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Trajectory Linearization Exhaust Manifold Observer Design for Diesel Engine control

Hyo-Min Jin and Sei-Bum Choi
Department of Mechanical Engineering
Korea Advanced Institute of Science and Technology
Daejeon, Korea
Email: {enfinirx7, sbchoi}@kaist.ac.kr

Min-Seok Ko
Institute of Technology
Doosan Infracore Co.,Ltd
Gyeonggi-Do, Korea
Email: Minseok.Ko@doosan.com

Abstract— Automobile industries and makers have devoted all theirs energy to deal with the exhaust gas regulations. Most of all, in point of controlling the engine, the simple PID control or map based control only has existed owing to the nonlinear properties of the engine and disturbances and reliabilities. However, model based study on the controlling the heavy duty diesel engine using the HP(High Pressure) EGR and VTG(Variable Turbine Geometry) has been introduced. Especially, the design problems of the exhaust manifold pressure observer which are dealt with controlling the diesel engine are investigated. Moreover, in modeling the engine desired value, there are some problems and trial error must be enacted. To solve these problems, trajectory linearization exhaust manifold observer is used. From the trajectory values, the uncertainty and disturbance of engine system are only considered in estimating the states with the assumption that the observer model is similar to the engine plant. Especially, in this paper, the trajectory observer is designed and verified in the fixed 1000rpm 50~100% loads.

***Index Terms*— model based sliding mode control, diesel engine control, exhaust manifold pressure observer, trajectory linearization method**

I. INTRODUCTION

The emission regulations in diesel engine are gradually strengthen to reduce PM(Particulate Matter) and NOx. According to the annual paper, the non-road diesel engine has to be applied to the Tier-4 final in 2014 which must be reduced to 96% comparing Tier-1's.

So, a variety of research, for example, exhaust gas after-treatment methods or injection control methods, are introduced and adapted to the real world. As compared with these methods, research of engine control fields has not been made in the past year. The reasons are the reliability and cost problems. However, in accordance with the development of data processing, the model based control method is possible to the engine control unit. This method shows the strongpoint in the transient situation and has the prediction control by using the engine model. So, innumerable research has been introduced and there are many control methods such as H-inf., sliding control, fuzzy and modified PID control. [1,2,3]However, unlike PID control which uses only error signals, these control methods based from the engine model need a variety of parameters and state values. Therefore, observer design and adaptation are issued in diesel engine control. Especially, in engine parts that it is difficult to sensing from the noise or heat, observer is an essential element in model based control. [4,5,6]

In this paper, we will show that the exhaust manifold pressure observer design using the reduced engine model. Especially, the trajectory linearization method will be adapted to the observer.

Trajectory linearization method can be used in nonminimum phase, unstable and fast time-varying system. Moreover, exponentially stability is proved. [7]

In fact, it is possible to design general exhaust manifold pressure observer. However it's very hard to design because of the coupling input states. There will be many trial and errors. So as to verify the trajectory linearization observer, 6000cc HP EGR-VTG based diesel engine will be used in the fixed 1000rpm 50~100loads.

II. ENGINE MODEL

The engine with HP EGR –VTG can be modeled by tenth or eleventh differential equations when the air flow management is only considered. However, spelling over the sensitivity of the engine characteristics, engine model can be reduced to seventh order equations as follows.

$$\frac{dm_1}{dt} = W_{c1} + W_{21} - W_{1e} - W_{12}$$
$$\frac{dm_2}{dt} = W_{12} + W_{e2} - W_{21} - W_{2t}$$
$$\dot{F}_1 = \frac{W_{21}(F_2 - F_1) - F_1 W_{c1}}{m_1}$$
$$\dot{F}_2 = \frac{W_{e2}(F_{e2} - F_2) - W_{12}(F_1 - F_2)}{m_2}$$
$$\frac{dT_1}{dt} = \frac{W_{21}(h_{21} - u_1) - W_{c1}(h_{c1} - u_1) - (W_{1e} - W_{12})R_1 T_1 - m_1 \chi_{F1} \dot{F}_1}{c_{v1} m_1} - \frac{\dot{Q}_1}{c_{v1} m_1}$$
$$\frac{dT_2}{dt} = \frac{W_{e2}(h_{e2} - u_2) - W_{12}(h_{12} - u_2) - (W_{21} - W_{2t})R_2 T_2 - m_2 \chi_{F2} \dot{F}_2}{c_{v2} m_2} - \frac{\dot{Q}_2}{c_{v2} m_2}$$
$$\frac{dP_c}{dt} = \frac{1}{\tau_{tc}}(-P_c + \eta_{tm} P_t) \tag{1}$$

