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PRESSURE MODEL BASED COORDINATED CONTROL OF VGT AND DUAL-LOOP EGR IN A DIESEL ENGINE AIR-PATH SYSTEM

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ABSTRACT−This paper describes a pressure-model-based coordinated control method of a variable geometry turbine (VGT) and dual-loop exhaust gas recirculation (EGR) in a diesel engine air-path system. Conventionally, air fraction or burnt gas fraction states are controlled for the control of dual-loop EGR systems, but fraction control is not practical since sensors for fractions are not available on production engines. In fact, there is still great controversy over how best to select control outputs for dual-loop EGR systems. In this paper, pressure and mass flow states are chosen as control outputs without fraction states considering the availability and reliability of sensors. A coordinated controller based on the simple control-oriented model is designed with practical aspects, which is applicable for simultaneous operations of high pressure (HP) EGR, low pressure (LP) EGR, and VGT. In addition, the controller adopts the method of input-output linearization using back-stepping to solve the chronic problems of conventional pressure-based controllers such as coupling effects between operations of HP EGR, and VGT. The control performance is verified by simulation based on the proven GT-POWER model of a heavy-duty 6000cc diesel engine air-path.

KEY WORDS : Diesel engine, Air-path control, Dual-loop EGR, Control-oriented model, VGT

NOMENCLATURE

A : effective area (m^2)
c_p : specific heat at constant pressure (kJ/kg·K)
F : burnt gas fraction (-)
m : mass (kg)
N : engine speed (RPM)
p : pressure (Pa)
P : power (kW)
R : ideal gas constant (kJ/kg·K)
T : temperature (K)
V : volume (m^3)
W : mass flow rate (kg/s)
γ : specific heat ratio (-)
η : efficiency (-)
λ_s : stoichiometric ratio (-)
τ_{tc} : turbocharger time constant (s)

SUBSCRIPTS

a : ambient
c : compressor
i : intake manifold
t : turbine
x : exhaust manifold
dt : downstream of turbine
uc : upstream of compressor
HPegr : high pressure EGR
LPegr : low pressure EGR

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1. INTRODUCTION

The environmental regulations for diesel engines have become increasingly strict, urging worldwide automotive industries to create more environmental-friendly engine technologies. Many environmental-friendly components of a diesel engine have been developed in violation of environmental regulations. For instance, most diesel engines are fitted with exhaust treatment systems including a diesel particulate filter (DPF), diesel oxidation catalyst (DOC), and selective catalytic reduction (SCR) to deal with harmful substances of emission. In addition, exhaust gas recirculation (EGR) and variable geometry turbine (VGT) systems are commonly equipped in diesel engines today. A great deal of research has been carried out on the coordinated operation of EGR and VGT because these technologies have the potential to reduce emissions considerably by treating the root cause of emissions. EGR systems recirculate exhaust gas into the intake manifold, which reduces nitrogen oxides (NOx), and VGT compresses the air into the intake manifold, even with a small amount of exhaust gas, which leads to the regulation of particulate matters (PM) as well as boosting the engine output torque. The combination of EGR and VGT in a diesel air-path system can adjust fundamental properties of combustion in the diesel engine by manipulating gas compositions in cylinders of the engine so that advanced combustion modes such as low temperature combustion (LTC) or homogeneous charge compression ignition (HCCI) are realizable, resulting in a significant reduction of emissions (Wang, 2008a; Wang, 2008b; Yoon, 2011; Yan & Wang, 2013). Since the late 2000s, diesel engines with dual-loop EGR systems have emerged in the market. A dual-loop EGR system uses two EGR loops of high pressure (HP) EGR and low pressure (LP) EGR simultaneously, which have quite different properties, for the effective utilization of EGR functions (Wang, 2008b; Yan and Wang, 2013; Castillo *et al*., 2013).

The problem is that, as configurations of a diesel engine are becoming more complicated, it is becoming harder to control the system precisely. The target system this paper deals with is a diesel engine air-path system with VGT and dual-loop EGR. The air path of a diesel engine is defined as the system consisting of paths that air and exhaust gas are flowing through in the engine, as depicted in Figure 1. By adjusting the valve positions of two EGR systems and vane position of VGT, the composition of gases into the cylinders can be changed, leading to a change of combustion properties for emission reduction. However, since EGR and air flows are highly coupled with each other, precise control of the flows is not an easy task. Various coordinated control methods of VGT and EGR system based on mathematical models have been investigated to handle the properties of the air-path system effectively and reduce calibration efforts. Many studies have been done on the single-loop EGR system (equipped with only HP EGR) (e.g. Upadhyay, 2001; Ammann *et al*., 2003), but for the dual-loop EGR systems, which is a relatively new research field, only a few studies were conducted.

Design of a control algorithm for the air-path system begins from the selection of set points that are determined by required engine performances such as desired engine power or emissions (Van Nieuwstadt *et al.*, 2000; Sarlashkar and Roecker, 2010). Specifically, since the air-path control is focusing on the simultaneous reduction of hazardous emitted materials of PM and NOx while continuing to generate the desired torque, the set-points are concerned with the desired values of air to fuel ratio (AFR), EGR rate, and engine torque. Once the set-points are given, it is necessary to select output states for proper feedback control to satisfy the set-points while keeping the whole system balanced. For the single-loop EGR and VGT systems, intake manifold pressure and compressor flow have been conventionally regarded as the main output states for control (e.g. Upadhyay, 2001; Van Nieuwstadt *et al*., 2000; Yoon, 2011). However, the way to select governing control outputs for the dual-loop EGR and VGT is still being disputed. In the vast majority of the papers dealing with the control of dual-loop EGR, the air fraction or burnt gas fraction state is controlled. Control algorithms for states including fraction states in dual-loop EGR systems are designed in several papers. A control method using motion planning was implemented for the dual-loop EGR system in Grondin *et al*. (2009). Another method using singular perturbation methodology was suggested in Yan and Wang (2013). The authors regarded the dual-loop EGR system as a singularly perturbed system based on the fact that the two EGR operations have very different time scales. In Mrosek and Isermann (2011), a decentralized PID control method was developed to control outputs including an LP EGR fraction using a local linearization of the semi-physical model around each operating point. It seems that the control of fraction states is essential for the dual-loop EGR air-path system. However, sensors for control of those fraction states are not available in production engines, and the accuracy of the sensors is doubtful (Upadhyay, 2001; Yoon, 2011; Wang, 2008a). Furthermore, the accuracy of the fraction model equations is vulnerable to model uncertainties of other primary states such pressure and temperature. Hence, a controller based on only pressure and flow states, which can be easily measured or estimated using mass produced sensors, as outputs is designed in this paper with practical concerns.

