

Model Based Air-path Management Control for Clean Diesel Engine

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ABSTRACT

Research has been conducted on a variety of combustion technologies in order to reduce diesel engine emissions. These technologies should precisely control the state of in-cylinder gas (EGR mass flow, air mass flow, and the like). However, because the controlled object is a multi-input, multi-output (MIMO) system and a coupled system, the use of control systems based on conventional methods that employ PID controllers represents a challenge. Model predictive control (MPC) is well known as a MIMO algorithm. An intake control system that could be applied to the intake system of a diesel engine was constructed by supplementing MPC with a feedback function using a disturbance observer and compensator for the nonlinear characteristic of the actuators. Performance tests using an actual vehicle verified that, when applied to a two-input (throttle valve, EGR valve), two-output (air mass flow, intake chamber pressure) system, the proposed MPC was able to rapidly control each output independently to the target value.

1. Introduction

Diesel engines have higher thermal efficiency and produce a lower amount of CO₂ exhaust than gasoline engines, and are thus gathering attention as an effective internal-combustion engine in helping avert global warming. However, the simple aftertreatment system used in gasoline engines that employ a three-way catalyst with a reduction efficiency close to 100% cannot be applied to lean combustion diesel engines. This makes it necessary to sufficiently reduce NO_x, HC, soot and other exhaust gas components during combustion within the cylinder.

The Exhaust Gas Recirculation (hereafter, "EGR") system is known to be an effective means of reducing NO_x emissions, which is particularly important for diesel engines. This system suppresses the generation of NO_x, which increases rapidly at high combustion temperatures, by mixing the inert gas included in the exhaust gas into the intake air to lower the oxygen concentration, thus lowering the maximum temperature during combustion. However, as the EGR ratio is increased, the amount of soot produced also tends to increase after a certain excess air factor is reached. Therefore, to effectively reduce NO_x emission while also suppressing soot generation, it is necessary to use EGR control and fuel injection control

to accurately control the combustion state, represented by the excess air factor and other parameters, to within the desired range.

Recent studies have focused on technology that can simultaneously reduce both NO_x and soot by using PCCI combustion, which introduces even larger amounts of EGR to aggressively extend the ignition delay period, and incorporates premixed combustion⁽¹⁾. However, the PCCI combustion test results in Figs. 1 and 2 show that the exhaust gas characteristics fluctuate greatly according to the EGR ratio, and practical application needs enhancement of fuel injection control and EGR control performance to enable construction of a high response control system that is capable of tracking transient operation.

On the other hand, research aimed at practical application is also actively underway to develop technology for further reducing tailpipe emissions using a NO_x catalyst and other exhaust aftertreatment systems in order to further reduce exhaust gas in the future. NO_x catalyst systems need regenerating combustion control, called rich control, to reduce the NO_x adsorbed during normal lean combustion⁽²⁾. Rich control needs high-level combustion state control to match the air-fuel ratio to the target value so as not to adversely affect the combustion

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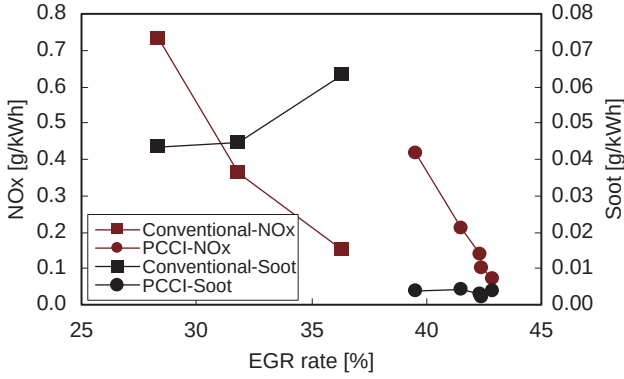


