Modeling and Control of Single Turbocharger with High Pressure Exhaust Gas Recirculation Diesel Engine

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ABSTRACT

The regulation of NOx emission from a diesel engine is becoming stricter by the environmental agency and furthermore, EGR has been adopted to reduce NOx emission. Air flow control of a diesel engine combined with turbocharger and EGR is complicated because turbocharger and EGR are coupled physically. This paper examines the design of a model based control structure for diesel engine with a turbocharger and EGR. Model based control provides desirable control strategies from the physical analysis of an engine model. Air fraction and exhaust manifold pressure target based control simulation was conducted in this research. Engine is highly nonlinear and susceptible to uncertainties. To overcome this problem, a robust controller was designed by using a sliding mode control method.

INTRODUCTION

Reducing NOx emission in a diesel engine is a major issue for its environmentally harmful influences. To cope with the problem, the current diesel engine is equipped with EGR. Actually, it is known that EGR can help to reduce NOx emission by limiting oxygen supplied into intake manifold, lowering the combustion temperature. However, it also reduces engine performance. Thus air flow control of a diesel engine is important to find an optimal trade-off point between engine performance and emission quantity.

NOx emission can be adjusted by regulating burnt gas fraction which goes into intake manifold. However, burnt gas fraction is not easily measured or estimated because of its physical definition. Therefore, modified target like pressure and compressor flow was used to control desired intake burnt gas fraction indirectly [1]. Although modified target control is not required for an accurate air fraction sensor, there is a limitation that discrepancy between real burnt gas fraction and desired burnt gas fraction exists as a result of indirect control method.

Recently, with the introduction of air fraction sensor like UEGO sensor, burnt gas fraction can be measured easily. Therefore, without converting control target variable, controller can be made which directly control burnt gas fraction, which makes precision control possible.

In this paper, a generic model based control structure is designed to control the diesel engine with EGR and turbocharger. The main advantage of the model based control is that it can easily respond to each operating point because the controller operation follows the physical model [2,6]. Compared with map based control its algorithm is logical. Thus the work of calibration can be reduced.

The paper consists of the following: First, thermodynamic equations are suggested to represent each part of engine. Given dynamics, differential equations are obtained to get a state space form, which is a conventional modeling method. Then how to get a desired value is discussed. From the state space form and desired value sliding mode controller which shows robust performance given uncertainties is designed for control target can follows each desired values. Finally, simulation results are presented.

SYSTEM DESCRIPTION

Engine Description

The engine plant considered is a 6.0L 6-cylinder heavy duty diesel engine. Each subpart was modeled to match given engine specifications. It is assumed that pressure at exhaust manifold, burnt gas fraction at intake manifold, burnt gas fraction at exhaust manifold can be measured and these measured values are used as inputs to the model. In Figure 1, dashed lines represent the control volumes of intake manifold and exhaust manifold.

