Exhaust Pressure Estimation for Diesel Engines Equipped With Dual-Loop EGR and VGT

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Abstract—This paper describes an observer design method that estimates the exhaust manifold pressure in a diesel engine with limited sensor information. In the modern model-based control of air flows, information on the exhaust pressure is often necessary for control stability. However, it is not easy to measure the exhaust pressure with sensors due to the unfavorable environment for measurement and the high cost of sensors. Therefore, a robust observer, which is applicable to engines equipped with a dual-loop exhaust gas recirculation (EGR) system and a variable geometry turbine, is designed to estimate the exhaust pressure. The observer takes the form of a Luenbergersliding mode observer in order to guarantee its asymptotic stability upon consideration of the model uncertainties. For research on state estimation and control of dual-loop EGR engines, information on the pressure states in the low pressure EGR loop must be provided. Hence, this paper also proposes an intuitive regression modeling method for the pressure states of the compressor inlet and turbine outlet with physical insights into the engine operation. Applying the regression models to the observer algorithm, the estimation performance of the exhaust pressure observer is verified experimentally in both steady state and transient conditions with a 6-L heavy-duty diesel engine.

Index Terms—Diesel engine control, dual-loop exhaust gas recirculation (EGR), exhaust pressure estimation, Lyapunov stability, sliding mode observer.

I. INTRODUCTION

S ENVIRONMENTAL regulations for diesel engines have become increasingly strict, the development of environmentally friendly engines has become a significant issue. In order to manage the harmful emissions such as hydrocarbon, nitrogen oxide, and particulate matter, various configurations have been developed for diesel engines. Nowadays, most diesel engines are equipped with variable geometry turbine (VGT) and exhaust gas recirculation (EGR) systems. With the combination of the EGR and VGT systems, the reduction of

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Color versions of one or more of the figures in this paper are available online at http://ieeexplore.ieee.org.

Digital Object Identifier 10.1109/TCST.2017.2665541

emissions without sacrificing vehicle performance is achievable in real applications. Recently, dual-loop EGR systems, which use high pressure (HP) EGR and low pressure (LP) EGR systems simultaneously, have been implemented in the air flow path of diesel engines in order to effectively use the EGR functions. With the direct recirculation of hot exhaust gases, the HP EGR increases the temperature and EGR rate in the intake manifold of diesel engines, and the LP EGR recirculates clean and relatively cool EGR gases that come through the after-treatment systems and an EGR cooler [1]-[3]. In terms of control, the air flow system is a highly coupled nonlinear system, and the concept of model-based control for engines was developed in order to treat these coupling properties of the system effectively. Many studies have been conducted on the model-based control of single-loop EGR systems [4]-[13] and a few studies have focused on dualloop EGR systems recently [1]–[3], [14], [15].

Information on the exhaust pressure is essential in the engine air flow control, because it contains important data on the engine operation status. For example, exhaust manifold pressure is primarily concerned with the flow rates of turbine and HP EGR, for which any variation has a significant influence on the entire system, and is regarded as one of states to be controlled in engines with VGT and EGR [7], [13]. Even though it is not directly related to specific engine performances, its value has crucial effects on other states. In addition, with respect to the design of the controller, the exhaust pressure should be selected as a control output in order to guarantee internal stability of the control system [7], [10], [15]. However, it is not easy to measure the exhaust pressure with a production sensor. The environment is not appropriate for measurements due to the large oscillations of the flow, high pressure, and high temperature; moreover, the sensor becomes cost prohibitive in production engines [16], [17].

Since the 2000s, various studies have been conducted on model-based estimation methods for the exhaust pressure in order to overcome the limitations of the conventional mapbased estimation methods. Andersson [18] and Andersson and Eriksson [19] proposed an estimation method for the exhaust pressure based on the first law of thermodynamics and a model of the mass of the residual exhaust gases in the cylinder. A nonlinear model-based observer was designed in [16]. In [20], the compressible flow equation was used to implement a mean-value model for the exhaust manifold pressure, and Buckland *et al.* [21] considered a reduced-order observer with a simple lead compensation in order to enhance the transient estimation performance. Chiara *et al.* [22] proposed

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Manuscript received June 25, 2016; revised November 21, 2016; accepted January 28, 2017. Date of publication February 21, 2017; date of current version February 8, 2018. Manuscript received in final form February 4, 2017. This work was supported in part by the Ministry of Science, ICT and Future Planning, South Korea, under the Information Technology Research Center support Program, supervised by the Institute for Information and Communications Technology Promotion under Grant IITP-2016-H8601-16-1005, in part by the BK21 plus program, and in part by the National Research Foundation of Korea funded by the Korean government under Grant 2010-0028680. Recommended by Associate Editor A. G. Stefanopoulou.

an estimation method for the exhaust manifold pressure in a diesel engine with a two-stage turbocharger based on the inversion of a turbocharger model. Recently, Siokos *et al.* [23] developed pressure models for exhaust manifold and turbine outlet in a gasoline engine with LP EGR system.