Fig. 1 Emission characteristics of conventional and PCCI combustion

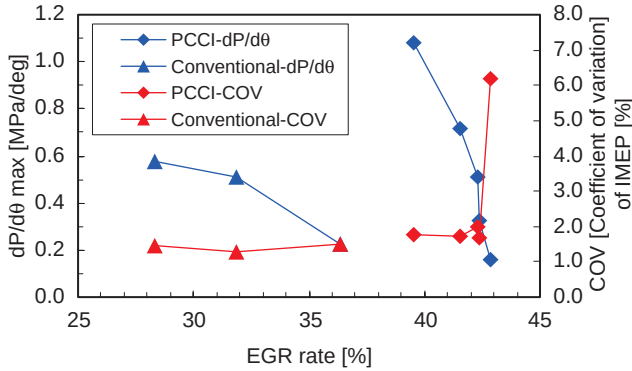


Fig. 2 Noise and vibration characteristics of conventional and PCCI combustion

noise and drivability. Therefore, in addition to fuel injection control, it is also important to increase the accuracy of the intake gas control that controls the distribution of the amount of air and the amount of EGR taken into the cylinder (hereafter “intake gas state”).

The above demands are increasing the need for more precise control of the intake gas state compared to previous systems.

2. Research Goals

This study designed and verified the effects of a control system that applies model predictive control (hereafter, “MPC”) to construct an intake gas state feedback control system in a diesel engine.

The physical amounts to be controlled by intake gas control are the amounts of oxygen and inert gas within the cylinder, that is to say the air mass and the amount of EGR gas. Here, among all the gas components taken into the cylinder, the air mass is the intake air component that excludes the gas returned by EGR. The directly operable intake actuators that control these physical amounts include the EGR valve and the throttle valve.

However, generally when any one of the intake actuators is changed, both physical amounts of the intake

gas state vary. This is the controlled object, called a coupled system, and construction of a feedback control system by conventional methods such as using a PID controller is known to be a challenge. Figure 3 shows the control effects when a general PID controller is applied to an engine simulator that simulates the above controlled object, and shows that coupling produces oscillating behavior.

On the other hand, MPC is known as a control algorithm that can be applied to a multi-input, multi-output system capable of feeding back multiple outputs to the target values. However, optimization calculations should be performed in real time in each sampling period, so MPC is mainly applied as in petrochemical plant systems and other systems that have long sampling periods.

However, in addition to increased ECU calculation processing speeds, remarkable progress is also being made on technology to further increase speeds by methods such as revising calculation algorithms⁽³⁾, and prospects look promising for MPC’s application to relatively fast systems⁽⁴⁾. Therefore, MPC was applied to design a feedback control system for the air mass flow and EGR mass flow, and the effects were confirmed.

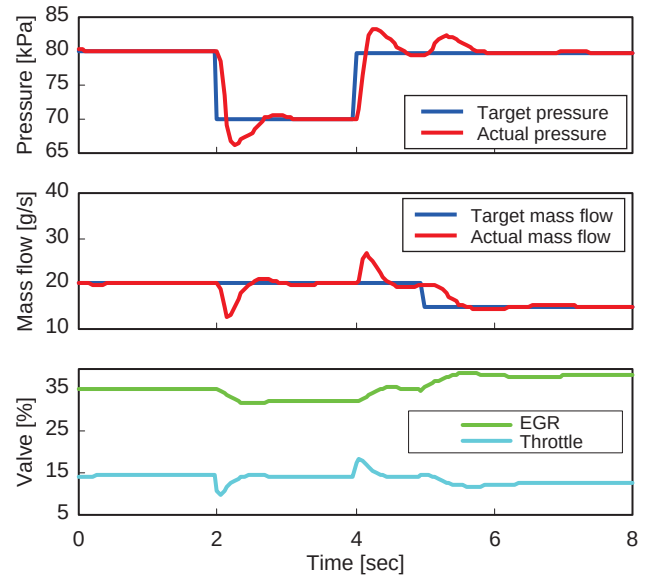


Fig. 3 PI control simulation results

3. Modeling

Figure 4 shows the intake and exhaust system configuration used in general diesel engines. This study considered the controlled object of a 2-input, 2-output system with the input and output variables defined as follows.

