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Multi-Variable Air-Path Management for a Clean Diesel Engine Using Model Predictive Control

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ABSTRACT

Recently, emission regulations have been strict in many countries, and it is very difficult technical issue to reduce emissions of diesel cars.

In order to reduce the emissions, various combustion technologies such as Massive EGR, PCCI, Rich combustion, etc. have been researched. The combustion technologies require precise control of the states of in-cylinder gas (air mass flow, EGR rate etc.). However, a conventional controller such as PID controller could not provide sufficient control accuracy of the states of in-cylinder gas because the air-pass system controlled by an EGR valve, a throttle valve, a variable nozzle turbo, etc. is a multi-input, multi-output (MIMO) coupled system. Model predictive control (MPC) is well known as the advanced MIMO control method for industrial process. Generally, the sampling period of industrial process is rather long so there is enough time to carry out the optimization calculation for MPC. However, due to the progress of the computer in the last decade and improvement of the optimization algorithm, the MPC can be applied to 'fast' process such as mechanical systems.

The air-pass management system for the diesel engine is assumed a two-input, two-output system in this research. The inputs of the system are the throttle valve and the EGR valve, and the outputs air mass flow and EGR rate. Consequently, MPC was applied to air-pass management system and was modified by adding the disturbance observer to eliminate steady-state error and the compensator for nonlinear characteristics of actuators.

The performance of the proposed control system was examined by using an actual testing vehicle. From the experiments, it was shown that an accurate decoupled control of two outputs, i.e. air mass flow and EGR rate, was accomplished by the proposed MPC control.

INTRODUCTION

Diesel engines display higher thermal efficiency and emit less CO_2 than gasoline engines, and have therefore attracted attention as a form of internal combustion engine that is effective in combating global warming. However, diesel engines are basically lean combustion engines, and the simple after-treatment systems involving three-way catalysts with close to 100% reduction efficiency that are employed in gasoline engines cannot be employed in them. This makes it necessary to reduce the amount of NOx, HC, soot and other emissions that are found in the exhaust gas in the combustion process in the cylinders.

The reduction of NOx in diesel engines is particularly important, and exhaust gas recirculation (EGR) systems are well known as an effective means of doing so. The higher the maximum temperature of combustion gas is the more NOx increases. Consequently, EGR reduces NOx by mixing the inert gases present in the exhaust with the intake air, reducing the oxygen concentration, and thus reducing the maximum temperature of combustion. However, as EGR is steadily increased, there is a rapid increase in the volume of soot produced above a certain excess air factor. In order to effectively reduce NOx while controlling soot production, therefore, it is necessary to accurately control the state of

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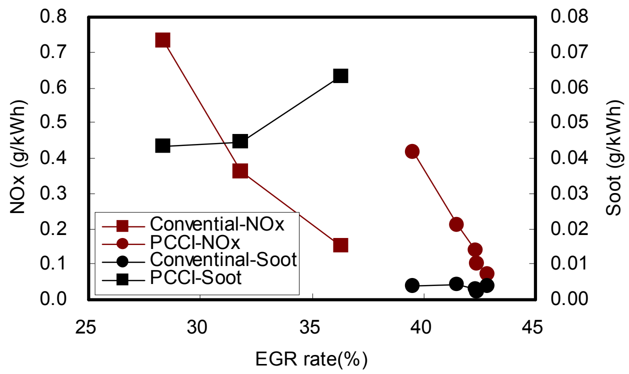

Figure 1 Emission characteristics of conventional and PCCI combustion

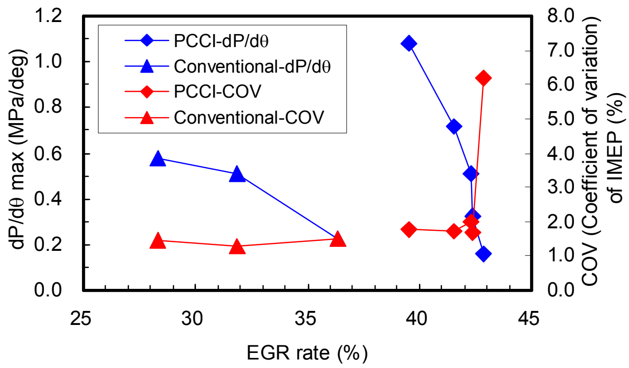

Figure 2 Noise and vibration characteristics of conventional and PCCI combustion

combustion (parameters such as the excess air factor, etc.) to within a desired range by controlling EGR and fuel injection.

PCCI combustion has been researched as a technology to reduce both NOx and soot emissions. PCCI combustion is premixed one achieved by increasing EGR amount and extending the ignition delay period. However, as indicated by the results of tests on steady-state PCCI combustion (Figures 1, 2), the state of the intake EGR produces significant variations in the characteristics of emissions. Therefore, the tracking response characteristics of fuel injection control and EGR control should be improved to achieve PCCI combustion during transient operating conditions.

There are, however, limits of further reductions in engine-out emissions. So, NOx exhaust after-treatment system, for example a NOx catalyst, is recently researching to enable further emissions reductions. The NOx catalyst system requires a regenerative combustion control known as rich control, which reduces the NOx that is adsorbed during lean combustion [1]. The rich control requires high-accuracy fuel injection control tracking air-fuel ratios on target values in order to keep combustion noise performance and drivability. Furthermore, high-accuracy intake EGR control is also important to realize this combustion control.

