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Pressure and Flow Based Control of a Turbocharged Diesel Engine Air-path System Equipped with Dual-Loop EGR and VGT*

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Abstract— **This paper describes a pressure and flow based control method of diesel engine air-path systems equipped with dual-loop exhaust gas recirculation (EGR) and variable geometry turbine (VGT). For the control of dual-loop EGR systems, fraction states are usually controlled, but how to select the control outputs is still controversial. In this paper, considering availability and reliability of sensors, pressure and mass flow states are selected as control targets instead of fraction states. A robust nonlinear control method, input-output linearization is applied for the states of intake manifold pressure, exhaust manifold pressure, and fresh air mass flow rate in order to satisfy the target emissions. The control performance is verified by simulation based on the valid model of a heavy-duty 6000cc engine air-path.**

I. Introduction

The increasingly strict environmental regulations for diesel engines have forced the worldwide automotive industries to develop environmental-friendly engine configurations. Nowadays, a diesel engine is called 'clean diesel' since its chronic problems such as emissions and noise have been improved a lot while it keeps its advantages of good vehicle performances as technology develops. For example, exhaust treatment systems including diesel particulate filter (DPF), diesel oxidation catalyst (DOC), and selective catalytic reduction (SCR) have been applied for diesel engines to cope with harmful emissions of a diesel engine. Besides, exhaust gas recirculation (EGR) and variable geometry turbine (VGT) systems are commonly equipped in most diesel engines today. EGR system has a role to recirculate exhaust gas into intake manifold, which reduces nitrogen oxides (NOx), and VGT is for enhancing the engine performance even in low-load conditions. In particular, the combination of EGR and VGT in a diesel air-path system has the potential to reduce emissions significantly since it can adjust fundamental properties of combustion in the diesel engine. Gas compositions in cylinders of the engine can be manipulated with EGR and VGT so that advanced combustion modes such as low temperature combustion (LTC) or homogeneous charge compression ignition (HCCI) are realizable resulting in significant improvement of emissions. Recently, dual-loop EGR systems, using high pressure (HP) EGR and low pressure (LP) EGR systems simultaneously, are applicable for the effective utilization of EGR functions. Mostly, HP EGR has a function to increase temperature in intake manifold of diesel engines, and LP EGR recirculates clean EGR gas which came through after-treatment systems and the cooler to offer high mass flow rate [8]-[12][14].

In accordance with the development of hardware configurations, precise control of the air-path system which is a highly-coupled system has been required. The well-known coupling effect between HP EGR system and VGT, both of which are driven by exhaust gas, is investigated in [6], and recently, static couplings of the dual-loop EGR system are dealt with in [8]. To handle control problems of the air-path system effectively, various coordinated control methods of VGT and EGR systems based on mathematical models have been investigated. Many studies have been conducted on the control of single-loop EGR (with only HP EGR) system [1]-[7], but, for dual-loop EGR systems, which are too complex systems to control precisely, there is not much research on it.

The air-path control is focusing on reducing the well-known hazardous substances of diesel engines, NOx and particulate matter (PM) simultaneously, which are in a trade-off relationship. Set-points determined from the target emission are concerned with air to fuel ratio (AFR) and EGR rate. Generally, AFR is related to the formation of PM, and the EGR rate directly affects the production of NOx [2]-[6]. In addition, it is suggested that intake manifold pressure should be considered in the set-points for generating desired torque and proper utilization of the turbocharger.

In terms of the air-path control, selecting the control targets is very crucial. Once a certain set-point for satisfying environmental regulations or other engine performances is given, we need to select the corresponding control states which are measurable. Conventionally, for the single-loop EGR and VGT system, intake manifold pressure and compressor flow have been regarded as main control states for control [2][4][5]. For the dual-loop EGR systems, which is a higher order system than the single-loop system, how to select governing control variables corresponding to the set-points is not yet definitely settled. In most related studies, oxygen or burnt gas fraction states with other intake manifold states are considered [9]-[12][14]. In [11] and [12], estimation methods of air fraction states or LP EGR flow rate are investigated. Control algorithm for states including air fractions in dual-loop EGR systems is designed using singular perturbation methodology in [9], and another control method using motion planning in [10]. However, in real applications, sensors for measuring fraction states are not available on production engines, and sensing speed of the sensor is relatively slow compared to other sensors so it is not suitable for transient control. Besides, fraction states are somewhat secondary states determined by other primary states such as pressure and flow in the air-path system. Hence, only pressure and mass flow states are selected as control outputs in this

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study considering the sensor reliability and expandability of the conventional controllers.

