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Transient Burned Gas Rate Control on VVA equipped Diesel Engines

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Abstract: This paper addresses the problem of cylinder burned gas rate transient control of VVA equipped Diesel engines. This problem is of importance for NOx emissions reduction. The proposed strategy coordinates existing low-level intake manifold burned gas rate and VVA controllers. To determine relevant control actions, a model of cylinder filling phenomenon is determined. It expresses, under the form of a discrete event dynamics, the behavior of the cylinder burned gas rate at intake valve closure. Interestingly, the model is invertible, thanks to its analytical nature. This property directly suggests an inverse formula that serves as control law. The strategy is efficient as is highlighted by experimental results obtained on a 1.6 L Diesel engine test bench. As a result, NOx emissions are largely diminished during transients.

Keywords: Diesel engine, NOx emissions, Burned Gas Rate control, VVA, Model-based control

1. INTRODUCTION

Upcoming Diesel engines emission standards such as Euro 6 require significant reduction of smoke, HC (Hydrocarbons), CO (Carbon monoxide) and above all NOx (Nitrogen Oxides) emissions. This has spurred an interest in new (cleaner) combustion modes. Usually, the EGR can be realized in two possible manners: through a High Pressure (HP) or a Low Pressure (LP) EGR circuit. The LP EGR technology (which is considered in this paper, as pictured in Figure 1) possesses many advantages over the HP EGR circuit. In this technology, the exhaust gases are picked up downstream the turbine and the after-treatment system. The fresh air and burned gases mix and then flow through the compressor. These modes consist of incorporating large amounts of Exhaust Gas Recirculation (EGR) into the combustion process. The premixed combustion modes (LTC: Low Temperature Combustion) represent a relatively simple and affordable solution to reduce the emissions while retaining the advantages of the Diesel engine. These modes of incorporating large amounts of EGR into the combustion process have been largely studied in the literature, see, e.g., Heywood (1988) or Toda et al. (1976) for example). In the engine under consideration, there are two ways of admitting the recirculated gases inside the cylinder. On the one hand, the LP EGR circuit permits to manage the BGR in the intake manifold by actuating both the EGR and the exhaust valves (see Jankovic and Kolmanovsky (1998), Chauvin et al. (2008)). Because of the length of the EGR circuit, the intake manifold BGR has a large response time (see Throop and Cook (1985)). On the other hand, the VVA actuator has a fast impact on the cylinder BGR thanks to its location in the vicinity of the cylinder.

As has been discussed previously, the LP EGR provides good results in terms of NOx emissions. Nevertheless, a reduced exhaust temperature resulting from the highly diluted combustion tends to produce a sharp increase in HC and CO emissions at part load because of after treatment catalyst deactivation (see Bression et al. (2008)). The VVA actuator permits to address this problem. Indeed, the variable valve actuation system traps the gases directly inside the combustion chamber (Internal Exhaust Gas Recirculation, IEGR), therefore limiting the heat exchanges. Hence, significantly higher gas temperature levels at valve closure can be reached. This permits to activate the catalyst, and, in turn, to reduce the HC and CO emissions.

In Diesel engines, the torque production control is achieved by varying the amount of fuel injected during each cycle. The NOx emissions is controlled by modulating the Burned Gas Rate (BGR) of the cylinder mixture before combustion (links between BGR and NOx emissions have been largely studied in the literature, see, e.g., Heywood (1988) or Toda et al. (1976) for example). In the engine under consideration, there are two ways of admitting burned gases inside the cylinder. On the one hand, the LP EGR circuit permits to manage the BGR in the intake manifold by actuating both the EGR and the exhaust valves (see Jankovic and Kolmanovsky (1998), Chauvin et al. (2008)). Because of the length of the EGR circuit, the intake manifold BGR has a large response time (see Throop and Cook (1985)). On the other hand, the VVA actuator has a fast impact on the cylinder BGR thanks to its location in the vicinity of the cylinder.
We propose a strategy to coordinate these two sources of recirculated burned gases, such that the cylinder filling is optimized, even during transients. This is the main contribution of the paper.

The paper is organized as follows. In Section 2, we expose existing control strategies and stress their limitations. Our proposed coordination controller is outlined. It relies on a cylinder filling model which is presented in Section 3. This model has invertibility properties which are exploited in the control design of Section 4. Finally, experimental results are reported in Section 5.