However, these works have not considered the simultaneous operations of an EGR and VGT system in the engine, which makes it much easier to calibrate the systems. Because the exhaust manifold pressure is directly related to the operations of the EGR and VGT, estimation methods that can consider these EGR and VGT operations simultaneously in real time are needed. Recently, some studies have been undertaken on the exhaust pressure estimation for engines with HP EGR and VGT [17], [24], [25]. In [17], a model-based method using a coordinate transformation to generate a turbine map was developed to estimate the exhaust pressure. Lee et al. [24] designed a nonlinear sliding mode observer, and Jin et al. [25] proposed an estimation method based on compressor dynamics. The limitation of these studies is that, even though dualloop EGR systems are commonly installed in diesel engines, practical studies on the estimation of the exhaust manifold pressure for dual-loop EGR engines remain absent. In [26], an exhaust pressure estimator using extrapolated VGT data maps with the effect of the turbine speed was proposed for a dual-loop EGR and VGT system; however, the authors used impractical sensor information such as the universal exhaust gas oxygen sensor signal. In this paper, the estimation algorithm of the observer for the exhaust manifold pressure that is applicable to a diesel engine equipped with a dual-loop EGR and VGT system is proposed using only limited sensor information.

In order to estimate the exhaust pressure of a dual-loop EGR system, information on the states of the LP EGR loop is essential. Because the mass flow rate of the LP EGR cannot be measured, the values for the pressure and temperature in the compressor inlet and turbine outlet, which represent the LP EGR loop, should be known in order to calculate the flow rate. However, the pressure states in the LP EGR loop, e.g., the turbine outlet pressure and compressor inlet pressure, are not measurable in production engines, which is a critical obstacle in research on dual-loop EGR engine control. Thus, this paper also develops a regression modeling method of the pressure states in the compressor inlet and turbine outlet that considers the physical characteristics of an engine with a dualloop EGR. The pressure values predicted by the regression models are used in the final observer algorithm for the exhaust manifold pressure.

The proposed observer is a robust sliding mode observer with a simple second-order form considering the simplicity of the implementation and model uncertainties. Using production sensors such as the mass air flow (MAF) and manifold absolute pressure sensors and a minimum number of map-based submodels, the observer can estimate the exhaust pressure accurately in various conditions, which also include transient operations.

The remainder of this paper is organized as follows. In Section II, a system overview and the principal equations of the model for the air flow system in a dual-loop



Fig. 1. Air flow system with a dual-loop EGR and VGT.

EGR system are described. Section III presents the design process of the sliding mode observer for the exhaust manifold pressure. In Section IV, concerns about the dual-loop EGR system for the observer design are investigated, and a new modeling method for the pressure states in the LP EGR loop is introduced. Finally, in Section V, the estimation performance of the observer with the pressure models of the LP EGR loop is experimentally evaluated in both steady state and transient conditions using a 6-L heavy-duty diesel engine with dual-loop EGR and VGT.

II. AIR FLOW SYSTEM MODEL

The base system is a 6-L heavy-duty diesel engine with a dual-loop EGR and VGT. The proposed observer is designed based on the model equations of the engine air flow system. The configuration of the engine air flow system is described in Fig. 1.

A. System Overview

The air flow system in Fig. 1 has a one-stage VGT and two EGR loops. Fresh air enters through the compressor operated by the VGT to the intake manifold and mixes with the exhaust gases from the HP EGR loop. After combustion, exhaust gas is either emitted from the exhaust manifold and provides energy to the turbine in order to operate the turbocharger or recirculated through the HP EGR loop. Differing from a singleloop EGR system, the exhaust gas from the turbine outlet is reused for the LP EGR functions. There are exhaust aftertreatment systems, such as a diesel oxidation catalyst and selective catalytic reduction (SCR) in the LP EGR loop. After passing through the after-treatment systems, the flow is divided into two subflows: one flows to the exhaust throttle through the SCR and the other flows to the compressor inlet. The mass flow through the LP EGR loop is mixed with the fresh air mass flow in front of the compressor inlet. It is clear that the dynamics of the flows are coupled and comprise one air flow system following the principle of mass balance. In turn, all flows affect the rate of pressure change, temperature, and oxygen fraction states in the corresponding locations.

The variables p, T, V, and W are the pressure, temperature, volume, and mass flow rate. The subscripts i, x, uc, and dt represent the intake manifold, exhaust manifold, upstream of the compressor, and downstream of the turbine, respectively, which are the primary locations that describe the system thermodynamic properties. With the LP EGR loop added, the dynamics at the compressor upstream and the turbine downstream are important in terms of the intake and exhaust manifolds. That is, the dual-loop EGR system is significantly more complicated than the single-loop system and thus is more difficult to analyze. The model equations that have been validated in previous papers [7], [8], [12] are introduced in this section. The principal equations are described in (1)–(8).

B. Pressure Dynamics

Based on the first law of thermodynamics, the pressure dynamics of the diesel engine air system with the dual-loop EGR are described as follows:

$$\dot{p}_i = \frac{R\gamma}{V_i} (T_c W_c + T_{\text{HPegr}} W_{\text{HPegr}} - T_i W_{ie})$$
(1)

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

$$\dot{p}_{uc} = \frac{R\gamma}{V_{uc}} (T_a W_{\text{air}} + T_{dt} W_{\text{LP}_{\text{egr}}} - T_c W_c)$$
(3)

$$\dot{p}_{dt} = \frac{R\gamma}{V_{dt}} (T_x W_t - T_{dt} W_{\text{LP}_{\text{egr}}} - T_{dt} W_{\text{out}})$$
(4)

where γ and *R* are the specific heat ratio and ideal gas constant. In (1)–(4), W_{ie} , W_{ex} , W_{air} , and W_{out} are the mass flow rates into the cylinders and out of the cylinders, the fresh air mass flow rate, and the flow rate through the exhaust throttle, respectively. In (1) and (2), T_{HPegr} is the HP EGR temperature and T_e is the cylinder exit temperature.