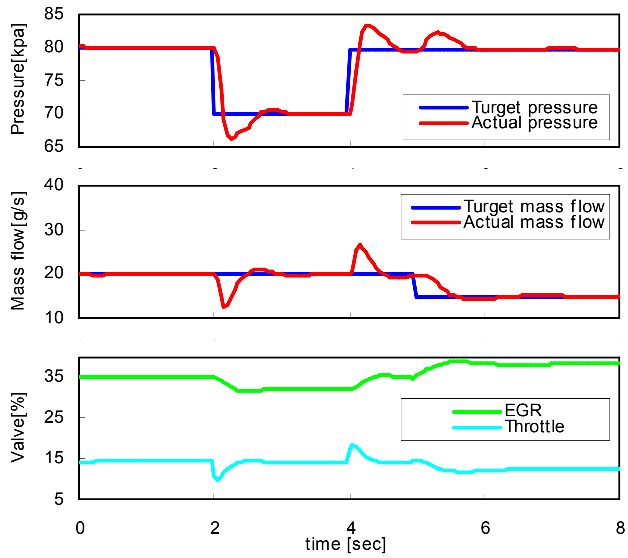

Figure 3 PI control simulation results

The various requirements discussed above indicate that the necessity for more precise control of the state of intake EGR is increasing.

1. DEVELOPMENT AIMS

The EGR valves, throttle valves, etc., are the intake actuators to directly operate the controlled object in intake control. The controlled outputs of the controlled object are oxygen and inert gases in the cylinder. The amounts or ratios meanings these two parameters also used as the controlled output.

However, in general, if one of the intake actuators is changed, both controlled outputs also change. The system indicates this behavior is known as a coupled system, and in such cases to design feedback control systems using conventional methods, such as PID controllers, is known to represent a challenge. Figure 3 shows oscillations caused by interference when a basic PID controller is employed in an engine simulator demonstrating behaviors of the controlled object discussed above.

Model predictive control (MPC), which enables multiple outputs to be controlled to target values, is known as a control algorithm that can be applied to multi-input, multi-output systems. Because MPC requires optimization calculations to be performed in real-time for each sampling period, it is mainly employed in applications in which the sampling period is long, such as in petrochemical plants.

However, increases in calculation speed with higher-speed processing of ECU and improvements in calculation algorithms [2] have increased the potential for the application of MPC to comparatively fast systems [3].

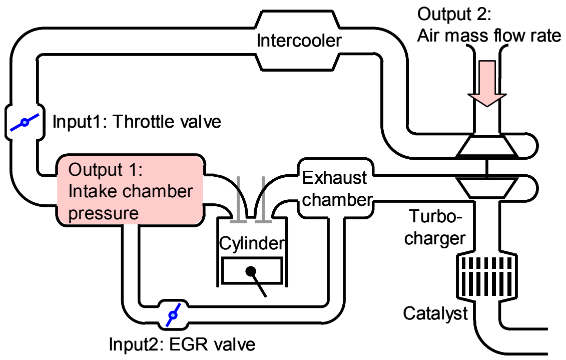

Figure 4 Air-path system of diesel engine and definition of the controlled object

The project discussed here applied MPC in the construction of an air mass flow/EGR feedback control system for a diesel engine. This paper will mention the basic configuration of the system and discuss the results of tests conducted to verify its effectiveness.

2. MODELING

Figure 4 shows the configuration of the intake and exhaust system of a diesel engine. The project discussed here assumed a two-input, two-output controlled object, with input and output parameters defined as follows:

- Inputs: EGR valve and throttle valve
- Outputs: Intake chamber pressure and mass flow rate of new air intake (air mass flow)

When employing MPC or any other model-based control theory such as optimal control or adaptive control, the dynamic characteristics of the controlled object must be modeled by using the simultaneous equations of differential or difference equations. In other word, the dynamic characteristics are expressed using state-space equations.

A model based on physical equations was therefore first constructed to enable the configuration of the controlled object to be understood, and the order of the state-space equations was estimated. Next, system identification was employed to determine the physical parameters in the state-space equations in order to obtain the state-space equations employed in the control theory incorporated in a real engine.

2.1. PHYSICAL MODELING - As Figure 4 shows, the intake and exhaust system of a diesel engine is made up of valve and chamber elements. Here, an example of modeling based on physical principles will be provided,

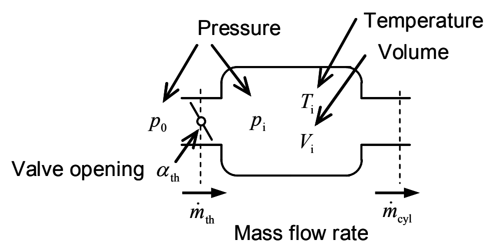

Figure 5 Element model of valve and chamber system

focusing on the valve and chamber in its wake shown in Figure 5.

The mass flow rate past the throttle valve is expressed using a convergent nozzle equation [4], [9].

$$\dot{m}_{th} = A_{th}(\alpha_{th}) \cdot \sqrt{2 p_0 \rho_0} \cdot \Phi$$
$$\Phi = \begin{cases} \sqrt{\dfrac{\kappa}{\kappa-1}\left\{\left(\dfrac{p_i}{p_0}\right)^{\frac{2}{\kappa}} - \left(\dfrac{p_i}{p_0}\right)^{\frac{\kappa+1}{\kappa}}\right\}} , & \left(\dfrac{p_i}{p_0}\right) > \left(\dfrac{2}{\kappa+1}\right)^{\frac{\kappa}{\kappa-1}} \\ \left(\dfrac{2}{\kappa+1}\right)^{\frac{\kappa}{\kappa-1}} \sqrt{\dfrac{\kappa}{\kappa+1}} , & \left(\dfrac{p_i}{p_0}\right) \le \left(\dfrac{2}{\kappa+1}\right)^{\frac{\kappa}{\kappa-1}} \end{cases} \quad (1)$$

where:

$\dot{m}_{th}$: Mass flow in throttle valve section, A_{th} : Effective opening area, α_{th} : Valve opening, p_0 : Upstream pressure, ρ_0 : Upstream density, κ : Specific heat ratio, p_i : Downstream pressure