2. CLASSICAL STRATEGY AND PROPOSED ADVANCED STRATEGY

2.1 Existing controllers

In classical VVA-equipped Diesel engine control strategies, a trade-off between external EGR and IEGR is formulated to minimize pollutant emissions at each engine operating point. In practice, this defines a VVA actuator position and an intake manifold BGR value. In other terms, from a torque request specified by the driver through the accelerator pedal, and from the engine speed, two-dimensional look-up tables give set points for the intake manifold BGR and the VVA actuator position controllers (see Figure 2). This model has invertibility properties which are exploited in the control design of Section 4. Finally, experimental results are reported in Section 5.
3.1 Aspirated masses model

The quantities of aspirated masses from the intake \( N_{int}^{asp} \) (which is composed of fresh air and burned gases coming from external EGR), and from the exhaust \( N_{exh}^{asp} \) (which is composed of fresh air and burned gases coming from IEGR) are determined by the thermal conditions of the intake and exhaust manifolds (pressure, \( P_{int} \) and \( P_{exh} \), and temperature, \( T_{int} \) and \( T_{exh} \)), the engine speed, \( N_e \), and the positions of the VVA actuator, \( \Phi_{vva} \). Detailed model has been presented in Leroy et al. (2009). It writes

\[
\begin{align*}
N_{int}^{asp} &= \alpha_1 P_{int} V_{vva} - \alpha_2 P_{exh} V_{vva}(\Phi_{vva}) \\
N_{exh}^{asp} &= \alpha_2 P_{exh} V_{vva}(\Phi_{vva})
\end{align*}
\]  

(1)

where \( \alpha_1 \) and \( \alpha_2 \) are known functions of \( N_e \) (this dependency is omitted hereafter for sake of conciseness). \( R \) is the ideal gas constant. \( V_{vva} \) is the cylinder volume at intake valve closing (ivc). Similarly, \( V_{vva} \) is the cylinder volume at exhaust valve closing (evc). It is a function of the VVA actuator position, \( \Phi_{vva} \) (see Figure 3(b)).

3.2 Cylinder BGR and temperature model

**Conditions at intake valve closing (ivc)** The cylinder BGR at ivc is equal to the mass-weighted average of the BGR of the flows contributing to the total trapped mass

\[
X_{ivc}(k) = \frac{X_{int}(k) N_{int}^{asp}(k) + X_{exh}(k) N_{exh}^{asp}(k)}{N_{int}^{asp}(k) + N_{exh}^{asp}(k)}
\]  

(2)

Similarly, we assume that the cylinder temperature at ivc is equal to the mass-weighted average

\[
T_{ivc}(k) = \frac{T_{int}(k) N_{int}^{asp}(k) + T_{exh}(k) N_{exh}^{asp}(k)}{N_{int}^{asp}(k) + N_{exh}^{asp}(k)}
\]  

(3)

**Exhaust BGR and temperature models** The models are detailed in Appendix A. For sake of conciseness, we introduce the mappings \( \xi \) and \( \theta \) such that the exhaust BGR and temperature given by models (A.2) and (A.3) write

\[
\begin{align*}
X_{exh}(k) &= \xi(X_{ivc}(k-1), T_{ivc}(k-1)) \\
T_{exh}(k) &= \theta(T_{ivc}(k-1))
\end{align*}
\]  

(4)

The discrete-time nature of system (4) stems from the valves closings which isolate the cylinder from the rest of the engine. Exhaust variables are dependent to the cylinder variables of the previous combustion.

3.3 Control-oriented rewriting of the model

Let \( f \) be a mapping representing the cylinder fraction of gases coming from the IEGR. Using equations (1) and (4), it comes

\[
f(\Phi_{vva}(k), \theta(T_{ivc}(k-1))) \triangleq \frac{N_{exh}^{asp}(k)}{N_{int}^{asp}(k) + N_{exh}^{asp}(k)}
\]  

(5)

Using mapping (5) into models (2) and (3) leads to the following cascade system

\[
\begin{align*}
X_{ivc}(k) &= f(\Phi_{vva}(k), \theta(T_{ivc}(k-1)))
\quad \cdot \xi(X_{ivc}(k-1), T_{ivc}(k-1)) \\
& \quad + (1 - f(\Phi_{vva}(k), \theta(T_{ivc}(k-1)))) X_{int}(k) \\
T_{ivc}(k) &= f(\Phi_{vva}(k), \theta(T_{ivc}(k-1))) \theta(T_{ivc}(k-1)) \\
& \quad + (1 - f(\Phi_{vva}(k), \theta(T_{ivc}(k-1)))) T_{int}(k)
\end{align*}
\]  