Note that it is assumed that the pressure drop through aftertreatment systems is negligible, i.e., $p_{dt} \approx p_{DOC}$ considering the characteristics of the target engine in this paper. In fact, the amount of pressure drop depends on types and conditions of after-treatment systems installed in the engine. For example, the pressure drop through a diesel particulate filter is affected by the collected soot in it, and cannot be neglected for modeling of the engine air path in some cases [27]. However, modern production engines with an LP EGR system are commonly equipped with a sensor for differential pressure across LP EGR, and the sensor can be easily utilized to estimate the pressure after the after-treatment systems [28].

C. Flow Equations

The mass flow rate into the cylinders from the intake manifold is modelled using a speed density equation as described in

$$W_{ie} = \frac{\eta_v p_i N V_d}{120 T_i} \tag{5}$$

where η_v , V_d , and N are the volumetric efficiency, displacement volume, and engine speed, respectively. The volumetric

efficiency is determined using the intake and exhaust manifold states and the engine speed based on the empirically calibrated map [6], [12].

The mass flow rate of the turbine can be modeled as in

$$W_t = \frac{p_x}{\sqrt{T_x R}} A_{t_{\rm max}} \sqrt{1 - \left(\frac{p_{dt}}{p_x}\right)^{K_t} g(u_t)}$$
(6)

where $A_{t_{max}}$ is the maximum effective area of VGT and $g(u_t)$ is the effective area function determined by the VGT rack position. The effect of the pressure ratio on the flow rate is described using the choking function with the positive exponent K_t in (6) [29].

The behavior of flows controlled by valves or throttles, such as the EGR flows, can be described using an orifice equation. Assuming that the reverse flow is neglected, the HP EGR flow is represented by

$$W_{\rm HPegr} = \frac{A_{\rm HP} p_x}{\sqrt{RT_x}} \sqrt{\frac{2\gamma}{(\gamma - 1)} \left(\left(PR_{\rm HP}\right)^{\frac{2}{\gamma}} - \left(PR_{\rm HP}\right)^{\frac{\gamma + 1}{\gamma}} \right)} \quad (7)$$

where

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

The minimum pressure ratio value across the flow is limited by $(2/(\gamma + 1))^{\gamma/(\gamma - 1)}$, which indicates the choking condition.

Likewise, the characteristic of the LP EGR flow rate is described in

$$W_{\text{LPegr}} = \frac{A_{\text{LP}} p_{dt}}{\sqrt{\text{RT}_{dt}}} \sqrt{\frac{2\gamma}{(\gamma - 1)}} \left(\left(P R_{\text{LP}} \right)^{\frac{2}{\gamma}} - \left(P R_{\text{LP}} \right)^{\frac{\gamma + 1}{\gamma}} \right)$$
(8)

where

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

In (7) and (8), $A_{\rm HP}$ and $A_{\rm LP}$ are the effective areas of the HP EGR and LP EGR flows, respectively, which are determined by the corresponding valve positions.

III. EXHAUST PRESSURE OBSERVER DESIGN

In this section, a second-order observer is developed in order to estimate the exhaust pressure of a diesel engine with a dualloop EGR and VGT. The observer can be used for a variety of applications, particularly for the engine air/gas flow control.

A. Design Considerations

The coupling effects of the engine air path system are very strong; thus, inaccurate modeling of one component can be detrimental to the estimation performance of the observer. The observer should have a robust form in order to consider the model uncertainties effectively. The availability and the reliability of the sensors are also important for real engine applications. It is desirable to use the reliable sensors already installed in production engines in order to reduce production costs. For the dual-loop EGR system, it is important to analyze the influences of the LP EGR operations on the exhaust pressure.



Fig. 2. Cylinder exit temperature versus exhaust manifold temperature.

Note that because the pressure states in the LP EGR loop, i.e., p_{uc} and p_{dt} , cannot be measured in mass-produced engines, estimation methods for these pressure states are also proposed in Section IV. First, the design procedure of the exhaust pressure observer is introduced under the assumption that the measurement information of p_{uc} and p_{dt} are given.

B. Design Procedure

Through differentiating the ideal gas equation for the intake manifold with respect to time under the assumption that the temperature varies slowly, the simplified dynamics of the intake pressure is derived as in (9) [11], [30]. This pressure dynamics has often been used as a control-oriented model in order to overcome measurement limitations

$$\dot{p}_i = k_1 (W_c + W_{\text{HPegr}} - W_{ie})$$

where $k_1 = \frac{RT_i}{V_i}$. (9)

The exhaust pressure dynamics in (2) is reduced to (10) assuming that the cylinder out temperature is approximately equal to the exhaust temperature for general operating conditions, i.e., $T_e \approx T_x$. This assumption is reasonable because the experimental data in Fig. 2 indicates that the temperatures are almost identical for numerous operating conditions with the maximum discrepancy being less than 2%

$$\dot{p}_x = k_2 (W_{ex} - W_{\text{HPegr}} - W_t)$$

where $k_2 = \frac{R\gamma}{V_x} T_x.$ (10)

