

Smooth In-Cylinder Lean-Rich Combustion Switching Control for Diesel Engine Exhaust-Treatment System Regenerations

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ABSTRACT

This paper describes an in-cylinder lean-rich combustion (no-post-injection for rich) switching control approach for modern diesel engines equipped with exhaust-treatment systems. No-post-injection rich combustion is desirable for regeneration of engine exhaust-treatment systems thanks to its less fuel penalty compared with regeneration approaches using post-injections and / or in-exhaust injections. However, for vehicle applications, it is desirable to have driver-transparent exhaust-treatment system regenerations, which challenge the in-cylinder rich-lean combustion transitions. In this paper, a nonlinear in-cylinder condition control system combined with in-cylinder condition guided fueling control functions were developed to achieve smooth in-cylinder lean-rich switching control at both steady-state and transient operation. The performance of the control system is evaluated on a modern light-duty diesel engine (G9T600). Experimental results show significant merits of the proposed control system compared against the results obtained by conventional calibration-based approaches in terms of smoothness of engine torque responses during lean-rich transitions.

INTRODUCTION

As diesel engine emission regulations are becoming more and more stringent worldwide, using exhaust treatment devices such as lean NO_x trap (LNT) and diesel particulate filter (DPF) etc., in conjunction with combustion improvements has become necessary [1-3]. However, in order to maintain the functionalities of these exhaust-treatment systems, intermittent regenerations are required. Particularly for the LNTs regenerations, rich exhaust gas so far is necessary in order to purge the stored NO_x and maintain their conversion efficiencies.

Post injection and/or in-exhaust injection are the popular approaches to create rich gas for LNT regenerations [4-6]. For these kinds of methods, producing of rich gas is independent of engine torque (fuel used for regeneration does not contribute to torque). Even though those methods can cover almost the entire engine operating

region and are easy to control, there are also some concerns. Besides the extra hardware requirement, post injection / in-exhaust injection regenerations also have the following issues:

- Could cause oil dilution problems due to unburned fuel.
- Lead to high temperature exotherms inside catalyst and cause thermal deterioration, which in turn could potentially shorten the catalyst lifetime.
- More importantly, the fuel penalties associated with these regeneration methods are not minimal [7].

To overcome the problems associated with post- and in-exhaust injection (with reducing fuel penalty being the main motivation), several in-cylinder no-post-injection rich combustion techniques were investigated in literature [7, 8]. No-post-injection rich combustion does not require any additional hardware and can fulfill LNT purging with less regeneration fuel penalty. However, the major challenge associated with the no-post-injection regeneration is how to achieve driver-transparent (smooth torque), stable, and fast rich-lean combustion switching at both steady-state and transient operation for vehicle applications. Since the in-cylinder conditions, fueling, and combustion behaviors are dramatically different between lean and rich states, it is then difficult for conventional calibration-based engine control approaches to achieve lean-rich switching with smooth torque responses and stable combustion. Sophisticated and advanced engine control approaches are then required [9]. To achieve solid lean-rich switching control, one of the key issues is thus how well to control the in-cylinder conditions. If we can ensure good control on in-cylinder conditions, we will have good control on engine combustion (knowing that fueling/injection control is nearly perfect), and in turn, good control on exhaust gas composition and engine performance.

In-cylinder condition process by nature is a multi-input-multi-output (MIMO), coupled nonlinear system. Classical linear control approaches (such as PID controller with lead / lag compensations) may not be able to produce satisfactory performance [9]. Advanced and sophisticated MIMO nonlinear control design

methods may offer great potentials in simultaneously meeting the real-world drivability and emission requirements. On the other hand, in order to achieve smooth and fast lean-rich switching, fueling needs to be elaborately controlled to match with the actual in-cylinder conditions. Since the fueling control is very fast, it is suitable to have an in-cylinder condition guided fueling control to make fueling parameters and in-cylinder conditions match well during transitions.