(6)

Owing to the structure of \( V_{vva} \) as function of \( \Phi_{vva} \), the function \( f \) defined in (5) and appearing as a weight

Fig. 4. Experimental IMEP transient (3-5 bar) at test bench on a 4-cylinders Diesel engine equipped with VVA actuator and LP EGR capability. The difference in response times between the intake manifold BGR and the VVA actuator leads to NO\textsubscript{x} emission peak during transient.

**2.3 Proposed advanced strategy**

We propose a controller specifically designed to handle transients of VVA equipped Diesel engines. This controller focuses on compensating the difference of response times of the various dynamics of the system. We use the VVA in a very active way.

Our control objective is the cylinder BGR (which is an image of the NO\textsubscript{x} emissions). The strategy we propose coordinates the two low-level control subsystems (intake manifold BGR and VVA actuator) to improve the cylinder BGR response. When the intake manifold BGR is too slow, the controller adjusts the VVA actuator position set point to speed up the air-feeding process, so that the cylinder BGR is satisfied. Explicitly, set point for the low-level VVA controller is determined in real time, based on estimation of the intake manifold BGR (several observers can be used to estimate this variable, as are proposed in Chauvin et al. (2008) or Wang (2008)).

3. SYSTEM MODELING

The control synthesis we propose is model-based. First, a cylinder filling model gives an estimation of the aspirated masses into the cylinder (from both the intake and the exhaust valves). Based on this analytic model, a dilution model permits to compute the cylinder BGR and temperature variables. As these dilution models require unmeasured exhaust manifold quantities (BGR and temperature), only a simple combustion model is derived.

3.1 Aspirated masses model

The discrete-time nature of system (4) stems from the valves closings which isolate the cylinder from the rest of the engine. Exhaust variables are dependent to the cylinder variables of the previous combustion.
for the balance equations (6) possesses some interesting invertibility properties. The following partial-invertibility property holds.

**Proposition 1.** For all \((T_{\text{ivc}}, X_{\text{egr}})\), there exists a unique \(\Phi_{\text{vva}}\) such that \(f(\Phi_{\text{vva}}, \theta(T_{\text{ivc}})) = X_{\text{egr}}\). We define the inverse function \(\mathcal{F}\) such that for all \(T_{\text{ivc}}\),

\[
f(\mathcal{F}(X_{\text{egr}}, \theta(T_{\text{ivc}})), \theta(T_{\text{ivc}})) = X_{\text{egr}}
\]

(7)

In practice, a VVA actuator can only admit bounded values. The inversion formula appearing in (7) may produce infeasible values that need to be saturated before they can be used as input signals to the VVA control system.

### 4. CONTROL STRATEGY

#### 4.1 Cylinder set points

Our control objective is to track a set point for the cylinder BGR. We define \(X_{\text{ivc}}^{sp}\) and \(T_{\text{ivc}}^{sp}\) as the set points for the cylinder BGR and temperature, respectively. They are computed from model (6), using experimentally derived look-up tables (giving set points for the intake manifold BGR and VVA actuator position, see Section 2.1). Then, the set points are obtained by solving the following equations

\[
\begin{align*}
X_{\text{ivc}}^{sp} & = f(\Phi_{\text{vva}}^{sp}, \theta(T_{\text{ivc}}^{sp}))(\xi(X_{\text{ivc}}^{sp}, T_{\text{ivc}}^{sp})
+ (1 - f(\Phi_{\text{vva}}^{sp}, \theta(T_{\text{ivc}}^{sp})))X_{\text{int}}^{sp} \\
T_{\text{ivc}}^{sp} & = f(\Phi_{\text{vva}}^{sp}, \theta(T_{\text{ivc}}^{sp}))(\theta(T_{\text{ivc}}^{sp})
+ (1 - f(\Phi_{\text{vva}}^{sp}, \theta(T_{\text{ivc}}^{sp})))T_{\text{int}}
\end{align*}
\]

(8)

The solution of these two equations can be analytically obtained because the system (8) is triangular. In practice, one can first compute the cylinder temperature set point \(T_{\text{ivc}}^{sp}\) by solving the second equation of system (8), using the function \(f\) defined in (5) and the aspirated masses model (1). Then, one can compute the cylinder BGR set point \(X_{\text{ivc}}^{sp}\) by substituting the obtained value of \(T_{\text{ivc}}^{sp}\) in the first equation of (8). In real applications, measurements are used in these computations (intake manifold pressure and temperature, exhaust manifold pressure and engine speed).