For a system with a single-loop EGR, the compressor flow is the same as the fresh air flow, which is easily measured using an MAF sensor. However, for a dual-loop EGR, the mixed flow of fresh air and LP EGR flow enters the compressor. Therefore, the flow rate through the compressor is approximated as the sum of the gas flow rates of LP EGR and fresh air, i.e., $W_c \approx$ $W_{\text{LPegr}}+W_{\text{air}}$. The flow rate exiting the cylinders is equal to the sum of the cylinder-in flow rate and the fuel rate, i.e., $W_{ex} =$ $W_{ie} + W_f$, where W_f is the mass flow rate of the fuel [11]. With these assumptions, a Luenberger-sliding mode observer is designed as in (11)

$$\dot{\hat{p}}_{i} = k_{1}(W_{\text{air}} + W_{\text{LPegr}} + \hat{W}_{\text{HPegr}} - k_{e}\hat{p}_{i}) + L_{1}(p_{i} - \hat{p}_{i}) + L_{2}\text{sgn}(p_{i} - \hat{p}_{i})$$

$$\dot{\hat{p}}_{x} = k_{2}(k_{e}\hat{p}_{i} + W_{f} - \hat{W}_{\text{HPegr}} - \hat{W}_{t}) + L_{3}(p_{i} - \hat{p}_{i}) + L_{4}\text{sgn}(p_{i} - \hat{p}_{i})$$
where $k_{e} = \frac{\eta_{v}NV_{d}}{120RT_{i}}$. (11)

In (11), L_1 , L_2 , L_3 , and L_4 are the observer gains to be determined. W_{air} and p_i can be easily measured using the already installed sensors on the production engine, and the injected fuel rate W_f is assumed to be known. However, the other flows, including the EGR and VGT flow, are not typically measurable and should be calculated from the model equations. Note that \hat{W}_{HPegr} and \hat{W}_t are direct functions of the estimated value of \hat{p}_x .

C. Stability Proof and Design of Observer Gains

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The estimation errors of p_i and p_x are defined as follows:

$$\varepsilon_1 = p_i - \hat{p}_i \tag{12}$$

$$\varepsilon_2 = p_x - \hat{p}_x. \tag{13}$$

The time derivative of (12) is expressed as (14) through combining (1) with (11), as follows:

$$\dot{\varepsilon}_{1} = \dot{p}_{i} - \dot{\hat{p}}_{i}$$

$$= k_{1} \left(\left(\frac{T_{c}\gamma}{T_{i}} W_{c} - \hat{W}_{c} \right) + \left(\frac{T_{\text{HPegr}}\gamma}{T_{i}} W_{\text{HPegr}} - \hat{W}_{\text{HPegr}} \right) \right)$$

$$- (\gamma W_{ie} - \hat{W}_{ie})$$

$$- L_{1}\varepsilon_{1} - L_{2} \text{sgn}(\varepsilon_{1})$$
(14)

where $\hat{W}_c = W_{air} + W_{LPegr}$. Likewise, the time derivative of (13) is described as in (15) based on the assumption that $T_e/T_x \approx 1$, as follows:

$$\dot{\varepsilon}_2 = \dot{p}_x - \hat{p}_x$$

$$= k_2(k_e \varepsilon_1 - (W_{\text{HPegr}} - \hat{W}_{\text{HPegr}}) - (W_t - \hat{W}_t))$$

$$^- L_3 \varepsilon_1 - L_4 \text{sgn}(\varepsilon_1). \tag{15}$$

Consider a Lyapunov function (positive definite and radially unbounded) for the error as in [31]

$$V = \frac{1}{2}\varepsilon_1^2 + \frac{1}{2}\varepsilon_2^2.$$
 (16)

The time derivative of (16) is described in (17) through substituting (14) and (15) into it

$$\dot{V} = \varepsilon_{1}\dot{\varepsilon}_{1} + \varepsilon_{2}\dot{\varepsilon}_{2}$$

$$= \varepsilon_{1} \left(k_{1} \left(\left(\frac{T_{c}\gamma}{T_{i}} W_{c} - \hat{W}_{c} \right) + \left(\frac{T_{\mathrm{HPegr}}\gamma}{T_{i}} W_{\mathrm{HPegr}} - \hat{W}_{\mathrm{HPegr}} \right) \right) \right) - k_{1}(\gamma W_{ie} - \hat{W}_{ie}) - L_{1}\varepsilon_{1} - L_{2}\mathrm{sgn}(\varepsilon_{1}) \right) + \varepsilon_{2} \left(k_{2}(k_{e}\varepsilon_{1} - (W_{\mathrm{HPegr}} - \hat{W}_{\mathrm{HPegr}}) - (W_{t} - \hat{W}_{t})) - L_{3}\varepsilon_{1} - L_{4}\mathrm{sgn}(\varepsilon_{1}) \right).$$

$$(17)$$

In (17), the effects of \hat{p}_x on \hat{W}_{HPegr} and \hat{W}_t complicate the proving procedure; therefore, these flow terms must be rearranged first. Using (7), the difference term of the actual and estimated HP EGR flows is rearranged as in

