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# Transient Burned Gas Rate Control on VVA equipped Diesel Engines

T. Leroy\* J. Chauvin\* N. Petit\*\*

\* IFP, Department of Engine Control, 1 et 4 Avenue de Bois Préau,  
92852 Reuil Malmaison, France.

\*\* Centre Automatique et Systèmes, Mines ParisTech, 60, bd St  
Michel, 75272 Paris, France.

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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  $\text{NO}_x$  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,  $\text{NO}_x$  emissions are largely diminished during transients.

*Keywords:* Diesel engine,  $\text{NO}_x$  emissions, Burned Gas Rate control, VVA, Model-based control

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## 1. INTRODUCTION

Upcoming Diesel engines emission standards such as Euro 6 require significant reduction of smoke, HC (Hydrocarbons), CO (Carbon monoxide) and above all  $\text{NO}_x$  (Nitrogen Oxides) emissions. This has spurred an interest in new (cleaner) combustion modes. 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 consist of incorporating large amounts of Exhaust Gas Recirculation (EGR) into the combustion process.

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. The advantages of LP EGR are threefold. First, due to the EGR circuit length, the intake manifold mixture temperature is lower than in the case of the HP EGR. This has a beneficial impact on the  $\text{NO}_x$  emissions. Second, higher EGR rates can be obtained because LP EGR does not have any discharge effect on the turbocharger. Third, because the recirculated gases are cleaned by the after-treatment system, there is no problem of exchanger fouling. However, one main drawback of the LP technology is the large settling time because of EGR pipe length and low inlet-outlet pressure difference.

Recently, a new way to generate EGR has emerged. On top of these technological solutions, additional improvements can be achieved using a VVA device (see Bression et al. (2008)). The VVA actuator is located on the camshaft

(see Figure 1). It permits a variable exhaust valve re-opening (pictured in Figure 3(b)) that drastically modifies the cylinder filling process.

As has been discussed previously, the LP EGR provides good results in terms of  $\text{NO}_x$  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  $\text{NO}_x$  emissions is controlled by modulating the Burned Gas Rate (BGR) of the cylinder mixture before combustion (links between BGR and  $\text{NO}_x$  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.

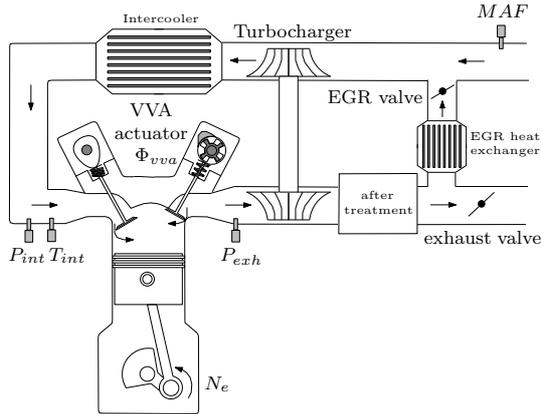


Fig. 1. Schematic illustration of a turbocharged, VVA-equipped Diesel engine. The intake manifold pressure and temperature  $P_{int}$  and  $T_{int}$ , the exhaust manifold pressure  $P_{exh}$  and the mass air flow at the intake  $MAF$  are measured by sensors.  $\Phi_{vva}$  is the position of the VVA actuator.

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). Implicitly, at each operating point, this defines a reference cylinder BGR and temperature.

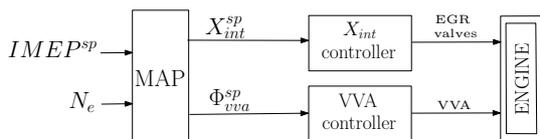
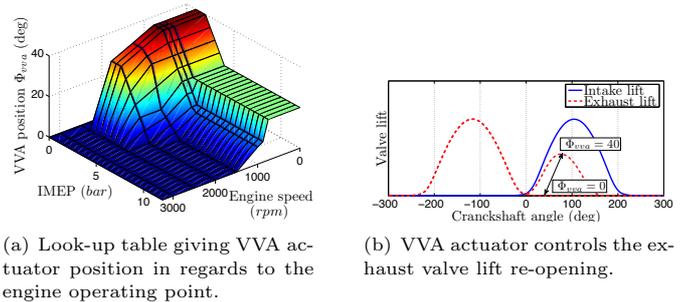


Fig. 2. Classical control strategy.  $IMEP^{sp}$  and  $N_e$  refer to the driver torque demand and the engine speed, respectively. Look-up tables determine set points for the intake manifold BGR and VVA actuator position,  $X_{int}^{sp}$  and  $\Phi_{vva}^{sp}$ .