There exist three control inputs: valve positions of HP, LP EGRs and vane position of VGT in the air-path system with dual-loop EGR and VGT. Compared to the large coupling property of HP EGR and VGT, the coupling effect of the LP EGR loop is not that much. It is possible to control the LP EGR valve separately. Hence, another good point of the pressure and flow based control is that if we control only the states of pressure and flow for the control of a dual-loop EGR system, we can expand availability of the conventional controllers from the single-loop EGR system to the dual-loop system by adding a control algorithm for the LP loop only.

In this paper, a model-based nonlinear controller is designed considering the complex nonlinearities of the air-path system equipped with the dual-loop EGR and VGT. One main characteristic of the dual-loop EGR system distinct from the single-loop EGR system is that fresh air flow is not the same as compressor flow any more. Fresh air flow and LP EGR flow are mixed into the compressor flow, so a new model for the fresh air flow rate only is suggested for implementing an effective controller. The control states are p_i (intake manifold pressure), p_x (exhaust manifold pressure), and W_{air} (fresh air mass flow rate). These states can be easily measured or estimated using sensors equipped on the production engine such as manifold absolute pressure (MAP) and mass air flow (MAF) sensors. By controlling p_i , p_x and simultaneously, it is possible to manipulate HP EGR flow and VGT flow properly to meet the emission goals [4][5][7]. If W_{air} is precisely controlled, we can adjust not only the LP EGR rate but also the total EGR rate.

This paper is organized as follows: in section II, main equations of a plant model of the air-path system with dual-loop EGR and VGT are described. In section III, control principles and design process of the controller based on the reduced order model for the system is represented. Finally, in section IV, the control performance is validated through simulations based on a valid engine model.

II. System Modeling

The air-path system model is implemented based on the specifications of a heavy-duty 6000cc diesel engine equipped with dual-loop EGR system and VGT. The configuration of the whole system is described in Fig. 1.

A. System Description

Different from the single EGR system, LP EGR loop is included in the system. There are exhaust after-treatment systems including DOC in the LP-EGR loop. After passing through the after-treatment systems, the flow is divided into two sub-flows: one is to the exhaust throttle through SCR and the other is to the compressor inlet. Mass flow through the LP EGR loop is mixed with fresh air mass flow through the intake throttle in front of compressor inlet. Likewise, HP EGR flow is mixed with the compressor flow in the intake manifold. Then, combustion reaction occurs in cylinders generating new exhaust gas at exhaust manifold. All the flows affect the rate of physical quantities change such as pressure and temperature in corresponding locations.