In this paper, a robust nonlinear lean-rich switching control approach is proposed. The control system can be decomposed into two parts: a nonlinear in-cylinder closed-loop controller and a dedicated in-cylinder condition guided fueling control, which collaboratively work towards achieving smooth and fast lean-rich combustion switching. Experimental results exhibit significant performance improvement compared against those of conventional calibration-based control approaches. The rest of the paper is organized as follows. In section II, the control-oriented dynamic modeling for an engine with dual-loop EGR system will be presented. The control system design and fueling control method are described in section III and the experimental results are presented in section IV. Finally, some conclusive remarks are given in section V.

SYSTEM MODELING

In order to exercise feedback control on in-cylinder conditions, the control-oriented dynamic models are indispensable. In this section, the dynamic models for engine intake / exhaust systems are constructed. The target platform is a modern light-duty 4-cylinder high-pressure common rail diesel engine. The engine is equipped with a variable geometry turbocharger (VGT), dual-loop EGR system (both high-pressure loop EGR and low-pressure loop EGR), high-pressure throttle valve, as well as exhaust-treatment systems such as diesel oxidation catalyst (DOC), DPF, and LNT. A schematic diagram of such set-up is shown in Figure 1. The sections of the intake and exhaust systems are numbered.

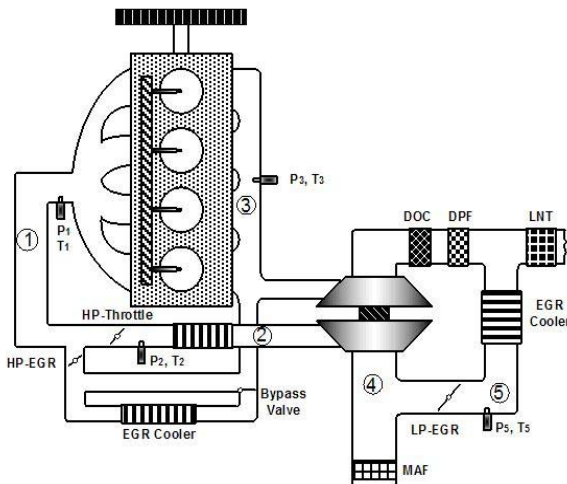


Figure 1. The schematic diagram of a modern light-duty diesel engine with dual-loop EGR system

Similar to the mean-value engine models which are widely used in production engine control systems, derivation of the models are based on the mass and energy conservation as well as ideal gas law. Details on the modeling techniques can be found in [9, 10, 12] and are omitted here. The resultant system dynamic models can be cast into state-space form as the following.

$$\begin{pmatrix} \dot{x}_1 \\ \dot{x}_2 \\ \dot{x}_3 \\ \dot{x}_4 \\ \dot{x}_5 \\ \dot{x}_6 \end{pmatrix} = \underbrace{\begin{pmatrix} -k_1 x_1 \\ k_2 x_4 \\ x_2^* - p_a^* \\ k_3 x_1 + k_f W_f \\ -x_4 \\ \tau_{ic} \\ 0 \\ k_5 (W_{MAF} - W_{ce} x_6) \end{pmatrix}}_{f(x)} + \underbrace{\begin{pmatrix} k_{ht1} & k_{r1} & 0 & 0 \\ -k_{ht2} & 0 & 0 & 0 \\ 0 & -k_{r3} & -k_{r3} & 0 \\ 0 & 0 & k_9 & 0 \\ k_4 (x_6 - x_5) & k_4 (F_3 - x_5) & 0 & 0 \\ 0 & 0 & 0 & k_5 F_3 \end{pmatrix}}_{g(x)} \begin{pmatrix} u_1 \\ u_2 \\ u_3 \\ u_4 \end{pmatrix} \quad (1)$$