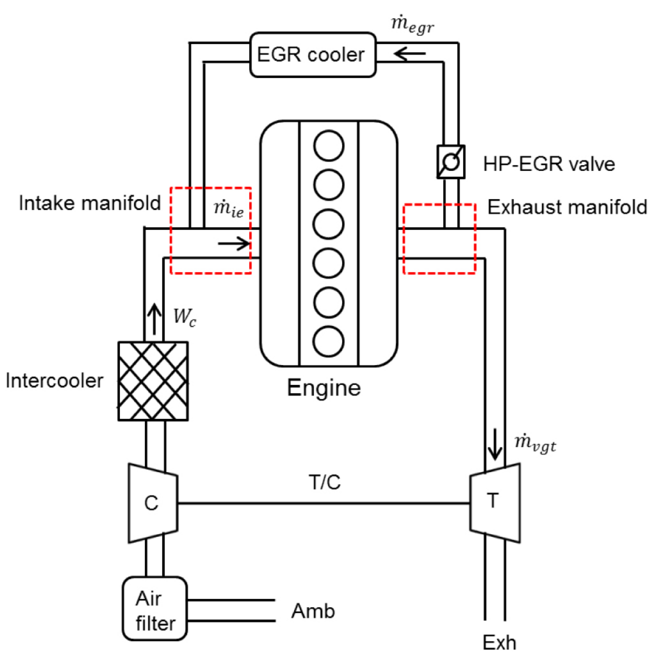

Figure 1. Diesel engine architecture with VGT and EGR systems

System Dynamics

Intake Manifold

From the mass flow equation and ideal gas law [1],

$$\begin{cases} \dot{m}_1 = (W_c + \dot{m}_{egr} - \dot{m}_{ie}) \\ \dot{p}_1 = \dfrac{\gamma R}{V_1}(T_{ic}W_c + T_r\dot{m}_{egr} - T_1\dot{m}_{ie}) \end{cases} \tag{1}$$

Where $\dot{m}_1$ represents rate of change of the mass stored in the intake manifold (see intake control volume in Figure 1). W_c is mass flow rate which comes from compressor to intake manifold. V_1 is volume of the intake manifold and γ is specific heat ratio. $\dot{m}_{egr}$ represents mass flow rate which comes from EGR to intake manifold. P_1 and T_1 are pressure and temperature of intake manifold each. R is the ideal gas constant. T_r is the temperature after EGR cooler and T_{ic} is the temperature after intercooler. It is assumed that thermal loss by work can be neglected.

$\dot{m}_{ie}$ is mass flow rate which goes into the cylinder. From the engine cylinder dynamics it is expressed as [2]:

$$\dot{m}_{ie} = \eta_{vol}\frac{m_1}{V_1}\frac{NV_d}{120} \tag{2}$$

where η_{vol} is the volumetric efficiency of the engine. η_{vol} is changeable parameter depending on operating point. Therefore, mapping data from experiment $\eta_{vol}(N, p_i)$was used in simulation. V_d is the engine displacement volume of all cylinders and N is the engine RPM.

Burnt gas fraction is defined as ratio mass of burnt gas to total mass of gas. Thus intake burnt gas fraction is given by:

$$F_1 = \frac{m_1 - m_{1,air}}{m_1} \tag{3}$$

where $m_{1,air}$ is mass of unburned gas at the intake manifold.

By differentiating (3) and using (1), the following differential equation can be obtained from the system physical structure [1]:

$$\dot{F}_1 = \frac{RT_1}{p_1V_1}(\dot{m}_{egr}(F_2 - F_1) - W_cF_1) \tag{4}$$

where F_2 is the burnt gas fraction at exhaust manifold, which is defined as:

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

Exhaust Manifold

Similar with intake manifold, if we assume thermal loss by work can be neglected [1],

$$\begin{cases} \dot{m}_2 = (\dot{m}_{ex} - \dot{m}_{egr} - \dot{m}_{vgt}) \\ \dot{p}_2 = \dfrac{\gamma R}{V_2}(\dot{m}_{ex}T_{ex} - \dot{m}_{egr}T_2 - \dot{m}_{vgt}T_2) \end{cases} \tag{6}$$

where $\dot{m}_2$ represents rate of change of the mass stored in the exhaust manifold. V_2 is volume of the exhaust manifold (see exhaust control volume in Figure 1). It is assumed that T_{ex}

and T_2 are same because temperature is relatively not significant variable which affects system response.

$\dot{m}_{ex}$ is rate of change of the mass which goes out to the exhaust manifold. $\dot{m}_{vgt}$ is rate of change of the mass which goes into the turbine. $\dot{m}_{ex}$ is described by:

$$\dot{m}_{ex} = \dot{m}_f + \dot{m}_{ie} \tag{7}$$

where $\dot{m}_f$ is mass flow rate which comes into the cylinder. In the desired value generation section, W_f will be used as a simple notation.

Similar with intake burnt gas fraction, differential equation about exhaust burnt gas fraction is obtained as follows [1]:

$$\dot{F}_2 = \frac{\dot{m}_{in}}{m_2} F_1 - \frac{\dot{m}_{in} + \dot{m}_f}{m_2} F_2 + \frac{\dot{m}_f}{m_2}(1+\lambda_s) \tag{8}$$

where λ_s is stoichiometric ratio. It is assumed that fuel is combusted at stoichiometric ratio. In simulation, 14.6 was used as a λ_s value.