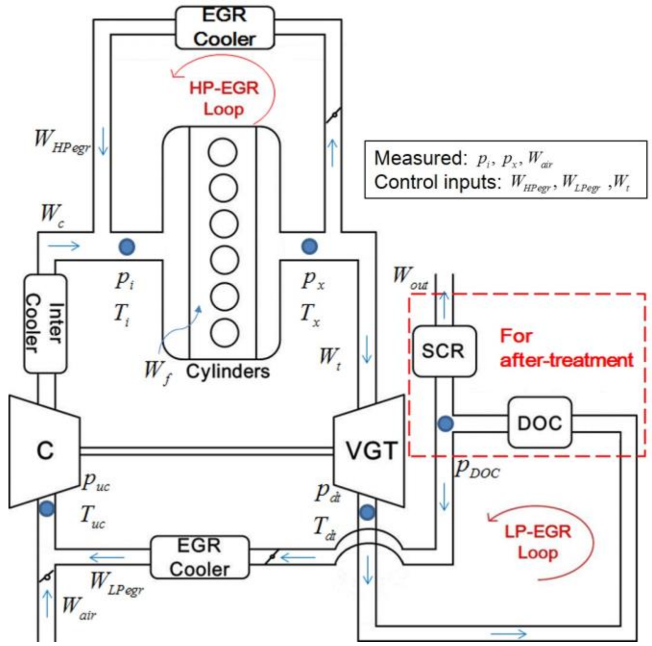

Fig. 1 The schematic of the air-path system with a dual-loop EGR system and VGT

The variables p_Δ , T_Δ , F_Δ , and V_Δ stand for pressure, temperature, burnt gas fraction, and volume in Δ . W_Δ means the mass flow rate through Δ . The subscripts, i, x, c, t, uc, dt are intake manifold, exhaust manifold, compressor, turbine, upstream of compressor, and downstream of turbine respectively, which are main locations or components to describe the system properties. With the LP EGR loop added, the dynamics at compressor upstream and turbine downstream carry an important meaning as much as intake and exhaust manifolds. Note that in the case of a single-loop EGR system, which does not have LP EGR loop, pressure or temperature developed on compressor upstream and turbine downstream can be regarded as constant. Model equations of the system with dual-loop EGR are represented as (1)-(15).

B. Pressure Dynamics

Based on the ideal gas law with mass flow, pressure dynamics are described as (1)-(4). The equations are under the assumption that the temperature of which the dynamics is relatively slow is not changed in same operating points.

$$\dot{p}_i = \frac{R\gamma}{V_i}\left(T_c W_c + T_x W_{HPegr} - T_i W_{ie}\right) \tag{1}$$

$$\dot{p}_x = \frac{R\gamma}{V_x}\left(T_e W_{ex} - T_x W_{HPegr} - T_x W_t\right) \tag{2}$$

$$\dot{p}_{uc} = \frac{R\gamma}{V_{uc}}\left(T_a W_{air} + T_{dt} W_{LPegr} - T_c W_c\right) \tag{3}$$

$$\dot{p}_{dt} = \frac{R\gamma}{V_{dt}}\left(T_x W_t - T_{dt} W_{LPegr} - T_{out} W_{out}\right) \tag{4}$$

where γ and R are the specific ratio and ideal gas constant, respectively. Note that DPF is not contained in the after-treatment systems of this system so we assume the pressure drop through the after-treatment system in LP-EGR loop is negligible stated as $p_{dt} \approx p_{DOC}$.

C. Temperature Dynamics

Temperature dynamics are described as (5)-(6). Note that the dynamics speed of temperature is much slower than that of other primary states.

$$\dot{T}_i = \frac{RT_i}{p_i V_i}\left[W_c(\gamma T_{ic} - T_i) + W_{HPegr}(\gamma T_{HPegr} - T_i)\right] \quad (5)$$

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

D. Power Dynamics

Power balance of a turbocharger mathematically described as (7).

$$\dot{P}_c = \frac{1}{\tau_{tc}}\left(-P_c + \eta_m P_t\right) \quad (7)$$

where P_c, P_t, τ_{tc}, 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 estimated by (8).

$$P_t = \eta_t c_p T_x \left(1 - \left(\frac{p_{dt}}{p_x}\right)^{\mu}\right) W_t \quad (8)$$

Where η_t is the turbine efficiency and c_p the specific heat at constant pressure.

E. Burnt Gas Fraction Dynamics

Burnt gas fraction is defined as the fraction of burnt gas of total gas composition. When m_Δ, m_{Δ_air}, and λ_s stand for mass of burnt gas, air in Δ, and stoichiometric ratio respectively, burnt gas fraction dynamics in intake, exhaust manifolds and compressor inlet are derived as (9)-(11) [14].

$$\dot{F}_i = \frac{1}{m_i}(F_{uc} - F_i)W_c + \frac{1}{m_i}(F_x - F_i)W_{HPegr} \quad (9)$$

$$\dot{F}_x = \frac{W_{ie}}{m_x}F_i - \frac{W_{ie} + W_f}{m_x}F_x + \frac{W_f}{m_x}\left(1 + \lambda_s\right) \quad (10)$$

$$\dot{F}_{uc} = \frac{1}{m_{uc}}\left[(F_x - F_{uc})W_{LPegr} - F_{uc}W_{air}\right] \quad (11)$$

$$where\ F_i = \frac{m_i - m_{i\_air}}{m_i},\ F_x = \frac{m_x - m_{x\_air}}{m_x},$$

$$\text{and } F_{uc} = \frac{m_{uc} - m_{uc\_air}}{m_{uc}}$$

F. Flow Equations

Compressor flow which mainly affected by the pressure ratio across the compressor and the compressor power is represented in (12).

$$W_c = \frac{\eta_c}{T_{uc}c_p}\frac{P_c}{\left(\frac{p_i}{p_{uc}}\right)^{\frac{\gamma-1}{\gamma}} - 1} \quad (12)$$

Where η_c is the compressor efficiency and γ the specific heat ratio.

