Model-Based Control System for Air Path and Premixed Combustion of Diesel Engine

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Abstract: Premixed charge compression ignition (PCCI) combustion has a potential to achieve the low emission level while the thermal efficiency remains high. It requires more precise combustion control due to the low robustness of the premixed combustion. In this study, a model-based control system for both air path system and combustion is developed for realizing the ideal PCCI combustion in future. For the air path system control, a feedback (FB) controller based on H_∞ control theory is used. In this controller, the nonlinearity of the plant is considered as the perturbation and robustness is ensured by H_∞ control. For the combustion control, a feedforward (FF) controller is designed as an inverse model of a discretized combustion model. In this controller, the inverse model is updated cycle by cycle on-board to consider the variation of combustion characteristics depending on operation condition. The performance of the overall control system including both controllers is evaluated by the reference following test under the transient mode operation in both the simulation and the experiment for PCCI-like combustion with advanced fuel injection timing. The control system showed good tracking performance both in air path and combustion systems and the availability of the model-based control system is shown.

Keywords: Diesel engine, Model-based control, Premixed combustion, Air path system.

1. INTRODUCTION

In diesel engines, to achieve low emission and high thermal efficiency a lot of devices and combustion technologies has been introduced, and then the engine system has become more complicated. Current engine control is conducted based on lookup tables constructed by a lot of experiments. The cost for preparing lookup tables has been increasing because of the complication of the engine system. In diesel engines, premixed charge compression ignition (PCCI) combustion gets attention as a next-generation combustion technology to achieve low emission and high thermal efficiency. PCCI combustion is the rapid combustion of the fuel-air mixture, which is achieved by the high exhaust gas recirculation (EGR) ratio and the advanced fuel injection timing (Okude et al., 2004). However, it is not robust and difficult to control by the conventional control system with lookup tables. With such backgrounds, the model-based control for engines is attracting attention as a new methodology. In this method, the engine model is installed to an engine control unit (ECU) and actuator control inputs are determined by the real-time calculation result of the model instead of the lookup tables.

Engine system mainly consists of two systems; combustion systems and air path systems, and model-based control methods for each system have been studied. As for the air path control, model-based controller design methods show a promising alternative to conventional control (Guzzella and Amstutz, 1998; Guzzela and H.Onder, 2010). Among them, a simplified physical model considering only the delay of the Turbo charger was developed and FF controller was designed by an inverse model (Hirata et al., 2017). In addition, MIMO FB controller based on H_∞ control theory was constructed by regarding the nonlinearity of the plant as the perturbation, and its performance was evaluated by control tests in GT-Power simulation and real engine bench experiment (Koizumi et al., 2017, 2018). As for the combustion control, there are a lot of previous researches of model-based control. The conventional diesel combustion is controlled by using physical and statistical model and numerical optimization to minimize emissions while providing the desired engine torque (Makowicki et al., 2017). The PCCI combustion is controlled by the predictive control with a neural network model (Drews et al., 2010). Moreover model-based control for both the air path and combustion system is also studied (Karlsson et al., 2010).

In most of the previous researches, the statistical methods or model identification method to build the models are mainly used. However these are effective only for the specific engine and calibrated operation region and it is not easy to be applied to other type engines and extrapolated region. From the viewpoint of the generality, it is desirable that the control-oriented model is based on the physics as possible. In our previous research, control-oriented combustion models, which is mainly composed of equations of physics and chemical reactions has been constructed (Yasuda et al., 2016; Yamasaki et al., 2017a). The model is applicable to multiple stage fuel injections and calculation load is very reasonable for on-board use. Feed-forward (FF) control system using the model on-

Table 1 Engine specification

Engine type	Inline 4 DI diesel
Bore × Stroke [mm]	92 × 103.6
Displacement [cm^3]	2754
Compression ratio	15.6
Combustion system	Direct injection
Injection system	Common rail
Air charging system	Inter-cooled turbocharger
Intake valve open / close	6 / 151 deg.BTDC
Exhaust valve open / close	132 / 10 deg.ATDC

