# Simplified Burnt Gas Fraction Estimation for Turbocharged Diesel Engine with Dual loop EGR System

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*Abstract*— **This paper describes simplified burnt gas fraction estimation method for turbocharged diesel engine with dual loop exhaust gas recirculation (EGR) system. According to the strengthening emission gas regulation in diesel engine, burnt gas fraction has become principal factor for diesel engine air path system. Previous burnt gas fraction estimation method was valid only for a few operating point because of many assumptions used to prove the observability. Or it limits the condition which is not appropriate for practical use. By introducing a simplified burnt gas fraction dynamics, observability can be proved only with a single assumption. Asymptotic stability of the burnt gas fraction observer was proved by lyapunov analysis. The observer was validated on a 6L heavy duty diesel engine GT-POWER model with various engine operating points.**

## I. Introduction

Recently, reducing Nitrogen Oxides (NOx) and Particulate Matter (PM) emissions which are by-products of diesel engine combustion is a major issue due to the growing interest of environmental problems.

As for diesel engine combustion system development, alternative combustion mode such as Low Temperature Combustion (LTC), Homogeneous Charge Compression Ignition (HCCI) and Premixed Controlled Compression Ignition (PCCI) was developed. Alternative combustion mode helps to meet emission regulation without losing desired engine performance much. As for the after treatment system, additional filters like Diesel Oxidation Catalysis (DOC) and Selective Catalytic Reduction (SCR) system were adopted. Both filters are usually located downstream of turbine, as shown in Fig 1.

As for diesel engine air path system development, Exhaust Gas Recirculation (EGR) system has been divided into HP (High Pressure) EGR and LP (Low Pressure) EGR path with EGR valves in each path. HP-EGR shows fast response of changes of EGR rate. But it highly affects the operation of turbocharger because it is located at turbine upstream. LP-EGR weakly affect the operation of turbocharger due to the location of its path. However, LP-EGR shows slow response because it recirculates burnt gas through long distance. A dual loop EGR system has been proven to be considerably effective way to reduce NOx and PM emission by combining alternative combustion modes. Thus the model based control of dual loop EGR system among various engine operating points is becoming important issue for efficient engine combustion. [4].

As for the control of air path system, Mitsuhiro Iwadare and Masaki Ueno suggested intake pressure and compressor mass flow rate target based control scheme using standard Manifold Air Pressure (MAP) and Mass Air Flow (MAF) sensor [14]. It is practical approach because it requires no additional sensors. However, to deal with strengthening environmental regulation in diesel engine, burnt gas fraction is important factor for efficient emission gas reduction. Recently, there has been approaches of diesel engine air path control method including burnt gas fraction as target state [7][8]. Actually, only one lambda sensor is equipped in diesel engine. Thus several burnt gas fraction states should be estimated to fully utilize the burnt gas fraction information. If burnt gas fraction observer is designed, it can be applied to not only for fault diagnosis of EGR path but also model based burnt gas fraction controller for optimal control of diesel engine air path system.

By considering above issues, there has been several studies for estimating burnt gas fraction states. Taking some representative previous publications, Junmin Wang suggested air fraction estimation method based on lyapunov analysis [5]. Although it applies with simultaneous HP-EGR and LP-EGR valve movements, several assumptions should be involved to assure the asymptotical stability of estimation error. F. Castillo and E. Witrant proposed a method of simultaneous air fraction and low-pressure EGR mass flow rate estimation [6]. It is meaningful that the method requires only standard sensors for commercial diesel engines. However, its application is quite limited because it assigns a condition that either HP-EGR or LP-EGR valve should be closed.

In this research, simplified burnt gas fraction estimation method is introduced. By selecting only three principal air path sections of interest in VGT/Dual loop EGR system, burnt gas fraction dynamics was designed. Suggested method reduces the computational burden of observer due to the simplified burnt gas fraction model. Also it does not assign constraints of HP-EGR and LP-EGR movement.