Inputs: EGR valve opening, throttle valve opening

Outputs: Intake chamber pressure, air mass flow

Normally, the air mass and the EGR amount taken into the cylinder should be selected as the output parameters, but it is a challenge to directly measure both of these, so the air mass flow and intake chamber pressure were selected as the respective substitute output parameters. Here, the air mass flow is the mass flow of the air measured at the position indicated in Fig. 4. In addition, the EGR amount can be estimated from the intake chamber pressure and the air mass flow, so the intake chamber pressure was selected as the other output parameter.

When applying MPC, optimal control, adaptive control, or other model-based control algorithms, the dynamic characteristics of the controlled object should be expressed by a state-space equation, that is to say a simultaneous equation using differential equations or difference equations.

First, modeling based on physical equations is performed to understand the structure of the controlled object, and the degree of the state-space equation is selected. Next, the physical parameters that prescribe the state-space equation are determined by system identification to obtain the state-space equation used by the control algorithm built into an actual diesel engine.

3.1. Physical Modeling

As shown in Fig. 4, the intake and exhaust system of a diesel engine is composed of a combination of valve elements and chamber elements. Therefore, modeling was performed based on physical principles, focusing on the throttle valve and its downstream chamber as shown by the example in Fig. 5.

Labeling the throttle valve opening as α_{th} , the upstream pressure and downstream pressure as p_0 and p_i , respectively, the upstream density as ρ_0 , and the specific heat ratio as κ , the air mass flow passing through the throttle valve is expressed by the equation of convergent nozzle^{(5), (6)}.

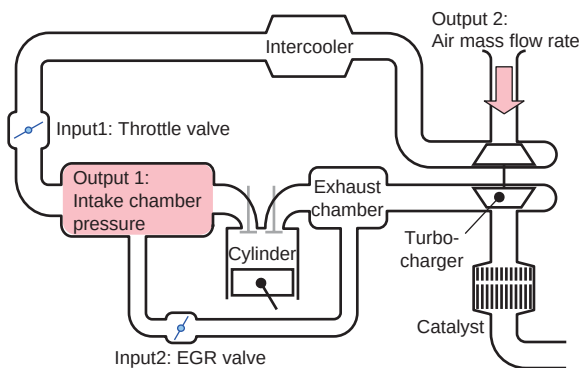


Fig. 4 Air-path system of diesel engine and definition of the controlled object

$$\dot{m}_{th} = A_{th}(\alpha_{th}) \cdot \sqrt{2p_0\rho_0} \cdot \Phi$$

$$\Phi = \begin{cases} \sqrt{\frac{\kappa}{\kappa-1} \left\{ \left(\frac{p_i}{p_0} \right)^{\frac{2}{\kappa}} - \left(\frac{p_i}{p_0} \right)^{\frac{\kappa+1}{\kappa}} \right\}}, & \left(\frac{p_i}{p_0} \right) > \left(\frac{2}{\kappa+1} \right)^{\frac{\kappa}{\kappa-1}} \\ \left(\frac{2}{\kappa+1} \right)^{\frac{\kappa}{\kappa-1}} \sqrt{\frac{\kappa}{\kappa+1}}, & \left(\frac{p_i}{p_0} \right) \leq \left(\frac{2}{\kappa+1} \right)^{\frac{\kappa}{\kappa-1}} \end{cases} \quad (1)$$

Here, $A_{th}(\alpha_{th})$ is the function that expresses the effective aperture area A_{th} relative to the opening α_{th} .

On the other hand, the change in the intake chamber pressure is expressed by the equation of ideal gas state within the chamber and the Law of Conservation of Mass⁽⁷⁾ as follows.

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

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

At this time, the change in the intake chamber pressure is expressed as follows.

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

Here, n is the polytropic index.

In addition, the cylinder intake air mass flow can be expressed as the 1st order function of the chamber internal pressure p_i as shown by the following equation.

$$\dot{m}_{cyl} = k_{\eta v} n_e p_i \quad (5)$$

Here, n_e is the engine speed, and $k_{\eta v}$ is a coefficient that includes the volumetric efficiency.

From the above, the following equation that is expressed by a 1st order lag system can be obtained from Eqs. (1), (4) and (5).

