

# Simplified Multiple Sliding Mode Transient Control with VGT and EGR Diesel Engine

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## ABSTRACT

This paper deals with the simplification and optimization methods in diesel engine control which is based on the air management system model of 1stage VTG and HP-EGR. Especially, simplified model based controller is suggested including more tracking performance of target values for comparison from previous multivariable sliding mode controller with intake and exhaust manifold output set. Moreover, in calculating desired exhaust manifold pressure, the simple process which only use target flow rates is suggested. There are fewer errors than general process's which have to add the target state process including the turbocharger parameter errors. Model based controller has assumption that mathematical model has to be very accurate, therefore, has to reflect the physical engine properties in every operating point. But, it is hard to satisfy this assumption because of the modeling uncertainties in mathematical point and a variety of environmental factor in real engine. Therefore, this paper suggests multiple sliding mode control with simplified process and robust scheme. And, this controller is verified in NRTC mode to analyze transient tracking performance, moreover, compared with previous 3<sup>rd</sup> order diesel engine model based sliding mode controller's performance.

## INTRODUCTION

With technologies advancing at a fast-growing rate to satisfy emission regulation of diesel engine, model based control which is based on mathematic equation has been developed. Especially, in diesel engine control of air management, model analysis and control studies of fixed type turbocharger or VTG type & HP-EGR based diesel engine were developed in early 2000[1]-[11]. In these days, a variety of combination with 2stage turbocharger system and LP-EGR are introduced and studied. This model based controller can replace the map based simple controller in ECU logic because of the robust characteristics and tracking performance of transient operating points. Moreover, model based controller has the advantage in coupling effect of a variety of increasing sensors and actuators. Actually, it is very hard to implement these mathematical

models to the real ECU because of the model uncertainties and sensor errors. Most of all, there are barriers of model calculations and ECU memory size in implementing ECU[15].

For more detailed specific analysis, there are two problems in this study. The first problem is the desired value errors. For example, it is exposed that desired state values which have to be tracked from the controller has the some errors. These errors bring about a result that model based controller's advantage mentioned above is diminished. Especially, in 3<sup>rd</sup> order diesel engine model based controller, it needs to design map interpolation and trial and error method for proper model based control, if we use turbocharger efficiencies and temperature in compressor power dynamics which are changed in real time. Especially, these problems are becoming the main in transient conditions.

The second is that state tracking errors of underactuated system. Sliding mode control which is the well-known nonlinear control is studied in this paper. It is widely known that sliding mode controller is very robust to the model uncertainties and sensor noise when the boundaries of these are known. These characteristics appeals to the engine's air management system. However, previous sliding mode controller with 1 stage VTG and HP-EGR is not changed to the regular form[16]. Therefore, input-output linearization method is used to delete the nonlinear characteristics, and, multivariable sliding surface design is suggested. Of course, according to the less actuator numbers comparing to states which have to be satisfied to the desired values, we need to check the stability of internal dynamics. However, nevertheless internal dynamics are stable, state tracking errors are existed. Moreover, it is very hard to implement long and complicated input equations to real ECU which is from coupled 3<sup>rd</sup> order diesel engine model based simultaneous equations.

So, we suggest that desired values are calculated from the intake manifold pressure dynamics which is main target state in diesel engine control, finally, the control calculation is simpler and more intuitive comparing with the previous multivariable sliding mode control's. Moreover, by

reinterpreting strict feedback form in diesel engine model, single input value can control the multiple sliding surfaces simultaneously. And then, main states and parameters can be tracked to the desired values.

In second chapter, we will show the 1 stage turbocharger and HP-EGR based diesel engine modeling process which is verified from the NRTC mode. In third chapter, the simplified sliding mode control design process will be introduced. Finally, to verify simplified sliding mode controller, we will adopt the controller to the NRTC 200second mode. And tracking performance of previous multivariable sliding mode controller with intake and exhaust manifold pressure output set is will be compared.

## DIESEL ENGINE MODELING

The engine model of this study is the heavy-duty 6000cc diesel engine with HP EGR and VTG system. Especially, 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. Moreover, filling and emptying method that the volume in which the gas is increased or decreased is utilized. The diesel engine mechanical parts and air flow directions are described in fig1. According to the directions of air mass flow, mechanical parts are composed of intake manifold, engine, exhaust manifold, turbo system and EGR valve.

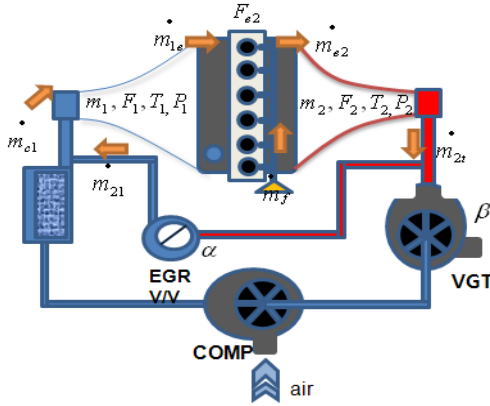


Figure 1. Air management diesel engine model

$$\frac{dm_1}{dt} = W_{c1} + W_{21} - W_{1e} - W_{12} \quad (1)$$

$$\frac{dm_2}{dt} = W_{12} + W_{e2} - W_{21} - W_{2t} \quad (2)$$

$$\dot{F}_1 = \frac{W_{21}(F_2 - F_1) - F_1 W_{c1}}{m_1} \quad (3)$$

$$\dot{F}_2 = \frac{W_{e2}(F_{e2} - F_2) - W_{12}(F_1 - F_2)}{m_2} \quad (4)$$

$$\dot{p}_i = \frac{\gamma R}{V_i} (T_c W_{ci} + T_x W_{EGR} - T_i W_{ie}) \quad (5)$$

$$\dot{p}_x = \frac{\gamma R}{V_x} (T_e W_x - T_x (W_{EGR} + W_{xt})) \quad (6)$$

$$\frac{dP_c}{dt} = \frac{1}{\tau_{tc}} (-P_c + \eta_m P_t) \quad (7)$$

The subscripts of the equations (1)...(7) are expressed as follows.