#### 4.2 Coordination strategy

We propose to use the inverse model (7) to compute set points for the low-level controllers. From a cylinder BGR set point \(X_{\text{ivc}}^{sp}\), we compute a VVA actuator position set point \(\Phi_{\text{vva}}^{sp}\) by inverting the dynamic model (6), using the function \(\mathcal{F}\) defined in (7). In this formula, we use an estimate of the intake manifold BGR, \(X_{\text{int}}^{sp}\). This gives

\[
\Phi_{\text{vva}}^{sp}(k) = f(X_{\text{ivc}}^{sp}(k) - X_{\text{int}}^{sp}(k), \xi(X_{\text{ivc}}^{sp}(k - 1), T_{\text{ivc}}^{sp}(k - 1) - X_{\text{int}}^{sp}(k), \theta(T_{\text{ivc}}^{sp}(k - 1))
\]

(9)

Notice that \(T_{\text{ivc}}^{sp}\) is used in place of its estimate \(T_{\text{ivc}}\) to keep a constant cylinder BGR. The goal of such an experiment is to validate the model validity and the control strategy under steady-state conditions. In this way, we guarantee that there is no mismatch between the set points computation and the control law.

#### 4.3 Block structure

The presented control law (9) can be implemented under the form of the global control scheme pictured in Figure 5. The set points formulas (8) and the control strategy are implemented in the high-level controller block. This block feeds the low-level controller handling the VVA actuator. Estimation of the intake manifold BGR \((X_{\text{int}})\) is used in the high-level controller to coordinate the low-level controllers by means of the reference signal \(\Phi_{\text{vva}}^{ref}\) determined according to (9).

### 5. EXPERIMENTAL RESULTS

We now report experimental results obtained with the control law (9).

#### 5.1 Engine setup

The engine used throughout the experiments is a 1.6 L four-cylinder direct injection HCCI-LTC Diesel engine. The second valve lift is provided by a Mechadyne™ system (see Lancefield et al. (2006)) allowing a second lift between 0 and 4 mm (corresponding to 0 to 40 deg VVA actuator position). External EGR is carried out by a LP circuit.

#### 5.2 Intake manifold BGR variation

The first sequence of experimental tests considers an intake manifold BGR variation, trying to keep a constant cylinder BGR. The goal of such an experiment is to validate the model validity and the control strategy under steady-state conditions. In Figure 6, the solid blue curves designate the reference control strategy (named “ref,” presented in Section 2.1) and the dashed red curves refer to the proposed control strategy (named “strat,” presented in Section 4.2).

Figure 6(a) reports the exhaust valve position variation trajectory that is imposed in both cases. Figure 6(c) presents the resulting intake manifold BGR response.
that the mismatch between the two curves arises from the differences in VVA positions between the two strategies. Figure 6(d) gives the VVA actuator trajectory. In the reference strategy, the VVA actuator is set to its mapped position according to the engine operating point. In the proposed strategy, the VVA actuator trajectory (given by the control law (9)) moves away from its reference position to satisfy the cylinder BGR demand. Figure 6(e) shows that the cylinder BGR request is kept constant during the whole test (except around 120 s because of the VVA actuator saturation). Figure 6(f) shows that the VVA actuator position variations lead to strong cylinder temperature variations. Finally, Figure 6(b) presents the NO\textsubscript{x} emissions obtained using the two strategies. The proposed strategy succeeds in keeping the emissions constants.

This experiment highlights two facts. First, the cylinder BGR is the relevant control variable for NO\textsubscript{x} emissions control purpose. Second, the designed cylinder composition model is consistent with reality.