The paper consists of the following: To reduce the number of assumptions for observer design, simplified burnt gas fraction dynamics is suggested. Then, observability is proved by lyapunov analysis. GT-POWER engine modeling which depicts real engine plant behavior was conducted for the validation of suggested burnt gas fraction observer. Finally, observer validation results among various engine operating points are presented.

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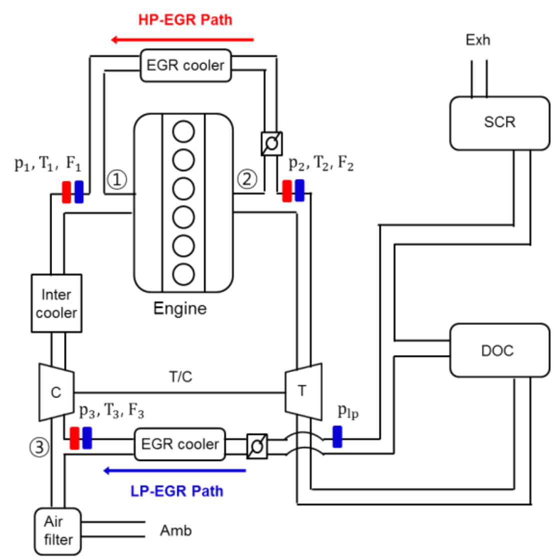
Figure 1. Schematic diagram of diesel engine air path system




The engine model for simulation is based on 6L heavy duty diesel engine with Variable Geometry Turbocharger (VGT) and Dual loop EGR system. Figure 1 represents its schematic diagram. Exh represents exhaust path and Amb means intake path of ambient air. T and C inside trapezoid indicate turbine and compressor each. It is denoted as intake manifold as section 1, exhaust manifold as section 2 and compressor upstream as section 3 for convenience. Subscript which is used throughout this thesis represents each section. Notations $p$, $T$ and $F$ in each section represent pressure, temperature and burnt gas fraction value each.

Simplified burnt gas fraction dynamics would be derived by focusing on three control volumes of intake manifold, exhaust manifold and compressor upstream. It is assumed that no residual mass exist in EGR pipe and no pressure drop happens whether gas passes cooler or not, which belongs to mean value modeling assumptions [1][4][9][12]. Then, burnt gas fraction at intake manifold, exhaust manifold and compressor upstream are obtained as follows:

A. *Burnt gas fraction at intake manifold*

By the definition of burnt gas fraction,

$$F_1 = \frac{m_1 - m_{1,air}}{m_1} \tag{1}$$

where $m_1$ is total mass of gas in the intake manifold and $m_{1,air}$ is mass of air in the intake manifold.

$$\begin{aligned} \dot{F}_1 &= \frac{\dot{m}_1 m_{1,air} - \dot{m}_{1,air} m_1}{m_1^2} \\ &= \frac{\dot{m}_1 m_1 (1 - F_1)}{m_1^2} \\ &+ \frac{-\left[W_c (1 - F_3) + W_{hpegr} (1 - F_2) - W_{ie} (1 - F_1)\right] m_1}{m_1^2} \end{aligned} \tag{2}$$

From the mass conservation law at intake manifold, $\dot{m}_1$ is obtained as follows:

$$\dot{m}_1 = W_c + W_{hpegr} - W_{ie} \tag{3}$$

where $W_c$ is the mass flow rate which comes from compressor and $W_{hpegr}$ is the mass flow rate which comes from HP-EGR pipe and $W_{ie}$ is the mass flow rate which comes into the cylinder.