This paper focuses especially on the coordinated control of VGT, HP EGR, and LP EGR for their combined operations. Studies on the coordinated control of VGT and dual-loop EGR operations are nearly absent. In Grondin *et al*. (2009), the control algorithm could be applied to the system when only one of the EGR operations is active, which makes the control problems much simpler. In Yan and Wang (2013), a high pressure throttle instead of a VGT vane was utilized as one of the main control actuators. The control algorithm developed in Mrosek and Isermann (2011) was for coordinated control of VGT and dual-loop EGR, but the controller was actually a linear controller that requires a great deal of calibration effort and was not suitable for the nonlinear air-path system with various operating points. In this paper, a full model-based control algorithm is developed appropriate for combined operations of VGT and dual-loop EGR considering the strong nonlinearities of the air path.

The chronic problem of the conventional pressure-based controllers for the air-path with EGR and VGT is that the controllers do not handle the coupling properties of the system effectively. For example, for an air path with a single EGR and VGT, where control actuators are an HP EGR valve and VGT vane, intake manifold pressure and exhaust manifold pressure are adjusted mainly by the turbocharging operations of VGT. However, EGR operations also affect the pressure states since EGR flow is strongly coupled with VGT flow, both of which are driven by exhaust gas flow (Kolmanovsky *et al*., 1997). Ideally, an EGR system should be operated not for adjusting pressures, but only for supplying the target EGR rate into cylinders. Due to the heavy coupling effects, controlling only pressure states may not uniquely determine the right combination of positions of the VGT vane and EGR valve. In many papers, controlling fraction states has been considered as the answer to solve the coupling problems, especially for the dual-loop EGR and VGT systems, which is not practical for the reasons mentioned earlier. Actually, the concept of a pressure- and flow-based control for the dual-loop EGR and VGT system was first proposed in the authors' previous conference paper (Kim *et al*., 2014). In this paper, a more advanced coordinated control method based on pressure model dynamics is suggested to effectively solve the coupling problems of HP EGR and VGT by using the method of input-output linearization with back-stepping. Note that the dual-loop EGR system is implemented by just expanding the single-loop EGR with LP EGR loop added. Applying the same principle of expansion, for the controller implementation, a model-based control algorithm for the LP EGR loop is also designed, and this sub controller acts on the system in cooperation with the control laws for HP EGR and VGT.

This paper is organized as follows. In Section 2, the main equations of a plant model of the air-path system with dual-loop EGR and VGT are described and a control-oriented model is introduced. In Section 3, the control principles and design procedure of the controller based on the control-oriented model for the system is represented. Finally, in Section 4, the control performance is validated through simulation based on a proven GT-POWER engine model.

2. AIR-PATH MODELING

The configuration of the air-path system is described in Figure 1. The main model equations for the system, which have been validated in many papers (e.g. Upadhyay, 2001;

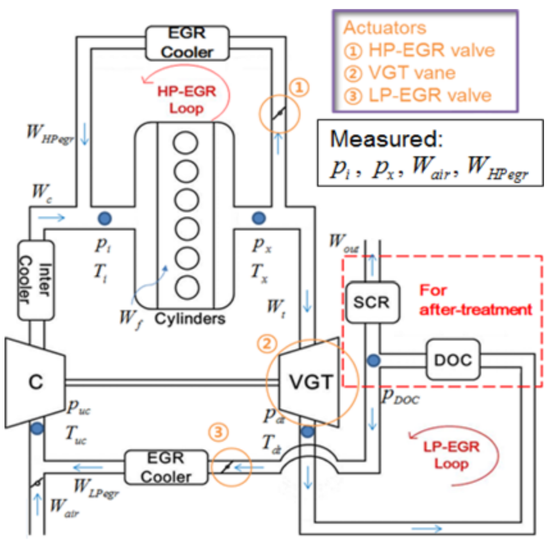

Figure 1. Schematic of the air-path system with a dual-loop EGR system and VGT.

Wahlström, 2006), are introduced in this section. Model-based control methods will be adopted under the assumption that the model equations describe properties of the air-path system well enough to control the system precisely.

2.1. System Overview

The air-path system covered in this paper has two EGR loops and a one-stage VGT. In the LP EGR loop, there are exhaust after-treatment systems including DOC to dispose of dangerous substances in exhaust gas flow coming from the turbine downstream. After being filtered in the after-treatment systems, the exhaust gas flows toward the compressor inlet through the LP EGR loop while some of it flows to the exhaust throttle through the SCR. Mass flow through the LP EGR loop is mixed with fresh air mass flow through the intake throttle at the compressor inlet. The mixed flow enters the intake manifold by a turbocharging operation of the compressor and is again mixed with another EGR flow from the HP EGR loop. The total flow enters the cylinders. The gas composition of this intake flow is the most important factor to determine the emission level of the engine. Then, combustion reaction occurs in cylinders generating new exhaust gas at the exhaust manifold. All the flows affect the change of physical quantities such as pressure and temperature in corresponding locations.

The model equations for the system consist of dynamics of pressure (p), temperature (T), burnt gas fraction (F), and other system variables such as mass flow rate (W). The subscripts, i, x, c, t, uc, and dt stand for intake manifold, exhaust manifold, compressor, turbine, upstream of compressor, and downstream of turbine respectively, which are the main components to describe the system properties. The dynamics at the compressor upstream and turbine downstream also carry an important meaning with the LP EGR loop included in the system.

2.2. Intake Manifold

Intake manifold pressure is modeled based on the ideal gas law with mass flow described as Equation (1)

$$\dot{p}_{\mathrm{i}} = \frac{R\gamma}{V_{\mathrm{i}}}\left(T_{\mathrm{ic}}W_{\mathrm{c}} + T_{\mathrm{x}}W_{\mathrm{HPegr}} - T_{\mathrm{i}}W_{\mathrm{ie}}\right), \qquad (1)$$

where R, W_{ie}, γ, and T_{ic} are the ideal gas constant, the mass flow rate into the cylinder, the specific heat ratio, and the temperature at the intercooler outlet, respectively.