$$W_{\rm HPegr} - \hat{W}_{\rm HPegr}$$

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

$$- A_{\rm HP} \hat{p}_x \sqrt{\frac{2\gamma}{RT_x(\gamma - 1)} \left(\left(\frac{p_i}{\hat{p}_x}\right)^{\frac{2}{\gamma}} - \left(\frac{p_i}{\hat{p}_x}\right)^{\frac{\gamma+1}{\gamma}} \right)}$$

$$= k_3(\alpha - \hat{\alpha}) \qquad (18)$$

where

$$\alpha = \sqrt{\left(\frac{p_i^2}{p_i^{\frac{2}{\gamma}-2}} - \frac{p_i^{\frac{1}{\gamma}+1}}{p_x^{\frac{1}{\gamma}-2}}\right)}, \quad \hat{\alpha} = \sqrt{\left(\frac{p_i^2}{p_x^{\frac{2}{\gamma}-2}} - \frac{p_i^{\frac{1}{\gamma}+1}}{p_x^{\frac{1}{\gamma}-1}}\right)}$$
$$k_3 = \sqrt{\frac{2\gamma A_{\rm HP}^2}{RT_x(\gamma - 1)}}.$$

Likewise, the error of the VGT flow is described in (19) using (6)

$$W_t - \hat{W}_t$$

$$= \frac{A_{t_\max}}{\sqrt{T_x R}} g(u_t) \left(\sqrt{1 - \left(\frac{p_{dt}}{p_x}\right)^{K_t}} - \sqrt{1 - \left(\frac{p_{dt}}{\hat{p}_x}\right)^{K_t}} \right)$$

$$= k_4 (\beta - \hat{\beta}) \tag{19}$$

where

$$\beta = \sqrt{1 - \left(\frac{p_{dt}}{p_x}\right)^{K_t}}, \quad \hat{\beta} = \sqrt{1 - \left(\frac{p_{dt}}{\hat{p}_x}\right)^{K_t}}$$
$$k_4 = \frac{A_{t_\max}}{\sqrt{T_x R}} g(u_t).$$

Substituting (18) and (19) into the second term of (17), (20) can be derived

$$\begin{split} \dot{V} &= \varepsilon_{1}\dot{\varepsilon}_{1} + \varepsilon_{2}\dot{\varepsilon}_{2} \\ &= \varepsilon_{1} \left(k_{1} \left(\left(\frac{T_{c}\gamma}{T_{i}} W_{c} - \hat{W}_{c} \right) + \left(\frac{T_{\mathrm{HPegr}}\gamma}{T_{i}} W_{\mathrm{HPegr}} - \hat{W}_{\mathrm{HPegr}} \right) \right) \right) \\ &- k_{1}(\gamma W_{ie} - \hat{W}_{ie}) - L_{1}\varepsilon_{1} - L_{2}\mathrm{sgn}(\varepsilon_{1}) \\ &+ \varepsilon_{2} \left(k_{2}(k_{e}\varepsilon_{1} - k_{3}(\alpha - \hat{\alpha}) - k_{4}(\beta - \hat{\beta})) \\ -L_{3}\varepsilon_{1} - L_{4}\mathrm{sgn}(\varepsilon_{1}) \right) \\ &= k_{1}\varepsilon_{1} \left(\frac{T_{c}\gamma}{T_{i}} W_{c} - \hat{W}_{c} \right) + k_{1}\varepsilon_{1} \left(\frac{T_{\mathrm{HPegr}}\gamma}{T_{i}} W_{\mathrm{HPegr}} - \hat{W}_{\mathrm{HPegr}} \right) \\ &- k_{1}\varepsilon_{1}(\gamma W_{ie} - \hat{W}_{ie}) - L_{1}\varepsilon_{1}^{2} - L_{2}\mathrm{sgn}(\varepsilon_{1})\varepsilon_{1} \\ &+ k_{2}k_{e}\varepsilon_{1}\varepsilon_{2} - k_{2}k_{3}\varepsilon_{2}(\alpha - \hat{\alpha}) - k_{2}k_{4}\varepsilon_{2}(\beta - \hat{\beta}) \\ &- L_{3}\varepsilon_{1}\varepsilon_{2} - L_{4}\mathrm{sgn}(\varepsilon_{1})\varepsilon_{2}. \end{split}$$

Let $k_1((T_c\gamma / T_i)W_c - \hat{W}_c)$, $k_1((T_{\text{HPegr}}\gamma / T_i)W_{\text{HPegr}} - \hat{W}_{\text{HPegr}})$, and $k_1(\gamma W_{ie} - \hat{W}_{ie})$ be bounded by $\Omega_1, \Omega_2, \Omega_3 > 0$, i.e., $|k_1||(T_c\gamma / T_i)W_c - \hat{W}_c| \leq \Omega_1, |k_1||(T_{\text{HPegr}}\gamma / T_i)W_{\text{HPegr}} - \hat{W}_{\text{HPegr}}| \leq \Omega_2$, and $|k_1||\gamma W_{ie} - \hat{W}_{ie}| \leq \Omega_3$, respectively.