Fig.1 Engine system

Fig. 2 Discretization of the model

Table 2 The summery of discretized points

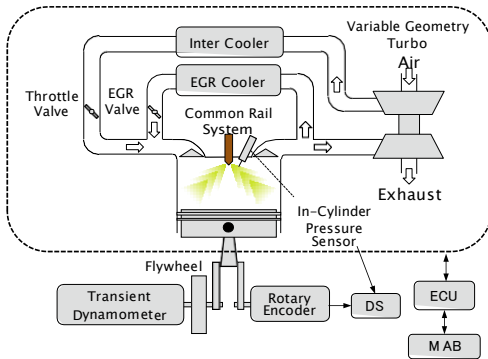| Discrete point	Definition
IVC	Intake valve closing
PILOT INJ	Pilot injection
PRE INJ	Pre injection
PRE IGN	Pre ignition
MAIN INJ	Main injection
MAIN IGN	Main ignition
PEAK	Pressure peak
EVO	Exhaust valve opening
IVO	Intake valve opening
EVC	Exhaust valve closing

board calculation has been developed (Yamasaki et al., 2017b). Multi-inputs and multi-outputs (MIMO) FF control system has been developed and its performance was evaluated under the transient operation test in a real engine bench. In these previous studies, the air path condition was controlled by lookup tables of a default ECU or with the valve profile fixed at the constant. In this paper the model-based control system for both air path and combustion is constructed by combining the model-based FB air path controller and the FF combustion controller. Its performance is evaluated by the control test. The control test is conducted by both simulation and experiment under the transient operation.

2. EXPERIMENTAL SETUP

The specification of a diesel engine for modelling and control tests are shown in Table 1, and the overall engine system including the measurement and control system is shown in Fig.1. The engine type is an inline 4-cylinder diesel engine and the displacement is about 2.8 liter. This engine has a common-rail injection system, a variable geometry turbocharger (VGT), and an external exhaust gas recirculation (EGR) system. Measurement and control of the injection condition and the air path condition are conducted by a rapid prototyping system (MicroAutoBoxII, dSPACE) with a default ECU. An In-cylinder pressure sensor and a rotary encoder are attached for analyzing combustion states. The pressure signals are recorded by a combustion analyzer (DS3000, Ono sokki) with the pulse of the rotary encoder.

3.CONTROL-ORIENTED COMBUSTION MODEL

In this study, simplified combustion model and air path model are used for the design of the controller and used as the plant models of the contr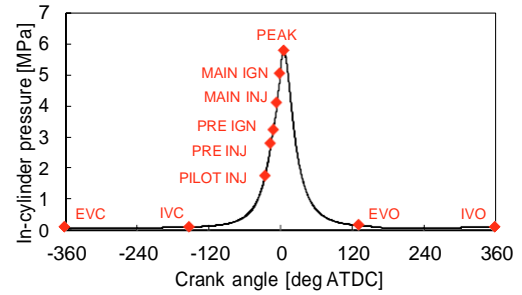ol test simulation. In this chapter our originally developed combustion model is described, and in the section 4.2 the design of the FF combustion controller. Regarding the air path model and FB controller, they are described in the section 4.3.

Model Concept

Low calculation load model is required for the on-board combustion control, and a combustion control model, which is based on the cycle discretization concept proposed by Ravi (2006), has been developed for the diesel engine by the authors. In this study, the model constructed in the previous research (Yamasaki et al., 2017a) is employed with a minor modification in the prediction of pressure peak timing. The essential points of the models are demonstrated in this chapter. An engine cycle is discretized into 10 representative points in this study as shown in Fig. 2. The meaning of them are shown in Table 2. The discretized points are valve opening/closing timing, each injection timing, each ignition timing, and in-cylinder pressure peak timing. In this model, it assumes that the pilot injection is not ignited by itself and ignited with the pre injection. This is because the role of pilot injection is forming the well-mixed fuel-air mixture and heat release is also not found from pilot injection to pre injection in experiments.

The gas state of each point is predicted by equations on the basis of physics and chemical kinetics, and minimum required statistical equations. This is because the model is easy to be applied to other engines when the model consists of versatile equations as possible.

Inputs and output of the combustion model

The inputs and outputs of the model are shown in Table 3. The inputs are the fuel injection condition, air path condition, and engine speed. All of these values can be obtained from ECU,