By substituting (3) into (2), $\dot{F}_1$ is obtained as follows:

$$\dot{F}_1 = \frac{W_c (F_3 - F_1) + W_{hpegr} (F_2 - F_1)}{m_1} \tag{4}$$

B. *Burnt gas fraction at exhaust manifold*

$$F_2 = \frac{m_2 - m_{2,air}}{m_2} \tag{5}$$

where $m_2$ is total mass of gas in the exhaust manifold and $m_{2,air}$ is mass of air in the exhaust manifold.

$$\begin{aligned} \dot{F}_2 &= \frac{\dot{m}_2 m_{2,air} - \dot{m}_{2,air} m_2}{m_2^2} \\ &= \frac{\dot{m}_2 m_2 (1 - F_2)}{m_2^2} \\ &+ \frac{-\left[W_{ex} (1 - F_e) - W_{hpegr} (1 - F_2) - W_{vgt} (1 - F_2)\right] m_2}{m_2^2} \end{aligned} \tag{6}$$

where $F_e$ is in cylinder burnt gas fraction which is defined as follows [1]:

$$F_e = \frac{W_{ie} F_1 - W_f (1 + \lambda_s)}{W_{ex}} \tag{7}$$

where $W_f$ is the fuel mass flow rate and $\lambda_s$ is stoichiometric ratio.

From the mass conservation law at exhaust manifold, $\dot{m}_2$ is obtained as follows:

$$\dot{m}_2 = W_{ex} - W_{hpegr} - W_{vgt} \tag{8}$$

where $W_{ex}$ is the mass flow rate which comes from cylinder.

By substituting (6) and (7) into (5), $\dot{F}_2$ is obtained as follows:

$$\dot{F}_2 = \frac{W_{ie} F_1 - W_{ex} F_2 + W_f (1 + \lambda_s)}{m_2} \tag{9}$$

C. *Burnt gas fraction at compressor upstream*

$$F_3 = \frac{m_3 - m_{3,air}}{m_3} \tag{10}$$

where $m_3$ is total mass of gas in the compressor upstream and $m_{3,air}$ is mass of air in the compressor upstream.

$$\begin{aligned}\dot{F}_3 &= \frac{\dot{m}_3 m_{3,air} - \dot{m}_{3,air} m_3}{m_3^2} \\ &= \frac{\dot{m}_3 m_3 \left(1 - F_3\right)}{m_3^2} \\ &+ \frac{-\left[W_{lpegr}\left(1 - F_2\right) + W_{air} - W_c\left(1 - F_3\right)\right] m_3}{m_3^2}\end{aligned} \tag{11}$$

From the mass conservation law at compressor upstream, $\dot{m}_3$ is obtained as follows:

$$\dot{m}_3 = W_{air} + W_{lpegr} - W_c \tag{12}$$

where $W_{air}$ is the mass flow rate which comes from air intake pipe and $W_{lpegr}$ is the mass flow rate which comes from LP-EGR pipe.

By substituting (6) and (7) into (5), $\dot{F}_3$ is obtained as follows:

$$\dot{F}_3 = \frac{W_{lpegr}\left(F_2 - F_3\right) - F_3 W_{air}}{m_3} \tag{13}$$

Generally, the gas flow rate through pipe can be calculated by orifice equation as follows:

$$W = C_d A p_{up} \sqrt{\frac{2\gamma}{R T_{up}\left(\gamma - 1\right)}\left[\left(p_r\right)^{\frac{2}{\gamma}} - \left(p_r\right)^{\frac{\gamma+1}{\gamma}}\right]} \tag{14}$$

where $p_{up}$ is pressure and $T_{up}$ is temperature at the pipe upstream. $C_d$ is discharge coefficient $A$ is cross sectional area of EGR pipe. $C_d A$ is usually expressed as effective area. In modeling, experimental data of effective area depending on EGR valve position was used. Pressure ratio $p_r$ is different whether fluid is sonic flow or not. $p_r$ is determined as follows:

$$p_r = \max\left(\frac{p_{down}}{p_{up}}, \left(\frac{2}{\gamma + 1}\right)^{\frac{\gamma}{\gamma - 1}}\right) \tag{15}$$

where $p_{down}$ is pressure at the pipe downstream.

## II. Observer Design

In this section, burnt gas fraction observer is designed to estimate burnt gas fraction at intake manifold and compressor upstream with information of burnt gas fraction at exhaust manifold.