If the gains are designed such that $L_1 \triangleq 0$, $L_2 > \Omega_1 + \Omega_2 + \Omega_3$, $L_3 \triangleq k_2 k_e$, and $L_4 \triangleq 0$, then (20) satisfies the inequality in (21)

$$\dot{V} = \varepsilon_1 \left(k_1 \left(\frac{T_c \gamma}{T_i} W_c - \hat{W}_c \right) + k_1 \left(\frac{T_{\text{HPegr}} \gamma}{T_i} W_{\text{HPegr}} - \hat{W}_{\text{HPegr}} \right) - k_1 (\gamma W_{ie} - \hat{W}_{ie}) - L_2 \text{sgn}(\varepsilon_1) \right) - k_2 k_3 \varepsilon_2 (\alpha - \hat{\alpha}) - k_2 k_4 \varepsilon_2 (\beta - \hat{\beta}) < 0$$
(21)

because $L_2 > \Omega_1 + \Omega_2 + \Omega_3$ and $k_1, k_2, k_3, k_4, k_e > 0$. Moreover, the last two terms in (21) are always smaller than 0, because $\alpha = \alpha$ (p_i, p_x) and $\beta = \beta$ (p_{dt}, p_x) are monotonically increasing functions of p_x , and $\hat{\alpha} = \hat{\alpha}(p_i, \hat{p}_x)$ and $\hat{\beta} = \hat{\beta}(p_{dt}, \hat{p}_x)$ are also monotonically increasing with respect to \hat{p}_x so that $-k_2k_3\varepsilon_2(\alpha - \hat{\alpha}) < 0$ and $-k_2k_4\varepsilon_2(\beta - \hat{\beta}) < 0$.

The relationships between p_x , and α and β indicate that as the exhaust manifold pressure of the diesel engine increases, the rates of the HP EGR and turbine flows are increased, given that the other states remain the same. Therefore, this stability proof method remains valid even if other models of EGR and turbine flows, which can describe this fundamental property, are adopted in the observer design. Consequently, the inequality in (21) implies that \dot{V} is a negative definite function, and it guarantees that the estimation errors ε_1 and ε_2 converge to zero asymptotically over time. Even though uncertainties exist in the observer model, the effects of the uncertainties can be canceled using the designed observer gains. The observer was initially designed with four gains, but only L_2 is a tuning parameter, which is a strong feature of the observer in practical aspects.

IV. PRESSURE MODELING FOR THE LP EGR LOOP

Information on the pressure states in the LP EGR loop is required in order to estimate the states and to control the air flow rates in the engine with a dual-loop EGR system. However, sensors for the states associated with the LP EGR loop are not available in production car engines. Thus, in this section, a simple method of modeling these pressure states using regression models is proposed.



Fig. 3. Turbine outlet pressure versus exhaust gas flow rate.

A. Overview

The main pressure states of the LP EGR loop are the turbine outlet pressure p_{dt} and the compressor inlet pressure p_{uc} . For a single-loop EGR system, which does not have a LP EGR loop, only fresh air flows into the compressor. In this case, it is possible to assume that the pressure at the compressor inlet is constant or a function of the air flow rate. However, mixed flows of fresh air and LP EGR gas enter the compressor in dual-loop EGR systems for which the compressor inlet pressure is varied based on the dynamics developed by the principle of flow balance. Likewise, the turbine outlet pressure can be assumed to be constant for a single-loop EGR system in which the exhaust gas is emitted through the turbine outlet.

However, for dual-loop EGR systems, the pressure is affected by the flow balance at the turbine outlet. Though (3) and (4) represent the pressure dynamics in the compressor inlet and turbine outlet, respectively, these equations are not practical because variables such as T_c and W_c are not measured in a dual-loop EGR system. Moreover, if (3) and (4) are used to estimate p_{dt} and p_{uc} , the whole estimation algorithm becomes too complex to be implemented, because the order of the algorithm would increase markedly. Physically, the variations of p_{dt} and p_{uc} are very small compared with the intake/exhaust manifold pressure for all operating points. Hence, it is beneficial to use the static regression model to estimate p_{dt} and p_{uc} instead of observer dynamics or recursive estimation methods. The modeling methods of these pressure states using polynomial regression models are proposed in the following sub-sections B, C.

B. Regression Modeling of the Turbine Outlet Pressure

It is intuitive that as the pressure at the turbine outlet increases, the exhaust gas flow rate increases for a fixed exhaust throttle position. To begin with, it is assumed that the pressure is a function of the exhaust gas flow rate only.

Note that the exhaust gas flow rate can be easily calculated through summing the fresh air flow rate and fuel rate: $W_{\text{out}} = W_{\text{air}} + W_f$. Fig. 3 shows the relationship between the turbine outlet pressure and the exhaust gas flow rate at various



Fig. 4. Turbine outlet pressure versus the corrected exhaust gas flow rate.

operating points. In Fig. 3, the circles indicate the relationship between the turbine outlet pressure and the exhaust gas flow rate in the HP EGR operations only, and the \times marks denote the relationship in the LP EGR operations only. As seen in Fig. 3, the assumption that the pressure is a function of the exhaust flow rate only is not correct. The ratio of the turbine outlet pressure and ambient pressure is the primary driving source of the exhaust flow, because these pressures represent the pressure states of the flow inlet and outlet. Then, a new variable is defined by combining the flow rate term with temperature difference of the flow inlet and outlet, as follows:

$$W_{\text{out_corr}} \triangleq W_{\text{out}} \sqrt{T_{dt} - T_a}$$
 (22)

where T_a is the ambient temperature (regarded as a constant) determined by the environment in which the engine operates. Then, it is proposed that the turbine outlet pressure be defined as a function of $W_{\text{corr_out}}$ as described in

$$p_{dt} = f(W_{\text{out_corr}}). \tag{23}$$

Equation (23) means that p_{dt} is a function of the exhaust flow rate and a square root of the difference of the turbine outlet and ambient temperature, not a function of the flow rate only. As Fig. 4 demonstrates that the turbine outlet pressure can be well represented by the corrected variable, it is deduced that the influence of LP EGR operations on pressure change in the LP EGR loop can be indicated by (23).