The equation is derived under the assumption that the temperature whose dynamics is relatively slow is not changing at the same operating points.

The temperature dynamics of intake manifold is represented as Equation (2) (Wang, 2008a; Yan and Wang, 2013; Jung, 2014).

$$\dot{T}_{\mathrm{i}} = \frac{RT_{\mathrm{i}}}{p_{\mathrm{i}}V_{\mathrm{i}}}[W_{\mathrm{c}}(\gamma T_{\mathrm{ic}} - T_{\mathrm{i}}) + W_{\mathrm{HPegr}}(\gamma T_{\mathrm{HPegr}} - T_{\mathrm{i}}) + W_{\mathrm{ie}}(T_{\mathrm{i}} - \gamma T_{\mathrm{e}})]. \qquad (2)$$

The burnt gas fraction, which is defined as the fraction of burnt gas out of the total gas composition, is modeled as Equation (3) for the intake manifold (Jung, 2014).

$$\dot{F}_i = \frac{1}{m_i}(F_{uc} - F_i)W_c + \frac{1}{m_i}(F_x - F_i)W_{HPegr}, \tag{3}$$

where $F_i = \dfrac{m_i - m_{i\_air}}{m_i}$.

In Equation (3), m_i and m_{i_air} stand for the mass of burnt gas and the mass of air in intake manifold, respectively.

The mass flow rate through the compressor can be described by Equation (4), which is determined mainly by pressure ratio across the compressor and compressor power.

$$W_c = \frac{\eta_c}{T_{uc}c_p}\frac{P_c}{\left(\dfrac{p_i}{p_{uc}}\right)^{\frac{\gamma-1}{\gamma}} - 1}, \tag{4}$$

where η_c is the compressor efficiency.

2.3. Exhaust Manifold

Likewise, the dynamics of pressure, temperature, and burnt gas fraction for exhaust manifold are described as Equations (5) ~ (7).

$$\dot{p}_x = \frac{R\gamma}{V_x}\left(T_e W_x - T_x W_{HPegr} - T_x W_t\right) \tag{5}$$

$$\dot{T}_x = \frac{RT_x}{p_x V_x}[W_x(\gamma T_e - T_x) + W_t(\gamma T_x - T_t) + W_{HPegr}(T_x - \gamma T_{HPegr})] \tag{6}$$

$$\dot{F}_x = \frac{W_{ie}}{m_x}F_i - \frac{W_{ie} + W_f}{m_x}F_x + \frac{W_f}{m_x}(1 + \lambda_s), \tag{7}$$

where $F_x = \dfrac{m_x - m_{x\_air}}{m_x}$.

λ_s is the stoichiometric ratio in Equation (7). HP EGR flow rate is modeled as Equation (8) using an orifice equation.

$$W_{HPegr} = \sqrt{\frac{2\gamma A_{HP}^2 p_x^2}{RT_x(\gamma-1)}\left((PR_{HP})^{\frac{2}{\gamma}} - (PR_{HP})^{\frac{\gamma+1}{\gamma}}\right)},$$
$$\text{where } PR_{HP} = \max\left(\frac{p_i}{p_x}, \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}\right). \tag{8}$$

In Equation (8), PR_{HP} is the pressure ratio across HP EGR flow and A_{HP} the effective area of HP EGR flow determined by the valve position.

2.4. Turbocharger Dynamics

The power dynamics of a turbocharger is mathematically described as Equation (9)

$$\dot{P}_c = \frac{1}{\tau_{tc}}\left(-P_c + \eta_m P_t\right), \tag{9}$$

where P_c, P_t, τ_{tc}, and η_m are the power generated by the compressor, the power delivered by the turbine, a turbocharger time constant, and mechanical efficiency, respectively. The equation of the turbine power can be addressed by Equation (10).

$$P_t = \eta_t c_p T_x\left(1 - \left(\frac{p_{dt}}{p_x}\right)^{\mu}\right)W_t, \tag{10}$$

where η_t, c_p, and μ are the turbine efficiency, the specific heat at constant pressure, and some positive exponent, respectively.

2.5. LP EGR Loop

To describe the properties of the LP EGR loop, model equations for a turbine outlet and compressor inlet are derived as Equations (11) ~ (13) (Jung, 2014).

$$\dot{p}_{uc} = \frac{R\gamma}{V_{uc}}\left(T_a W_{air} + T_{dt}W_{LPegr} - T_c W_c\right) \tag{11}$$

$$\dot{p}_{dt} = \frac{R\gamma}{V_{dt}}\left(T_x W_t - T_{dt}W_{LPegr} - T_{out}W_{out}\right) \tag{12}$$

$$\dot{F}_{uc} = \frac{1}{m_{uc}}\left[(F_x - F_{uc})W_{LPegr} - F_{uc}W_{air}\right], \tag{13}$$

where $F_{uc} = \dfrac{m_{uc} - m_{uc\_air}}{m_{uc}}$.

In Equation (12), T_{out} and W_{out} stand for the temperature at the exhaust throttle and the exhaust mass flow rate, respectively.

Similar to Equation (8), the mass flow rate of LP EGR is described as Equation (14).

$$W_{LPegr} = \sqrt{\frac{2\gamma A_{LP}^2 p_{dt}^2}{RT_{dt}(\gamma-1)}\left((PR_{LP})^{\frac{2}{\gamma}} - (PR_{LP})^{\frac{\gamma+1}{\gamma}}\right)},$$
$$\text{where } PR_{LP} = \max\left(\frac{p_{uc}}{p_{dt}}, \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}\right). \tag{14}$$

In Equation (14), the parameter PR_{LP} represents the pressure ratio across the LP EGR flow, and A_{LP} is the effective area of LP EGR flow, which is determined by the valve position.

To model the mass flow rate of pure fresh air flow in the dual-loop EGR system, a new model for the air flow rate in dual-EGR systems is designed as Equation (15) using the orifice equation.

$$W_{air} = A_{air}p_a\sqrt{\frac{2\gamma}{RT_a(\gamma-1)}\left(\left(\frac{p_{uc}}{p_a}\right)^{\frac{2}{\gamma}} - \left(\frac{p_{uc}}{p_a}\right)^{\frac{\gamma+1}{\gamma}}\right)}. \tag{15}$$

Note that it is assumed that the intake throttle is fully opened throughout this paper. Because the position of intake throttle is fixed, the effective area A_{air} is regarded as a constant in Equation (15).