EGR

$\dot{m}_{egr}$ is governed by orifice equation as follows:

$$\dot{m}_{egr} = C_d A p_2 \sqrt{\left[\frac{2\gamma}{R_2 T_2(\gamma-1)}\left(\left(\frac{p_1}{p_2}\right)^{\frac{2}{\gamma}} - \left(\frac{p_1}{p_2}\right)^{\frac{\gamma+1}{\gamma}}\right)\right]} \tag{9}$$

where C_d is discharge coefficient and A is EGR pipe cross sectional area. It is generally defined as C_dA is effective area. In the simulation, valve position is converted to effective area by look-up table to control $\dot{m}_{egr}$.

Turbine and Compressor

Turbine and compressor are connected by shaft. Thus related equations between turbine and compressor are as follows:

$$\frac{d}{dt}\left(\frac{1}{2} J_{tc} n_{tc}^2\right) = P_t - P_c \tag{10}$$

$$\dot{P}_c = \frac{1}{\tau}(\eta_{tm} P_t - P_c) \tag{11}$$

where P_t and P_c are turbine and compressor power each. J_{tc} represents turbine shaft inertia. τ is time constant. η_{tm} is modified turbine efficiency, defined as $\frac{P_c}{P_{t,s}}$. $P_{t,s}$ is turbine power at isentropic process. η_{tm} is same as η_t when it reaches a steady state.

From the pressure ratio between upstream and downstream of turbine, temperature at the turbine and turbine power are defined by:

$$T_t = T_2 - \eta_t T_2 \left(\left(\frac{p_t}{p_2}\right)^{\frac{\gamma-1}{\gamma}} - 1\right) \tag{12}$$

$$P_t = \dot{m}_{vgt} c_p (T_2 - T_t) = \dot{m}_{vgt} c_p \eta_t T_2 \left(1 - \left(\frac{p_t}{p_2}\right)^{\frac{\gamma-1}{\gamma}}\right) \tag{13}$$

where p_2 is pressure of exhaust manifold (upstream of turbine). p_t is pressure at turbine. c_p is specific heat at constant pressure and η_t is turbine efficiency, defined as $\frac{P_t}{P_{t,s}}$. Similary, temperature at the compressor and compressor power are defined by:

$$T_c = T_a + T_a \frac{1}{\eta_c}\left(\left(\frac{p_c}{p_a}\right)^{\frac{\gamma-1}{\gamma}} - 1\right) \tag{14}$$

$$P_c = \dot{m}_c c_p (T_c - T_a) = W_c c_p \frac{1}{\eta_c} T_a \left(1 - \left(\frac{p_c}{p_a}\right)^{\frac{\gamma-1}{\gamma}}\right) \tag{15}$$

where p_c is pressure at compressor and p_a is ambient pressure. η_c is compressor efficiency.

Compressor mass flow rate is obtained by rearranging above compressor power equation:

$$W_c = \frac{\eta_c}{T_a c_p} \frac{P_c}{\left(\frac{P_1}{P_a}\right)^{\frac{\gamma-1}{\gamma}} - 1} \tag{16}$$

State Space Model

State space modeling was done to verify the system, with state variables of interest. From the burnt gas fraction and exhaust manifold pressure differential equations, state space model can be obtained as follows:

$$\begin{bmatrix}\dot{F}_1\\ \dot{F}_2\\ \dot{p}_2\end{bmatrix}=\begin{bmatrix}-\frac{W_cF_1}{m_1}\\ \frac{\dot{m}_{in}}{m_2}F_1-\frac{\dot{m}_{in}+\dot{m}_f}{m_2}F_2+\frac{\dot{m}_f}{m_2}(1+\lambda_s)\\ \frac{\gamma R}{V_2}T_2\dot{m}_{ex}\end{bmatrix}+\begin{bmatrix}\frac{(F_2-F_1)}{m_1}u_1\\ 0\\ -\frac{\gamma RT_2}{V_2}\left(\dot{m}_{egr}+\dot{m}_{vgt}\right)\end{bmatrix}$$

$$\begin{bmatrix}y_1\\ y_2\end{bmatrix}=\begin{bmatrix}1&0\\0&1\end{bmatrix}\begin{bmatrix}F_1\\ p_2\end{bmatrix}$$

