results</u> - Model (9) has been calibrated on a set of approximately 600 points (engine speed ranging from 1500 to 2750 rpm, intake manifold pressure ranging from 1 to 1.6 bar). For each operating point (engine speed / intake manifold pressure), VVA actuator position covers all the operating range. The two look-up tables α_1 and α_2 are determined as follows. Each point of the look-up tables (constant intake manifold pressure and constant engine speed), are obtained by fitting model (9) to reference data m_{asp}, through a standard linear regression. Look-up tables α_1 and α_2 are presented in Figure 5. Their relative smoothness and small variations prove the consistency of the model structure (9).

Figure 6 presents a comparison between the developed mean-value model and the high-frequency refer-



(a) α_1 look-up table (b) α_2 look-up table

Figure 5: Optimized model parameters α_1 and α_2.

ence simulator presented above. Figure 6a shows linear regression results on the aspirated mass. A good estimation is obtained on all the considered range. The three following figures give estimation results on the recirculated gas mass (Figure 6b), the in-cylinder mass at ivc (Figure 6c), and the in-cylinder temperature at ivc (Figure 6d). Comparison with reference simulator data show the good behavior of the model. Indeed, *while parameters α_1 and α_2 are obtained by minimizing the cost $(m_{asp} - \hat{m}_{asp})^2$, a good estimation of the distribution m_{ivc} / m_{iegr} is obtained as well*. Consequently, the in-cylinder BGR estimation presented in Figure 6e is good.

Model robustness - In-cylinder BGR model (3) is based on exhaust pressure and temperature measurements. To take into account measurement biases (which are expected with commercial-line sensors, or any exhaust pressure or temperature models (as proposed in [11] or [10]) for example), we introduce (constant) biases of 0.1 bar and 50 K and investigate robustness. Figure (7) reports model results in the presence of exhaust manifold pressure and temperature biases. Observed modeling errors remains within acceptable ranges (lower than 3% absolute error).

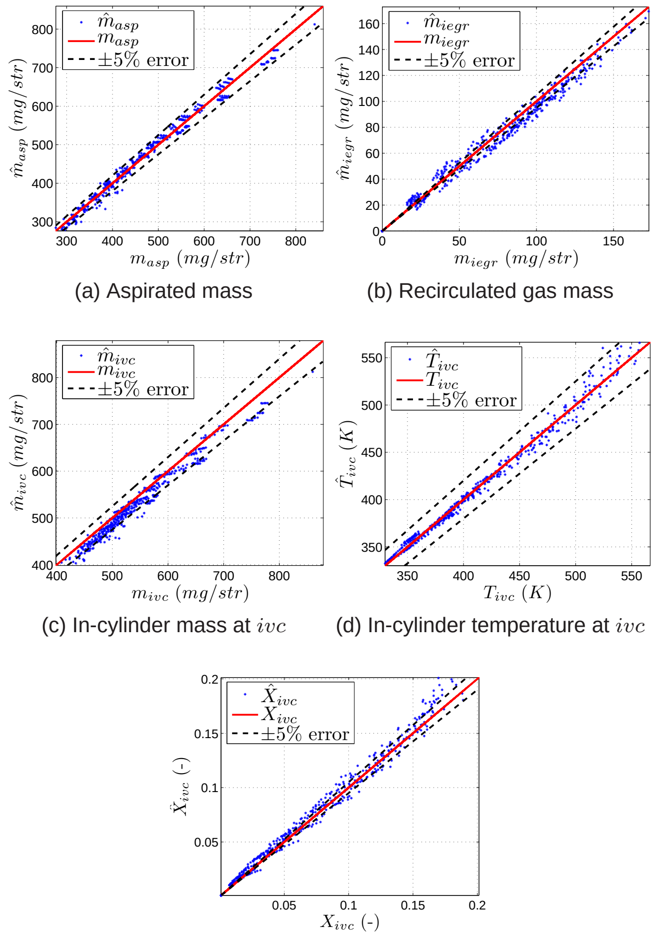
IN-CYLINDER BURNED GAS RATE CONTROL

CONTROL STRATEGY - Our control objective is to speed-up the in-cylinder BGR transients by coordinating the two low-level controllers (intake manifold BGR and VVA actuator position).