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

$$a = -n \frac{RT_i}{V_i} k_{\eta v} n_e, \quad b = n \frac{RT_i}{V_i} \sqrt{2p_0\rho_0} \cdot \Phi \quad (6)$$

However, delay parameter a and gain parameter b are both expressed by non-linear functions, so this is a so-called non-linear system in which the dynamic behavior varies according to the operation point. Therefore, linear

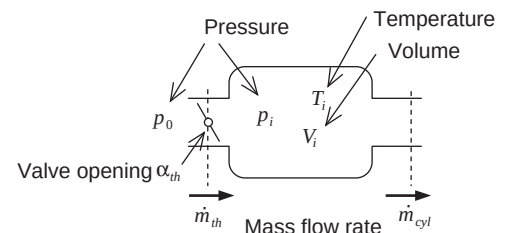


Fig. 5 Element model of valve and chamber system

control theory that includes MPC is unable to be applied in this format.

For this reason, an attempt was made to create a linear system using methods that establish input-state linearization⁽⁸⁾. Figure 6 shows the control system configuration achieved by input-state linearization. When input linearity transformation using the equation of convergent nozzle [Eq. (1)] is positioned at the controller output, the controlled object as viewed from the controller can be considered linear. This enables configuration of a linear controller that uses Eqs. (4) and (5) as the state equations.

The controlled object also has element systems other than those shown in the example above. The dynamic characteristics of each actuator are the dead time element and the 1st-order lag element, and the pressure lag from the air flow meter position to the throttle valve due to gas charging of the chamber can be approximated using the 1st-order lag element. Therefore, the controlled object is considered to be a system that combines dead time elements and lag elements.

3.2. System Identification

The characteristics of the controlled object were examined in the previous section, but the parameters included in the controlled object model are composed of a combination of numerous physical constants, and it is a challenge to physically determine these values. Therefore, system identification⁽⁹⁾ was performed to obtain the state-space equation used by the control theory applied to an actual engine.

Inputs: Command value for mass flow in throttle valve section m_{th_cmd} [g/s]

Command value for mass flow in EGR section m_{egr_cmd} [g/s]

Outputs: Intake chamber pressure p_i [kPa]

Air mass flow m_a [g/s]

The input and output data used for system identification were measured by applying a binary pseudo random signal called a signal of maximal-length sequences⁽¹⁰⁾ to the two input values noted above, and measuring the two output values at that time. Figure 7 shows the input and output time series data at that time.

The dead time T of the controlled object was determined from the measured input and output time series data using a method that estimates the impulse response⁽⁹⁾.

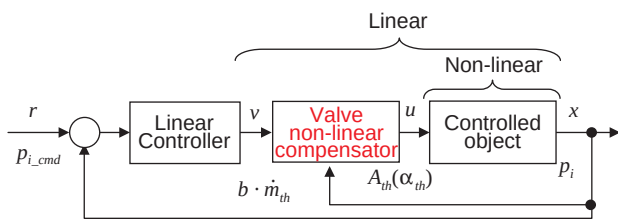


Fig. 6 Control structure with non-linear compensator

Next, the model structure used the 2-input, 2-output multivariable Auto Regressive with eXtra input model (hereafter “ARX model”) expressed by Eq. (7), and system identification was performed using the measured input and output time series data.

$$y(k+1) + A_1 y(k) + A_2 y(k-1) = B_1 u(k-T) + B_2 u(k-T-1) \quad (7)$$

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

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

Figure 8 shows the time series data of the actual output and the ARX model output to verify the accuracy of the identified 2-input, 2-output ARX model. The ARX model output accurately represents the actual output behavior, indicating that the obtained ARX model has sufficient accuracy. This is thought to be because inserting a non-linear compensator between the system identification input values and the input values applied to