Table 3 Inputs and outputs of the discretized combustion model

| Inputs from ECU	
$N_{enggiine}$	Engine speed [rpm]
P_{raiil}	Fuel injection pressure [MPa]
QQ_{total}	Total injection quantity [mm^3]
QQ_{piilot}	Pilot injection quantity [mm^3]
$\theta\theta_{PILPPT\ INJ}$	Pilot injection timing [deg.ATDC]
QQ_{pre}	Pre injection quantity [mm^3]
$\theta\theta_{PRE\ INJ}$	Pre injection timing [deg.ATDC]
QQ_{maiin}	Main injection quantity [mm^3]
$\theta\theta_{MAIN\ INJ}$	Main injection timing [deg.ATDC]
P_{boost}	Boost pressure [kPa]
r_{EGR}	External EGR ratio [-]
$T_{iinmanii}$	Intake manifold temperature [K]
Inputs from previous cycle	
QQ_{prev}	Previous total injection quantity [mm^3]
$n_{x,RG}$	Mole of each gas in residual gas [mol]
Prediction outputs	
$\theta\theta_{PEAK}$	Peak pressure timing [deg.ATDC]
P_{PEAK}	Peak pressure [MPa]
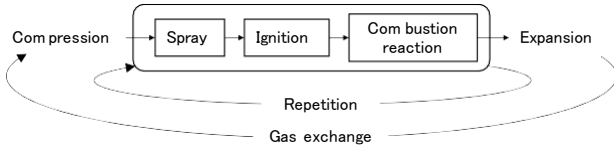
Fig. 3 Flow of the calculation

The temperature and the gas composition of the residual gas calculated by the model are also used as inputs because the residual gas state has the effects on next cycle combustion.

Model outputs are pressure peak timing and pressure peak value. These values are related to the engine efficiency and the combustion noise and determined as the control targets in the control test.

Prediction of ignition and combustion

The calculation flow is shown in Fig. 3, at first, the mixture gas composed of intake air, residual gas, and EGR gas is compressed, this process is described as the polytropic change. Next, the calculation step goes to the ignition and combustion process, where fuel spray formation, ignition of premixed gas and combustion reaction are considered. And then, calculation step goes to the expansion process, this is also described as the polytropic change. Finally, gas state at the end of the cycle is taken over as the residual gas to the next cycle, and the gas state at the start of next cycle is predicted in the gas exchange process.

Some representative equations of the ignition and combustion process are introduced in this section. The combustion state with the multiple fuel injection is described by the repetition of the 3 sub-models calculation; spray model, ignition model, and chemical reaction model.

In the spray model, the spray shape is calculated as (1) and (2) based on the model by Reitz and Bracco, (1979).

$$L_{Spray} = 2.95 \left(\frac{\Delta P}{\rho_{Fuel}}\right)^{0.25} \sqrt{d_{Hole} t_{spray}} \tag{1}$$

$$\tan(\varphi_{Spray}) = \{3.0 + 0.28 \left(\frac{L_{zzle}}{d_{Hole}}\right)\}^{-1} 4\pi\sqrt{\frac{\rho_{Gas}}{\rho_{Fuel}}} \frac{\sqrt{3}}{6} \tag{2}$$

where, L_{Spray} is the fuel penetration distance, φ_{Spray} is the spray angle, ΔP is the gap between fuel injection pressure and in-cylinder gas pressure, ρ_{Fuel} and ρ_{Gas} are the density of fuel and in-cylinder gas, d_{Hole} is the nozzle hole diameter, t_{spray} is the injection duration and L_{Nozzle} is the injector nozzle length. The spray volume can be obtained from L_{Spray} and φ_{Spray}, and the fuel concentration calculated from the spray shape and amount of injected fuel is used in the ignition model and the chemical reaction model.

In the ignition model, the ignition delay time is calculated by simplified Livengood and Wu integration shown as (3). (Livengood and Wu, 1955).

$$K = A[Fuel]_{*iinii}[O_2]_{*iinii}^C \exp\left(-\frac{E}{RT_{*iinii}}\right) \Delta t_{delay}$$

where $K, A, B, C,$ and E are calibration parameters and calibrated by about 160 experimental results in steady states, $[X]$ indicates the concentration of X, R is the molar gas constant, Δt_{delay} is the ignition delay time, and the subscript $*iinii$ is the value of each injection timing.