From the equation of state for an ideal gas in the cylinder and the principle of conservation of mass [10]

$$p_i V_i = M_i R_i T_i \quad (2)$$

$$\frac{dM_i}{dt} = \dot{m}_{th} - \dot{m}_{cyl} \quad (3)$$

where:

$\dot{m}_{cyl}$: Mass flow in cylinder inflow part

Pressure variation in the intake chamber can be expressed by the following equation:

$$\frac{dp_i}{dt} = n\frac{RT_i}{V_i}\left(\dot{m}_{th} - \dot{m}_{cyl}\right) \quad (4)$$

where:

n : Polytrophic index

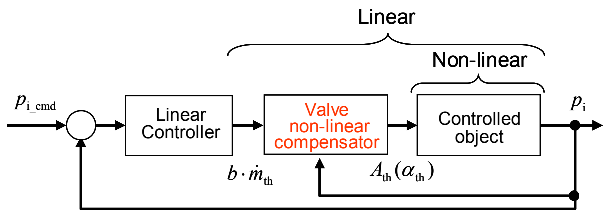

Figure 6 Control structure with non-linear compensator

In addition, because the cylinder intake mass flow rate, m_{cyl}, is expressed using a function almost exactly proportional to the chamber pressure, p_i, the following equation

$$\dot{m}_{cyl} = k_{\eta_v} n_e p_i \tag{5}$$

where:

n_e : Engine speed, k_{η_v} : Factor including volume efficiency

From equations (1), (4) and (5), enabling expression as a first-order lag system.

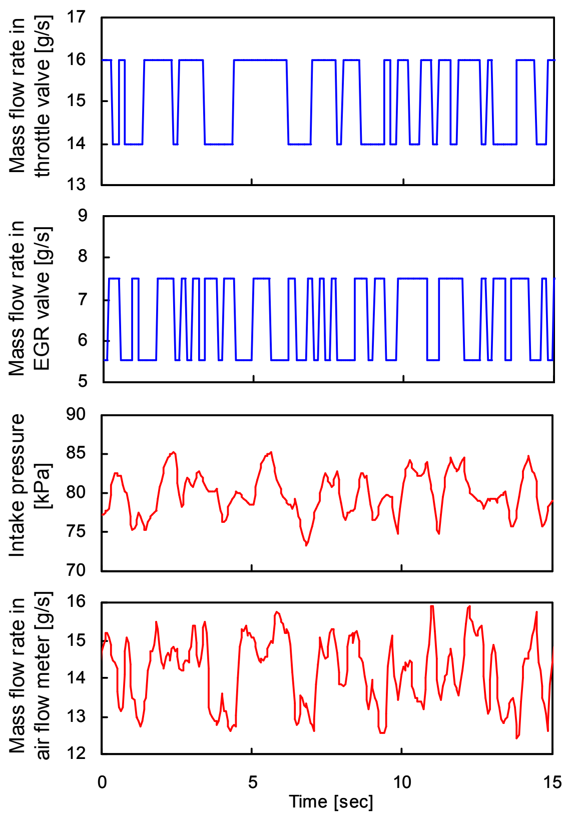

Figure 7 Input and output measurement data

$$\frac{dp_i}{dt} = a \cdot p_i + b \cdot A_{th}(\alpha_{th}) \tag{6}$$

$$\because \quad a = -n\frac{RT_i}{V_i} k_{\eta_v} n_e, \quad b = n\frac{RT_i}{V_i}\sqrt{2p_0\rho_0} \cdot \Phi$$

However, because both the lag parameter a and gain parameter b are expressed as nonlinear functions, the system is a nonlinear system in which dynamic behavior varies depending on the point of the system under consideration. A linear control theory incorporating MPC could not be applied to the nonlinear system.

Linearization of the system was therefore attempted using a method of linearizing input [4]. Figure 6 shows the control system formed by linearizing input. By positioning the linearization of input using convergent nozzle equation (1) in the controller output, the controlled object from the perspective of the controller can be regarded as linear. It is therefore possible to construct a linear controller with the equations above as the equations of state.

The controlled object in this paper discussed here incorporated element systems other than the example offered above, but they were all able to be expressed as dead time or first-order lag systems, as follows:

- Actuator : first-order lag system with dead time
- From position of air flow meter via chamber to throttle valve: first-order lag system.

Therefore, the controlled object system for this project is described by lag systems with dead time.

2.2. SYSTEM IDENTIFICATION - Using aforementioned techniques was possible to understand the characteristics of the controlled object, however the parameters incorporated in the controlled object model were made up of a large number of physical constants, making physical determination of their value a challenge. System identification [6] was therefore conducted in order to obtain the equations of state used in the control theory incorporated in a real engine based on the following input-output data.

- Input: Command value for mass flow in throttle valve section (m_{th_cmd} [g/s]) and Command value for mass flow in EGR section (m_{egr_cmd} [g/s])

- Output: Intake chamber pressure (p_{i_act} [kPa]) and air mass flow (m_{a_act} [g/s])

The method employed to measure input and output data for system identification was to apply pseudo random binary signal (PRBS) [7], and to measure the resulting

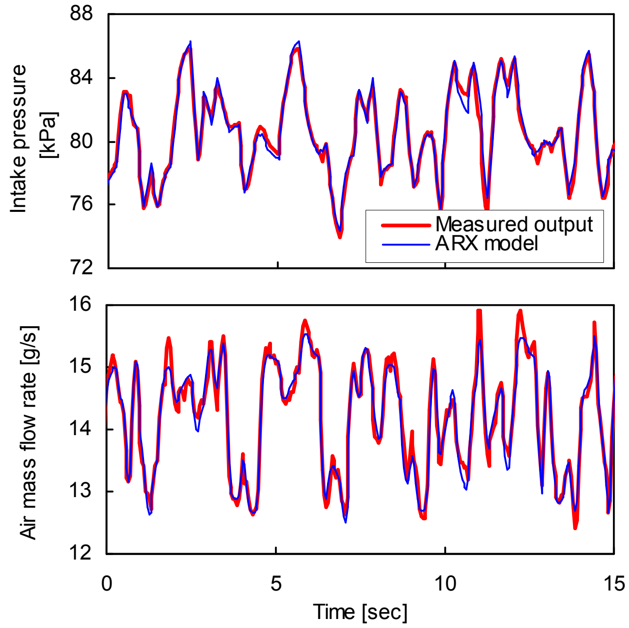

Figure 8 Time series result

two output values. Figure 7 shows the input and output data.