c : compressor e : engine t : turbine

ij : Bottom subscript expresses flow from volume i to volume j.

1 or i : intake manifold 2 or x : exhaust manifold

F : Fraction of air mass to the total mass of air EGR mixture

p : pressure P : power W : flow

For more information on parameters, find a nomenclature from the end of the paper. Each parts or components of the air management engine models are analyzed and designed from the equation (1)~(7) and the integrated engine model is completed.

Each manifold pressures and remained flows are calculated from the ideal gas equations and flow dynamics such as equation (1) (2). The fraction dynamics are from the oxygen concentration of control volumes and mass conservation such as equation (3),(4). Temperature dynamics of each manifold are designed from the specific energy of each component and fraction dynamics. Turbine flow and efficiencies are designed from experiment data and supplier map data. And these models can be implicated to the turbocharger power dynamics in equation (7). More specific modeling methods are studied in a variety of papers[6][11][16]. To verify the modeling and controller, we compose the HiLS system.

Diesel engine air-management system based on Matlab/Simulink is designed by real experiments and WAVE simulation. This model is embedded on RTPC and connected ECU and CAN bus system, calibration and measuring tests are operated by INCA tool. In controller verification, the

AUTOBOX in which the controller based on Simulink can be operated is connected to the RTPC and CAN bus system. The whole model making and verification process are expressed from fig 2.

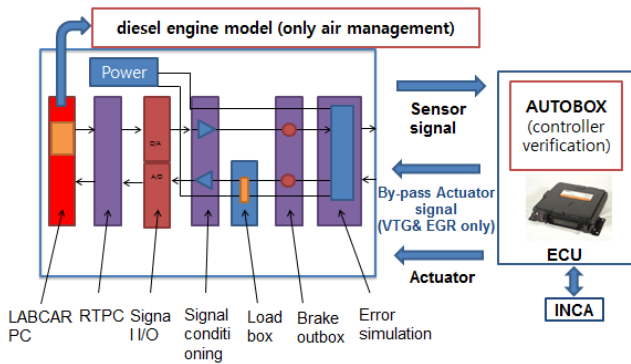


Figure 2. HiLS component diagram

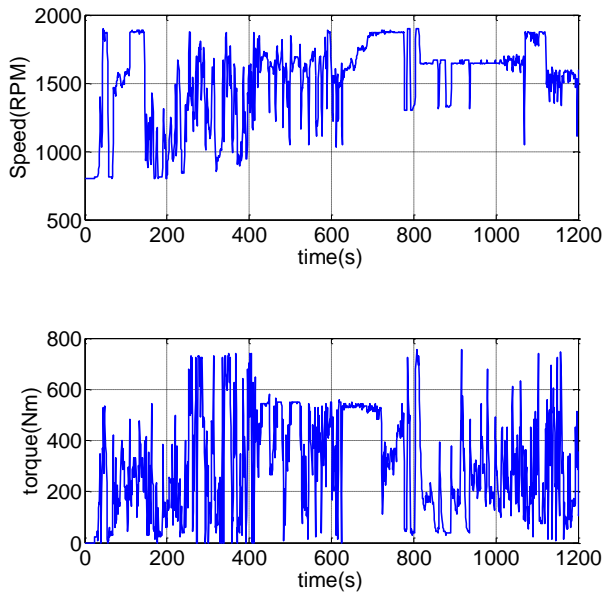


Figure 3. NRTC(Non Road Transient Cycle) mode

The verification driving mode is NRTC(Non Road Transient Cycle) which is a representative driving mode in transient state. Total driving time is about 1200 seconds. Engine RPM and load are independently changed a short time about total engine operating driving points which data are expressed in figure 3. These data is acquired by test engine cell about 6000cc heavy duty diesel. From these properties, it is widely known that it is very hard to track the desired values using conventional control system. Therefore, these driving mode represents the standard in comparing model based controller and conventional controller.

In this part, engine fueling is realized from the engine load and RPM, the controllers of EGR and VTG which are not coupled

each other are the PID base with production ECU. The results graphs are the intake & exhaust pressure, EGR flow, compressor flow and intake & exhaust temperature respectively.

From these figures, there are about 20 percent RMS errors. Especially, flow parameter, compressor flow, have about 10 percent errors in each component.