In the chemical reaction model to express the combustion, the fuel consumption rate is expressed by the Arrhenius equation (4) for assuming the premixed combustion in this study,

$$-[Fuel]_{ii\sim ii+1} = \alpha[Fuel]_{ii}^\beta[O_2]_{ii}^{\gamma\gamma} \exp\left(-\frac{\varepsilon\varepsilon}{RT_{ii}}\right) \Delta t_{ii\sim ii+1}$$

where, $\alpha, \beta, \gamma\gamma,$ and $\varepsilon\varepsilon$ are also calibration parameter. $[Fuel]_{ii\sim ii+1}$ and $\Delta t_{ii\sim ii+1}$ indicate the fuel consumption and the duration from i-th discretized point to $i+1$-th point. The amount of burned fuel is calculated by these equations. Then the gas state of each discretized point is calculated by the law of energy conservation (5),

$$n_{ggas,ii} C_{v,ii}(T_{ii} - T_{rerr}) + WW_{ii\sim ii+1} + dQQ_{ii\sim ii+1} = n_{ggas,+1} C_{v,ii+1}(T_{ii+1} - T_{rerr})$$

where, $n_{ggas,ii}$ indicates the amount of substance at i-th discretized point, C_v molar heat capacity at constant volume, WW indicated work, dQQ heat release, and T_{rerr} the reference temperature.

In this model, peak pressure timing is predicted by a statistic model (6), where the A, B, C, D and E are the also calibration parameter.

$$t_{MAIN\ IGN\sim PEAK} = A + B\Delta t_{delay} + CQQ_{maiin} + DN_{enggiine} + ET_{MAIN\ IGN}$$
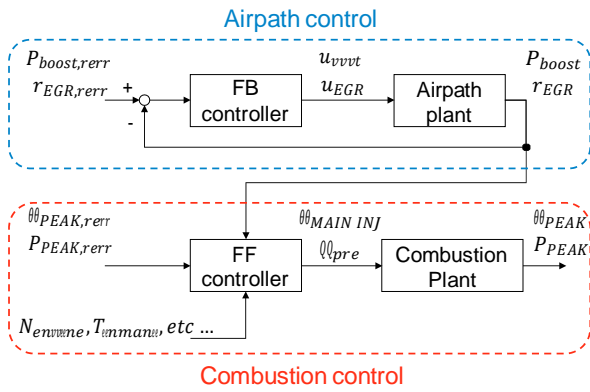
Fig. 4 Flow of the signals in the overall control system

Table 4 Inputs and targets value of each control system

	Targets	Control inputs
Air path	Boost pressure [kPa] External EGR ratio [-]	VGT valve [% closed] EGR valve [% open]
Comb.	Pressure peak timing [deg. ATDC] Pressure peak [MPa]	Main injection timing [deg. ATDC] Pre injection quantity [mm^3/st]

4. ENGINE CONTROL SYSTEM DESIGN

Overall control system

An overall control system including the combustion control system and the air path control system is shown in Fig. 4 and inputs and targets of each controller are shown in Table 4. Control system in this study consists of FB air path controller and FF combustion controller. Firstly, the air path FB control is done. In the air path control, the control target variables are boost pressure (P_{boost}) and external EGR ratio (r_{EGR}), control inputs are the closing ratio of the VGT valve (u_{VGT}) and the opening ratio of the EGR valve (u_{EGR}). Next, combustion FF control is done by considering the real-time engine operation condition including control result of boost pressure and external EGR ratio by the air path control system. Here, the control targets are peak pressure timing (θ_{PEAK}) and peak pressure value (P_{PEAK}), and control inputs are main injection timing ($\theta_{MAIN\ INJ}$) and pre-injection quantity (Q_{pre}). θ_{PEAK} seems to have a correlation with the thermal efficiency and P_{PEAK} seems to have a correlation with combustion noise. These values are strongly coupled and could not necessarily be controlled to arbitrary values independently. In the combustion control, the total fuel injection quantity is fixed at the at first determined value, then the main injection quantity is changed along with the change of pre-injection quantity. The detailed design of each controller is demonstrated in the following sections.FF combustion controller

FF combustion controller is designed by the same method as the previous research (Yamasaki et al., 2017b) with discretized
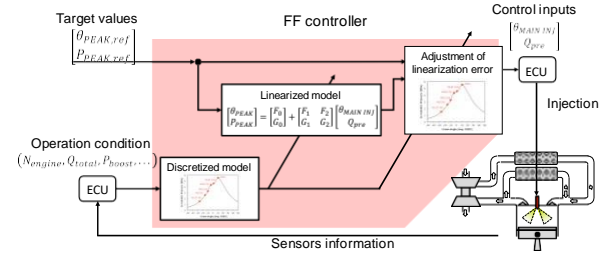