The subscripts of the equations (1) are expressed as follows

1 : intake manifold 2 : exhaust manifold

c : compressor e : engine

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ij : flow from volume i to volume j

This model shows that thermodynamics with insulation condition and the principle of the conservation of energy and mass are used in designing the air management system of the mean value engine model. The diesel engine mechanical parts and each state are expressed in figure 1.

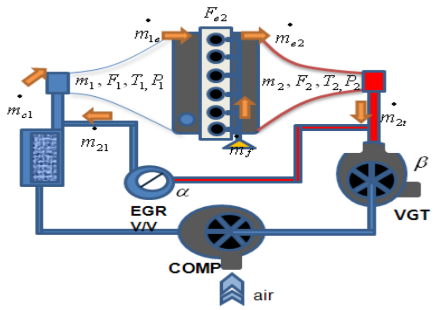


Figure1-Diesel engine model

The model based controller in this paper is based from the nonlinear Multi Input Multi Output (MIMO) system with combination of EGR and VTG flow inputs. This controller can be worked in transient states and predicts future works comparing PID controller. However, the controller can't consider every state like equation 1 because of the control stability analysis and ECU implementation. So, we use the reduced order engine model equation which is composed of the intake and exhaust manifold pressure and compressor power differential equations as follows equation 2.

$$\dot{P}_1 = k_1\left(\frac{\eta_c}{C_P T_a}\frac{P_C}{\left(\left\{\frac{P_1}{P_a}\right\}^{\mu}-1\right)} + W_{egr} - k_e P_1\right),$$

$$\dot{P}_2 = k_2(k_e P_1 + W_f - W_{egr} - W_{2t}),$$

$$\dot{P}_c = \frac{1}{\tau}\left(-P_c + \eta_{tm}\eta_{tis} C_{pa} T_2\left(1-\left(\frac{P_a}{P_2}\right)^{\mu}\right) W_{2t}\right),$$

$$where\ k_1 = \frac{R_a T_1}{V_1},\quad k_e = \frac{\eta_v N V_d}{R_a T_1},\quad k_2 = \frac{R_a T_2}{V_2} \qquad (2)$$

There are two inputs EGR flow and VTG flow respectively on the assumption that value actuator and conversion flow equation are omitted.

Before designing the controller, the comparison of engine model and reduced engine model must be taken precedence because reduced engine model is based in designing the controller and observer.

Verification process is from the 1000RPM 50~100% load transient situation which shows that operation point 1, 2 and 1 for each ten seconds respectively except in the early simulation for ten seconds. Reduced engine model and engine model are named third order model and seventh order model respectively.