The dead time of the controlled object, T, was determined from the measured input and output time series data using a method of estimating impulse response [6].

Next, system identification was conducted using the measured input and output data, with the model assumed to be a two-input, two-output multivariable ARX model, expressed by the equations as follows.

$$
\begin{aligned}
&y(k+1)+A_1y(k)+A_2y(k-1)\\
&\quad =B_1u(k-T)+B_2u(k-T-1)
\end{aligned} \tag{7}
$$

where

$$
y(k)=\begin{bmatrix} y_1(k) \\ y_2(k) \end{bmatrix},\ u(k)=\begin{bmatrix} u_1(k) \\ u_2(k) \end{bmatrix} \tag{8}
$$

$$
\begin{aligned}
A_1&=\begin{bmatrix} a_{111} & a_{112} \\ a_{121} & a_{122} \end{bmatrix}, \quad A_2=\begin{bmatrix} a_{211} & a_{212} \\ a_{221} & a_{222} \end{bmatrix},\\
B_1&=\begin{bmatrix} b_{111} & b_{112} \\ b_{121} & b_{122} \end{bmatrix}, \quad B_2=\begin{bmatrix} b_{211} & b_{212} \\ b_{221} & a_{222} \end{bmatrix}
\end{aligned} \tag{9}
$$

In order to verify the accuracy of the identified two-input, two-output ARX model, the output of the model was compared with the output of an actual air mass flow for time series data (Figure 8) and step response (Figure 9). In results for time series data, a fit rate of 85.8% was obtained for air mass flow (output 1) and a fit rate of 80.27% was obtained for pressure (output 2), indicating that the ARX model displays excellent accuracy. This is believed to be due to the fact that an ideally-functioning nonlinear compensator enabled input and output data with a high S/N ratio to be obtained.

The frequency characteristics obtained from the ARX model and the spectrum actually measured using spectral analysis [6] are almost same over the entire frequency range shown in Figure 10, demonstrating that the ARX model displays excellent accuracy even in the frequency domain. The results of frequency analysis also indicated the following regarding this controlled object:

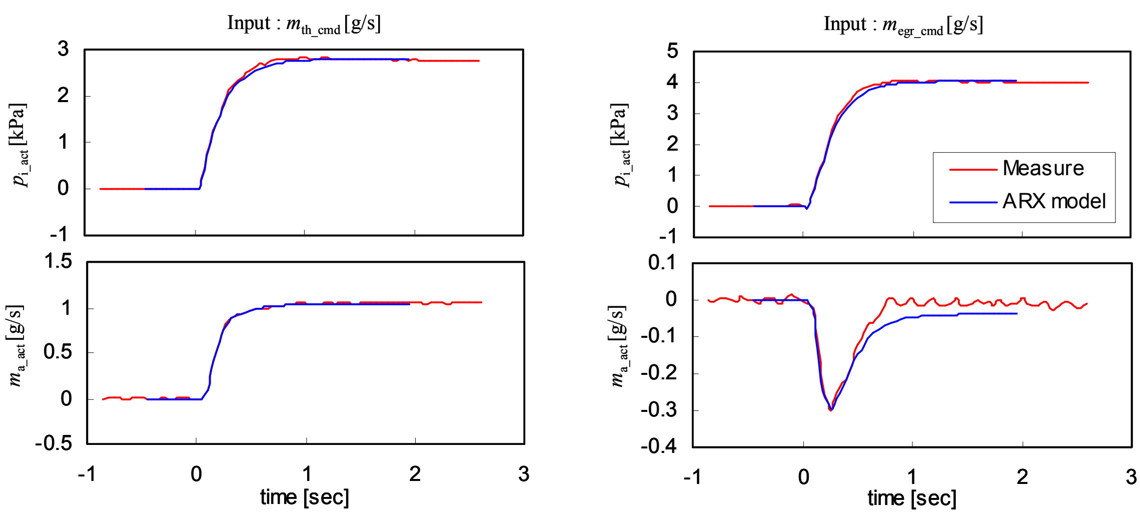

Figure 9 Step response

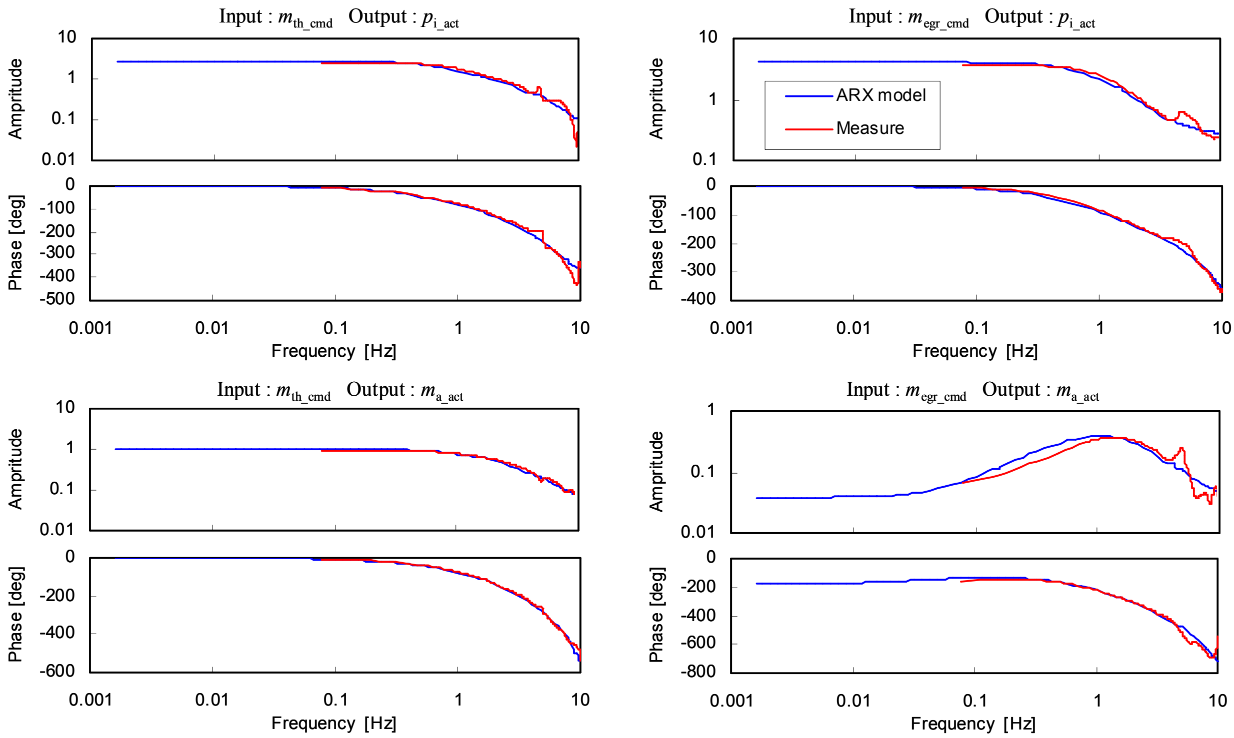

Figure 10 Bode diagram

- Phase lags as the frequency becomes higher, and the controlled object is therefore a lag system
- The inclination of gain in the high-frequency range other than the path of air mass flow from the EGR flow rate is -40 dB/dec, and the relative degree of the controlled object is therefore 2
- Because the relative degree of the controlled object is 2, the phase lag in the high-frequency range should be –90*2 = -180 deg, but higher phase delays also exist

These characteristics show that the controlled object is a non-minimum phase system with dead time. The fact that the controlled object is basically a lag system containing dead time indicates the validity of the characteristics obtained from physical analysis in the previous section.

2.3. STATE-SPACE MODELING OF AN AUGMENTED SYSTEM - The ARX model obtained from system identification was transformed into state-space equations to enable it to be used as the internal model in MPC.

Expressing equations (7), (8) and (9) as a matrix from

$$X(k)=\begin{bmatrix} y(k) \\ y(k-1) \\ u(k-1) \end{bmatrix}, A=\begin{bmatrix} A_1 & A_2 & B_2 \\ I & O & O \\ O & O & O \end{bmatrix}, B=\begin{bmatrix} B_1 \\ O \\ I \end{bmatrix}, \quad (10)$$
$$C(k)=\begin{bmatrix} I & O & O \end{bmatrix}$$

In an augmented state assuming constant output disturbance,

$$\begin{cases} \begin{bmatrix} X(k+1) \\ d(k+1) \end{bmatrix} = \begin{bmatrix} A & 0 \\ 0 & I \end{bmatrix} \begin{bmatrix} X(k) \\ d(k) \end{bmatrix} + \begin{bmatrix} B \\ 0 \end{bmatrix} u(k-T) \\ y(k) = \begin{bmatrix} C & I \end{bmatrix} \begin{bmatrix} X(k) \\ d(k) \end{bmatrix} \end{cases} \quad (11)$$

Using a disturbance observer, the constant disturbance term, $d(k)$, is successively corrected in relation to the augmented system model.

Now, the input and output vectors are defined below.

$$u(k)=\begin{bmatrix} m_{th\_cmd}(k) \\ m_{egr\_cmd}(k) \end{bmatrix}, \quad y(k)=\begin{bmatrix} p_i(k) \\ m_a(k) \end{bmatrix} \quad (12)$$

The state-space equations can be expressed as follows.

$$
\left\{
\begin{aligned}
&\begin{bmatrix} p_{\mathrm{i}}(k+1) \\ m_{\mathrm{a}}(k+1) \\ p_{\mathrm{i}}(k) \\ m_{\mathrm{a}}(k) \\ m_{\mathrm{th\_cmd}}(k) \\ m_{\mathrm{egr\_cmd}}(k) \\ d_{p_{\mathrm{i}}}(k+1) \\ d_{m_{\mathrm{a\_Afm}}}(k+1) \end{bmatrix}
= \begin{bmatrix} A_1 & A_2 & B_2 & O \\ I & O & O & O \\ O & O & O & O \\ O & O & O & I \end{bmatrix}
\begin{bmatrix} p_{\mathrm{i}}(k-T) \\ m_{\mathrm{a}}(k-T) \\ p_{\mathrm{i}}(k-T-1) \\ m_{\mathrm{a}}(k-T-1) \\ m_{\mathrm{th\_cmd}}(k-T-1) \\ m_{\mathrm{egr\_cmd}}(k-T-1) \\ d_{p_{\mathrm{i}}}(k-T) \\ d_{m_{\mathrm{a\_Afm}}}(k-T) \end{bmatrix} \\
&\qquad\qquad + \begin{bmatrix} B_1 \\ O \\ I \\ O \end{bmatrix} \begin{bmatrix} m_{\mathrm{th\_cmd}}(k-T) \\ m_{\mathrm{egr\_cmd}}(k-T) \end{bmatrix} \\
&\begin{bmatrix} p_{\mathrm{i}}(k) \\ m_{\mathrm{a}}(k) \end{bmatrix} = C \begin{bmatrix} p_{\mathrm{i}}(k) \\ m_{\mathrm{a}}(k) \\ p_{\mathrm{i}}(k-1) \\ m_{\mathrm{a}}(k-1) \\ m_{\mathrm{th\_cmd}}(k-1) \\ m_{\mathrm{egr\_cmd}}(k-1) \\ d_{p_{\mathrm{i}}}(k) \\ d_{m_{\mathrm{a\_Afm}}}(k) \end{bmatrix}
\end{aligned}
\right. \tag{13}
$$