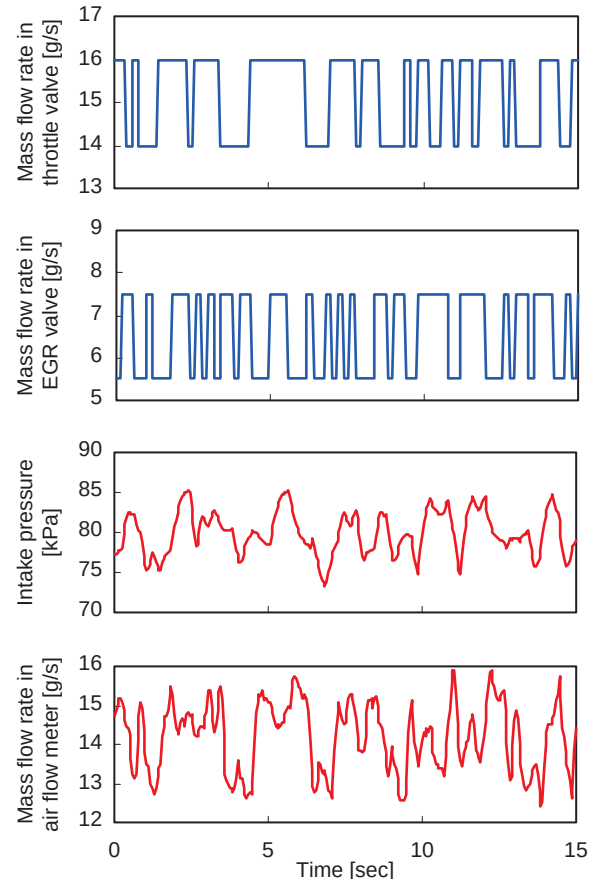


Fig. 7 Input and output measurement data

the actuators enabled the non-linear characteristics of the controlled object to be sufficiently canceled, and thus provided input and output data with high signal to noise ratios.

In addition, verification using the spectral analysis method⁽⁹⁾ also confirmed that the spectrum of the measured values and that of the obtained ARX model matched closely, which indicated that the ARX model is also accurate in terms of frequency range.

3.3. Augmented State-space Modeling

The ARX model obtained by system identification was transformed into a state-space equation for use as an MPC internal model.

Eqs. (7), (8), and (9) can be expressed in matrix form as follows.

$$\begin{cases} X(k+1) = AX(k) + Bu(k-T) \\ y(k) = CX(k) \end{cases}$$

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

$$C = [I \quad O \quad O]$$

In addition, the following augmented model is defined by adding a constant output disturbance to the states.

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

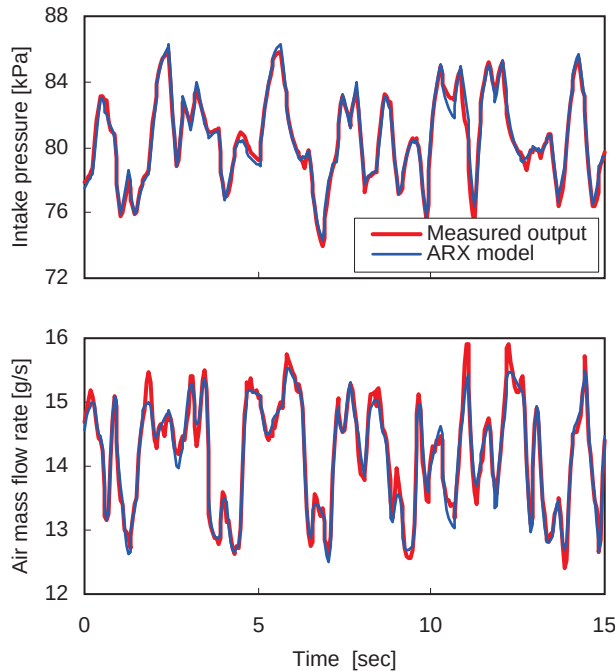


Fig. 8 Time series result

The constant disturbance item $d(k)$ of this augmented model is successively corrected using a disturbance observer.