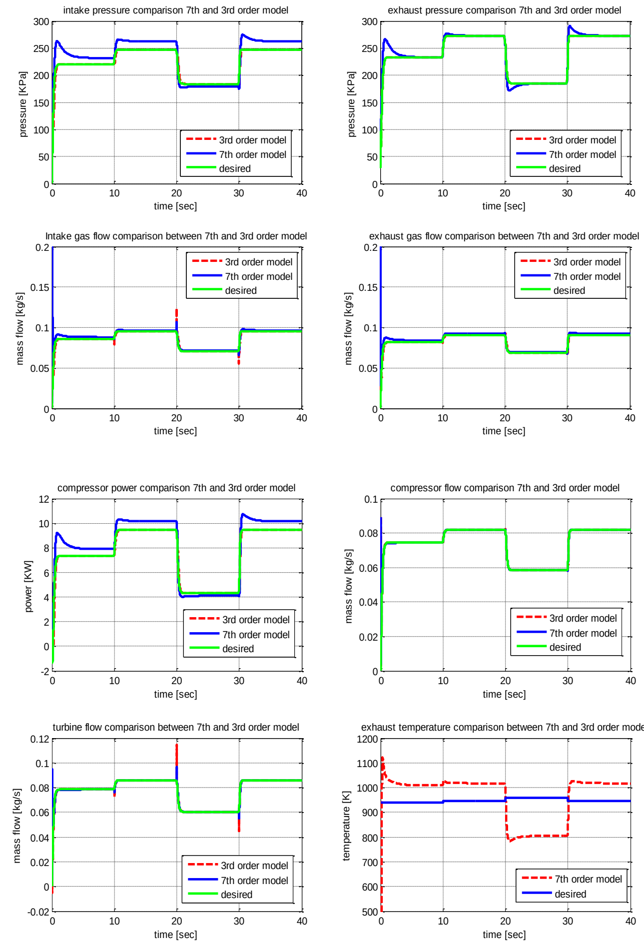


Figure 2. Comparison of 3rd order and 7th order model

Reduced order model and seventh order model have the similar results from the figure 2. Especially, desired value and reduced order model use the constant intake and exhaust temperature and efficiency values of turbine systems, more tracking performance is presented comparing the seventh order model's.

III. OBSERVER DESIGN

From the figure 1 results, reduced order model is used in designing the controller. among the a variety of the model based controller, we consider only sliding mode controller with VTG flow and EGR flow as input states, Yoon[8 proves that output set of exhaust manifold pressure and compressor flow shows the best performance in the considering output sets using input-output linearization method. However, though the sensor with compressor flow is possible, exhaust manifold pressure sensing is difficult because of the reliability about the heat and pressure. Most of all, the cost increase militates as a disadvantage factor in the engine manufacturing.

Therefore, the exhaust manifold pressure observer has to be designed in model based controller. the observer dynamics use the reduced order model equation in the same with controller's.

Before observer is designed, we need to check the observability in this system. Using the Lie derivative method in nonlinear system, the observability can be verified as follows equation 3. The output set is in observer is intake manifold pressure and compressor flow. There is assumption that the intake manifold pressure and compressor flow are measured by the MAP and MAF respectively.

$$y_1 = P_i = h_1(x)$$
$$y_2 = P_C = h_2(x)$$
$$dh = \begin{bmatrix} 1 & 0 & 1 \end{bmatrix}$$
$$L_f h = dh \cdot f = -\frac{N}{V_i}\frac{\eta_v V_d}{2\cdot 60} P_i + \frac{RT_i}{V_i}\frac{\eta_c}{C_P T_a}\frac{1}{\left(\left\{\frac{P_1}{P_a}\right\}^{\mu} - 1\right)} P_c$$
$$-\frac{1}{\tau} P_c + \frac{\eta_m}{\tau}\eta_t c_p T_x \left(1 - \left(\frac{P_a}{P_x}\right)^{\mu}\right) W_{xi}$$
$$d(L_f h) = \left[-\frac{N}{V_i}\frac{\eta_v V_d}{2\cdot 60} + \frac{RT_i}{V_i}\frac{\eta_c}{C_P T_a}\frac{\mu P_1^{\mu-1}}{P_1^{2\mu} - 2P_1^{\mu}P_a^{\mu} + P_a^{2\mu}} P \right.$$
$$\left. \frac{\eta_m}{\tau}\eta_t c_p T_x \left(1 - \mu\left(\frac{P_a}{P_x}\right)^{\mu-1}\right) W_{xi} \quad \frac{RT_i}{V_i}\frac{\eta_c}{C_P T_a}\frac{1}{\left(\left\{\frac{P_1}{P_a}\right\}^{\mu} - 1\right)} + -\frac{1}{\tau} \right]$$
$$L_f L_f h = d(L_f h)\cdot f = \left[\left(-\frac{N}{V_i}\frac{\eta_v V_d}{2\cdot 60} + \frac{RT_i}{V_i}\frac{\eta_c}{C_P T_a}\frac{\mu P_1^{\mu-1}}{P_1^{2\mu} - 2P_1^{\mu}P_a^{\mu} + P_a^{2\mu}} P_c \right)\cdot f \right.$$
$$\left. \left(\frac{\eta_m}{\tau}\eta_t c_p T_x \left(1 - \mu\left(\frac{P_a}{P_x}\right)^{\mu-1}\right) W_{xi} \right)\cdot f_2 \quad \left(\frac{RT_i}{V_i}\frac{\eta_c}{C_P T_a}\frac{1}{\left(\left\{\frac{P_1}{P_a}\right\}^{\mu} - 1\right)} + -\frac{1}{\tau} \right)\cdot f_3 \right] \cdots$$
$$O(x) = \begin{bmatrix} dh \\ d(L_f h) \\ d(L_f L_f h) \end{bmatrix} \Longrightarrow rank(O(x)) = 3 \tag{3}$$