The relationship can be approximated using a polynomial regression model with a least squares curve fitting method as described in

$$\hat{p}_{dt} = 0.7584(W_{\text{out_corr}}^2) + 0.7853(W_{\text{out_corr}}) + 103.08.$$
 (24)

 R^2 of the regression model is 0.9766, which indicates a good fit.

C. Regression Modeling of the Compressor Inlet Pressure

In the same manner, the compressor inlet pressure can be modeled using the information of the fresh air flow rate and compressor inlet temperature. For the compressor inlet pressure, it is clear that the fresh air flow rate increases as



Fig. 5. Compressor inlet pressure versus fresh air flow rate.

the pressure decreases for a fixed intake throttle position in a single-loop EGR system. Although the LP EGR flow is added to the fresh air flow rate at the compressor inlet, which establishes a slightly different relationship between the flow and the pressure in a dual-loop EGR system, the compressor inlet pressure is relevant to the fresh air flow rate. It should be noted that the fresh air flow rate can also be measured by sensors in the production engine. First, assume that the pressure is a function of the fresh air flow rate only. In Fig. 5, the compressor inlet pressure is plotted against the fresh air flow rate for various operating points. As for the HP EGR operations (circles), the compressor inlet pressure is inversely proportional to the fresh air flow rate as expected, because this case is equivalent to an engine with a single EGR system. However, it is difficult to determine a similar relationship with the LP EGR operations (\times marks). In order to determine an appropriate relationship for both the HP EGR and LP EGR operations, the compressor inlet pressure is assumed to be a function of the corrected air flow rate, which reflects the effect of temperature change due to LP EGR operations, as follows:

$$p_{uc} = f(W_{\text{air_corr}}) \tag{25}$$

where $W_{\text{air}_\text{corr}} \triangleq W_{\text{air}} \sqrt{T_{uc} - T_a}$.

Through considering the temperature terms T_{uc} and T_a , it is expected that the effects of the LP EGR rate on p_{uc} can be indicated using (25).

Fig. 6 demonstrates that it is reasonable that p_{uc} is a function of the corrected air flow rate. The relationship can also be approximated using a second-order regression model, as follows:

$$\hat{p}_{uc} = -5.1351 \left(W_{\text{air_corr}}^2 \right) + 0.7751 \left(W_{\text{air_corr}} \right) + 101.14$$
(26)

whose R^2 is 0.9845.

Note that it is assumed that the intake and exhaust throttles of the engine are fixed in this paper. However, the suggested modeling approach can be extended to the cases of variable throttle positions by including effective area term of the throttle in the regressor.



Fig. 6. Compressor inlet pressure versus the corrected air flow rate.



Fig. 7. Estimation results of the turbine outlet pressure.



Fig. 8. Estimation results of the compressor inlet pressure.

D. Experimental Verification

The regression models for the pressure states of the LP EGR loop were experimentally validated based on the 6-L heavyduty diesel engine with a dual-loop EGR and VGT. In order to prove that the regression models work well without a recursive algorithm, the verification was undertaken for numerous operating points.

Figs. 7 and 8 compare the estimated data from the regression models in (24) and (26) with the experimental data under the broad operating conditions. The maximum steady-state error was less than 0.65% for both pressure states. Consequently, the pressure states at the turbine outlet and compressor inlet were modeled precisely using the simple regression models and the information already available on the flow rate and temperature at the corresponding locations. These regression



Fig. 9. Final exhaust pressure observer structure.

models will be used in the implementation of the exhaust pressure observer designed in Section III.

V. EXPERIMENTAL VERIFICATION OF THE FINAL EXHAUST PRESSURE OBSERVER

A. Observer With LP EGR Pressure Models

In this section, the estimation performance of the exhaust pressure observer with the LP EGR pressure models is evaluated experimentally. The entire structure of the observer is shown in Fig. 9.

In Fig. 9, the current values for the compressor inlet pressure and turbine outlet pressure are computed from the polynomial regressions designed in Section IV. With these pressure models added to the observer, a more practical estimation algorithm for the exhaust pressure is implemented with less sensor information. In fact, the accuracy of differential pressure values across the LP EGR is crucial in estimating the LP EGR flow rate, and using the steady compressible orifice equation further deteriorates the estimation accuracy of it [28], [32]. However, if errors of the flow models are bounded, and the sliding mode gain is sufficiently high, the effect of the model uncertainties can be eliminated. In other words, while using the pressure regression models and the steady orifice equation has adverse effects on the model accuracy of the LP EGR flow \hat{W}_{LPegr} , it does not affect the convergence stability of the observer, because the uncertainty is canceled by the sliding mode observer gain L_2 . Then, the mass flow rates are calculated using model (5)-(8) with other measurements. Finally, the exhaust manifold pressure is estimated by the observer. The final exhaust pressure observer equipped with the LP EGR pressure models is implemented as described in