The behavior of EGR flows controlled by valves such as W_{HPegr} and W_{LPegr} can be described using the orifice equation (13)-(14).

$$W_{HPegr} = A_{HP}p_x\sqrt{\frac{2\gamma}{RT_x(\gamma-1)}\left(\left(PR_{HP}\right)^{\frac{2}{\gamma}} - \left(PR_{HP}\right)^{\frac{\gamma+1}{\gamma}}\right)}$$

$$where\ \ PR_{HP} = \max\left(\frac{p_i}{p_x}, \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}\right) \quad (13)$$

$$W_{LPegr} = A_{LP}p_{dt}\sqrt{\frac{2\gamma}{RT_{dt}(\gamma-1)}\left(\left(PR_{LP}\right)^{\frac{2}{\gamma}} - \left(PR_{LP}\right)^{\frac{\gamma+1}{\gamma}}\right)}$$

$$where\ \ PR_{LP} = \max\left(\frac{p_{uc}}{p_{dt}}, \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}\right) \quad (14)$$

In (13)-(14), the parameters of PR_{HP} and PR_{LP} represent the pressure ratio across the flows, and A_{HP}, A_{LP} the effective areas of the flows which are determined by the valve positions each.

To describe the behavior of fresh air flow in the dual-loop EGR system, which is distinct from compressor flow, a new model for the air flow rate in dual-EGR systems is designed as (15) using the orifice equation. In (15), we assumed that the position of intake throttle is fixed so the effective area A_{air} is constant.

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

Based on (1)-(15), the 10th order plant model (full order model) of the air-path system is implemented and will be validated in the next section.

III. Controller Design

A. Control Principles

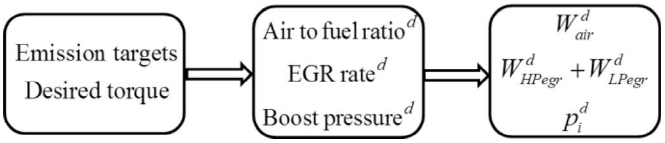

Fig. 2 The set-point transformation

Fig. 2 shows the set-point transformation to get the desired values of control states. From the target emissions and desired values of torque, the desired values of AFR, EGR rate, and boost pressure are determined as a set-point. Then, desired values of fresh air flow rate and total EGR flow rate can be computed from the set-point. This procedure implies that combustion variables such as EGR rate or AFR are not usually measurable so these variables should be transformed into variables of flow or pressure, which can be treated in the air-path system for control. In turn, p_i , p_x , and W_{air} are selected as control outputs for feedback control, which can satisfy with the target values of flow and pressure. Consequently, we can achieve the emission targets by controlling p_i , p_x , and W_{air} . The diagram of the whole control system is described in Fig. 3.

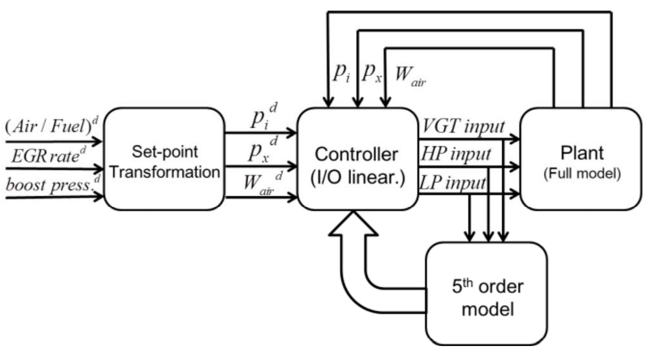

Fig. 3 Principle of the air-path control

Except the set-point transformation block, the system has the look of a general MIMO (multi-input multi-output) feedback control system. The controller, which is developed based on the reduced-order model, controls the full model. 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 in an inverse model of the orifice equation, and the position commands actuate the engine plant.