2.6. Control-oriented Model

The fifth-order control-oriented model of the air-system with dual-loop EGR and VGT was implemented as in Equations (16) ~ (20) for the controller design. The third-order model consisting of Equations (16), (17), and (20) was a well-known control-oriented model for the single-loop EGR and VGT system validated in many papers (Upadhyay, 2001; Yoon, 2011; Jin *et al*., 2014). The fifth-order model was designed by adding two states Equations (18) and (19), which describe the properties of the LP EGR loop, to the third-order model. The control-oriented model is implemented under the following assumptions (Jin *et al*., 2013; Wang, 2008a);

- Burnt gas fraction states, which are difficult to measure and not directly coupled with other dynamics, are omitted.
- Temperature states whose dynamics are relatively slow are ignored.
- Temperatures of all flows in the same location are on the same level.

$$\dot{p}_{\mathrm{i}} = k_1\left(W_{\mathrm{c}} + W_{\mathrm{HPegr}} - k_{\mathrm{e}} p_{\mathrm{i}}\right) \tag{16}$$

$$\dot{p}_{\mathrm{x}} = k_2\left(W_{\mathrm{x}} - W_{\mathrm{HPegr}} - W_{\mathrm{t}}\right) \tag{17}$$

$$\dot{p}_{\mathrm{uc}} = k_3\left(W_{\mathrm{air}} + W_{\mathrm{LPegr}} - W_{\mathrm{c}}\right) \tag{18}$$

$$\dot{p}_{\mathrm{dt}} = k_4\left(W_{\mathrm{t}} - W_{\mathrm{LPegr}} - W_{\mathrm{out}}\right) \tag{19}$$

$$\dot{P}_{\mathrm{c}} = \frac{1}{\tau_{\mathrm{tc}}}\left(-P_{\mathrm{c}} + \eta_{\mathrm{m}}\eta_{\mathrm{t}} c_{\mathrm{p}} T_{\mathrm{x}}\left(1-\left(\frac{p_{\mathrm{dt}}}{p_{\mathrm{x}}}\right)^{\mu}\right)W_{\mathrm{t}}\right), \tag{20}$$

$$\text{where } k_1 = \frac{RT_{\mathrm{i}}}{V_{\mathrm{i}}},\ k_2 = \frac{RT_{\mathrm{x}}}{V_{\mathrm{i}}},\ k_3 = \frac{RT_{\mathrm{uc}}}{V_{\mathrm{uc}}},$$

$$k_4 = \frac{RT_{\mathrm{dt}}}{V_{\mathrm{dt}}},\ k_{\mathrm{e}} = \frac{\eta_{\mathrm{v}} N V_{\mathrm{d}}}{120 R T_{\mathrm{i}}}.$$

In Equation (16), the mass flow rate of the cylinder flow W_{ie} is represented as $k_e p_i$ by the speed-density equation where η_v, V_d, and N are the volumetric efficiency, displacement volume, and engine RPM, respectively.

In fact, referring to Equations (1) ~ (15), the number of system states for the air-path system with dual-loop EGR is generally more than 10 to describe the system accurately. The higher the order the model has, the more information it can have in it, and the reduced order model may not have all the information of the system. However, when considering real applications, implementation of the controller in an engine control unit (ECU) would be very difficult when the order of the model where a controller is developed is too high. It is important to develop a practical controller with high performances using only limited information of the reduced order model. By designing a robust control algorithm, uncertainties of the reduced model should be addressed. Hence, a control-oriented fifth-order model of the air-path with dual-loop EGR and VGT is implemented consisting of only primary states of the system. The model is composed of four pressure states and one power dynamics. By omitting fraction states, the order of the model can be reduced by more than 3. In this study, exhaust temperature is assumed to have only two constant values, and the other states including fraction states omitted in the reduced model are assumed to be constant.

3. CONTROLLER DESIGN

3.1. Set-point Transformation

Figure 2 shows the set-point transformation to get the desired values of control states (Upadhyay, 2001; Yoon, 2011). After a set-point concerning the target emissions and desired values of torque is determined, the desired values of fuel rate, AFR, EGR rate, and boost pressure are derived corresponding to the set-point. Note that, because the target engine of this research is a heavy-duty diesel engine, the engine speed is usually fixed during the operation of the engine. Then, when the fuel rate values are determined in accordance with the desired torque, the desired values of the fresh air flow rate and total EGR flow rate can be computed from the set-point. This procedure implies that combustion variables such as the EGR rate or AFR are not usually measurable so these variables should be transformed into control variables like flow rate or pressure, which can be treated in the air-path system for control.

The objective of the air-path control is to achieve the target set-point by letting the three control variables track the desired values of each them through the proper control algorithm. The structure of the whole control scheme is depicted in Figure 3.

3.2. Control Principles of the Model-based Controller

In Figure 3, the one coordinated controller is composed of three sub controllers. The controller, which is developed based on the reduced-order model, controls the engine plant. The measured variables are the intake pressure, exhaust pressure, fresh air flow rate, and HP EGR flow rate. The information of the other parameters or states is obtained from the control-oriented model during the

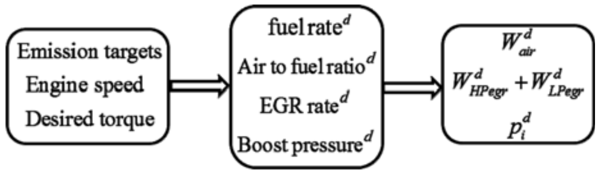

Figure 2. Set-point transformation.

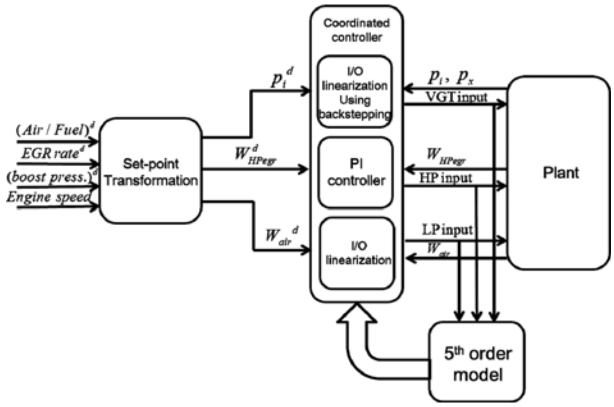

Figure 3. Control structure.

control process. Since the model dynamics describe the actual physical properties of the system, the estimated values of system parameters can be known from the model in real time. The model-based controller utilizes the information of the model and effectively controls the plant dealing with the coupling problems of it based on the information.