The discussed intake manifold BGR and VVA actuator controllers can be considered as low-level controllers. We do not give any detail about the derivation of such controllers. Some intake manifold BGR controllers are given in Jankovic and Kolmanovsky (1998), Van Nieuwstadt et al. (2000) or Chauvin et al. (2008). Some VVA actuators controllers can be found in Chauvin and Petit (2007) or Genç et al. (2001).

Figure 3(a) represents the look-up table that gives VVA actuator position on the whole engine operating range. Figure 3(b) shows the impact of the VVA actuator on the second exhaust valve lift. It clearly appears that the second valve lift is particularly used ( $\Phi_{vva} > 0$ ) at low and partial load to compensate for too low exhaust temperatures obtained with only external EGR.



(a) Look-up table giving VVA actuator position in regards to the engine operating point.

(b) VVA actuator controls the exhaust valve lift re-opening.

Fig. 3. Static positioning of the VVA actuator ( $\Phi_{vva}$  stands for the VVA position) in regards to the engine operating points.

### 2.2 Issues during transients

Under steady operation, the above-described strategy is optimal. However, the engine is not very often under steady operation. Importantly, the dynamic responses of the intake manifold BGR and the VVA actuator are rather different. On the one hand, the response of the intake manifold BGR is quite slow (because of the EGR pipe length and the low inlet-outlet pressure difference). On the other hand, the VVA actuator dynamics is fast.

These discrepancies between the two subsystems (the intake manifold BGR and the VVA actuator) during transients lead to undesired engine behavior. Figure 4 gives results on  $NO_x$  emissions during a 3-5 bar IMEP transient. Detrimental emission peak can be observed. The two figures at the top give the intake manifold BGR and the VVA actuator position. The figure on at the bottom reports the resulting  $NO_x$  emissions. Here, the change of engine operating point modifies the intake BGR and VVA actuator position set points (more external EGR, and, consequently, less IEGR is required). Because of the intake manifold BGR sluggishness corresponding to a slow external EGR response (compared to a fast IEGR response because of the fast VVA actuator position actuation), the cylinder BGR objective is not satisfied.  $NO_x$  emission peak follows from this poor cylinder BGR control. Here, VVA actuator is not optimally used because it remains on its reference position during the transient (used as passive actuator).

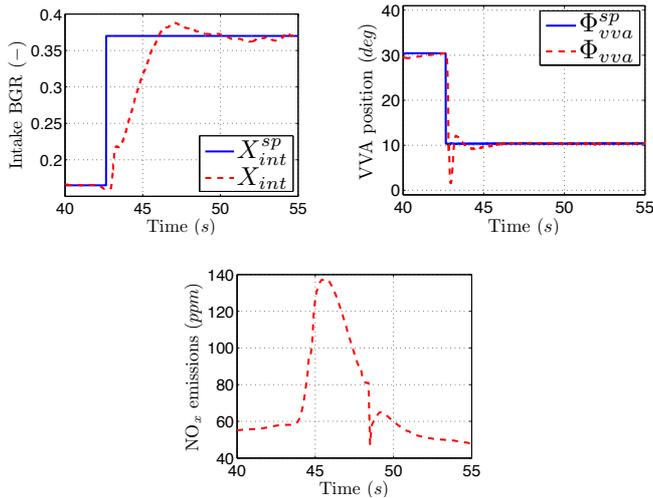


Fig. 4. Experimental IMEP transient (3-5 bar) at test bench on a 4-cylinders Diesel engine equipped with VVA actuator and LP EGR capability. The difference in response times between the intake manifold BGR and the VVA actuator leads to  $\text{NO}_x$  emission peak during transient.