Notations - Let $\Omega_\Theta = [\underline{\Theta}; \overline{\Theta}]$ be the set of feasible VVA actuator positions. Let Ω_X be the set of considered BGR (either in the intake manifold or in the cylinder). Referring to model (3), we note $f : \Omega_X \times \Omega_\Theta \to \Omega_X$ such that

$$\hat{X}_{ivc} \triangleq f(X_{int}, \Theta) \tag{10}$$

Due to the structure of the analytic expressions (3) and (7), and the one-to-one property of function V_{evc}, the following inversion assumption holds.

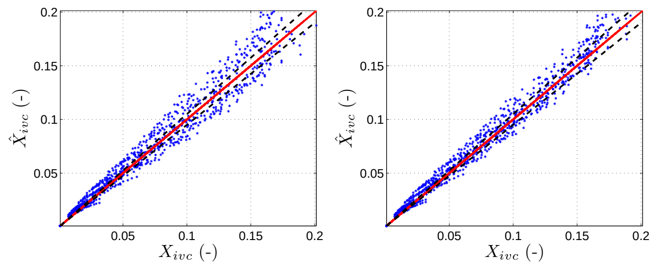


(a) Aspirated mass (b) Recirculated gas mass

(c) In-cylinder mass at ivc (d) In-cylinder temperature at ivc

(e) In-cylinder BGR at ivc

Figure 6: Mean-value model comparison with the high-frequency reference simulator.



(a) $\Delta P_{exh} = \pm 0.1$ bar (b) $\Delta T_{exh} = \pm 50$ K

Figure 7: Model robustness in relation to exhaust pressure and temperature variations.

Assumption 1 *For all $(X_{int}, \hat{X}_{ivc}) \in \Omega_X^2$, there exists a unique $\Theta \in \Omega_\Theta$ such that $f(X_{int}, \Theta) = \hat{X}_{ivc}$.*

From the above assumption, inverse function of f can be defined. We note $f^{-1}_{X_{int}} : \Omega_X \to \Omega_\theta$ such a function. Then, for all $(X_{int}, \Theta) \in \Omega_X \times \Omega_\Theta$, the following relationship holds.

$$\hat{X}_{ivc} = f(X_{int}, \Theta = f^{-1}_{X_{int}}(\hat{X}_{ivc})) \quad (11)$$

It means that we are able to find a VVA actuator position, at any intake BGR, so that the in-cylinder BGR is satisfied. In practice, VVA actuator can only admit bounded values. The inversion formula used in (11) may produce infeasible values that need to be saturated before they can be used as input signals to the VVA control system. This point is addressed in the next section.

Controller design - Our contribution is the in-cylinder BGR control scheme pictured in Figure 8. It is contained in block A which simultaneously feeds two low-level controllers, the intake BGR controller in block B and the VVA controller in block C. From a torque requested by the driver through the accelerator pedal, a static look-up table is used to compute an in-cylinder BGR set point, X^{sp}_{ivc}. Physically, the in-cylinder BGR depends on the intake BGR and the VVA actuator position (see model (10)). Block A computes corresponding intake BGR and VVA set points, X^{sp}_{int} and Θ^{sp}. The closed-loop (from block B to A) feeds back the current values of the intake manifold BGR.

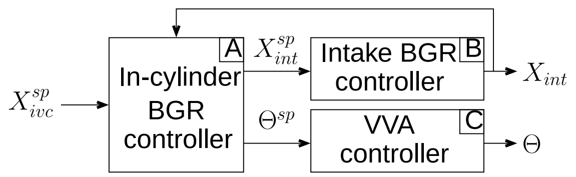


Figure 8: In-cylinder BGR control scheme. Block A contains the contribution of the paper. Blocks B and C are the low-level intake manifold BGR and VVA controllers.

We now give an explicit procedure to compute both intake manifold BGR and VVA actuator position set points in order to reach the desired amounts of in-cylinder BGR.

The in-cylinder BGR set point, X^{sp}_{ivc}, is turned into physical variables set points on interest : an intake manifold BGR, X^{sp}_{int} and a VVA actuator position, Θ^{sp}. To maximize the control performance, in terms of in-cylinder BGR tracking, it is important to notice that intake manifold BGR control is very slow compared to internal gas recirculation (because of the EGR loop sizing). Thus, we propose to use the inverse model derived above to compute set points for the low-level controllers. To account for the respective time responses of the corresponding subsystems, we coordinate them by replacing target set point by real-time measurement.