$$
C = [I \quad O \quad O \quad O],\ I = \begin{bmatrix} 1 & 0 \\ 0 & 1 \end{bmatrix},\ O = \begin{bmatrix} 0 & 0 \\ 0 & 0 \end{bmatrix} \tag{14}
$$

In this paper these equations are used in designing an MPC controller.

3. CONTROL SYSTEMS DESIGN

3.1. CONTROL REQUIREMENT – The requirements in the design of a control system for the controlled object that formed the subject of this research can be summarized as follows:

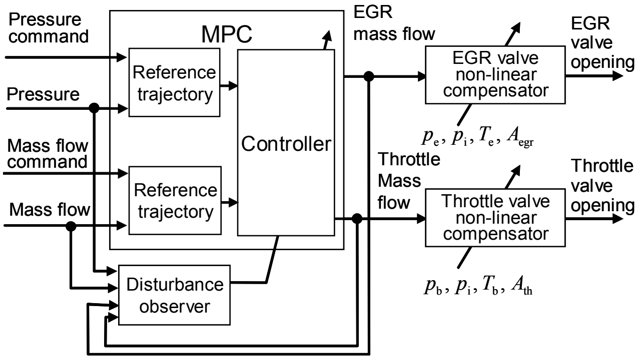

Figure 11 Block diagram of controller

1. Increase tracking speed of outputs to the changes in target values
2. Control of the effect of mutual interference between input and output systems
3. Correction of nonlinearity of valve flow rate characteristic
4. Compensation for lag characteristic of pressures and flow rates generated by actuators or volume of chambers
5. Maintenance of control performance and stability in relation to manufacturing variations and decline in flow rate characteristic of actuators, etc.
6. Maintenance of stability in relation to disturbances originating in environmental changes, etc.

3.2. CONFIGURATION OF CONTROL SYSTEM - The design of the control system responded to the requirements outlined in the section above using the following elements. Figure 11 shows the configuration of the control system.

Element	Requirement
• Model predictive controller Internal model: Multivariable ARX model	1, 2, 4
• Reference trajectory	1, 6
• Disturbance observer	5, 6
• Static nonlinear compensator for valve section	3

3.2.1. Model predictive control - The fact that MPC is able to deal entirely naturally (effortlessly) with multi-input/multi-output systems represents a considerable advantage from the perspective of practical use. In addition, the facts that its basic operating concepts are easy to understand, and that adjustment of the control in a real engine is simple and intuitive, are advantageous factors from the perspective of use in practical applications. Focusing on these advantages, MPC is applied as a controller.

Figure 12 shows a conceptual diagram of the basic operation of the controller. The order of procedures is as

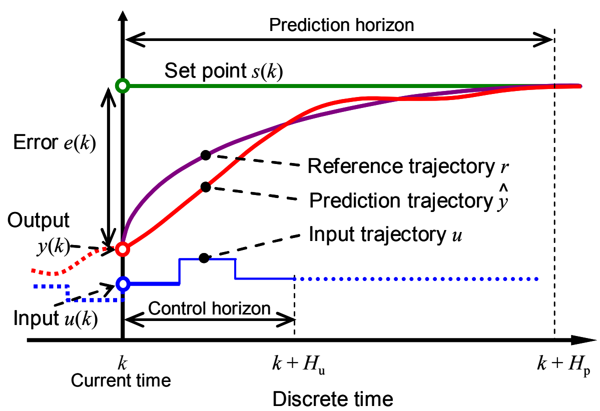

Figure 12 Basic concept of MPC

follows:

1. Formulation of reference trajectory: The speed at which the plant converges on the set value is specified using the real time error, $e(k)$.
2. Calculation of prediction trajectory: Behavior of output in the prediction horizon is predicted from the input to the internal model and control horizon.
3. Setting input trajectory: Using optimization calculations, an optimum combination of inputs (input trajectory) is calculated that will achieve as close a match as possible between the reference trajectory, 1, and prediction trajectory, 2.
4. Application of input value: The first element of the input trajectory, 3, $u(k)$, is applied to the actual controlled object.

The entire cycle from steps 1 to 4 is repeated after one sampling cycle.

3.2.2. Disturbance observer - Because MPC is a model-based control theory, modeling error in the controlled object model can result in deviations from the intended behavior of the system. In particular, if the steady-state gain in the model is inaccurate, steady-state deviation will occur in the controlled object output. As a result, performance may decline due to variations or deterioration in device characteristics or unexpected changes in the environment, etc. A disturbance observer was therefore introduced to the system in order to control effects from disturbances that would be challenging to incorporate in the model.

DMC [8], as shown in the procedures below, is known as a method enabling a disturbance observer to be employed in MPC. Using DMC, this project incorporated a disturbance observer in the system that would exclude the effect of constant disturbance.