B. The Control-oriented Model

For the controller design, we implement the 5th order model of the air-system with dual-loop EGR and VGT as (16)-(20) under the assumptions as follows:

- Burnt gas fraction states, which are difficult to measure and not directly coupled with other dynamics, are omitted.

- Transition of temperature of which dynamics is relatively slow is ignored.

- Temperatures of all flows in the same location are in the same level.

$$\dot{p}_i = k_1\left(W_c + W_{HPegr} - k_e p_i\right) \tag{16}$$

$$\dot{p}_x = k_2\left(W_{ex} - W_{HPegr} - W_t\right) \tag{17}$$

$$\dot{p}_{uc} = k_2\left(W_{air} + W_{LPegr} - W_c\right) \tag{18}$$

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

$$\dot{P}_c = \frac{1}{\tau_{tc}}\left(-P_c + \eta_m \eta_t c_p T_x \left(1 - \left(\frac{p_{dt}}{p_x}\right)^{\mu}\right) W_t\right) \tag{20}$$

$$where\, k_1 = \frac{RT_i}{V_i},\ k_2 = \frac{RT_x}{V_i},\ k_3 = \frac{RT_{uc}}{V_{uc}},$$

$$k_4 = \frac{RT_{dt}}{V_{dt}},\ k_e = \frac{\eta_v N V_d}{120 R T_i}$$

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

In general, model simplicity is in a trade-off relationship with model accuracy. The more complicated the model is, the more information it can have in it. However, when considering real applications, implementation of the controller in an electronic control unit (ECU) would be very difficult when the order of the model where a controller is developed is too high. Besides, it is clear that we are not be able to use all the accurate information about states or parameter from the plant in real applications. The control-oriented model is also needed in verifying robustness of the suggested controller. Hence, a control-oriented 5th order model of the air-path with dual-loop EGR and VGT is introduced consisting of only primary states of the system.

In the reduced order model, less information is available for identifying system parameters such as compressor efficiency and turbine efficiency. These two parameters have enormous effects on the whole air-path system because they are directly related with turbocharger dynamics. Fraction states and temperature dynamics are also ignored in the reduced model.

In this study, compressor efficiency in the reduced model is assumed to have two constant values: one is for relatively high load conditions and the other for low loads. Likewise, turbine efficiency in the reduced model is assumed to have constant values in the same VGT vane positions. In addition, exhaust temperature is assumed to have only two constant values, and the other states omitted in the reduced model are assumed to be constant. The reduced order model shows poorer model accuracy than the full model mainly caused by parameter errors described in Fig. 4.