The rank is three which shows the same number of the observer equations. It means that this system is locally observable system.

So, the Luenberger observer is designed and then, the results are expressed in the figure 3.

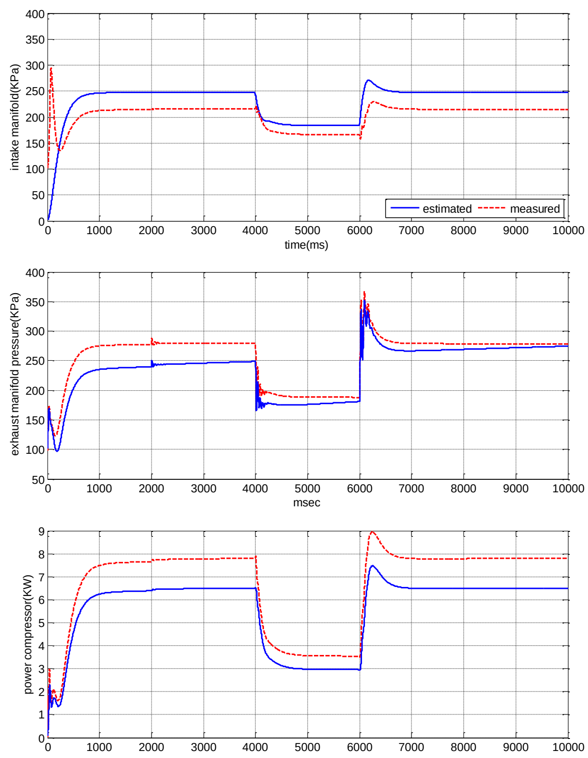


Figure 3. Luenberger observer simulation results

This result shows that the intake manifold pressure and compressor power is similar to the measured data except to the exhaust manifold pressure. The exhaust manifold pressure is also possible to the real value from the many trial and error of observer gain tuning. There are no feedback term and only intake manifold pressure and compressor power observer gain term in exhaust manifold pressure differential equation. So, it is difficult to control the observer error term perfectly. It shows underactuated system's drawback.

Therefore, we use the trajectory linearization method to solve this problem. Most of all, the trajectory linearization method has the assumption that the model can be presented an almost complete similarity to the plant that is good to the engine plant because of the robust characteristics. The trajectory linearization method is only considered to the error term which is similar to the disturbance observer's feature except to the trajectory state. The outline of trajectory linearization method is as follows figure 4.

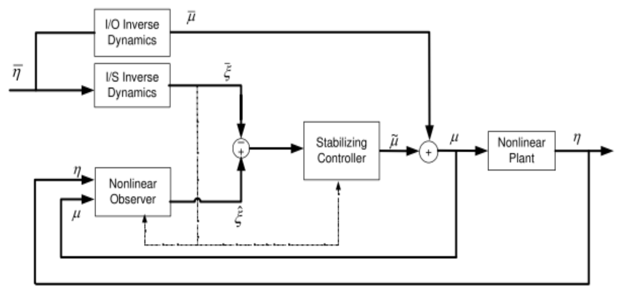


Figure 4. Trajectory linearization scheme

Given the nonlinear system is expressed as follows.