### 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  $\text{NO}_x$  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 quantities of aspirated masses from the intake  $m_{asp}^{int}$  (which is composed of fresh air and burned gases coming from external EGR), and from the exhaust  $m_{asp}^{exh}$  (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

$$\begin{cases} m_{asp}^{int} = \alpha_1 \frac{P_{int} V_{ivc}}{RT_{int}} - \alpha_2 \frac{P_{exh}}{RT_{int}} V_{evc}(\Phi_{vva}) \\ m_{asp}^{exh} = \alpha_2 \frac{P_{exh}}{RT_{exh}} V_{evc}(\Phi_{vva}) \end{cases} \quad (1)$$

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.  $V_{ivc}$  is the cylinder volume at intake valve closing (*ivc*). Similarly,  $V_{evc}$  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

$$X_{ivc}(k) = \frac{X_{int}(k)m_{asp}^{int}(k) + X_{exh}(k)m_{asp}^{exh}(k)}{m_{asp}^{int}(k) + m_{asp}^{exh}(k)} \quad (2)$$

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

$$T_{ivc}(k) = \frac{T_{int}(k)m_{asp}^{int}(k) + T_{exh}(k)m_{asp}^{exh}(k)}{m_{asp}^{int}(k) + m_{asp}^{exh}(k)} \quad (3)$$

*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

$$\begin{cases} X_{exh}(k) \triangleq \xi(X_{ivc}(k-1), T_{ivc}(k-1)) \\ T_{exh}(k) \triangleq \theta(T_{ivc}(k-1)) \end{cases} \quad (4)$$

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

$$f(\Phi_{vva}(k), \theta(T_{ivc}(k-1))) \triangleq \frac{m_{asp}^{exh}(k)}{m_{asp}^{int}(k) + m_{asp}^{exh}(k)} \quad (5)$$

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

$$\begin{cases} X_{ivc}(k) = f(\Phi_{vva}(k), \theta(T_{ivc}(k-1))) \\ \quad \cdot \xi(X_{ivc}(k-1), T_{ivc}(k-1)) \\ \quad + (1 - f(\Phi_{vva}(k), \theta(T_{ivc}(k-1))))X_{int}(k) \\ T_{ivc}(k) = f(\Phi_{vva}(k), \theta(T_{ivc}(k-1)))\theta(T_{ivc}(k-1)) \\ \quad + (1 - f(\Phi_{vva}(k), \theta(T_{ivc}(k-1))))T_{int}(k) \end{cases} \quad (6)$$

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



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  $\text{NO}_x$  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  $\text{NO}_x$  emissions control purpose. Second, the designed cylinder composition model is consistent with reality.

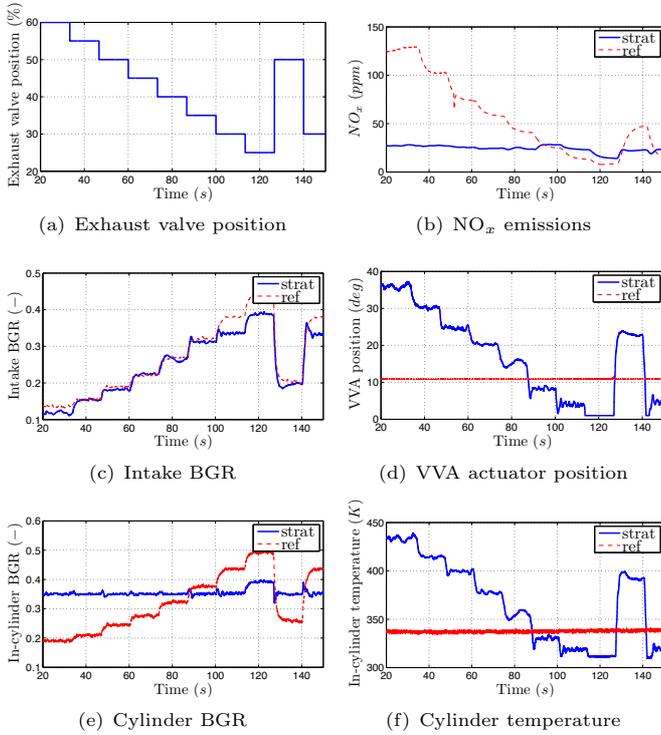


Fig. 6. Test-bench results for a four-cylinder Diesel engine for exhaust valve position variation at constant speed of 1500 *rpm* with and without the proposed control strategy. The controller succeeds in keeping constant the cylinder BGR, and consequently the  $\text{NO}_x$  emissions.