### A. Observability Analysis

From the burnt gas fraction dynamics of previous section, state space form of burnt gas fraction dynamics is obtained as follows:

$$\underbrace{\begin{bmatrix}\dot{F}_1 \\ \dot{F}_2 \\ \dot{F}_3\end{bmatrix}}_{\dot{F}} = \underbrace{\begin{bmatrix} -\frac{W_c + W_{hpegr}}{m_1} & \frac{W_{hpegr}}{m_1} & \frac{W_c}{m_1} \\ \frac{W_{ie}}{m_2} & -\frac{W_{ie} + W_f}{m_2} & 0 \\ 0 & \frac{W_{lpegr}}{m_3} & -\frac{W_{air} + W_{lpegr}}{m_3}\end{bmatrix}}_{A(\rho)} \underbrace{\begin{bmatrix}F_1 \\ F_2 \\ F_3\end{bmatrix}}_{F} + \cdots \tag{16}$$

$$+ \underbrace{\begin{bmatrix} 0 \\ \frac{\dot{m}_f}{m_2}\left(1 + \lambda_s\right) \\ 0 \end{bmatrix}}_{W}$$

Luenberger-like observer is as follows:

$$\dot{\hat{F}} = A(\rho)\hat{F} + W + L(\rho)\left(y - \hat{y}\right) \tag{17}$$

where

$$y = \begin{bmatrix}0 & 1 & 0\end{bmatrix}\begin{bmatrix}F_1 \\ F_2 \\ F_3\end{bmatrix}$$

Equation (19) is Linear Parametric Variable (LPV) system. Thus observer gain can be determined by ensuring asymptotic stability of following lyapunov function candidate,

$$V = \frac{1}{2}\tilde{F}_1^2 + \frac{1}{2}\tilde{F}_2^2 + \frac{1}{2}\tilde{F}_3^2 \tag{18}$$

where

$\tilde{F}_1 = F_1 - \hat{F}_1$, $\tilde{F}_2 = F_2 - \hat{F}_2$, $\tilde{F}_3 = F_3 - \hat{F}_3$.

Then, derivative of lyapunov function is developed as follows:

$$\begin{aligned}\dot{V} &= \tilde{F}_1\dot{\tilde{F}}_1 + \tilde{F}_2\dot{\tilde{F}}_2 + \tilde{F}_3\dot{\tilde{F}}_3 \\ &= \tilde{F}_1\left[\left(-\frac{W_c + W_{hpegr}}{m_1}\tilde{F}_1 + \frac{W_{hpegr}}{m_1}\tilde{F}_2 + \frac{W_c}{m_1}\tilde{F}_3\right) - L_1\tilde{F}_2\right] + \cdots \\ &\quad \tilde{F}_2\left(\frac{W_{ie}}{m_2}\tilde{F}_1 - \frac{W_{ie} + W_f}{m_2}\tilde{F}_2 - L_2\tilde{F}_2\right) + \cdots \\ &\quad \tilde{F}_3\left(\frac{W_{lpegr}}{m_3}\tilde{F}_2 - \frac{W_{air} + W_{lpegr}}{m_3}\tilde{F}_3 - L_3\tilde{F}_2\right)\end{aligned} \tag{19}$$

Equation (18) can be rearranged as follows:

$$\begin{aligned}\dot{V} &= -\frac{W_{ie}}{m_1}\tilde{F}_1^2 - \frac{W_{ie} + W_f}{m_2}\tilde{F}_2^2 - \frac{W_c}{m_3}\tilde{F}_3^2 + \cdots \\ &\quad \frac{W_{hpegr}}{m_1}\tilde{F}_1\tilde{F}_2 + \frac{W_c}{m_1}\tilde{F}_1\tilde{F}_3 + \frac{W_{ie}}{m_2}\tilde{F}_1\tilde{F}_2 + \frac{W_{lpegr}}{m_3}\tilde{F}_2\tilde{F}_3 + \cdots \\ &\quad - L_1\tilde{F}_1\tilde{F}_2 - L_2\tilde{F}_2^2 - L_3\tilde{F}_2\tilde{F}_3 \\ &= -\frac{W_c}{m_1}\left(\tilde{F}_1 - \frac{\tilde{F}_3}{2}\right)^2 - \frac{W_{hpegr}}{m_1}\tilde{F}_1^2 - \frac{W_{ie} + W_f}{m_2}\tilde{F}_2^2 + \cdots \\ &\quad -\frac{W_c}{m_1}\left(\frac{m_1}{m_3} - \frac{1}{4}\right)\tilde{F}_3^2 + \frac{W_{hpegr}}{m_2}\tilde{F}_1\tilde{F}_2 + \frac{W_{ie}}{m_2}\tilde{F}_1\tilde{F}_2 + \cdots \\ &\quad \frac{W_{lpegr}}{m_3}\tilde{F}_2\tilde{F}_3 - L_1\tilde{F}_1\tilde{F}_2 - L_2\tilde{F}_2^2 - L_3\tilde{F}_2\tilde{F}_3\end{aligned}$$

Here we applied following equations which are hold at steady state.

$$W_c = W_{air} + W_{lpegr} \tag{20}$$

$$W_{ie} = W_c + W_{hpegr} \tag{21}$$

Obviously, following inequalities is valid for general engine operating point.

$$\frac{m_1}{m_3} - \frac{1}{4} > 0$$

Thus asymptotic stability is proved by choosing observer gain as follows:

$$L(\rho) = \left[ \frac{W_{hpegr}}{m_1} + \frac{W_{ie}}{m_2} \quad L_2 \quad \frac{W_{lpegr}}{m_3} \right], \; L_2 > 0 \tag{22}$$

Observer gain which is same with (22) makes the $\dot{V}$ to be negative definite. To calculate observer gain mass flow rate was obtained by orifice equation and mass contained in each section was obtained by ideal gas equation. But $W_{ie}$ is estimated from the following equation [1][12]:

$$W_{ie} = \eta_v \frac{p_1}{RT_1} \frac{NV_d}{120} \tag{23}$$

where $\eta_v$ is the volumetric efficiency, $V_d$ is displacement volume and $N$ is engine rpm. $p_1$ and $T_1$ are pressure and temperature at intake manifold each.

Figure 2. Diagram of burnt gas fraction estimation

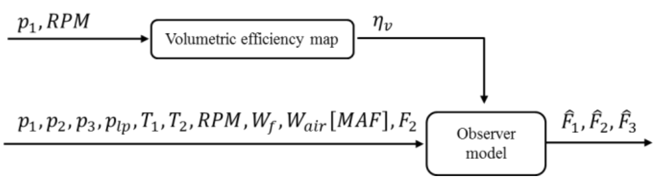



Figure 2 shows the prerequisites of suggested burnt gas fraction estimation. It was assumed that pressure, temperature and MAF data is obtained through sensors. Volumetric efficiency map was designed by experiment as a function of intake manifold pressure and RPM.