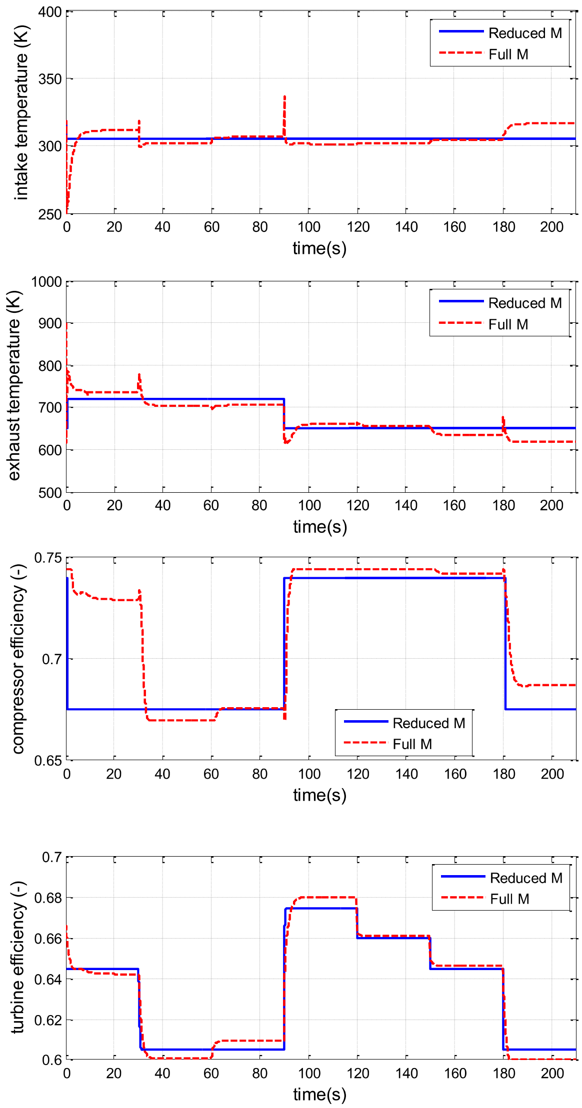

Fig. 4 Parameter errors between the full and reduced models

The full model and reduced model are validated for steady state with experimental data under operating conditions as follows: 1400 rpm/377.7 Nm, 1400 rpm/529.4 Nm, 1800 rpm/460.6 Nm, and 1800 rpm/329.0 Nm. The results are shown in Fig. 5

C. Controller Design

We will apply input-output linearization for the three states, and the error terms are defined as:

$$\begin{aligned} e_1 &= p_i - p_i^d \\ e_2 &= p_x - p_x{}^d \\ e_3 &= W_{air} - W_{air}{}^d \end{aligned} \tag{21}$$

where the superscript 'd' stands for desired value. The time derivatives of (21) are described in (22) using the control oriented model equations (16)-(20). Note that the dynamics of fresh air flow W_{air} is derived from the time derivative of (15) where p_{uc} is the only variable.

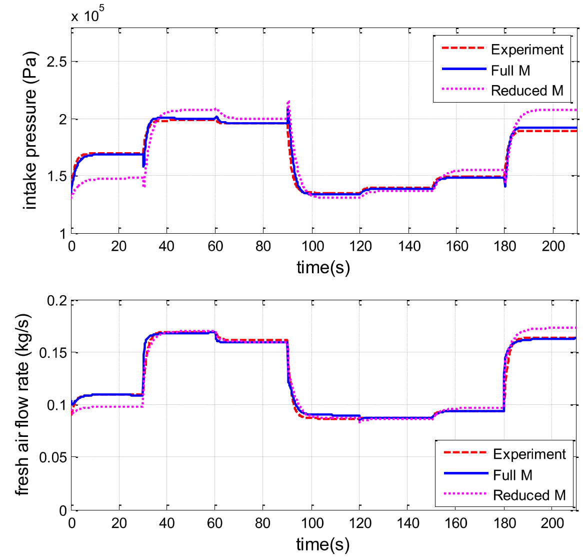

Fig. 5 Model validation (for steady state)

$$\begin{aligned} \dot e_1 &= k_1\left(W_c - k_e p_i\right) + k_1 u_1 - \dot p_i^d = -\lambda_1 e_1 \\ \dot e_2 &= k_2(W_{ex} - u_1 - u_2) - \dot p_x^d = -\lambda_2 e_2 \\ \dot e_3 &= \frac{a}{p_a} k_3 (W_{air} + u_3 - W_c)\cdots \\ &\times\left(\frac{r-1}{-2r} b^{-0.5} + \frac{1}{r}\left(\frac{p_{uc}}{p_a}\right)^{\frac{1-r}{r}} b^{0.5}\right) - \dot W_{air}{}^d = -\lambda_3 e_3 \end{aligned} \tag{22}$$

$$where\ a = A_{air} p_a \frac{1}{\sqrt{RT_a}} \sqrt{\frac{2\gamma}{\gamma - 1}},\ b = 1 - \left(\frac{p_{uc}}{p_a}\right)^{\frac{\gamma-1}{\gamma}}$$

Equation (22) stands for the error dynamics of the control states. Using the input-output linearization, we can manipulate the error dynamics to be converged. Here λ_1, λ_2, and λ_3 are the gains determining the convergent speed of the error dynamics. The corresponding control laws are derived as:

$$\begin{aligned} \therefore u_1 &= -\left(W_c - k_e p_i\right) - k_1^{-1}\lambda_1 e_1 + k_1^{-1}\dot p_i^d \\ u_2 &= W_{ex} - u_1 + k_2^{-1}\lambda_2 e_2 - k_2^{-1}\dot p_x^d \\ u_3 &= -\frac{\lambda_3 e_3 p_a}{k_3 a c} - W_{air} + W_c + k_3^{-1}\dot W_{air}^d \end{aligned} \tag{23}$$

$$where\ c = \left(\frac{r-1}{-2r} b^{-0.5} + \frac{1}{r}\left(\frac{p_{uc}}{p_a}\right)^{\frac{1-r}{r}} b^{0.5}\right)$$