The objective of this paper is to show the potential impact of in-cylinder BGR control using VVA actuators in transients. Thus, we impose no re-opening of the exhaust valve ($\Theta = 0$) under steady-states conditions. Consequently, it follows

$$X^{sp}_{int} = X^{sp}_{ivc} \quad (12)$$

From an in-cylinder BGR set point, we compute a VVA actuator position set point, Θ^{sp}, using equation (11). In this formula, *we use intake manifold BGR measurement*. It gives

$$\Theta^{sp} = \psi\left(f^{-1}_{X_{int}}\left(X^{sp}_{ivc}\right)\right) \quad (13)$$

where ψ is the saturation function. $\psi(x) = x$ if $x \in \Omega_\Theta$, $\psi(x) = \underline{\Theta}$ when $x < \underline{\Theta}$ and $\psi(x) = \overline{\Theta}$ when $x > \overline{\Theta}$.

From (11), it appears that, provided that the VVA actuator position is not saturated, the observed in-cylinder BGR reaches its set point.

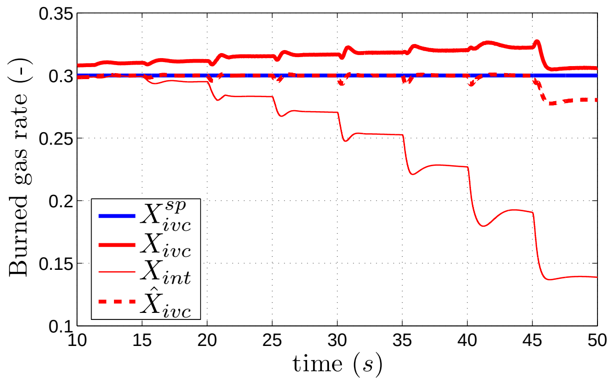
Explicitly, using (3) and (7), the implemented open-loop control law (13) writes

$$\Theta^{sp} = \psi\left(V^{-1}_{evc}\left(\frac{RT_{exh}}{\alpha_2 P_{exh}}\frac{X^{sp}_{ivc}\hat{m}_{ivc} - X_{int}\hat{m}_{asp}}{X_{exh}}\right)\right) \quad (14)$$

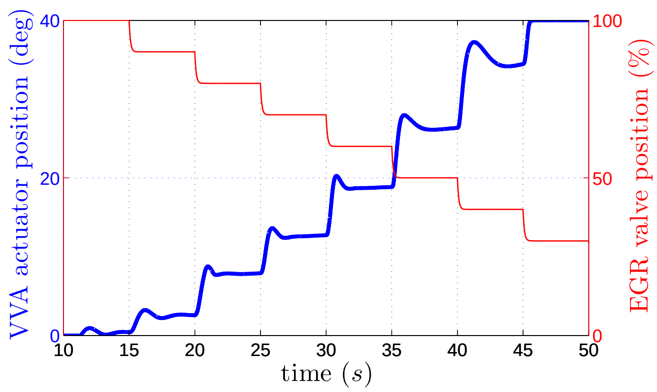
SIMULATION RESULTS - Control strategy (14) has been implemented and tested on the reference simulator.

Intake manifold BGR variations - The first scenario under consideration in Figure 9 consists of requesting a constant in-cylinder BGR set point, changing EGR valve position.

Figure 9a represents the in-cylinder BGR set point, X^{sp}_{ivc} (constant), the high-frequency reference simulator in-cylinder BGR, X_{ivc}, the intake manifold BGR, X_{int} (estimated by an observer presented in [5]), and the estimated in-cylinder BGR given by model (3), $\hat{X}_{ivc}$. Figure 9b gives airpath actuators positions (EGR valve and VVA actuator). Every $5s$ step, we decrease EGR valve opening area, the consequence is that intake manifold BGR decreases. If no control strategy acts on VVA actuator position, in-cylinder BGR should also decrease (one should keep in mind that the intake manifold BGR is equal, on engines not equipped with VVA, to in-cylinder BGR). However, we succeed in keeping the in-cylinder burned gas rate to its set point (see control-law (14)). This can be explained by the fact that the open-loop control strategy (14) uses intake manifold BGR information to compute VVA actuator position set point. As we can see in Figure 9b, VVA actuator is positioned in such a way that $\hat{X}_{ivc}$ equals to X^{sp}_{ivc}. One can observe that the VVA actuator position follows the intake BGR dynamics so that the in-cylinder BGR no overshoots. After $45s$, intake manifold BGR is so low that internal recirculation cannot answer to in-cylinder BGR demand (VVA actuator saturates). This simulation proves that in-cylinder BGR control strategy is efficient. It also validates the mean-value model behavior (comparing $\hat{X}_{ivc}$ and X_{ivc}).