Above matrix is rewritten for simple representation:

$$\begin{bmatrix}\dot{F}_1\\ \dot{F}_2\\ \dot{p}_2\end{bmatrix}=\begin{bmatrix}-aW_cF_1+a(F_2-F_1)u_1\\ bF_1-cF_2+d\\ e\left[\dot{m}_{ex}-\left(u_1+u_2\right)\right]\end{bmatrix}$$

$$\begin{bmatrix}y_1\\ y_2\end{bmatrix}=\begin{bmatrix}1&0\\0&1\end{bmatrix}\begin{bmatrix}F_1\\ W_c\end{bmatrix}$$

where $a=\frac{1}{m_1},\ b=\frac{\dot{m}_{in}}{m_2},\ c=\frac{\dot{m}_{in}+\dot{m}_f}{m_2},\ d=\frac{\dot{m}_f}{m_2}(1+\lambda_s),\ e=\frac{\gamma RT_2}{V_2}$

Here, $\dot{m}_{egr}$ was set to u_1 and $\dot{m}_{vgt}$ was set to u_2.

System total order is 3rd and there are two control inputs. It is nonlinear MIMO system. Also it is under actuated system because the number of control inputs is less than system order.

Stability Analysis

There exist internal dynamics in under actuated system. Thus stability check is prior to controller design. It is easily found that exhaust burnt gas differential equation is internal dynamics because there is no control input term. By rearranging $\dot{F}_2$ equation,

$$\dot{F}_2-\dot{F}_{2,e}=-c(F_2-F_{2,e})\ \ (c>0)$$

Where $F_{2,e}=\frac{bF_1+d}{c}$.

Thus it is verified that internal dynamics is stable.

Desired Value Generation

The purpose of diesel engine air flow control is to achieve desired engine performance as well as NOx emission regulation. It can be realized for the engine to operate satisfying two conditions:

$$AF_{ref}=AF_{ref}(N,W_f),\ EGR_{ref}=EGR_{ref}(N,W_f)$$

where AF_{ref} is air to fuel ratio reference. EGR_{ref} is EGR fraction reference which is denoted as:

$$EGR_{ref}=\frac{W_{egr}}{W_c+W_{egr}} \tag{17}$$

Given that AF_{ref} and EGR_{ref} values, desired values of several states can be obtained as follows:

$$W_{cd}=\frac{W_f}{2}\left[\beta+\sqrt{\beta^2+4\left(1-EGR_{ref}\right)AF_{ref}}\right] \tag{18}$$

W_{cd} is desired compressor mass flow rate. Coefficient β is defined as [1]:

$$\beta=AF_{ref}(1-EGR_{ref})+(AF_{ref}+1)EGR_{ref}-1 \tag{19}$$

From equation (17) desired EGR mass flow rate is obtained as follows:

$$W_{egrd}=\frac{EGR_{ref}}{1-EGR_{ref}}W_{cd} \tag{20}$$

Thus from equation (19) and (20), F_{1d} is obtained as follows:

$$F_{1d}=\frac{\left(\lambda_s+1\right)W_fW_{egrd}}{W_{ied}\left(W_{cd}+\dot{m}_f\right)} \tag{21}$$

where $W_{ied}=W_{cd}+W_{egrd}$.

Also, p_{2d} can be obtained as:

$$p_{2d}=p_a\left[1-\frac{W_{cd}}{W_{cd}+W_f}\cdot\frac{T_a}{T_2\eta_t\eta_m\eta_c}\left(\left(\frac{p_{1d}}{p_a}\right)^{\frac{\gamma-1}{\gamma}}-1\right)\right]^{-\frac{\gamma}{\gamma-1}} \tag{22}$$

Details of getting desired value are presented at Appendix I of [9].

Controller Design

With two state variables other than exhaust burnt gas fraction, controller can be designed for F_1 and p_2 follow each desired values.