Here, the input and output vectors are defined as shown in the following equation.

$$u(k) = \begin{bmatrix} m_{th_cmd}(k) \\ m_{egr_cmd}(k) \end{bmatrix}, y(k) = \begin{bmatrix} p_i(k) \\ m_a(k) \end{bmatrix} \quad (12)$$

At this time, the state-space equation is expressed by Eqs. (13) and (14), so this study used these equations to design the MPC controller.

$$\begin{cases} \begin{bmatrix} p_i(k+1) \\ m_a(k+1) \\ p_i(k) \\ m_a(k) \\ m_{th_cmd}(k) \\ m_{egr_cmd}(k) \\ d_{pi}(k+1) \\ d_{ma}(k+1) \end{bmatrix} = \begin{bmatrix} A_1 & A_2 & B_2 & O \\ I & O & O & O \\ O & O & O & O \\ O & O & O & I \end{bmatrix} \begin{bmatrix} p_i(k-T) \\ m_a(k-T) \\ p_i(k-T-1) \\ m_a(k-T-1) \\ m_{th_cmd}(k-T-1) \\ m_{egr_cmd}(k-T-1) \\ d_{pi}(k-T) \\ d_{ma}(k-T) \end{bmatrix} \\ + \begin{bmatrix} B_1 \\ O \\ I \\ O \end{bmatrix} \begin{bmatrix} m_{th_cmd}(k-T) \\ m_{egr_cmd}(k-T) \end{bmatrix} \\ \begin{bmatrix} p_i(k) \\ m_a(k) \end{bmatrix} = C \begin{bmatrix} p_i(k) \\ m_a(k) \\ p_i(k-1) \\ m_a(k-1) \\ m_{th_cmd}(k-1) \\ m_{egr_cmd}(k-1) \\ d_{pi}(k) \\ d_{ma}(k) \end{bmatrix} \end{cases} \quad (13)$$

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

4. Control System Design

4.1. Control Goals

The following requirements were given for the design of a control system for the controlled object in this study.

- (1) Increasing the output tracking speed with respect to changes in target values
- (2) Suppressing the effects of coupling in a multi-input, multi-output system
- (3) Removing the non-linearity of the valve mass flow characteristics
- (4) Compensating the pressure and mass flow lag characteristics produced by the actuator and chamber capacities
- (5) Achieving control performance and stability with respect to manufacturing variance and changes in the mass flow characteristics of actuators and other parts

- (6) Achieving stability with respect to disturbances produced by environmental changes, and other factors

4.2. Control System Configuration

This study addressed the requirements listed in the previous section using the following elements. Figure 9 shows the control system configuration.

Elements	Requirements
(a) Model predictive controller (MPC)	(1), (2), (4)
(b) Reference trajectory	(1), (6)
(c) Disturbance observer	(5), (6)
(d) Valve block static non-linear compensator (3)	

4.2.1. Model predictive control

One of the practical advantages of MPC is the ability to naturally handle multi-input, multi-output control. Additional practical advantages are that the basic operation concept is easy to understand, and adjustment in an actual engine is easy and intuitive. Therefore, focusing on these advantages, MPC was applied as the controller.

Figure 10 shows a concept drawing of basic operation, and the operation procedure is described below.