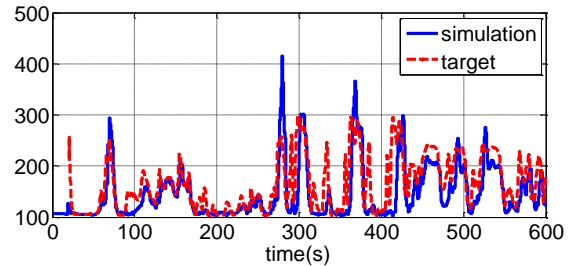


Figure 4. Model verification with intake pressure

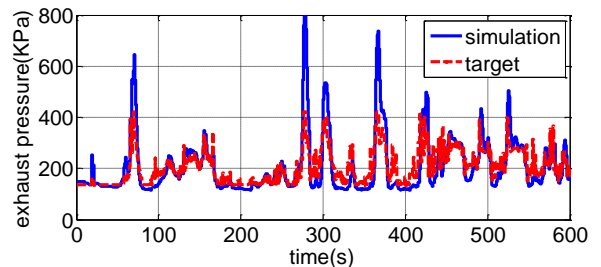


Figure 5. Model verification with exhaust pressure

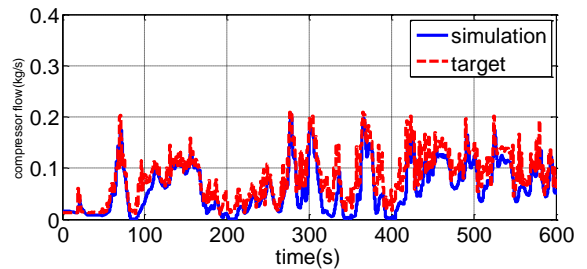


Figure 6. Model verification with compressor flow

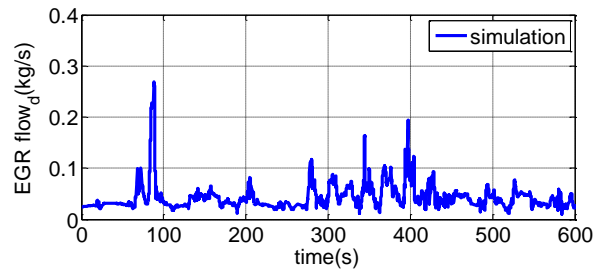


Figure 7. Model verification with EGR flow

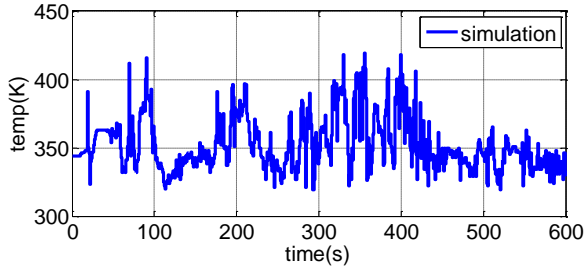


Figure 8. Model verification with intake temperature

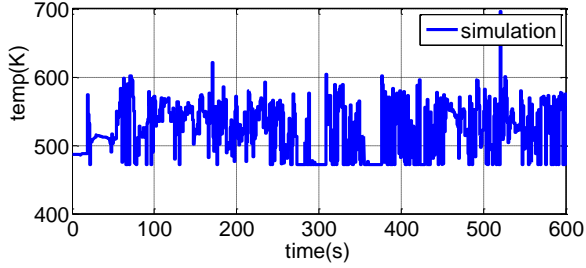


Figure 9. Model verification with exhaust temperature

Other parameters such as temperature, EGR flow can't be compared with target data. These results show that the appropriateness of air system modeling on which we can verify the proposed controller in next chapter. The rests of errors may be interpreted in adding the sensor noise or model uncertainties and can be compensated or solved by the controller on the assumption that the controller is very robust.

## **SIMPLIFIED CONTROLLER DESIGN**

In this paper, The base model for designing the controller with HP-EGR and 1 stage VTG air management is the 3<sup>rd</sup> order diesel engine model. The 3<sup>rd</sup> order model is expressed in equation (8)~(10).

$$\dot{p}_i = k_1 \left( \frac{\eta_c}{C_p T_a} \frac{P_c}{\left( \left\{ \frac{p_i}{p_a} \right\}^\mu - 1 \right)} + W_{EGR} - k_e p_i \right) \quad (8)$$

$$\dot{p}_x = k_2 (k_e p_i + W_f - W_{EGR} - W_{VTG})$$

where  $k_1 = \frac{\gamma R}{V_i}$ ,  $k_2 = \frac{\gamma R}{V_x}$  (9)

$$\dot{P}_c = \frac{1}{\tau} \left( -P_c + \eta_{im} \eta_{iis} C_{pa} T_x \left( 1 - \left( \frac{p_a}{p_x} \right)^\mu \right) W_{VTG} \right) \quad (10)$$

This model composed of three dynamics, the intake & exhaust manifold pressure dynamics which are based from the flow balance and turbocharger power dynamics. The fraction and temperature dynamics of air management diesel engine model can be skipped from these assumptions.

We compose the simplified model based control only considering intake pressure and exhaust pressure dynamics which is the main contribution of this paper.

In case of existing 3<sup>rd</sup> model based controller, according to the coupling effect between VTG valve command and EGR valve command, the calculation process which is simultaneous equation is complicated. This form should make the difficulty in processing ECU implementation and ask for an algorithm embodiment. Moreover, in mathematical analysis, this control system has a general underactuated system property that means two inputs make system states stay equilibrium points or neighborhood of an equilibrium points as early as possible.