## III. Observer Validation

### A. Test Environmen

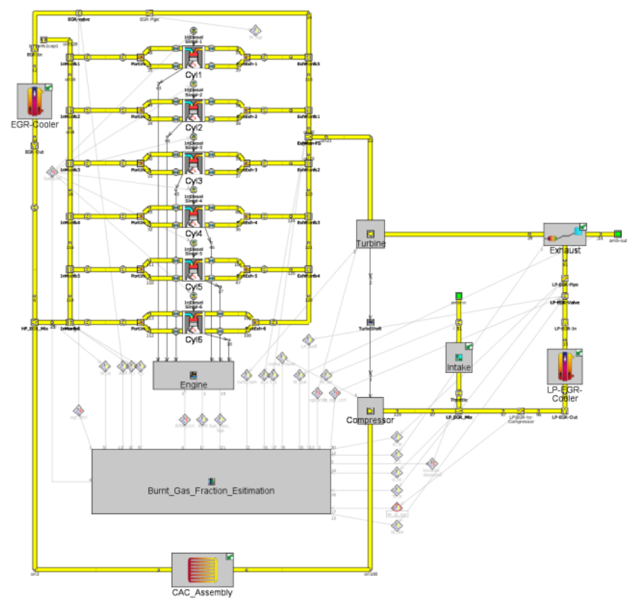
TABLE I shows the specification for 6.0L heavy duty diesel engine with VGT/Dual loop EGR system. Based on specification data GT-POWER engine modeling was conducted as shown figure 3. In modeling, DOC filter, intercooler, HP-EGR cooler, LP-EGR cooler were also modeled to consider pressure drop effect.

For the calculation of real burnt gas fraction value, proved fourth order burnt gas fraction model was used [5][6]. Then it was compared with the value of burnt gas fraction observer which was designed in section II.

TABLE I. SPECIFICATION OF TURBOCHARGED DIESEL ENGINE

| ***System specification*** | ***Unit*** | ***Value*** |
|---|---|---|
| Bore | mm | 100 |
| Stroke | mm | 125 |
| Number of cylinder | E.A. | 6 |
| Compression ratio | - | 17.5 |
| Firing order | - | 1-5-3-6-2-4 |
| Displacement volume | $m^3$ | 0.006 |
| Intake manifold volume | $m^3$ | 0.001920 |
| Exhaust manifold volume | $m^3$ | 0.0006352 |
| Compressor upstream volume | $m^3$ | 0.0054 |
| HP EGR pipe diameter | mm | 45 |
| LP EGR pipe diameter | mm | 45 |

Figure 3. GT-POWER engine model for simulation




### B. Operating Points

Figure 4. Operating points for observer validation

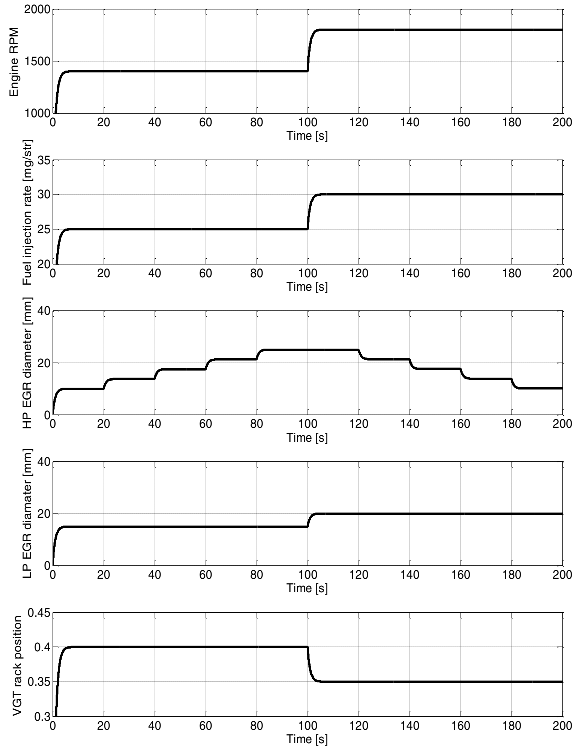



Figure 4 shows the operating points for observer validation, which covers most frequent actuator positions. HP-EGR valve changes in every 20s and LP-EGR valve changes in every 100s. Engine RPM was set to 1400 and 1800, which are common for heavy duty diesel engine. In simulation, HP-EGR pipe and LP-EGR pipe diameter which are controlled by valve were set to inputs. VGT rack controls gas flow rate which comes into turbine. In simulation, it was assumed that there is 3% white noise of burnt gas fraction sensor. Also, simulation sampling time was set to 10ms by considering ECU implementation of algorithm.