The procedures conducted by a disturbance observer in MPC are as follows for time k:

1. Measures actual output, $y(k)$
2. Estimates disturbance as the difference between actual output and estimated output
3. Uses estimated value of disturbance to predict output between prediction horizons

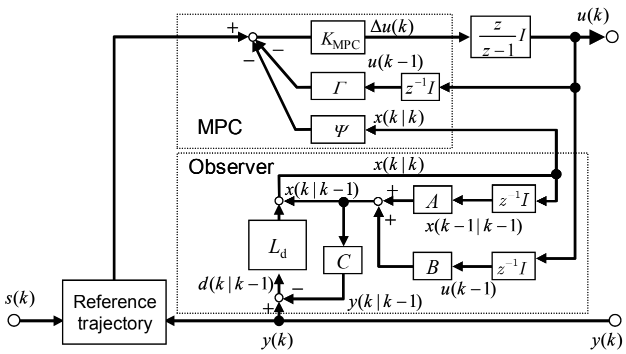

Figure 13 Block diagram of MPC with disturbance observer

This method requires data for all of the state variables of the controlled object, necessitating estimation of variables that cannot be directly measured, i.e., the use of a state observer.

Using the augmented state model incorporating the output disturbance model introduced via equation (13), state vectors that include constant disturbance can be estimated. Terming the state estimation gain matrix L, the standard observer equations can be expressed as follows:

$$\begin{bmatrix} \hat{x}(k+1|k) \\ \hat{d}(k+1|k) \end{bmatrix} = \left(\begin{bmatrix} A & 0 \\ 0 & I \end{bmatrix} - \begin{bmatrix} L_x \\ L_d \end{bmatrix} \begin{bmatrix} C & I \end{bmatrix} \right) \begin{bmatrix} \hat{x}(k|k-1) \\ \hat{d}(k|k-1) \end{bmatrix} + \begin{bmatrix} B \\ 0 \end{bmatrix} u(k) + \begin{bmatrix} L_x \\ L_d \end{bmatrix} y(k) \tag{15}$$

The entire configuration formed from the MPC and the observer is expressed as shown in Figure 13.

4. PERFORMANCE VERIFICATION USING SIMULATION

Simulations of responses to target values were conducted under conditions of constant engine speed and amount of fuel injection.

Multiple calibration parameters exist for MPC. These calibration parameters were estimated on the basis of generally known MPC tuning parameters [8], and were ultimately determined by trial and error in the simulation. The effect of the coincidence point parameter is discussed here as an example. Figures 14 and 15 show the results of simulations of responses to target values.

As Figure 14 shows, when coincidence points are set across the entire range between the predictions horizons, output response is rapid, but input is unstable.

Figure 15 shows that when coincidence points are set only at the beginning and end of the prediction horizon, results display a comparatively good balance, with good response and stability for both input and output. In addition, minimizing the number of coincidence points reduces the calculation load on the ECU.

The simulation results show rapid convergence of two output values on two independent changes in target values without oscillation due to interference, demonstrating that the MPC behaves as required.

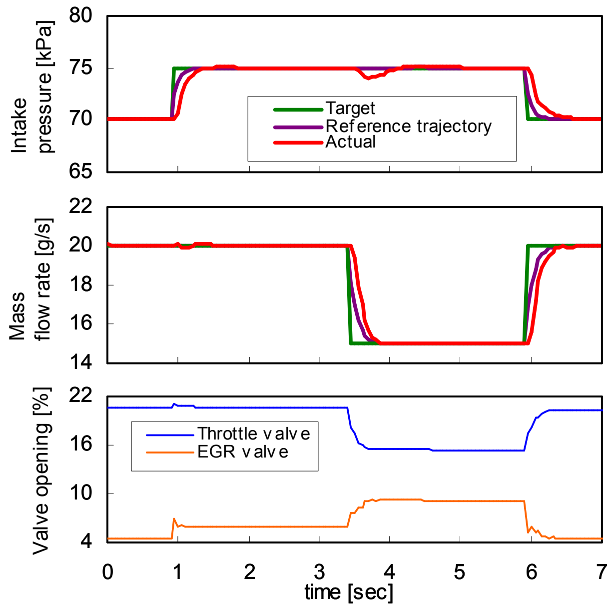

Figure 14 Simulation results for variable coincidence point

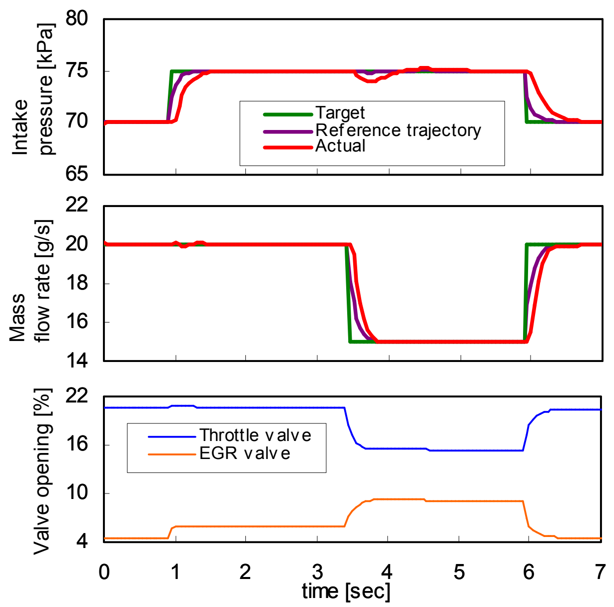

Figure 15 Simulation results with variation of target value

5. PERFORMANCE VERIFICATION IN REAL ENGINE

Tests of responses to target values were conducted using MPC and a conventional control method in a real vehicle. Steady-state operating conditions were employed, with engine speed and the amount of fuel injection held constant.