Therefore, solving these problems, EGR flow is substituted as intake and exhaust manifold pressure ratio nonlinear function which is from the orifice equation, not an input value. From substituted EGR flow system analysis, the intake pressure differential equation is composed of only states and parameters except the input values, EGR flow. And exhaust pressure differential equation is composed of states and single input, VTG flow.  $f_{0m}$  expresses nonlinear functions composed of intake & exhaust manifold pressure and compressor power states respectively.

$$\dot{p}_i = k_1 \cdot f_{0m}(p_i, p_x, P_c) = k_1 [-W_{ie} + W_{EGR} + W_{ci}] \quad (11)$$

$$\begin{aligned} \dot{p}_x &= k_2 \cdot f_{1m}(p_i, p_x, P_c) + k_2 u_1 \\ &= k_2 [W_{ie} + W_f - W_{EGR} + W_{VTG}] \end{aligned} \quad (12)$$

To control the underactuated system, intake pressure and exhaust pressure have to be controlled by a fictitious input value [17] and real input value simultaneously under the chain rule. Therefore, fictitious input value which deals with the intake pressure dynamics can be calculated from the desired exhaust manifold pressure function. This process shows how to chain up between intake pressure and exhaust pressure dynamics. Desired exhaust pressure is implicated in EGR orifice flow equation. From this processing, the system can be expressed as a strict-feedback form.

By starting next calculations, we will shows how to calculate fictitious input (equation 18) and real input (equation 21) from only intake pressure and exhaust pressure dynamics.

$$\text{If } k_1 \left[ -\frac{\eta_v NV_d}{120RT_i} p_i + \frac{\eta_c}{c_p T_a \left( \frac{p_i}{p_a} \right)^\mu - 1} P_c \right] = f_0, \quad ,$$

$k_1 W_{EGR} = g_0 p_x = g_0 z_1$  in intake manifold pressure dynamics,

$$k_2 \left[ \frac{\eta_v NV_d}{120RT_i} p_i + W_f - W_{EGR} \right] = f_1, \quad k_2 W_{VTG} = g_1 u_1 \quad \text{in}$$

exhaust manifold pressure dynamics, we will express system dynamics such as

$$\begin{aligned} \dot{x} &= f_0(x) + g_0(x)z_1 \\ \dot{z}_1 &= f_1(x, z_1) + g_1(x, z_1)u_1 \end{aligned} \quad (13)$$

And then, we can design the control input when system form satisfies the assumptions as follows.

- $u$  is a scalar input to the system
- $f_0, f_1$  vanish at the origin. (i.e..  $f_i(0,0)=0$ )
- $g_1$  are nonzero over the domain of interest (i.e..  $g_i(0,0) \neq 0$ )
- here, strict feedback refers to the that the nonlinear functions  $f_i$  and  $g_i$  in the  $\dot{z}_i$  equation
- only depend on states  $x, z_1$  that are fed back to that subsystem.

The system form expressed from the equation (13) satisfies these assumptions with engine operating points. Especially, according to physical properties of engine systems, for example, no ignition condition, the system form satisfies second, third and final assumptions.

So, multiple sliding control design is started by defining the sliding surface as intake manifold pressure error.

$$s_1 = p_i - p_{id} \quad (14)$$

To satisfy  $\dot{s}_1 s_1 \leq -\eta s_1^2 / \phi_1$ , where  $\eta$  and  $\phi_1$  are sliding surface gain and model uncertainties bounded value which are determined from the actuator bandwidth and model uncertainty respectively.

Sliding surface error dynamics are developed as follows.

$$\begin{aligned} \dot{s}_1 &= \dot{p}_i - \dot{p}_{id} = k_1 [-W_e + W_{EGR} + W_c] - \dot{p}_{id} \\ &= k_1 \left[ -\frac{\eta_v NV_d}{120RT_i} p_i + W_{EGR} + \frac{\eta_c}{c_p T_a \left( \frac{p_i}{p_a} \right)^\mu - 1} P_c \right] - \dot{p}_{id} \end{aligned} \quad (15)$$

There is assumption that the model uncertainty mentioned above can bounded such as equation (16).

$$\dot{s}_1 = f_0 + \Delta f - \dot{p}_{id}, \quad \Delta f \approx \Delta f_0 + \Delta g_0 \quad (16)$$

Nonlinear equation  $f_0(p_i, p_x)$  can be substituted to

$$\begin{aligned} &f_{0d}(p_i, p_{xd}) \quad \text{for chain rule using } f_{0d}(p_i, p_{xd}) \\ &\equiv \dot{p}_{id} - \frac{k_1 s_1}{\phi_1} \end{aligned}$$

To make fictitious input which is related to the exhaust pressure, upstream and downstream pressure ratio function in EGR flow orifice equation is used. Moreover, to calculate desired intake pressure, the cylinder flow expressed by speed density equation and volumetric efficiency is substituted from the fraction function of AFR and EGR reference. Therefore, we can get the independent desired EGR calculation process based upon the AFR and EGR reference, and satisfying the desired intake and exhaust manifold pressure.

$$\begin{aligned} p_{xd} &= \frac{\left( a_1 \frac{\eta_v NV_d}{120RT_i} p_{i-d} - \frac{k_1 s_1}{\phi_1} - a_1 \frac{\eta_c}{c_p T_a \left( \frac{p_i}{p_a} \right)^\mu - 1} P_c + \dot{p}_{id} \right) \sqrt{T_x R_e}}{a_1 A_{EGR} \psi_{EGR}} \\ &= \frac{W_{EGR-d} \sqrt{T_x R_e}}{a_1 A_{EGR} \psi_{EGR}} \end{aligned} \quad (17)$$

Equation (17) is based on the EGR flow reverse calculation.