Note that the control inputs computed from the controller have the dimension of flows, not the valve positions. The commands of flows are transformed into position commands using an inverse model of the orifice equation, and the final position commands actuate the engine plant.

3.3. Design Procedure

A proper control algorithm should be designed for an engine air-path system that has good tracking control performance. In the procedure in Figure 2, the target control variables are the intake manifold pressure, mass flow rate of EGR flows, and fresh air flow rate. When the desired values of the three variables are given according to the target performances corresponding to operating points, the control algorithm should make the variables track their desired values.

As shown in Figure 3, three sub-controllers act as one coordinated controller. The sub controllers make input commands for VGT, HP EGR, and LP EGR systems, respectively, in cooperation with one another based on the model dynamics of the system. The design procedures will be explained separately.

3.3.1. VGT controller: Input-output linearization using back-stepping

Before beginning the controller design, it is needed to look into the structure of Equation (16). Even though it is clear that intake manifold pressure should be controlled by adjusting VGT input in the physical sense, the VGT input (VGT flow rate) is not explicitly shown in Equation (16). Instead, HP EGR input is present in Equation (16) explicitly, which should be operated only for EGR rate regulations, not for controlling intake manifold pressure. In the authors' conference paper (Kim *et al*., 2014), the control algorithm was developed simply based on Equations (16) and (17) without any modification, where the operations of HP EGR and VGT directly affect each other. Here, a VGT controller will be developed to regulate the intake manifold pressure decoupled from HP EGR input. The controller is designed using the back-stepping principle, where exhaust pressure is regarded as a fictitious input for the control of intake manifold pressure.

First, the error of intake pressure control is defined as Equation (21).

$$e_1 = p_i - p_i^d. \tag{21}$$

Differentiating Equation (21) with respect to time and combining with Equation (16), the error dynamics is presented as Equation (22).

$$\dot{e}_1 = k_1\left(W_c - k_e p_i\right) + k_1 W_{HPegr} - \dot{p}_i^d \triangleq -\lambda_1 e_1. \tag{22}$$

Applying input-output linearization for the error dynamics to be converged, the control law is derived as Equation (23).

$$\begin{aligned} z_1 &= W_{HPegr}^d = W_{HPegr}^d(p_i, p_x^d) \\ &= -(W_c - k_e p_i) - k_1^{-1}\lambda_1 e_1 + k_1^{-1}\dot{p}_i^d. \end{aligned} \tag{23}$$

Note that the control law is denoted as z_1 instead of u_1 standing for a fictitious input.

If Equation (8) is arranged with respect to p_x, Equation (24) is derived.

$$p_x = \left(\frac{-p_i^{1+\frac{1}{\gamma}} + \sqrt{p_i^{2+\frac{2}{\gamma}} + 4\alpha p_i^{\frac{2}{\gamma}}}}{2\alpha} \right)^{\frac{\gamma}{1-\gamma}}, \tag{24}$$

$$\text{where } \alpha = \frac{(W_{HPegr})^2 R T_x (\gamma - 1)}{A_{HP}{}^2 2\gamma}.$$

By combining Equations (23) and (24), the exhaust pressure value corresponding to the fictitious input z_1 is derived as described in Equation (25).

$$p_x^d = \left(\frac{-p_i^{1+\frac{1}{\gamma}} + \sqrt{p_i^{2+\frac{2}{\gamma}} + 4\alpha p_i^{\frac{2}{\gamma}}}}{2\alpha} \right)^{\frac{\gamma}{1-\gamma}}, \tag{25}$$

$$\begin{aligned} \text{where } \alpha &= \frac{(z_1)^2 R T_x (\gamma - 1)}{A_{HP}{}^2 2\gamma} \\ &= \frac{(-(W_c - k_e p_i) - k_1^{-1}\lambda_1 e_1 + k_1^{-1}\dot{p}_i^d)^2 R T_x (\gamma - 1)}{A_{HP}{}^2 2\gamma}. \end{aligned}$$

Equation (25) implies what value the exhaust pressure should be for p_i to track the target when current value of intake pressure is known. Consequently, the equation of Equation (25) has the role of assigning proper exhaust pressure targets in order to enhance the tracking performance of intake pressure. Unlike intake pressure, exhaust pressure is not a performance variable of the system (Upadhyay, 2001; Jin *et al.*, 2014). The desired values of exhaust pressure in Equation (25) are assigned to the controller for precise control of intake pressure.

In turn, a new algorithm is designed for exhaust pressure to track the target of Equation (25) in a similar way to Equations (21) ~ (23).

$$e_2 = p_x - p_x^d \tag{26}$$

$$\dot{e}_2 = k_2(W_{ex} - u_1 - W_{HPegr}) - \dot{p}_x^d \triangleq -\lambda_2 e_2, \tag{27}$$

$$u_1 = W_t^d = W_{ex} - W_{HPegr} + k_2^{-1}\lambda_2 e_2 - k_2^{-1}\dot{p}_x^d. \tag{28}$$

In Equation (28), the actual input of VGT is derived for the control of exhaust pressure. As a result, if the VGT operates properly for exhaust pressure to track the target of Equation (25), simultaneous control of intake pressure also can be achieved by the principle of back-stepping. As a result, using the form of input-output linearization through back-stepping, it is possible to regulate intake manifold pressure by the VGT input u_1. It should be noted that HP EGR input is not involved in the control of intake pressure.

3.3.2. HP EGR controller: PI controller

Since the HP EGR input is mathematically decoupled from the intake manifold pressure from the design procedure of the VGT controller, it is possible to utilize HP EGR input only for regulating the EGR rate. A simple PI controller with no feed-forward terms is suggested for the control of HP EGR flow since there is no state representing the EGR flow directly.