$$\dot{\tilde{\xi}} = f(\bar{\xi}(t) + \tilde{\xi}(t), \bar{\mu}(t) + \tilde{\mu}(t), t) - f(\bar{\xi}(t), \bar{\mu}(t), t)$$
$$\tilde{\eta} = h(\bar{\xi}(t) + \tilde{\xi}(t), \bar{\mu}(t) + \tilde{\mu}(t), t) - h(\bar{\xi}(t), \bar{\mu}(t), t) \tag{4}$$

And the nonlinear observer is designed as follows.

$$\dot{\hat{\tilde{\xi}}} = f(\bar{\xi}(t) + \hat{\tilde{\xi}}(t), \bar{\mu}(t) + \tilde{\mu}(t), t) - f(\bar{\xi}(t), \bar{\mu}(t), t) + K_0(\cdot)(\hat{\tilde{\eta}} - \tilde{\eta})$$
$$\hat{\tilde{\eta}} = h(\bar{\xi}(t) + \hat{\tilde{\xi}}(t), \bar{\mu}(t) + \tilde{\mu}(t), t) - h(\bar{\xi}(t), \bar{\mu}(t), t) \tag{5}$$

Drive nonlinear observer error $\hat{\tilde{x}} = \hat{\tilde{\xi}} - \tilde{\xi} \to 0$ as $t \to \infty$.

By choosing the appropriate parameters of $H(\bullet)$.

$$\dot{\hat{\tilde{x}}} = A_0\hat{\tilde{x}} + H\hat{\tilde{y}} = (A_0 + K_0(\cdot)C_0)\hat{x}$$
$$y = C_0\hat{\tilde{x}} \tag{6}$$
$$where\ \hat{\tilde{x}} \approx \hat{\tilde{\xi}} - \tilde{\xi},\ \hat{\tilde{y}} \approx \hat{\tilde{\eta}} - \tilde{\eta}$$
$$A_0 = A(\tilde{\xi}, \mu)\frac{\partial}{\partial \hat{\tilde{\xi}}} f(\bar{\xi}, \mu)_{|\xi,\mu}$$
$$C_0 = C(\tilde{\xi}, \mu)\frac{\partial}{\partial \hat{\tilde{\xi}}} h(\bar{\xi}, \mu)_{|\xi,\mu}$$

Using the error state $\hat{\tilde{x}} = \hat{\tilde{\xi}} - \tilde{\xi}$, the trajectory linearization is enacted as follows equation 7.

$$\begin{bmatrix}\dot x_1\\ \dot x_2\\ \dot x_3\end{bmatrix} = \begin{bmatrix} -K_1K_e + K_1\frac{\eta_c}{C_pT_a}\frac{\mu x_1^{\mu-1}}{x_1^{2\mu}-2x_1^\mu p_a^\mu + p_a^{2\mu}}x_3 & 0 & K_1\frac{\eta_c}{C_pT_a}\frac{1}{\left(\left\{\frac{x_1}{P_a}\right\}^\mu -1\right)} \\ K_2K_e & 0 & 0 \\ 0 & -\frac{\eta_m}{\tau}\eta_t c_p T_x W_{xt}\frac{\mu x_2^{\mu-1}}{x_2^{2\mu}} & -\frac{1}{\tau}\end{bmatrix}\begin{bmatrix}x_1\\x_2\\x_3\end{bmatrix} + \begin{bmatrix}\frac{RT_i}{V_i} & 0 \\ -\frac{RT_x}{V_x} & -\frac{RT_x}{V_x} \\ 0 & \frac{\eta_m}{\tau}\eta_t c_p T_x\left(1-\left(\frac{P_a}{x_2}\right)^\mu\right)\end{bmatrix}\begin{bmatrix}W_{xi}\\W_{xt}\end{bmatrix}$$

where $K_e = \frac{N}{RT_i}\frac{\eta_v V_d}{2\cdot 60},\quad K_1 = \frac{RT_i}{V_i},\quad K_2 = \frac{RT_x}{V_x},\ x_1 = \hat p_i - \bar p_i,\ x_2 = \hat p_x - \bar p_x, x_3 = \hat P_c - \bar P_c$

$$\begin{bmatrix}y_1\\y_2\end{bmatrix} = \begin{bmatrix}1&0&0\\0&0&1\end{bmatrix}\begin{bmatrix}x_1\\x_2\\x_3\end{bmatrix} \tag{7}$$