Fig. 5 FF combustion control system

combustion model described in chapter 3. The outputs of the model are the pressure peak and its timing, and these variables are also control targets. Then the inverse model can be used as the FF controller. However, the structure of the model is very complicated and it is difficult to obtain the inverse model by the mathematical method such as Taylor expansion. Then, taking the advantage of the low calculation load of the model, the inverse model is obtained on-board by using the calculation results with perturbations of inputs. At first, a linearized model (7) is obtained, where F_k and G_k are determined by the multiple regression analysis of 9 sets of inputs and output. These parameters differ depending on the engine operation condition, and to consider the change of condition, this procedure is conducted cycle by cycle. Then, inverse model (8) is derived by deformation of (7).

$$\begin{bmatrix} \theta_{PEAK} \\ P_{PE} \end{bmatrix} = \begin{bmatrix} F_0 \\ G_0 \end{bmatrix} + \begin{bmatrix} F_1 & F_2 \\ G_1 & G_2 \end{bmatrix} \begin{bmatrix} \theta_{MAIN\ INJ} \\ Q_{pre} \end{bmatrix}$$

$$\begin{bmatrix} \theta_{MAIN\ INJ} \\ Q_{pre} \end{bmatrix} = \begin{bmatrix} F_1 & F_2 \\ G_1 & G_2 \end{bmatrix}^{-1} \begin{bmatrix} \theta_{PEAK,rr} - F_0 \\ P_{PEAK,rr} - G_0 \end{bmatrix}$$

In addition, in order to adjust linearization errors, the control inputs derived from (8) is delivered into the original discretized model and the control inputs are corrected in response to the amount of the error. The overview of the FF controller is shown in Fig. 5.

FB air path controller

In previous research (Koizumi et al., 2018), FB air path controller based on H_∞ control theory was designed. A nonlinear physical model was constructed, which was calibrated by 40 experimental data sets. Then, the nonlinear air path model was linearized at 40 operation points and 40 liner models were obtained, and this model set was used for H_∞ controller design. The H_∞ controller was designed to stabilize all the models in the model set. The frequency responses of the 40 linear models are examined and one of the models located near the center value was selected as a nominal model.

Characteristic change of the plant due to the nonlinearity is considered as the multiplicative perturbation to the nominal model. When a perturbed model $\tilde{P}$ is described by the nominal model P and perturbation Δ_m as (9), Δ_m can be estimated as (10) and they are shown in Fig. 6 as the dashed lines.
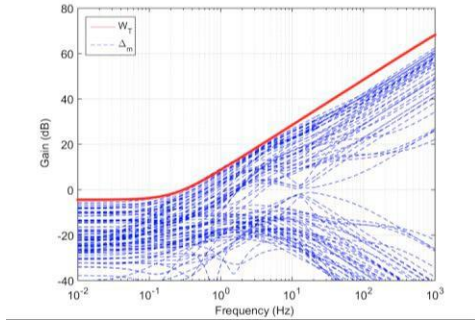
$$\tilde{P} = P(I + \Delta_m)$$

Fig. 6 Multiplicative perturbations (dashed line) and weighting function WW_T (solid line)
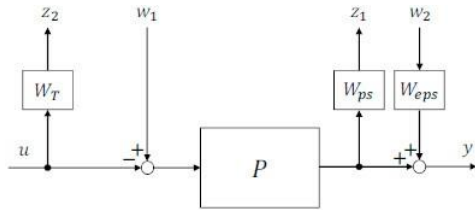

Fig. 7 Generalized plant for design of H_∞ controller.
(: nominal model, WW_T, WW_{ps}, WW_{eps} : weighting functions)
(Koizumi et al, 2018)

$$\Delta(jj\omega) = P^{-1}(jj\omega)(\tilde{P}(jj\omega) - P(jj\omega)) \qquad (10)$$

The generalized plant for the H_∞ air path controller design is shown in Fig. 7, which is the typical mixed sensitivity type problem. WW_T was selected to cover the multiplicative perturbation to achieve robustness as (11), which is shown in Fig. 6 as the solid line. The weighting function WW_{ps} for disturbance attenuation was selected through trial and error as (12), and WW_{eps} was selected as a high-pass type filter as (13).