In this sub-controller, it is assumed that the current values of HP EGR flow rate can be measured with precision, and the target values of the flow, which considers system coupling properties, are also given. The control law is derived as Equation (30) using Equation (29).

$$e_3 = W_{HPegr} - W_{HPegr}^d, \tag{29}$$

$$u_2 = W_{HPegr}^{in} = -\lambda_3 e_3 - \lambda_4 \int e_3 dt. \tag{30}$$

Note that the current HP EGR flow is controlled by Equation (30), and this EGR flow is reflected on the feed-forward terms of other sub-controllers u_1, u_3 in real time.

3.3.3. LP EGR controller: Input-output linearization

A new way to control LP EGR flow is also suggested in this paper. Fresh air flow is blended with LP EGR flow at the compressor inlet in the air-path system with a dual-loop EGR, and the mixed flow is turbocharged into intake a

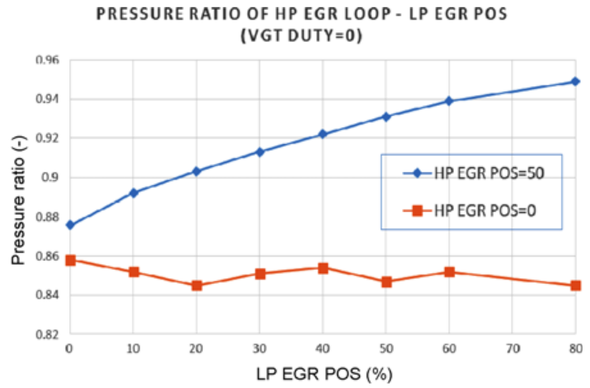

Figure 4. Influence of LP EGR operation on the pressure ratio of the HP EGR loop (p_i/p_x).

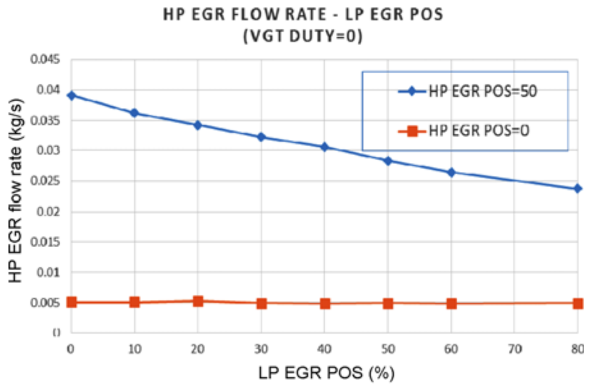

Figure 5. Influence of LP EGR operation on the HP EGR flow rate.

manifold through the compressor. The suggested method is to control the LP EGR valve depending on the information of the fresh air flow rate. Instead of utilizing the LP EGR flow rate information for which sensors are not available in the production engine, the developed control method uses information of the fresh air flow rate for feedback control, which is easily measured, even in the production engine. For a detailed explanation, let's examine Figures 4 ~ 6.

Figures 4 ~ 6 represent the influences of LP EGR operation on the pressure ratio across the HP EGR loop, HP EGR flow rate, and fresh air flow rate, respectively. All the data of the figures are obtained from the experiments at steady states when the engine speed is 1,400 RPM and the load is 413 N·m. Two lines exist in each figure, and those lines show different values of HP EGR valve when VGT duty is zero. Referring to Figures 4 and 5, LP EGR operations seldom affect the pressure ratio of the HP EGR loop and HP EGR flow rate when the HP EGR valve is closed. However, when the HP EGR valve is opened by 50 %, the pressure ratio is increased, which leads to a decrease of HP EGR flow rate, in accordance with the increase of the LP EGR valve position.

To deal with the coupling properties of HP EGR and LP

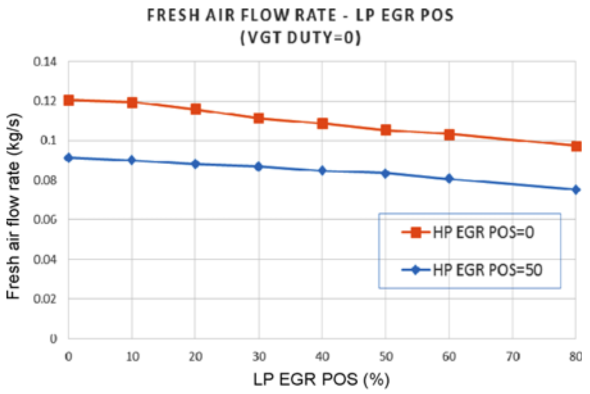

Figure 6. Influence of LP EGR operation on the fresh air flow rate.

EGR systems, a model-based control algorithm for the LP EGR loop is needed. Figure 6 shows that as the LP EGR valve is opened more, the fresh air flow rate is decreased whether the HP EGR valve is opened or not. Note that the operating conditions are exactly the same as those of Figures 4 and 5. Thus, the LP EGR valve can be adjusted based on the information of the fresh air flow rate to supply the desired flow rate of the fresh air flow when the position of the HP EGR valve and VGT duty are fixed. Making use of the properties, the model-based LP EGR controller is designed using the information of fresh air flow rate as follows.

A new state of fresh air flow rate for the air path with dual-loop EGR is derived as in Equation (31) by differentiating Equation (15) with respect to time.

$$\dot{W}_{\mathrm{air}} = \frac{a}{p_{\mathrm{a}}} k_3 (W_{\mathrm{air}} + W_{\mathrm{LPegr}} - W_{\mathrm{c}}) \cdots \times \left(\frac{\gamma - 1}{-2\gamma} b^{-0.5} + \frac{1}{\gamma} \left(\frac{p_{\mathrm{uc}}}{p_{\mathrm{a}}}\right)^{\frac{1-\gamma}{\gamma}} b^{0.5} \right), \tag{31}$$

where $k_1 = \frac{RT_{\mathrm{i}}}{V_{\mathrm{i}}}$, $k_2 = \frac{RT_{\mathrm{x}}}{V_{\mathrm{x}}}$, $k_{\mathrm{e}} = \frac{\eta_{\mathrm{v}} N V_{\mathrm{d}}}{120 RT_{\mathrm{i}}}$,

$$a = A_{\mathrm{air}} p_{\mathrm{a}} \frac{1}{\sqrt{RT_{\mathrm{a}}}} \sqrt{\frac{2\gamma}{\gamma - 1}}, \quad b = 1 - \left(\frac{p_{\mathrm{uc}}}{p_{\mathrm{a}}}\right)^{\frac{\gamma - 1}{\gamma}}.$$