Generally, true value of parameter η_{vol} is not easy to estimate. But it can be assumed that η_{vol} is bounded, $\left|\eta_{vol}-\hat{\eta}_{vol}\right|\le\Delta\eta_{vol}$.

Where $\hat{\eta}_{vol}$ is estimated value.

To achieve control purpose, multivariable sliding mode control method [10] was considered:

Sliding surface s_1 is defined as:

$$s_1 = F_1 - F_{1d} \tag{23}$$

From (4), $\dot{s}_1$ is as follows:

$$\dot{s}_1 = -aW_c F_1 + a(F_2 - F_1)u_1 - \dot{F}_{1d} \tag{24}$$

Control input u_1 is obtained as follows for s_1 to converge zero,

$$u_1 = \frac{1}{a(F_2 - F_1)}\left(aW_c F_1 + \dot{F}_{1d} - \lambda_1 \operatorname{sgn}(s_1)\right), \lambda_1 > 0 \tag{25}$$

where sgn is sign function defined as follows:

$$\operatorname{sgn}(x) := \begin{cases} -1 \text{ if } x < 0, \\ 0 \text{ if } x = 0, \\ 1 \text{ if } x > 0 \end{cases}$$

In simulation, it is assumed that $\dot{F}_{1d} = 0$.

Sliding surface s_2 is defined as:

$$s_2 = p_2 - p_{2d} \tag{26}$$

From (5), $\dot{s}_2$ is as follows:

$$\dot{s}_2 = \frac{\gamma R}{V_2}\left[T_e W_{ex} - T_2\left(u_1 + u_2\right)\right] - \dot{p}_{2d} \tag{27}$$

Control input u_2 is obtained as follows for s_2 to converge zero,

$$u_2 = \left[\left(-\frac{V_2}{\gamma R}\lambda_2 \operatorname{sgn}(s_2) + \frac{V_2}{\gamma R}\dot{p}_{2d}\right) - T_e W_{ex}\right]\left(-\frac{1}{T_2}\right) - u_1, \lambda_2 > 0 \tag{28}$$

In simulation, it is assumed that $\dot{p}_{2d} = 0$.

By substituting (28) to (27),

$$\dot{s}_2 = -\lambda_2 \operatorname{sgn}(s_2) + \frac{\gamma R}{V_2} T_e\left(W_{ex} - \hat{W}_{ex}\right)$$

From (2) and boundary condition of $\left|\eta_{vol} - \hat{\eta}_{vol}\right| \le \Delta\eta_{vol}$,

$$\dot{s}_2 = -\lambda_2 \operatorname{sgn}(s_2) + \frac{\gamma R}{V_2} T_e \frac{m_1}{V_1}\frac{NV_d}{120}\Delta\eta_{vol}$$

Hence, controller gain $\lambda_2 > \frac{\gamma R}{V_2} T_e \frac{m_1}{V_1}\frac{NV_d}{120}\Delta\eta_{vol}$ was selected to cover worst case parameter uncertainty.

In simulation, sgn(s) was replaced to sat(s) because of chattering problem in sliding mode controller.

Where sat is saturation function defined as follows:

$$sat(x) := \begin{cases} x \text{ if } -1 \le x \le 1, \\ 1 \text{ if } x > 1, \\ -1 \text{ if } x < -1, \end{cases}$$

Simulation

Simulation was done with simulink engine model. Engine model was made based on 7th order engine dynamics. Pressures, temperatures and mass flow rate are calculated at certain engine RPM and fuel rate. These values are transferred to controller then desired $\dot{m}_{egr}$ and $\dot{m}_{vgt}$ (which are u_1 and u_2 each in state space model) are suggested as explained above sliding mode control. Through conversion model based on EGR orifice equation and turbine dynamics, desired HP EGR valve position which adjust effective area and desired turbine rack position are calculated.