And, cylinder flow,  $W_{ie-d} = \frac{\eta_v NV_d}{120RT_i} p_{i-d}$  is substituted

from  $\frac{W_{EGR-d}}{\phi_g}$ . Where  $\phi_g = EGR_{ref} = \frac{W_{EGR-d}}{W_{ci-d} + W_{EGR-d}}$ .

Finally, the fictitious input, desired exhaust pressure is calculated as follows equation (18). Equation (17) and (18) is same property. But, equation (18) can be expressed by EGR reference form which is independent target value in ECU.

$$p_{xd} = \frac{\left( -\frac{k_1 s_1}{\phi_1} - k_1 \frac{\eta_c}{c_p T_a \left( \frac{p_i}{p_a} \right)^\mu} P_c + \dot{p}_{id} \right) \sqrt{T_x R_e}}{k_1 \left( \frac{1}{\phi_g} - 1 \right) A_{EGR} \psi_{EGR}} \quad (18)$$

Equation  $\frac{\sqrt{T_x R_e}}{A_{EGR} \psi_{EGR}}$  mentioned above is from

$$W_{EGR} = \frac{A_{EGR} \psi_{EGR} P_x}{\sqrt{T_x R_e}}, \text{ effective area } A_{EGR} \text{ can be modeled}$$

as a polynomial function of the EGR valve position in a suitable operating points.

The second sliding surface is defined from the exhaust manifold pressure error such as equation (19), the differential equation of the second sliding surface can be developed by the exhaust manifold pressure dynamics model uncertainty such as equation (20).

$$s_2 = p_x - p_{xd} \quad (19)$$

$$\begin{aligned} \dot{s}_2 &= \dot{p}_x - \dot{p}_{xd} = k_2 \left[ \frac{\eta_v N V_d}{120 R T_i} p_i + W_f - W_{EGR} \right] \dots \\ &\quad - k_2 W_{VTG} - \dot{p}_{xd} \\ &= f_{2m} + \Delta f_2 + k_2 u_1 - \dot{p}_{xd} \end{aligned} \quad (20)$$

Nonlinear equation  $k_2 \left[ \frac{\eta_v N V_d}{120 R T_i} p_i + W_f - W_{EGR} \right]$  can be

expressed by  $f_{2m} + \Delta f_2$ . Single input, VTG flow, can be developed from the desired exhaust manifold pressure of equation (18) and sliding surface error differential equation (20), EGR flow is developed from the PID control according to  $\phi_g$  values. If  $\dot{s}_2 s_2 \leq -\eta s_2^2 / \phi_2$  is satisfied and model uncertainty in equation (20) can be guaranteed by assumption with  $\Delta f_2 \equiv \frac{k_1 s_1}{\phi_1}$ . Finally, we can calculate the input value such as

$$u_1 = \frac{a_2 \frac{\eta_v N V_d}{120 R T_i} p_i + a_2 W_f - a_2 W_{EGRd} - \dot{p}_{xd} - \frac{k_2 s_2}{\phi_2}}{a_2} \quad (21)$$

$$\text{where } \dot{p}_{xd} \equiv \frac{p_{xd}(n) - p_{xd}}{\Delta t} \quad (22)$$

VTG flow input value in equation (21) can guarantees high robustness to parametric uncertainties by substituting the input value and fictitious input value to the  $\dot{s}_2$  and  $\dot{s}_1$  respectively such as equation (23) and (24).

$$\dot{s}_2 = -\frac{k_2 s_2}{\phi_2} + \Delta f_2 \quad (23)$$

$$\dot{s}_1 = f_{1m}(p_i, p_x) - f_{1m}(p_i, p_{xd}) + \Delta f_1 - \frac{k_1 s_1}{\phi_1} \quad (24)$$

For a stability analysis, Lyapunov function candidate is suggested such as

$$V = \frac{1}{2} s_1^2 + \frac{1}{2} s_2^2 \quad (25)$$

Lyapunov function V satisfies the positive definite with  $s_1, s_2 \in \beta(r)$ , where  $\beta(r)$  is bounded continuous function from the engine operating. And, differential Lyapunov function V can be developed such as equation (26) by using the single input value of equation (21) and fictitious input value of equation (18). This analysis is verified by the backstepping method such as a recursive procedure combining Lyapunov stability theory in systematic approach of a variety of papers[17][18].

$$\dot{V} = -\frac{k_1 s_1^2}{\phi_1} - \frac{k_2 s_2^2}{\phi_2} \quad (26)$$

The overall block diagram of the closed-loop system is shown in figure 10.

In case of 3<sup>rd</sup> order engine model based controller, the desired exhaust manifold pressure calculation process has to be needed. The turbine model data with equation (10) is urgently needed in such a process. However, in simplified model based controller, desired exhaust manifold pressure process is simplified by transforming the intake pressure differential equation which is possible to prove convergence of error. This main contribution is expressed in desired value calculator block of figure 10. Next chapter, we will verify the simplified sliding mode controller in NRTC mode, compare and analyze with 3<sup>rd</sup> order diesel model based controller's tracking performance.