Where, $x_1 = p_1$ is the pressure of section 1 (intake manifold), $x_2 = p_2$ is the pressure of section 2 (between high-pressure throttle valve and turbocharger), $x_3 = p_3$ is the exhaust manifold pressure, $x_4 = p_c$ is the compressor power, $x_5 = F_1$ is the air fraction (including contribution from the unburned air in EGR gas) in the intake manifold, $x_6 = F_4$ is the air fraction (including contribution from LP-EGR) of section 4 (between MAF sensor and compressor). The system inputs are: $u_1 = W_{HT}$, the gas flow rate through high-pressure throttle valve; $u_2 = W_{HEGR}$, the gas flow rate through HP-EGR valve; $u_3 = W_t$, the gas flow rate through turbine; and $u_4 = W_{LEGR}$, the gas flow rate through LP-EGR valve. W_{ce} is the gas flow rate through the compressor. W_f is the fueling rate which is determined by the engine maps at both lean and rich combustion modes along with sophisticated guiding functions during lean-rich transitions. Ambient pressure is denoted by p_a . The parameters are defined as $k_1 = \frac{\gamma \eta_v N_e V_d}{120 V_1}$

where γ is the specific heat ratio, η_v is the volumetric efficiency, N_e is engine speed, V_d is engine displacement volume, and V_1 is the intake manifold volume.

$$\begin{aligned} k_2 &= \frac{R \gamma T_c \eta_c p_a^{(\gamma-1)/\gamma}}{V_2 C_p T_a}, & k_{ht1} &= \frac{R \gamma T_{HT}}{V_1}, \\ k_{r1} &= \frac{R \gamma T_{HEGR}}{V_1}, & k_{ht2} &= \frac{R \gamma T_{HT}}{V_2}, & k_3 &= \frac{\gamma T_{eo} \eta_v N_e V_d}{120 V_3 T_1}, \\ k_4 &= \frac{R T_1}{p_1 V_1}, & k_5 &= \frac{R T_4}{p_4 V_4}, & k_6 &= \frac{R T_3}{p_3 V_3}, & k_7 &= \frac{R T_2}{p_2 V_2}, \\ k_{r3} &= k_{t3} = \frac{R \gamma T_3}{V_3}, & k_f &= \frac{R \gamma T_{eo}}{V_1} \end{aligned}$$

$$k_{t2} = \frac{C_p T_3 \eta_t (p_a + \Delta p_{ex})^{(\gamma-1)/\gamma}}{\tau_{tc}}, \text{ and } \kappa = \frac{\gamma-1}{\gamma}. \quad \Delta p_{ex} \text{ is}$$

the pressure difference across the exhaust-treatment systems.

The gas temperature reductions due to EGR coolers and intercooler are approximated using the linear heat exchanger effectiveness as well as the measured upstream and coolant temperatures as below [12].

$$T_{down} = \eta_{he} T_{coolant} + (1 - \eta_{he}) T_{up}, \quad (2)$$

where T_{up} is the upstream temperature and T_{down} is the downstream temperature. So, for HP-EGR gas, when the passage with EGR cooler is turned on by the switching valve, the temperature at HP-EGR valve becomes,

$$T_{HEGR} = \eta_{ec} T_{coolant} + (1 - \eta_{ec}) T_3. \quad (3)$$

The switching of the bypass valve depends on engine operating conditions. The temperature of the gas passing through the high-pressure throttle valve is,

$$T_{HT} = \eta_{ic} T_{coolant} + (1 - \eta_{ic}) T_2, \quad (4)$$

where $T_{coolant}$ is the engine coolant temperature. η_{ec} and η_{ic} are the heat exchanger efficiencies for EGR cooler and inter-cooler, respectively.