## C. *Simulation Results*

Figure 5. Estimation results of burnt gas fraction at intake manifold

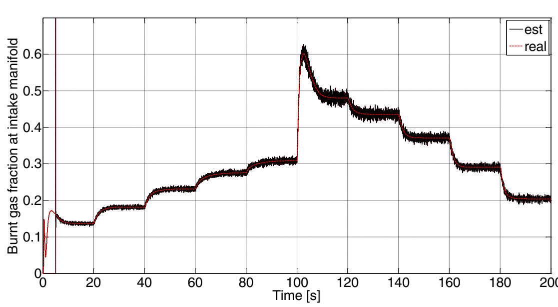



Figure 6. Estimation results of burnt gas fraction at exhaust manifold

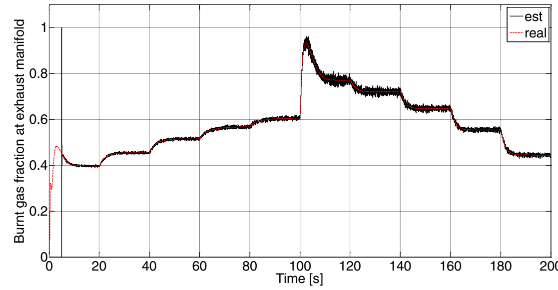
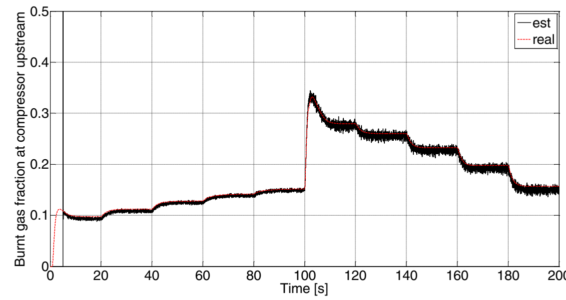



Figure 7. Estimation results of burnt gas fraction at compressor upstream




Given the operating points in figure 4, figure 5 to 7 show the simulation results of burnt gas fraction estimation. Here black solid line is real value and red dotted line is estimated value. Due to recirculation of exhaust burnt gas through HP-EGR pipe, $F_1$ was higher than $F_3$. Also $F_2$ was higher than $F_1$ because more burnt gas is formed in engine cylinder. Mostly, burnt gas fraction quickly converged to real values at each operating points. In simulation, observer was turned on at 5 sec. It is noteworthy that chattering boundary is different depending on operating points because eigenvalues vary due to LPV coefficients of matrix.

Figure 8. LP-EGR mass flow rate

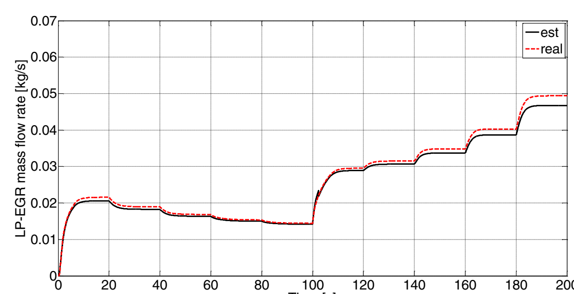



Figure 8 shows the difference of LP-EGR mass flow rate depending on operating points. Precise estimation of LP-EGR mass flow rate is difficult because it was assumed that temperature sensor is not equipped in LP-EGR pipe. Uncertainty of LP-EGR pipe is connected to model uncertainty. Model uncertainty can affect asymptotical stability of the observer. In figure 7, slight offset appeared due to inaccuracy of LP-EGR mass flow rate.

## IV. CONCLUSION

In this paper, burnt gas fraction estimation method with simplified burnt gas fraction model was proposed for turbocharged diesel engine with dual loop EGR system. With the lyapunov approach, LPV observer gains were determined to guarantee the asymptotical stability of error dynamics of burnt gas fraction model.

With the reduced burnt gas fraction model designed observer effectively estimated burnt gas fraction states, which contributes to reduce the amount of calculation for burnt gas fraction estimation as well as less sensor usage than higher order model. Simulation results of GT-POWER engine model validated the observer performance.