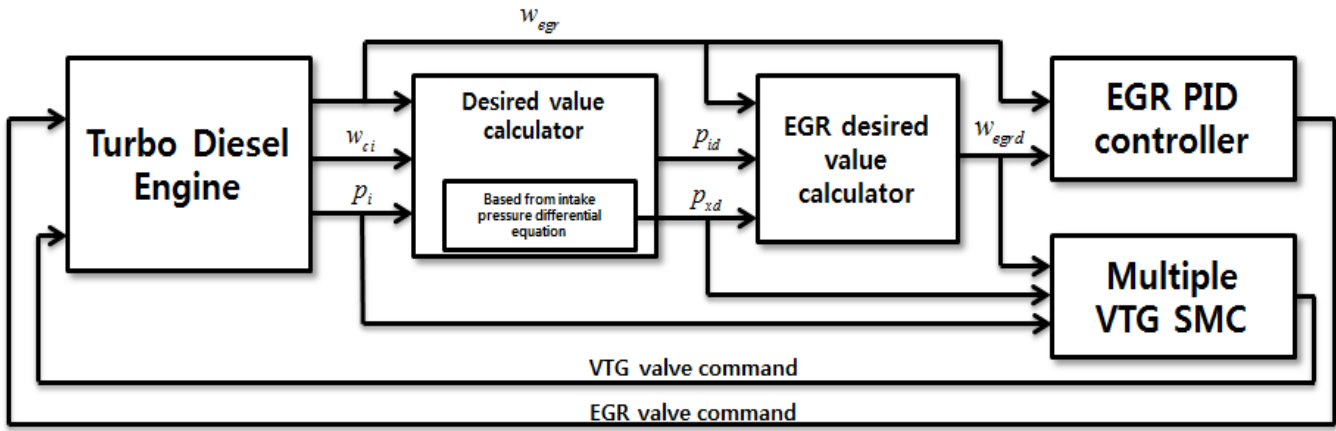


Fig.10. Block diagram of simplified SMC for turbo diesel engine air management

## CONTROLLER VERIFICATION

To verify the simplified sliding mode controller, the NRTC mode which is used in diesel engine model verification is suggested. Especially, to examine the tracking performance of controller in detail, only early 200 seconds of total NRTC mode are adopted.

The intake and exhaust manifold control results which are main target states are as follows figure 11 and 12.

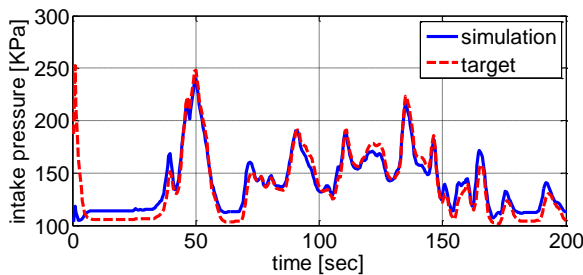


Figure 11. Simplified SMC simulation results in intake pressure

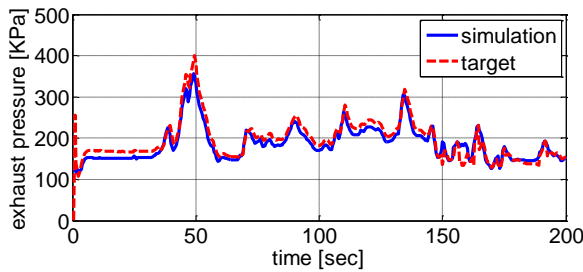


Figure 12. Simplified SMC simulation results in exhaust pressure

Total simulation operating points are 2000 as 0.1 second of sampling time. Maximum error of intake manifold pressure is about 10.4 % in row load and rpm operating points, and is about 6.0% in mid-high load and rpm operating points respectively. Similarly, maximum error of exhaust manifold pressure is about 8.9% in row load and rpm operating points, and is about 7.6% in mid-high load and rpm operating points respectively. There are about 10% errors in lower 120KPa of intake manifold pressure. Because the compressor flow tracking errors occur as follows figure 13.

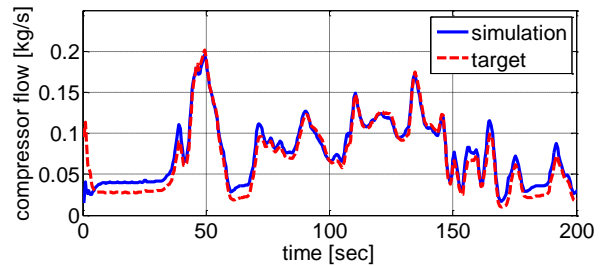


Figure 13. Simplified SMC simulation results in compressor flow

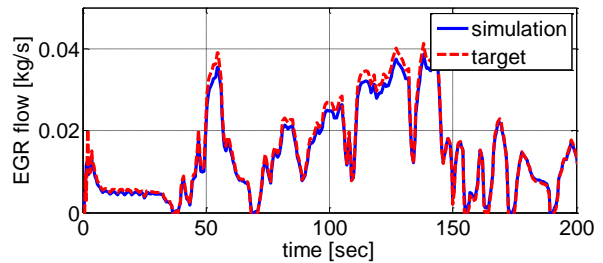


Figure 14. Simplified SMC simulation results in EGR flow

In compressor flow, the transient error of lower 0.05kg/s is about 0.01kg/s. This error is from model uncertainty of turbine model coupling effect between turbocharger and compressor.