$$\hat{p}_{i} = k_{1}(W_{\text{air}} + \hat{W}_{\text{LPegr}} + \hat{W}_{\text{HPegr}} - k_{e}\hat{p}_{i}) + L_{1}\varepsilon_{1} + L_{2}\text{sgn}(\varepsilon_{1}) \hat{p}_{x} = k_{2}(k_{e}\hat{p}_{i} + W_{f} - \hat{W}_{\text{HPegr}} - \hat{W}_{t}) + L_{3}\varepsilon_{1} + L_{4}\text{sgn}(\varepsilon_{1})$$
(27)

where $\hat{W}_{\text{LPegr}} = \hat{W}_{\text{LPegr}}(\hat{p}_{uc}, \hat{p}_{dt})$ and $\hat{W}_t = \hat{W}_t(\hat{p}_x, \hat{p}_{dt})$.

B. Experimental Setup

A heavy-duty six-cylinder 6-L diesel engine was used for the experimental validation of the observer; it is depicted



Fig. 10. Test engine setup.



Fig. 11. Exhaust pressure estimation in steady state conditions. (a) Fuel rate. (b) EGR valve positions and VGT duty. (c) Exhaust pressure.

in Fig. 10. The LP EGR loop was added to the original single EGR system for the dual-loop EGR operations. The sensor information already available on the production engine, e.g., engine speed, fuel rate, fresh air flow rate, and intake pressure, was acquired by the engine control unit in real time. Additional pressure sensors were installed at different locations including the exhaust manifold in order to validate the observer and pressure models. MicroAutobox dSPACE 1401 was used to run the observer algorithm with the pressure models and to process the signals from all sensors.

C. Steady-State Verification

The developed observer and the regression models of the turbine outlet and compressor inlet pressure states were validated with the test engine. The verification in steady



Fig. 12. Exhaust pressure estimation in transient scenario 1). (a) Fuel rate. (b) EGR valve positions and VGT duty. (c) Exhaust pressure.

states was conducted under the conditions described in Fig. 11(a) and (b). At each point of 10 s, the engine speed, injected fuel rate, positions of EGR valves, and VGT duty were maintained constant. The measured data of the exhaust pressure were steady-state values at the corresponding point. This verification was conducted in order to determine whether the estimated values of the observer would rapidly converge to near the measured values even though the operating points were changed significantly. Note that because the reference engine in this research is a heavyduty diesel engine, the engine speed is generally fixed during operation, and the fuel rate is determined in accordance with the desired engine load. Here, the engine speed is fixed at either 1400 or 1800 r/min. The experimental results are presented in Fig. 11(c). The maximum steady-state error was less than 1.6%. Although the conditions for the verification were numerous and changed abruptly, the estimated pressure values quickly converged to near the true values. Note that the change in LP EGR valve position had little effect on the exhaust pressure here, because the engine load was relatively low with the HP EGR valve closed during the LP EGR operations [15]; thus, the exhaust pressure was mostly dependent on the VGT duty.

D. Transient State Verification

The observer performance was also verified for transient conditions that contain various load conditions.



Fig. 13. Exhaust pressure estimation in transient scenario 2). (a) Fuel rate. (b) EGR valve positions and VGT duty. (c) Exhaust pressure.

The experimental evaluations were undertaken in extremely transient conditions called the nonroad transient cycle (NRTC) mode for heavy-duty diesel engines. The operating conditions, including the fuel rate, EGR valve positions, and VGT duty, were continuously varied during the transient tests, while the engine speed was maintained at near 1800 r/min. It should be noted that the NRTC mode includes the combined operations of both EGR valves. The transient validations were performed for two different scenarios: 1) low/medium engine loads with a full opening of the LP EGR valve and 2) high engine loads with a large sweep of the LP EGR valve position. The estimation results for scenario 1) are described in Fig. 12.

It is clearly seen that the estimated values follow the true values well, as expected from the stability proof of the observer. The maximum estimation error was less than 6%. Another experiment was conducted under more severe conditions [scenario 2)] where the engine load was significantly higher and the LP EGR valve positions were considerably and continuously varied. The estimation results are presented in Fig. 13.

It should be noted that most values of the fuel rate and exhaust pressure were higher than those of scenario 1) due to the higher engine loads. The mean estimation error was slightly larger for scenario 2) than for scenario 1), but the maximum error remained less than 6%. Considering that scenario 2) contained severe operating conditions, the experimental results verify the observer performance well.

VI. CONCLUSION

In this paper, an exhaust pressure observer for diesel engines with a dual-loop EGR and VGT was designed with limited measurements. In order to overcome the lack of sensors for the pressure states in the LP EGR loop, polynomial regression models for the compressor inlet and turbine outlet pressures were proposed, and the model accuracy was validated experimentally with the test engine. The developed pressure models have simple and intuitive structures that can be utilized for various control applications in diesel engines with dual-loop EGR systems. The proposed exhaust pressure observer with the regression models was experimentally validated using a test engine with a dual-loop EGR and VGT for numerous operating points covering both steady state and transient conditions. The results demonstrate that the observer is robust to model uncertainties and sufficiently accurate to replace the exhaust pressure sensor. For future work, research on developing adaptive algorithms for the system parameters will be conducted in order to manage changes in the environmental conditions.

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