The intake manifold BGR is controlled by its low-level controller. It behaves similarly with the two different control strategies (“ref” and “strat”, respectively). Yet, the VVA actuator trajectories are strongly different in the two cases. Indeed, in the case of the reference controller, the VVA trajectory \( \Phi_{vva}^{ref} \) is only based on a static look-up table depending on the engine operating point. By contrast, in the proposed strategy, the VVA trajectory \( \Phi_{vva}^{strat} \) is continuously updated during transients to satisfy the cylinder BGR objective. This can be seen in Figure 7(e) where the cylinder BGR tracks well its set point (which is not the case for the reference controller). Simultaneously, the cylinder temperature drifts away from its set point due to the change in the EGR/IEGR trade-off (see Figure 7(f)). Finally, Figure 7(b) gives results on NO\textsubscript{x} emissions. One can see that the whole system tends to move closer to the steady-state conditions with the proposed strategy. This permits to prevent the undesired NO\textsubscript{x} emissions peaks during torque tips-in (especially during increasing intake manifold BGR demands).

5.3 Torque transients

The second sequence of experimental tests consists of torque trajectories composed of increasing and decreasing torque requests. These experiments are conducted at constant engine speed of 1000 rpm (Figure 7). The reference control strategy (named “ref”) and the proposed control strategy (named “strat”) are compared. The low-level control variables (the intake BGR and the VVA actuator) are given in Figure 7(c) and Figure 7(d).

6. CONCLUSIONS

A control strategy specifically designed to handle transients of turbocharged Diesel engines equipped with VVA actuator and LP EGR loop has been presented. To reduce pollutant emissions, we control the BGR contained in the combustion chamber at ivc. A model of cylinder BGR and temperature has been presented. It uses a breathing model along with a dilution and a combustion
models. This model is shown to have interesting invertibility properties from which a control strategy has been developed. The strategy permits to accelerate the cylinder BGR response by coordinating the two variables acting on the breathing process, i.e. the intake manifold BGR and the position of the VVA actuator.

Experimental results obtained on a test bench have stressed the relevance of the control strategy. A significant reduction of the NO$_x$ emissions during transients has been obtained.

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Appendix A. EXHAUST MODELING

A.1 Exhaust BGR modeling

Assuming complete and lean combustion, the exhaust BGR only depends on the composition inside the cylinder before the combustion (at ivc) and on the quantity of injected fuel, it comes

$$X_{exh}(k+1) = X_{ivc}(k) + \frac{(AFR_s + 1) \cdot m_f(k)}{m_{ivc}(k)} \quad (A.1)$$

where $X_{ivc}$ and $m_{ivc}$ are the cylinder BGR and mass at ivc, respectively, $m_f$ is the injected fuel mass, and $AFR_s$ is the stoichiometric air/fuel ratio. The cylinder mass writes (see Leroy et al. (2009))

$$m_{ivc} = \frac{\alpha_1 P_{int} V_{ivc}}{\gamma R T_{ivc}} \quad (A.2)$$

A.2 Exhaust temperature modeling

We assume an adiabatic compression, an instantaneous heat release and a polytropic expansion. After exhaust valve closing, the exhaust temperature of the gas leaving the cylinder during the exhaust blowdown is given by the adiabatic expansion of the gas from the combustion chamber to the exhaust manifold. Noting $T_{bd}$ the exhaust temperature during the exhaust blowdown, and using the cylinder mass expression given above, one obtains (see Rausen et al. (2005))

$$T_{bd} = T_{ivc} \left( \frac{P_{exh}}{\alpha_1 P_{int}} \right)^{\frac{\gamma-1}{\gamma}} \left( 1 + \mu_1 \frac{R Q L H V m_f}{\alpha_1 \gamma c_v V_{ivc} P_{int}} \right)^{\frac{1}{\gamma}}$$

where $\mu_1 = \left( \frac{V_{comb}}{V_{ivc}} \right)^{\frac{\gamma}{\gamma-1}}$ with $V_{comb}$ the cylinder volume when combustion appears. To simplify the model, $V_{comb}$ can be taken as constant. Finally, taking into account the heat losses in the exhaust runner (as is done in Rausen et al. (2005), e.g.), the dynamics of the temperature in the exhaust manifold can be written as

$$T_{exh}(k+1) = \mu_2 T_{ivc}(k) \left( \frac{P_{exh}(k)}{\alpha_1 P_{int}(k)} \right)^{\frac{\gamma-1}{\gamma}} \cdot \left( 1 + \mu_1 \frac{R Q L H V m_f(k)}{\alpha_1 \gamma c_v V_{ivc} P_{int}(k)} \right)^{\frac{1}{\gamma}} \quad (A.3)$$

where $\mu_2 < 1$ is a positive constant.