Especially, model uncertainty of VTG valve effective area and turbine efficiency values in transient state transient have to be modified by a trial and error method or systematic approach in lower load operating points. Outside those problems, there is no problem in tracking the target values with high load and speed turbine operating points which show 5.4 % maximum error. Moreover, the results of EGR valve control with PID are shown as follows figure 14.

The EGR flow target tracking performance is comparatively higher comparing multivariable SMC's as showing in figure 18. Especially, in lower load and RPM operating points, there are little errors. However, transient error occurs in high load and RPM compared with lower load errors. This phenomenon is from the physical engine characteristics that turbine flow influences to the engine in high load and speed and EGR flow influences to the engine in low load and speed respectively.

Meanwhile, to analysis 3<sup>rd</sup> order engine model based controller with MIMO system, we suggest Devish's control design in multivariable sliding mode control development[16]. Generally, it is needed to design regular form for using the sliding mode control. However, 3<sup>rd</sup> order model does not satisfy the integrability criterion, alternate approaches, input-output linearization, can be adopted for designing the sliding surface. More specific design processing and analysis are represented in [16].

Multivariable sliding mode control with output set of intake pressure and exhaust manifold pressure shows the simulation results under the same conditions as follows figure 15 and 16.

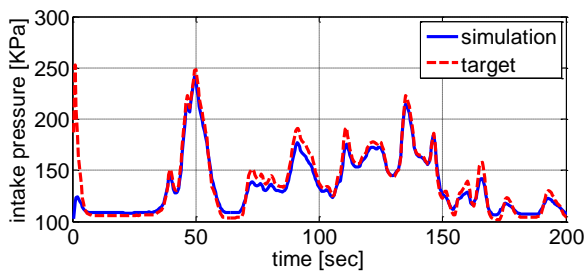


Figure15. Multivariable SMC simulation results in intake pressure

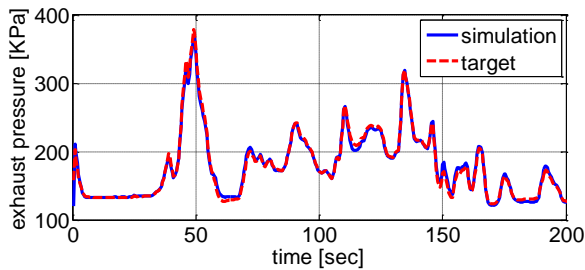


Figure 16. Multivariable SMC simulation results in exhaust pressure

In case of Intake pressure and exhaust pressure simulation results, maximum errors are 7.5percent, 4.2percent respectively. Moreover, mean errors are 4.4percent, 2.0percent respectively. These results present that multivariable sliding mode controller is greater than simplified sliding mode controller by reason of direct feedback output state control. However, indirect feedback state or indirect control target parameters, flows simulation results are shown as follows.

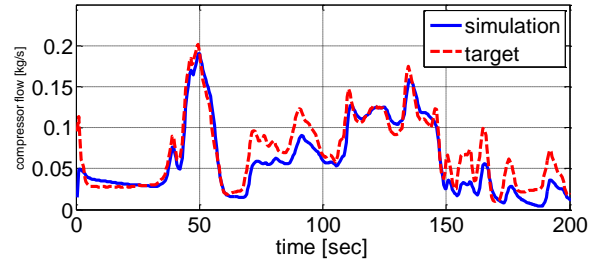


Figure 17. Multivariable SMC simulation results in compressor flow

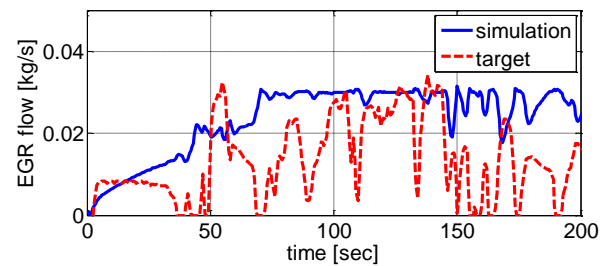


Figure 18. Multivariable SMC simulation results in EGR flow

According to coupling effect between VTG flow and EGR flow, before VTG flow actuator is operated in low load operating points EGR flow actuator is already activated. Moreover, VTG gas is more flow in some operating points. These erroneous valve commands raise the wrong gas fraction in engine cylinder. So, we need to design the some systematic solutions such as the flow saturation method in EGR flow actuator or adaptation method in estimating parameters related to the compressor flow. However, these solutions require much effort to design the filter or precondition of adaptation. In addition, multivariable sliding mode controller designed from the 3<sup>rd</sup> order engine model most probably make more errors in calculating desired exhaust manifold pressure. This target value is calculated from the last target states block as follows figure 19. However, in simplified sliding mode controller, desired exhaust pressure can be calculated from only target flow rates block from which we can skip the target states block and can reduce the errors. Moreover, we can optimize the total engine operating system process which is the most important factor in this paper. The table 1 shows that simulation results summary data between simplified multiple sliding surface design and multivariable sliding surface respectively.