(a) Burned gas rate



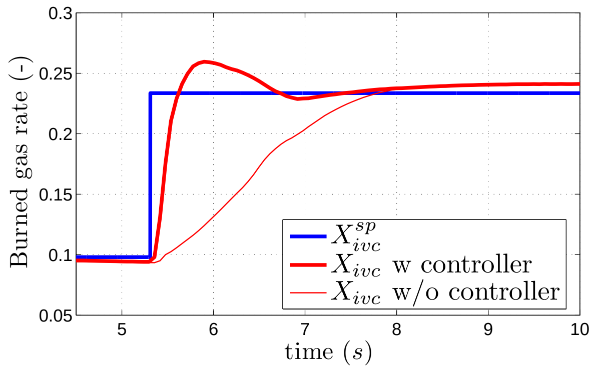
(b) VVA actuator position

Figure 9: Simulation results on a variation of the intake manifold BGR, keeping constant in-cylinder BGR by controlling the exhaust valve re-opening.

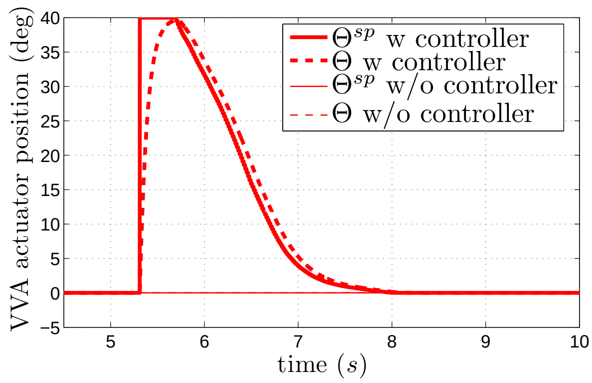
Load transient - The transient response observed in Figure 10 is particularly interesting. In this scenario, a torque transient is requested by the driver (from 1 to 3 bar of IMEP - Indicated Mean Effective Pressure). This defines an increase in the in-cylinder BGR set-point. In Figure 10a, two simulations are compared. One of them uses the presented controller and the other one does not (no VVA actuator control). Figure 10b gives VVA actuator set point Θ^{sp} and position Θ for both cases.

In the first case, our proposed controller coordinates both the intake manifold BGR and the VVA controllers. VVA actuator position quickly changes to take into account intake manifold BGR lag. It results in a very fast in-cylinder BGR response. The small overshoot arises from an error in the intake manifold BGR observer that under-estimates intake BGR during transient.

In-cylinder BGR response is much faster when we use the proposed transient control strategy of VVA actuator position. In that kind of transient, we can expect to substantially reduce NO_x emissions peaks.



(a) Burned gas rate



(b) VVA actuator position

Figure 10: Simulation results on a load transient (1 to 3 bar of IMEP). Dynamic control of the VVA actuator permits to highly accelerate the in-cylinder BGR response.

CONCLUSION AND FUTURE WORK

The paper presents a control strategy to manage the in-cylinder BGR on engines equipped with VVA actuator. We first present a mean-value model of the in-cylinder composition. It permits, without any in-cylinder variable sensor, to estimate the aspirated mass from the intake, the in-cylinder total mass and the temperature at ivc, the recirculated gas mass, and the in-cylinder burned gas rate. A control strategy has been proposed. The method consists of computing intake manifold BGR and VVA actuator set points to coordinate the low-level controllers. Experimental results obtained on a reference simulator stress the relevance of the control strategy.

This study presents a methodology that can be applied on Diesel engines equipped with VVA actuators. The following step will be an experimental validation at test bench, carefully looking at NO_x emissions to evaluate the potential of the proposed method. In order to incorporate this work on a more complete engine control strategy, we are working on how the systems VVA-EGR-turbocharger are coupled.

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ACRONYMS

BGR	Burned Gas Rate
EGR	Exhaust Gas Recirculation
evc	Exhaust Valve Closing
HPC	Highly Premixed Combustion
IEGR	Internal Exhaust Gas Recirculation
IMEP	Indicated Mean Effective Pressure
ivc	Intake Valve Closing
ivo	Intake Valve Opening
LP	Low Pressure
LTC	Low Temperature Combustion
TDC	Top Dead Center
VVA	Variable Valve Actuation

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