To transform the observer canonical form, apply coordinate transformation to the observer model as follows.

$$T = \begin{bmatrix} A_3A_4 & -A_1A_4 + A_2A_4 & 0 \\ -A_5 & A_4 & -A_1 + A_2 \\ 1 & 0 & 1\end{bmatrix} \tag{8}$$

$$A_z = TA_0T^{-1} = \begin{bmatrix}0&0&A_2A_3A_4\\1&0&-A_1A_5\\0&1&A_1+A_5\end{bmatrix}$$

$$z_i = Tx_i$$

$$C_z = \begin{bmatrix}0&0&0\\0&0&1\end{bmatrix}$$

where

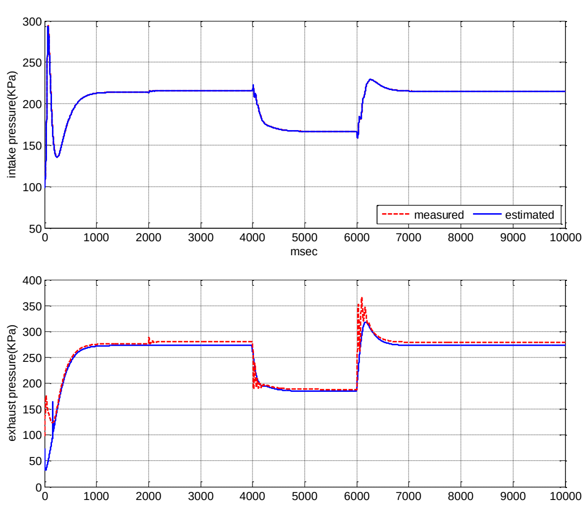
$$A_1 = -K_1K_e + K_1\frac{\eta_c}{C_pT_a}\frac{\mu x_1^{\mu-1}}{x_1^{2\mu}-2x_1^\mu p_a^\mu + p_a^{2\mu}}x_3,\quad A_2 = K_1\frac{\eta_c}{C_pT_a}\frac{1}{\left(\left\{\frac{x_1}{P_a}\right\}^\mu -1\right)}$$

$$A_3 = K_2K_e,\qquad A_4 = -\frac{\eta_m}{\tau}\eta_t c_p T_x W_{xt}\frac{\mu x_2^{\mu-1}}{x_2^{2\mu}},\qquad A_5 = -\frac{1}{\tau}$$

We choose observer gain as appropriate parameters to stabilize $A_z + H_iC_z$ as follows.

$$\dot z_i = \begin{bmatrix}0&0&A_2A_3A_4\\1&0&-A_1A_5\\0&1&A_1+A_5\end{bmatrix}\begin{bmatrix}z_1\\z_2\\z_3\end{bmatrix} + \begin{bmatrix}H_1z_3\\H_2z_3\\H_3z_3\end{bmatrix} \tag{9}$$

To verify the observer performance, the same situation is adapted to the simulation. The result is as follows figure 5.



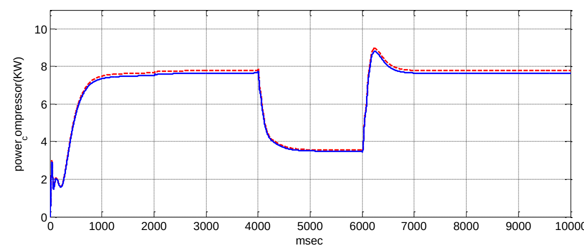


Figure 5. Trajectory linearization observer simulation results

In this figure, the estimated states can be tracked to the real valued. Especially, the exhaust manifold pressure has the 2.8% tracking error. Owing to the observer canonical form by trajectory linearization, the exhaust manifold pressure is decoupled to the observer gain of measured states, is only influenced to the modified output z_3.

IV. RESULTS

In this paper, we conclude that the exhaust manifold observer is adapted to the model based sliding mode control with diesel engine air management. Especially, using the trajectory linearization method, the exhaust manifold observer gain is decided easily comparing to the Luenberger observer gain. Moreover, the observer dynamics can be expressed to the simple form which is only influenced to the one output.

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