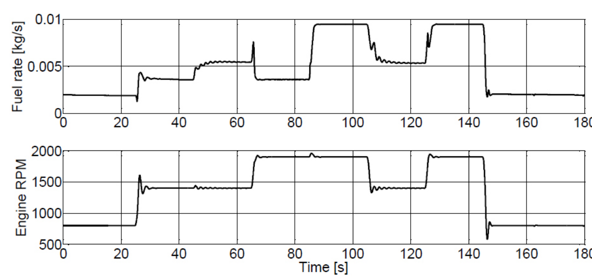

Figure 2. C1-08 mode of engine RPM and fuel rate

Simulation was done with C1-08 mode, which covers several engine operating points. In the C1-08 mode, engine RPM and fuel mass flow rate vary with time. C1-08 mode values are shown in Figure 2.

Burnt gas fraction at intake manifold and burnt gas fraction at exhaust manifold were calculated from state space form, when it reaches at steady state:

$$F_1 = \frac{\left(\lambda_s + 1\right)W_f W_{egr}}{W_{ie}\left(W_c + W_f\right)}$$

$$F_2 = \frac{\left(\lambda_s + 1\right)W_f}{W_c + W_f}$$

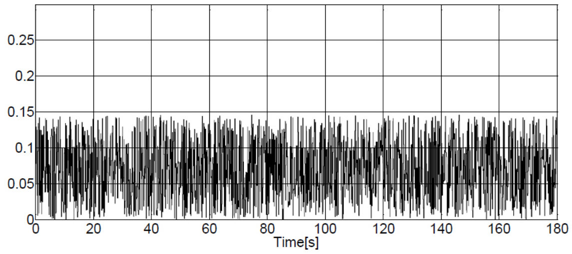

Figure 3. $|\Delta\eta_{vol}|/\eta_{vol}$

To verify robustness given parameter uncertainty, $|\Delta\eta_{vol}| < 0.15\eta_{vol}$ was given to the controller. As Figure 3 shows, uncertainty of volumetric efficiency was set randomly with time.

In case of desired value calcualtion, (22) is too ideal to apply. It's difficult to find out exact value of p_{2d} due to parameter uncertainties involved in the equation. Thus WAVE [11] simulation data run by conventional map based controller in C1-08 mode was used as a desired reference. From WAVE, W_{cd}, p_{1d}, p_{2d} map data were given. Then only F_{1d} was calculated as explained in desired value generation section. Simulation sampling time was set at 1ms.

Results

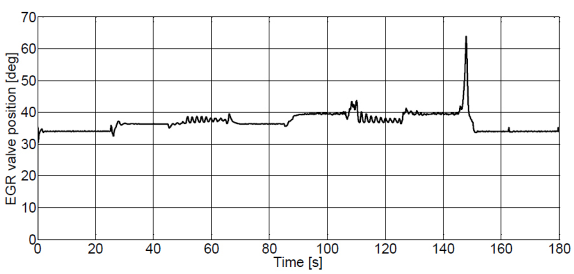

Figure 4. EGR valve position

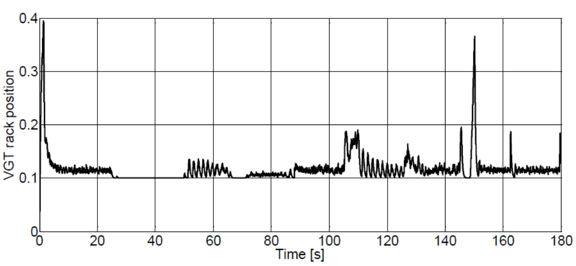

Figure 5. VGT rack position

EGR valve position and VGT rack position are shown Figure 4 and Figure 5. Considering valve response calculated position signals were filtered by low pass filter. EGR valve actuator bandwidth was set to 3.3rad/s and VGT actuator bandwidth was set to 5rad/s.

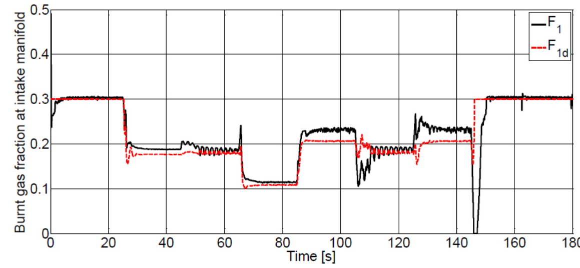

Figure 6. Burnt gas fraction at intake manifold

The graph of burnt gas fraction at intake manifold is shown in Figure 6. Considering engine combustion property, maximum F_{1d} was set to 0.3. Dotted red line represents the desired value and solid black line is the simulated value. Generally, F_1 followed the F_{1d} well. Although actuator bandwidth and parameter uncertainty might affect the controller performance, there were some implausible discrepancies appeared at some ranges.