The error dynamics of the fresh air flow rate is taken as Equation (32), and input-output linearization is applied to the equation as in Equation (33).

$$e_4 = W_{\mathrm{air}} - W_{\mathrm{air}}^{\mathrm{d}}, \tag{32}$$

$$\dot{e}_4 = \frac{a}{p_{\mathrm{a}}} k_3 (W_{air} + u_3 - W_{\mathrm{c}}) \cdots \times \left(\frac{r - 1}{-2r} b^{-0.5} + \frac{1}{r} \left(\frac{p_{\mathrm{uc}}}{p_{\mathrm{a}}}\right)^{\frac{1-r}{r}} b^{0.5} \right) - \dot{W}_{\mathrm{air}}^{\mathrm{d}} \triangleq -\lambda_5 e_4. \tag{33}$$

Then, the corresponding input for LP EGR is derived as Equation (34).

$$u_3 = W_{\mathrm{LPegr}}^{\mathrm{in}} = -\frac{\lambda_5 e_4 p_{\mathrm{a}}}{k_3 ac} - W_{\mathrm{air}} + W_{\mathrm{c}} + \frac{p_{\mathrm{a}}}{k_3 ac} \dot{W}_{\mathrm{air}}^{\mathrm{d}}, \quad \text{where } c = \left(\frac{r - 1}{-2r} b^{-0.5} + \frac{1}{r} \left(\frac{p_{\mathrm{uc}}}{p_{\mathrm{a}}}\right)^{\frac{1-r}{r}} b^{0.5} \right). \tag{34}$$

In Equation (34), u_3 determines the desired position of the LP EGR valve for the fresh air flow rate to track its desired values when the current values of the fresh air flow rate are known.

The final coordinated controller has the form of a controller consisting of feedforward terms and feedback terms with constant gains. u_1, u_2, and u_3 stand for the control inputs of VGT, HP EGR, and LP EGR, respectively. u_1 controls the VGT vane for regulating exhaust manifold pressure to ensure precise control of the intake manifold pressure. u_2 makes the input command for the HP EGR valve to supply the proper EGR rate into cylinders. Finally, u_3 controls the mass flow rate of LP EGR flow to maintain the desired flow rate of fresh air into the intake manifold. Those control inputs act on the plant in cooperation with one another based on the model dynamics.

4. SIMULATIONS

4.1. GT-POWER Model of the System

The GT-POWER model of the target diesel engine was developed and its configuration is depicted in Figure 7. GT-POWER is an industry-standard engine simulation tool, and it is widely used not only by every major engine manufacturer but also by control system engineers for ECU development and controller design for engines. GT POWER provides various sophisticated modeling tools appropriate for controller design with very realistic engine environments.

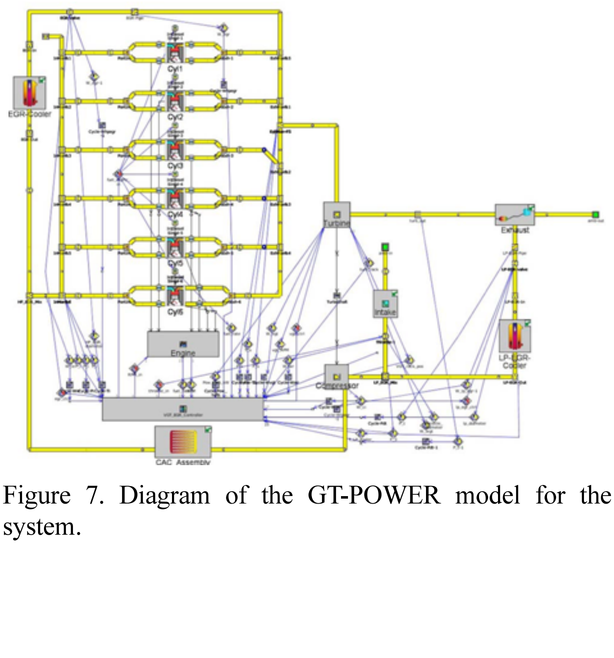

Figure 7. Diagram of the GT-POWER model for the system.

Table 1. Specifications of the base engine.

System specification	Unit	Value
Bore	mm	100
Stroke	mm	125
Number of cylinder	EA	6
Compression ratio	-	17.5
Firing order	-	1-5-3-6-2-4
Displacement volume	m^3	$6 \cdot 9.87148 \cdot 10^{-4}$
Intake manifold volume	m^3	0.001920
Exhaust manifold volume	m^3	0.0006352
Compressor upstream volume	m^3	0.0054

The base engine is a 6,000 cc heavy-duty diesel engine with the specifications described in Table 1, and the GT-POWER model was experimentally validated with the test-bench.

4.2. Control Verification

To verify the controller, several simulations were carried out through the GT POWER model for various operating points. The operating conditions are described in Figure 8. Due to the operating characteristics of this heavy-duty engine, the engine speed remains constant at each operating point, and the fuel rate is determined by the desired load values. To verify the control performance of the controller with precision, the simulation was conducted for many points covering all operating conditions.

The verification results are shown in Figures 9 ~ 11 under the operating conditions.

In Figures 9 ~ 11, the desired values of performance variables such as intake manifold pressure or fresh air flow rate satisfying the set-point were obtained experimentally from the engine test bench for each operating point. On the other hand, the desired values of exhaust manifold pressure

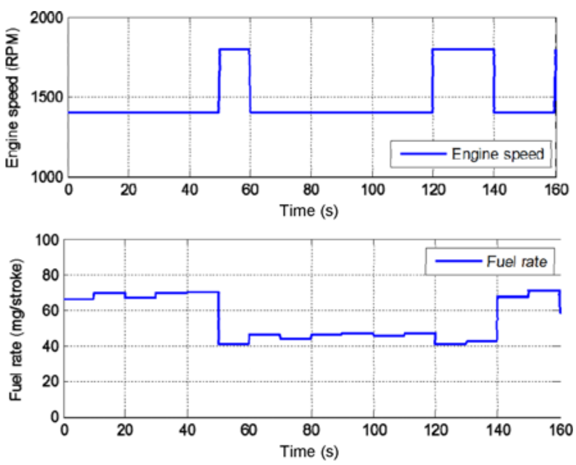

Figure 8. Operating conditions for control verification.