The final controller has the form of a controller consisting of feedforward terms and a feedback term with constant gain. In (23), u_1, u_2, and u_3 stand for control inputs of HP EGR, VGT, and LP EGR respectively. u_1, u_2 adjust intake and exhaust pressures simultaneously with keeping the system balanced. While the combination of u_1 and u_2 is determining the amount of turbocharged flow into the intake manifold, u_3 will control mass flow rate of LP EGR to maintain desired flow rate of fresh air into the compressor inlet.

IV. SIMULATIONS

To verify the controller, the simulation is carried out based on the full model. The desired values of the control states are obtained experimentally from the engine test bench. The engine is operated on the conditions where desired loads (in Nm) are varied by the dynamometer and RPM is either 1800 or 1400. The results are shown in Fig 6-8 where the simulation conditions are the same as those of Fig. 5.

Referring to Fig. 6-8, even though the controller is developed on the reduced order model, it shows high tracking performance. The intake pressure and fresh air flow, in particular, track the desired trajectories with few overshoot or oscillatory behaviors. For the exhaust pressure, which does not determine certain engine performances, the control performance is not as good as the other states, but the steady state errors are in an acceptable range [13]. The maximum error in steady states is 2.88% for intake pressure, 1.70% for fresh air rate, and 6.54% for exhaust pressure. The largest error of exhaust pressure occurs near 200sec. which is caused by poor accuracy of the reduced order model at the condition.

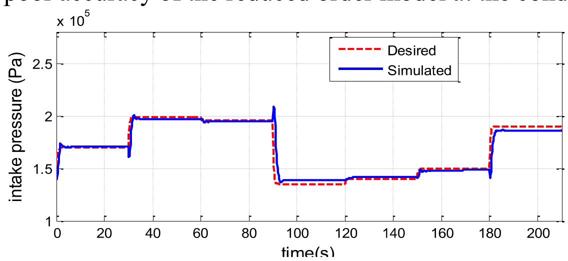

Fig. 6 Intake manifold pressure for the controller

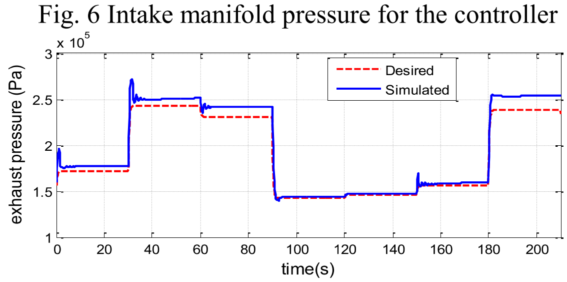

Fig. 7 Exhaust manifold pressure for the controller

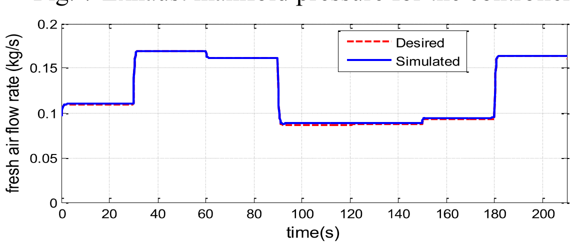

Fig. 8 Fresh air flow rate for the controller

V. CONCLUSION

This paper deals with a pressure and mass flow based control method of the air-path system with dual-loop EGR and VGT. Without taking the fraction states, which are hard to measure in real applications, the control performance is verified with the simulation on a valid model. The future works will include the experimental verification of the control system. The estimation methods of exhaust pressure also can be investigated. With the estimation methods, the whole system can be validated through HILS (hardware in the loop) tests or experiments. Besides, research can be done about the expansion of the conventional controller which was for the single-loop EGR system using the principle of a pressure and flow based control.

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