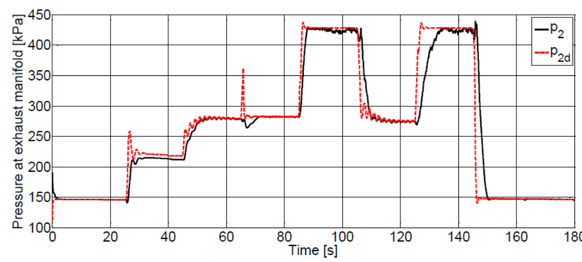

Figure 7. Pressure at exhaust manifold

The graph of pressure at exhaust manifold is shown in Figure 7. p_2 followed p_{2d} well in most of operating points. Only obvious discrepancies appeared at ranges where p_{2d} changes rapidly.

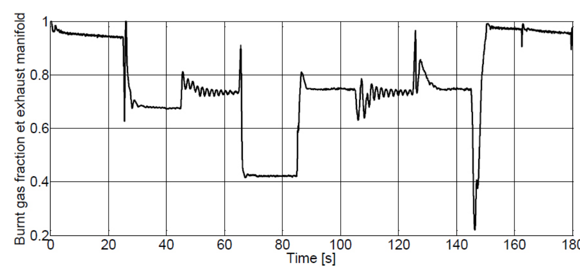

Figure 8. Burnt gas fraction at exhaust manifold

Also, the graph of burnt gas fraction at exhaust manifold is shown in Figure 8. As explained in the stability analysis section, it is internal dynamics which is not directly affected by the inputs. However, it remains stable at all operating points because of stable property.

SUMMARY/CONCLUSIONS

In this paper, a model structure based on burnt gas fraction and pressure at exhaust manifold is suggested for a diesel engine with VGT and EGR. By using the sliding mode control scheme, the controller was verified through simulation. Despite there exist coupling effects between VTG and EGR, two states F_1 and p_2 followed the desired values well. Also, given feasible parameter uncertainty, controller showed robust result.

To explain implausible discrepancies in F_1, F_1 and p_2 are need to be analyzed simultaneously because of coupling effect.

In case of time range and 125-130s, there continues undesirable discrepancy between F_1 and F_{1d}. To reduce F_1, EGR valve starts to close. Meanwhile, p_2 has to be increased to follow p_{2d} in the same time range, which causes F_1 to increase.

In case of time range 105-110s and 145-150s are the opposite case. To increase F_1, EGR valve starts to open. Meanwhile, p_2 has to be decreased sharply to follow p_{2d} in the same time range, which causes F_1 to decrease. Thus, because of change in p_2, there are some range where F_1 cannot follow F_{1d} immediately. Discrepancy becomes apparent when pressure drop is drastic.

Burnt gas fraction sensor can be implemented in a diesel engine but it is expensive. Generally, sensors are not equipped both of intake and exhaust manifold in the engine. There are several studies of estimating burnt gas fraction [7,8]. It is left as a future work to make an observer which estimates burnt gas fraction at one of manifolds. Moreover, a simulink engine model was made to verify the controller. To cover a variety of properties of engine, Simulation needs to be dealt with more reliable engine simulation software to certify a controller.

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DEFINITIONS/ABBREVIATIONS

EGR - Exhaust Gas Recirculation

NOx - Nitrogen Oxide

UEGO - Universal Exhaust Gas Oxygen

MIMO - Multi-Input and Multi Output

APPENDIX

The coefficients used in this paper are as follows:

$$a = \frac{1}{m_1}$$

$$b = \frac{\dot{m}_{in}}{m_2}$$

$$c = \frac{\dot{m}_{in} + \dot{m}_f}{m_2}$$

$$d = \frac{\dot{m}_f}{m_2}(1 + \lambda_s)$$

$$e = \frac{\gamma R T_2}{V_2}$$

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