AIR FRACTION ESTIMATION

For diesel engines, the combustion is usually lean, which means there is excessive air than the stoichiometric amount in the engine cylinder mixture. Thus, the exhaust gas contains unburned air and it could be re-circulated back into intake manifold and combustion chambers through both HP- and LP-EGR valves. For engine combustion and emissions, not only the in-cylinder total EGR gas amount but also the ratio between LP-EGR gas amount and HP-EGR gas amount are very important because of their very different gas temperatures and compositions. Temperature of HP-EGR gas is high, which is helpful to increase intake manifold temperature and stabilize alternative combustion at light-load. LP-EGR gas is effectively cooled by EGR cooler and cleaned by DPF, therefore it can provide sufficient amount of cleaner EGR gas at medium load region to effectively reduce engine NOx emissions [2, 13]. Controlling the intake system air fraction (including air from EGR gas) has been proven as an effective means to control the in-cylinder EGR gas amount for engines with single-loop EGR systems [9-12]. For engines with dual-loop EGR systems as the one shown in Figure 1, the dynamic model of the air fractions in different sections can be derived in a similar manner as,

$$\begin{pmatrix} \dot{F}_3 \\ \dot{F}_4 \\ \dot{F}_2 \\ \dot{F}_1 \end{pmatrix}_g = \underbrace{\begin{pmatrix} -k_6 W_{eo} & 0 & 0 & 0 \\ k_5 W_{LEGR} & -k_5 W_{ce} & 0 & 0 \\ 0 & k_7 W_{ce} & -k_7 W_{HT} & 0 \\ k_4 W_{HEGR} & 0 & k_4 W_{HT} & -k_4 (W_{HT} + W_{HEGR}) \end{pmatrix}}_{A(p)} \begin{pmatrix} F_3 \\ F_4 \\ F_2 \\ F_1 \end{pmatrix}_g + \underbrace{\begin{pmatrix} k_6 W_{eo} F_{eo} \\ k_5 W_{MAF} \\ 0 \\ 0 \end{pmatrix}}_{W(p)}, \quad (5)$$

where, F_{1-4} are the air fractions of the section 1 – 4 in Figure 1. The air fraction coming out of the cylinder can be estimated based on the fueling rate and stoichiometric air-to-fuel ratio (AFR) λ_s as,

$$F_{eo} = \frac{W_e F_1 - W_f \lambda_s}{W_e + W_f} = \frac{W_f (\lambda_m - \lambda_s)}{W_e + W_f}, \quad (6)$$

where, λ_m is the measured exhaust gas AFR by a universal exhaust gas oxygen (UEGO) sensor. Typical UEGO sensor signal has a time delay and an observer was used to estimate the actual exhaust gas AFR assuming a fixed sensor response time constant. W_e is the engine intake gas flow rate, which can be calculated by the speed-density equation as $W_e = \frac{\eta_v P_1 N_e V_d}{RT_1 120}$.

This is a linear parameter-varying (LPV) system, whose parameter vector is defined as $\rho = (k_4 \ k_5 \ k_6 \ k_7 \ W_{eo} \ W_{LEGR} \ W_{ce} \ W_{HEGR} \ W_{HT} \ F_{eo} \ W_{MAF})^T$. A practically easy-to-implement observer was designed in [14] to estimate the air fractions F_{1-4} in real-time for closed-loop control purpose and used in this paper. The details of the observer design are omitted here.

DESIGN OF CONTROL SYSTEM

The overall goal of the control system for LNT regeneration lean-rich switching is to produce rich exhaust gas while maintaining or following the desired engine torque during the transitions, and thus cause no sacrifice of vehicle drivability. As stated earlier, in this paper, no-post-injection rich combustion is considered. It certainly makes the control challenges much higher than post-injection and/or exhaust-injection methods because the rich combustion (close to the stable edge) not only needs to produce the rich gas but also the desired engine torque. To achieve smooth transitions between lean and rich combustions, both in-cylinder conditions and fueling need to be controlled elaborately.

IN-CYLINDER CONDITION CONTROLLER

As discussed previously, control of in-cylinder conditions is important to achieve seamless lean-rich combustion transitions because of its slow, nonlinear, and coupled dynamics. Two nonlinear in-cylinder condition controllers were designed for lean combustion and rich combustion, respectively. The control design methods are similar as those described in [9] except that low-pressure loop EGR is also used in this paper. The in-cylinder control laws are briefly described while