It will be anticipated that simplified burnt gas fraction observer can be applied to both observer based burnt gas fraction control and fault diagnosis of dual loop EGR in diesel engine air path system.

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## REFERENCES

[1] Yoon, Young Sik (2010). “A Study of Turbocharged Diesel Engine Modeling and Robust Model Based Sling Mode Controller Design”, Master’s Thesis, Korea Advanced Institute of Science and Technology, Daejeon, Republic of Korea, 122 pages.

[2] Bradely C. Glenn (2005), “COORDINATED CONTROL OF THE TURBO ELECTRICALLY ASSISTED VARIABLE GEOMETRY TURBOCHARGED DIESEL ENGINE WITH EXHAUST GAS RECIRCULATION”, Ph.D Dissertation, The Ohio State University, USA, 178 pages..

[3] Oliver Grondin, Philippe Moulin, Jonathan Chauvin, “Control of a Turbocharged Diesel Engine Fitted with High Pressure and Low Pressure Exhaust Gas Recircualtion Systems, IEEE Conference on Decision and Control, 2009

[4] Benjamin Haber (2010), “A Robust Control Approach on Diesel Engines with Dual-Loop Exhaust Gas Recirculation Systems”, Master’s Thesis, The Ohio State University, USA

[5] Junmin Wang, “Air fraction estimation for multiple combustion mode diesel engines with dual-loop EGR systems”, Control Engineering Practice, 16, pp. 1479-1486, 2008

[6] F. Castillo, E. Witrant, V. Talon, L. Dugard, “Simultaneous Air Fraction and Low-Pressure EGR Mass Flow Rate Estimation for Diesel Engines”, 5th IFAC Symposium on System Structures and Control, 2013.

[7] Ingo Friedrich, Chia-Shang Liu and Dale Oehlerking, “Coordinated EGR-Rate Model-Based Controls of Turbocharged Diesel Engines via an Intake Throttle and an EGR Valve”, IEEE 2009, pp. 340-347

[8] Fengjun Yan (2012), “Diesel Engine Advanced Multi-Mode Combustion Control and Generalized Nonlinear Transient Trajectory Shaping Control Methods”, Ph.D Dissertation, The Ohio State University, USA, 232 pages.

[9] J Wahlström and L Eriksson, “Modeling diesel engines with a variable-geometry turbocharger and exhaust gas recirculation by optimization of model parameters for capturing non-linear system dynamics”, Proceedings of the Institution of Mechanical Engineers, Part D, Journal of Automobile Engineering, 225(7):960-986, 2011.

[10] Jayant Sarlashkar, Ryan Roecker and Theodore Kostek, “Sliding Mode Control For Diesel Engines with Airflow Dominant Fueling”, JSAE Annual Congress Proceedings, 2010, pp.7-12.

[11] Fengjun Yan and Junmin Wang, “Control of Dual Loop EGR Air-Path Systems for Advanced Combustion Diesel Engines by a Singular Perturbation Methodology”, American Control Conference, 2011, San Francisco, CA, USA, pp. 1561-1566.

[12] Merten Jung (2003), “Mean-Value Modelling and Robust Control of the Airpath of a Turbocharged Diesel Engine”, Ph.D Dissertation, University of Cambridge, United Kingdom, 147 pages.

[13] Alois Amstutz, Luigi R. Del Re, “EGO Sensor Based Robust Output Control of EGR in Diesel Engines”, IEEE TRANSACTIONS ON CONTROL SYSTEMS TECHNOLOGY, Vol. 3, No. 1, pp. 39-47, 1995.

[14] Mitsuhiro Iwadare, Masaki Ueno, Shuichi Adachi (2009), “Multi-Variable Air-Path Management for a Clean Diesel Engine Using Model Predictive Control”, SAE Int. J. Engines 2(1), pp. 764-773, 2009-01-0733.