Figure 16 shows the results of a test of step response to intake chamber pressure target values with the target value for air mass flow held constant. Despite a certain degree of overshoot, MPC converges on the target value in around 300 ms. The conventional control method was able to converge on the target value in 500 ms when the value was reduced, but a long convergence time – approximately 2.5 sec – was required when the value was increased. In addition, the conventional control method displayed a certain amount of fluctuation in actual output in relation to the target value for air mass flow, and exact convergence was not achieved.

Figure 17 shows the test results of step response to air mass flow target values with the target value for intake chamber pressure held constant. MPC converged on the target value in around 300 ms, while convergence in the conventional control method took longer at approximately 800 ms. In addition, the intake chamber target pressure was constant, but the conventional control method produced one significant reduction in the pressure, which is believed to have been a result of interference.

As in the case of the engine simulator results, these results indicate that the use of MPC has enabled the achievement of rapid convergence on two outputs for independent target values.

Under the test conditions used here, the time required for the conventional control method to converge on the target values was 2.5 to 8 times that required by MPC, and this phenomenon was particularly conspicuous when the intake chamber pressure target value was increased, and when the air mass flow target value was changed.

As indicated in Chapter 1, deviation in the intake state can negatively affect engine performance, for example by increasing emissions and combustion noise. The application of intake control using MPC, which is able to increase the speed of response of the intake state, can be expected to result in a significant improvement of emissions, combustion noise and other performance parameters under transient operating conditions.

6. CONCLUSION

An intake control system using MPC has been applied to the intake system of a diesel engine, a multi-input, multi-output coupled system. The following results have been achieved:

1. The internal model used in MPC was constructed using a multivariable ARX model produced by means of system identification and correction of the nonlinearity of the actuator flow rate characteristic based on a convergent nozzle equation. This enabled an extremely accurate internal model to be obtained.

2. MPC has been applied to intake control in a diesel engine. The system is able to achieve rapid convergence of two outputs on independent target values, and convergence is 2.5 to 8 times faster than when a conventional control method is employed.

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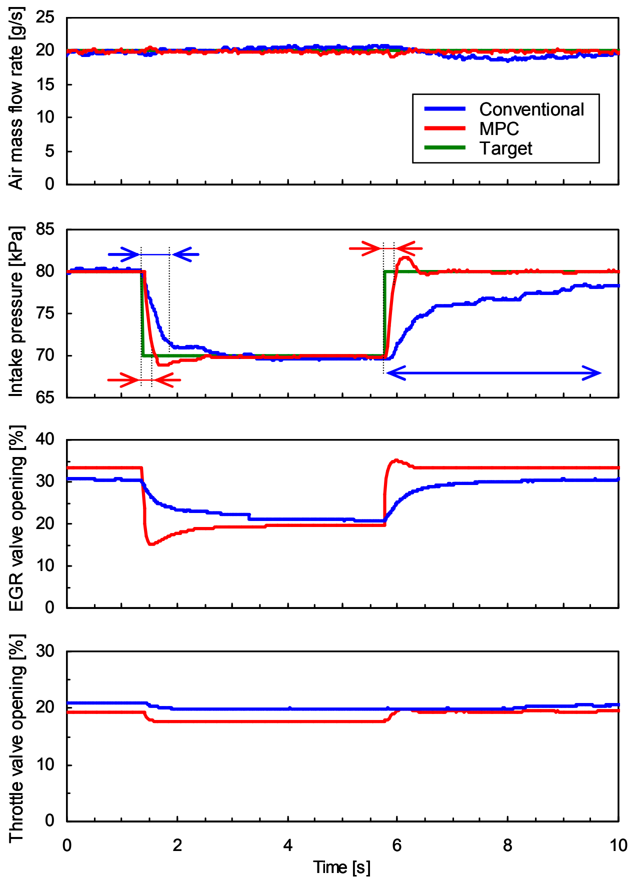

Figure 16 Test results at intake chamber pressure target values with the target value for air mass flow held constant

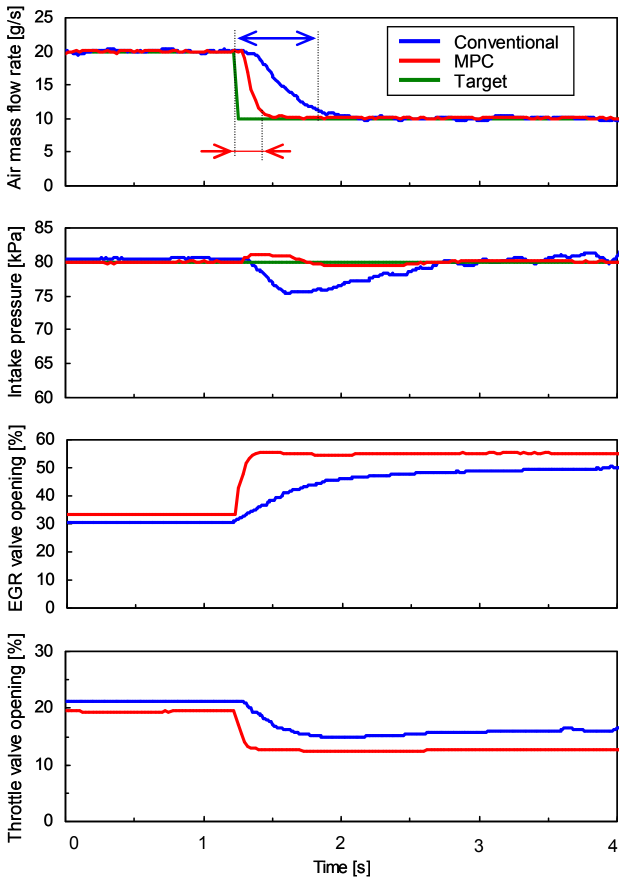

Figure 17 Test results at air mass flow target values with the target value for intake chamber pressure held constant