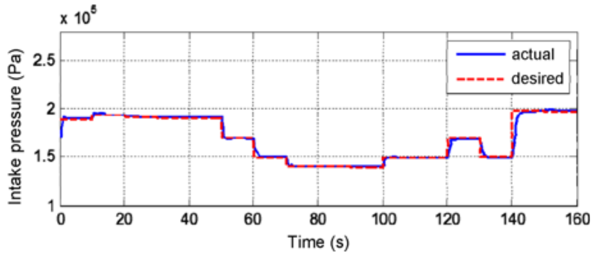

Figure 9. Intake manifold pressure for the suggested control.

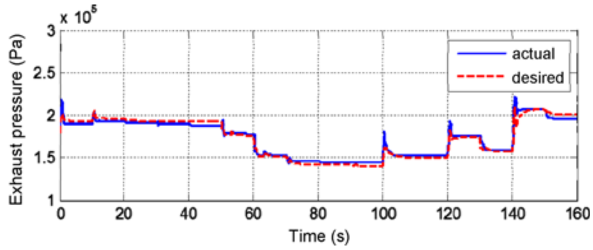

Figure 10. Exhaust manifold pressure for the suggested control.

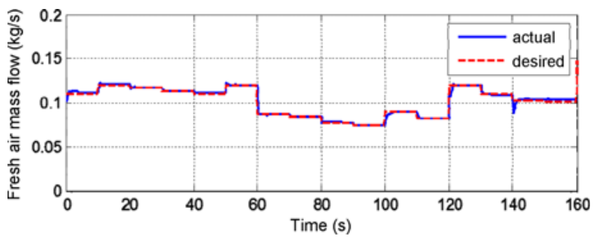

Figure 11. Fresh air flow rate for the suggested control.

were given by Equation (25) in real time for ensuring good tracking performance of the intake pressure.

As for the results, although the controller is developed using only the fifth-order model, which has much poorer accuracy than that of the full plant model, it shows high tracking performance without breaking the system balance. The intake pressure and fresh air flow track the desired trajectories very accurately without any overshoot or oscillatory behaviors. The mean tracking error is just 0.90 % for the intake pressure and 1.15 % for the fresh air flow

Table 2. Tracking error for the suggested controller.

Control state	Mean tracking error (%)	Max. steady state error (%)
Intake manifold pressure	0.90	1.20
Exhaust manifold pressure	1.84	2.68
Fresh air flow rate	1.15	1.60

rate. The maximum error in steady states is 1.20 % for the intake pressure and 1.60 % for the fresh air rate. Referring to the result of Figure 10, the shape of the desired trajectory itself for exhaust pressure is somewhat rough since this trajectory consists of an implicit target calculated in Equation (25). VGT controls the exhaust pressure to track the desired trajectory, and intake manifold pressure, the true control target, is controlled simultaneously with high accuracy.

4.2.1. Comparison with the PID controller

Additional simulations were conducted to compare the performance of the suggested model-based controller with that of a PID controller. The results of the PID control under the same operating conditions are shown in Figures 12 and 13.

Compared to the results of the suggested control, the control performance, particularly for the transient states, is much poorer in the case of the PID control. Especially for the abruptly changed operating points, the response shows large overshoot or undershoot (50 s ~ 60 s), or the response is too slow to track the desired response (130 s ~ 140 s). Note that the control gains for the PID control are intentionally set to be fixed in this simulation for direct comparison of control performances. For the PID control, a fine tuning process for each operating point is needed to improve the control performance, which means the values of the control gains may be varied for each operating point. However, it is obvious that fine tuning for each operating point becomes much more difficult as the number of operating points to be tuned increases. On the other hand, the response of the suggested model-based control shown in Figures 9 and 11 shows high tracking performance for both transient and steady states, even though its control gains are fixed throughout all of the operating points. The tracking error for the PID control is presented in Table 3, which shows that both the mean tracking error and the maximum steady state error of the suggested control are much smaller than those of the PID control method.

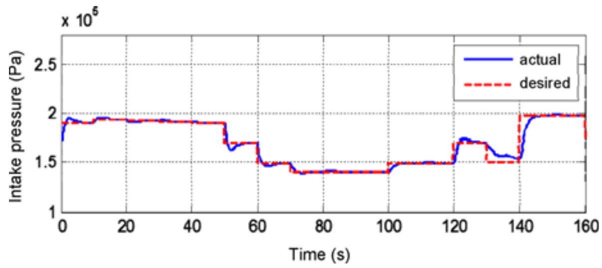

Figure 12. Intake manifold pressure for the PID control.

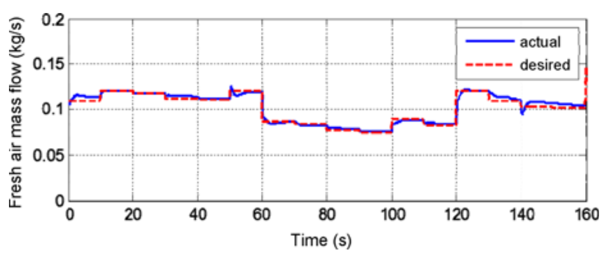

Figure 13. Fresh air flow rate for the PID control.

Table 3. Tracking error for the PID controller.

Control state	Mean tracking error (%)	Max. steady state error (%)
Intake manifold pressure	1.28	2.40
Fresh air flow rate	3.51	4.78

5. CONCLUSION

This paper suggests a coordinated control method for the combined operations of VGT, HP EGR, and LP EGR in a diesel engine air-path system. The control algorithm was designed based on a control-oriented model consisting of several pressure dynamics equations. It was assumed only four pressure and flow variables were measured from the plant with practical considerations. The coordinated control algorithm was composed of three sub-controllers managing the operations of VGT, HP EGR, and LP EGR, respectively. Each sub-controller was designed considering the coupling effects of EGR and air flows based on the model dynamics equations. Even though the control algorithm is developed based on the simple control-oriented model with poor accuracy, the control performance was verified through the simulation of a proven GT-POWER engine model of much higher order.

Certainly, the accuracy of a model-based control method for a diesel engine air path can be vulnerable to the uncertainties of its base model. However, it is now clear that model-based controllers are very easy to tune with significantly reduced tuning parameters and are also implementable in production engines without using prohibitively expensive sensors. In that sense, this paper shows the promise of practical applications of the developed model-based controller.

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