### 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).

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  $\Phi_{vva}^{sp}$  is only based on a static look-up table depending on the engine operating point. By contrast, in the proposed strategy, the VVA trajectory  $\overline{\Phi}_{vva}$  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  $\text{NO}_x$  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  $\text{NO}_x$  emissions peaks during torque tips-in (especially during increasing intake manifold BGR demands).

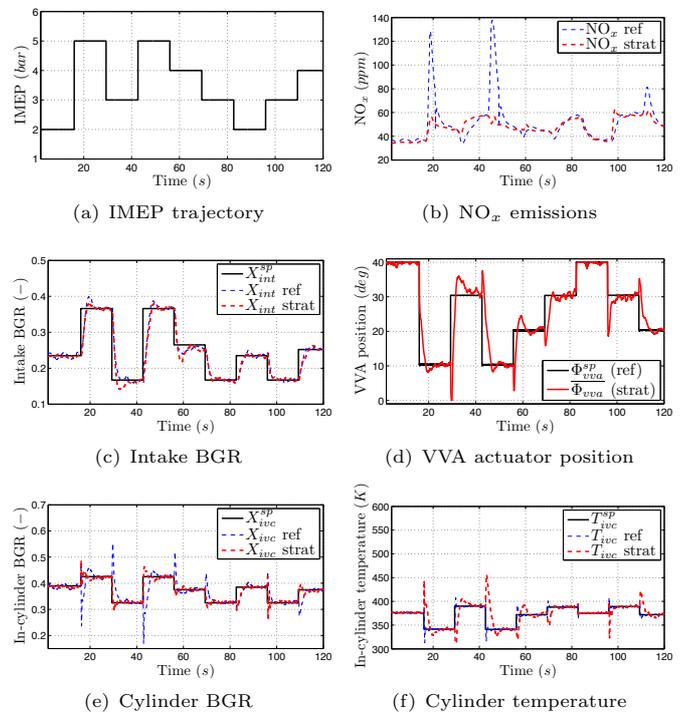


Fig. 7. Test-bench results for a four-cylinder Diesel engine for step torque demand at constant speed of 1000 *rpm* with and without the proposed control strategy. The controller succeeds in tracking the cylinder BGR set point. Consequently, the  $\text{NO}_x$  emissions peaks are reduced.

## 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  $\text{NO}_x$  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

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

where  $X_{ivc}$  and  $m_{ivc}$  are the cylinder BGR and mass at  $ivc$ , respectively.  $m_f$  is the injected fuel mass, and  $\text{AFR}_s$  is the stoichiometric air/fuel ratio. The cylinder mass writes (see Leroy et al. (2009))  $m_{ivc} = \alpha_1(P_{int}V_{ivc})/(RT_{ivc})$ , then substituting into (A.1), one finally obtains

$$X_{exh}(k+1) = X_{ivc}(k) + \frac{(\text{AFR}_s + 1) \cdot m_f(k)R}{\alpha_1 P_{int}(k) V_{ivc}} T_{ivc}(k) \quad (\text{A.2})$$

### 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))

$$T_{bd} = T_{ivc} \left( \frac{P_{exh}}{\alpha_1 P_{int}} \right)^{\frac{\gamma-1}{\gamma}} \left( 1 + \mu_1 \frac{RQ_{LHV}m_f}{\alpha_1 c_v V_{ivc} P_{int}} \right)^{\frac{1}{\gamma}}$$

where  $\mu_1 = \left( \frac{V_{comb}}{V_{ivc}} \right)^{\gamma-1}$  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

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

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