	Control target	Control models	Intake pressure	Exhaust pressure	Compressor flow	EGR flow
Simplified multiple SMC	EGR flow Intake pressure (one sliding surface)	Intake pressure Exhaust pressure	10.4	8.9	10.5	5.4
Multivariable SMC	Exhaust pressure Intake pressure (two sliding surfaces)	Intake pressure Exhaust pressure Compressor power	7.5	4.2	15.8	X

\* Max. errors of NRTC 200sec. standards

Table1. Simulation results between simplified multiple SMC and Multivariable SMC

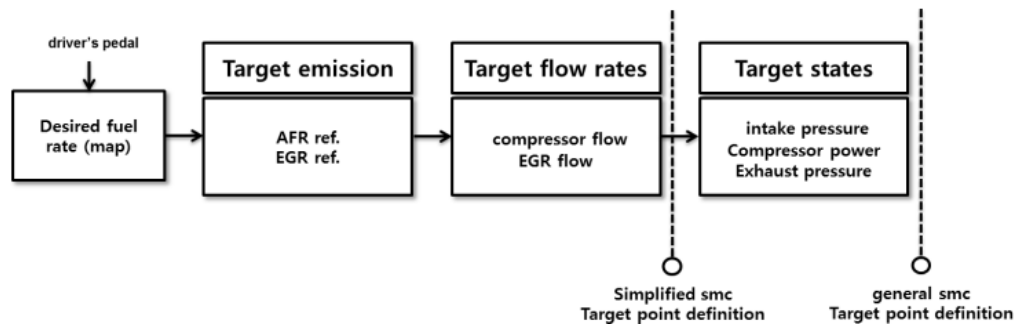


Figure19. Simplified control target generation process

## CONCLUSIONS

This paper deals with diesel engine control design of 1stage VTG and HP-EGR based. Especially, we make to effort to realize the problem with model based multivariable sliding mode controller, and to implement the ECU more efficiently. This paper's contributions are as follows.

At first, to reduce the model errors in calculating desired value, desired exhaust manifold pressure can be calculated through the data of desired EGR flow value and EGR valve actuator opening angle. From this simple process, gas fraction in cylinders and intake manifold pressure can be tracked to the desired values respectively.

At second, by interpreting 3<sup>rd</sup> order engine model single input strict feedback form as an underactuated system, control input is simpler and more intuitive. And, it is easy to design control tuning gains comparing to the multivariable sliding mode controller's. Moreover, according to the chain rule of underactuated system, intake and exhaust manifold pressure which are multiple sliding surfaces, can be tracked to the desired values simultaneously.

Therefore, the simplified sliding mode controller is verified in NRTC mode, main parameters and states of diesel engine,

pressure and flow have the lower 10 percent errors comparing multivariable sliding mode controller's performance with intake and exhaust manifold pressure output set. Especially, EGR flow tracking performance can be upgraded and gas fraction in engine cylinder can be satisfied.

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## NOMENCLATURE

EGR Exhaust gas recirculation  
 VTG Variable geometry turbocharger  
 MAF Manifold absolute pressure sensor

MAP Mass air flow sensor  
 EXMAP Exhaust manifold absolute pressure sensor  
 $AFR$  Air to fuel ratio, [-]  
 $A_{EGR}$  EGR valve effective area, [ $m^2$ ]  
 $F_i$  Air fraction in intake manifold, [-]  
 $F_x$  Air fraction in exhaust manifold, [-]  
 $F_e$  Air fraction in cylinder, [-]  
 $m_i, m_1$  Intake manifold mass, [kg]  
 $m_x, m_2$  Exhaust manifold mass, [kg]  
 $N_e$  Engine speed, [RPM]  
 $N_{tc}$  Turbocharger rotational speed, [RPM]  
 $p_a$  Ambient pressure, [kpa]  
 $p_i, p_1$  Intake manifold pressure, [kpa]  
 $p_x, p_2$  Exhaust manifold pressure, [kpa]  
 $P_c$  Compressor power, [KW]  
 $P_t$  Turbine Power, [KW]  
 $R$  Specific gas constant, [KJ/kg · K]  
 $T_a$  Ambient temperature, [K]  
 $T_i$  Intake manifold temperature, [K]  
 $T_x$  Exhaust manifold temperature, [K]  
 $T_e$  Engine out temperature, [K]  
 $\tau_{tc}$  Turbine to compressor power transfer time constant (Turbocharger time constant), [s]  
 $V_i$  Volume of intake manifold, [ $m^3$ ]  
 $V_x$  Volume of exhaust manifold, [ $m^3$ ]  
 $V_d$  Total displacement volume, [ $m^3$ ]  
 $W_{ie}$  Mass flow rate from intake manifold to cylinder, [kg/s]  
 $W_{EGR}$  EGR mass flow rate, [kg/s]  
 $W_{VTG}$  Turbine mass flow rate, [kg/s]  
 $W_{ci}$  Mass flow rate from compressor to intake manifold, [kg/s]

$W_f$	Injected fuel mass flow into cylinder, [kg/s]	$\phi_g$	EGR reference, [-]
$W_x$	Mass flow rate from cylinder to exhaust manifold, [kg/s]	$\eta$	Sliding gain, [-]
$\gamma$	Specific heat ratio ( $c_p / c_v$ ), [-]	$\phi_1, \phi_2$	Model uncertainties bounded gains, [-]
$\eta_m$	Turbocharger mechanical efficiency, [-]		
$\eta_c$	Compressor efficiency, [-]		
$\eta_t$	Turbine efficiency, [-]		
$\eta_{vol}$	Volumetric efficiency, [-]		