$$W = \text{diag}\,[w_t\ w_t]$$

$$w_T = \frac{s + 1.414}{s + 2.356 \times 10^4} \times 10^4 \qquad (11)$$

$$WW = \text{diag}\,[1.2\ w_{ps}\ w_{ps}]$$

$$w_{ps} = \frac{1}{s + 0.001} \qquad (12)$$

$$WW_e = \text{diag}\,[w_{eps}\ w_{eps}]$$

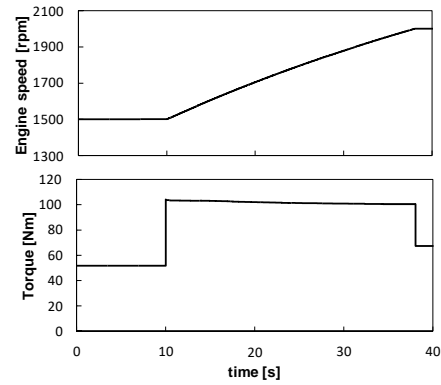$$w_{eps} = \frac{15s + 16.97}{s + 1.697 \times 10^4} \times 10^{-3} \qquad (13)$$

Fig. 8 Transient test mode in the control test

Table 5 Fixed fuel injection condition

Fuel injection pressure [MPa]	80
Pilot injection timing [deg. ATDC]	-25
Pilot injection quantity [mm^3/st]	2
Pre injection timing [deg. ATDC]	-15

are greatly affected by the air path control error, then in order to avoid entering the impossible combustion target values, they are corrected on-board in response to the air path condition. The fuel injection condition other control inputs are fixed at the constant value shown in Table 5.

Simulation

The performance of the control system is evaluated by the simulation before the control experiment. In the simulation, the physical air path model (Koizumi et al., 2018) is used as the air path plant model and discretized combustion model is used as the combustion plant model.

The simulation result is shown in Fig. 9, where upper 4 graphs indicate air path system results, from the top the graphs indicate boost pressure, external EGR ratio, VGT valve and EGR valve respectively, and bottom 4 indicate the combustion system results, from the top the graphs indicate pressure peak timing, pressure peak value, main injection timing and pre injection quantity. Regarding the air path control system, in the steady area and a few seconds after a step change of operation condition, boost pressure and the EGR ratio are adjusted to the target value. On the other hand, immediately after the step change, there are some delays, and EGR ratio moved opposite to the reference after the second step change at 38 s. Regarding the combustion control system, the graphs show that pressure peak timing and pressure peak value follow the target values under the existence of delay or tracking error in the air path

5. CONTROL TEST

In this section, the control performance of the control system with the combustion controller and the air path controller is evaluated by the transient mode control test in both simulation and experiment. The transient mode used in this study is shown in Fig. 8. In this study, the target following test is conducted. The air path target values are determined by the look-up tables of default ECU. Combustion target values are at first determined by hand in advance but combustion characteristics

system. In this simulation, the discretized combustion model is used as the plant model, which is also used for the design of the FF controller, then this simulation result indicates that theinverse model is obtained precisely.

Experiment

The control system is introduced to the test engine bench and its performance is evaluated. MicroAutoBoxII (dSPACE) was used as a rapid prototyping system with a default ECU.

The experimental results are shown in Fig. 10. Top 4 graphs indicate that boost pressure and EGR ratio follow the target values in the experiment, the air path target values after the step change of the injection quantity differ from the simulation ones. This is because the ECU has the function to compensate the transient air path target values depending on the operation condition such as the gas flow condition (Nakayama, 2003), and this is not equipped to the simulation. Then, the response of EGR value is also different from the simulation one. Regarding the combustion control, there is the block to correct the combustion target values depending on the air path value, and air path control result is somewhat different from the one in simulation, then the target values in the combustion control are not exactly the same as the simulation ones. In the steady area, from 0 s to 10 s, there is about 2 degrees fluctuation of peak pressure timing. This seems to be because that the low oxygen concentration due to the high EGR ratio makes the large cycle validation of the pre combustion. In the acceleration period, from 10 s to 38 s, pressure peak timing and value follow targets with little error. Immediately after second step change at 38 s, there is a big undershoot in the pressure peak timing, this seems to be caused by a drastic increase of EGR ratio to about 0.7. The error between the combustion targets and actual values in the experimental results are derived from the error of combustion model prediction. The model is not calibrated for the 0.7 EGR ratio operation range and the prediction error at this point is reflected in the target following performance.

6. CONCLUSIONS

In this study, model-based air path control system and combustion control system are combined and the performance of the control system is evaluated by both simulation and experiment. In the control test, as for the air path system, boost pressure and external EGR ratio follow the target though there are some error or delay, and as for combustion system, pressure peak timing and value can follow the target without

any delay although there is some tracking error due to the incomplete prediction accuracy of the model and the availability of the model-based control system is shown.

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