- (1) Generation of reference trajectory: Designate the speed

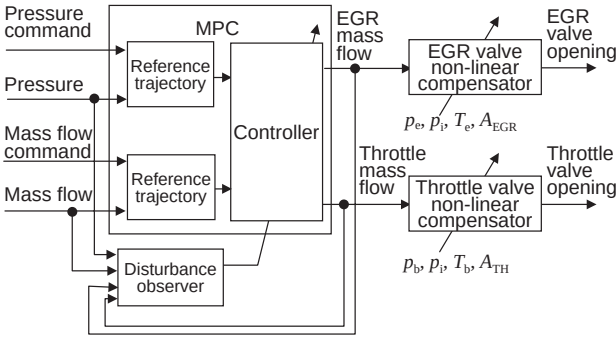


Fig. 9 Block diagram of controller

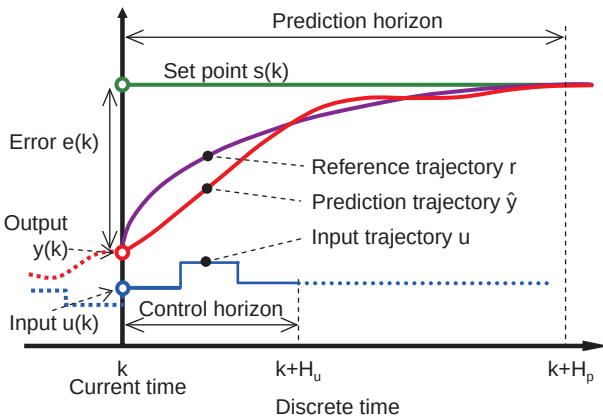


Fig. 10 Basic concept of MPC

at which the plant moves towards the setting value using the current time error $e(k)$.

- (2) Calculation of prediction trajectory: Predict the behavior of the output during the prediction horizon period from the internal model and the input during the control horizon period.
- (3) Determination of input trajectory: Perform optimization calculation to determine the ideal combination of inputs (input trajectory) so that the reference trajectory (1) and prediction trajectory (2) match as closely as possible.
- (4) Application of input values: Apply the first element $u(k)$ of the input trajectory (3) to the actual controlled object.

The entire cycle from step (1) through step (4) is repeated after a single sampling period.

4.2.2. Disturbance observer

MPC is a control theory based on a state equation (hereafter, "model") that expresses the dynamic characteristics of the controlled object, so if that model contains errors, it will produce error relative to the intended behavior. In particular, if the model steady state gain is inaccurate, steady state error will occur in the controlled object output. As a result, the device characteristics may change or have variance, or the control performance may drop due to unpredictable changes in the environment, and other factors. Therefore, a disturbance observer was introduced to suppress the effects of disturbance, which are a challenge to include in the model.

Dynamic Matrix Control (DMC)⁽¹¹⁾ is known as a method that applies a disturbance observer to MPC by the procedure shown below. This study used DMC to apply a disturbance observer that helps eliminate the effects of constant disturbance.

The disturbance observer in MPC performs the following standard procedure at time k .

- (1) Measure the actual output $y(k)$.
- (2) Estimate the disturbance as the difference between the actual output and the estimated output.
- (3) Use that estimated disturbance value to predict the output during the prediction horizon period.

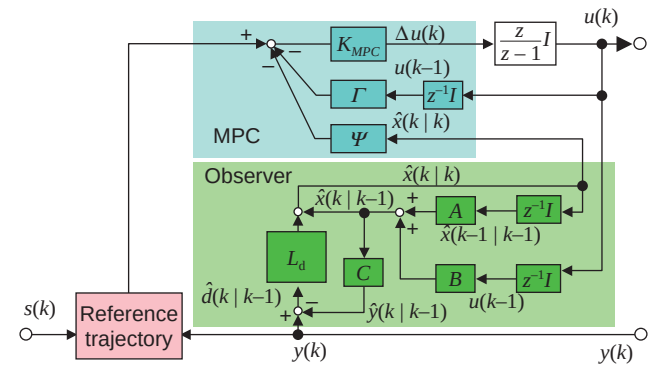


Fig. 11 Block diagram of MPC with disturbance observer

This method should have information on all state variables of the controlled object, so a so-called state observer is needed to estimate state variables that cannot be directly measured.

The state vector that includes the constant output disturbance can be estimated using an augmented state-space model that includes the output disturbance model derived by Eq. (13). Labeling the state estimation gain matrix as L , the standard observer equation is expressed by the following equation.

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

From the above, the overall configuration that combines MPC with the observer is as shown in Fig. 11.

5. Verification of Operation by Simulations

5.1. Conditions

The MPC logic is large-scale and complex, so operation was verified using an engine simulator to debug the algorithm and perform pre-calibration before application to an actual engine. The conditions of the engine simulator used are as follows.

- One-dimensional fluid simulation software GT-Power Ver. 6.1 for analyzing engine performance
- An average value model using a neural network (MeanV Cyl)

Figure 12 shows the simulation environment configuration. In the environment shown in the figure, the control algorithm used in the actual engine is described by MATLAB/Simulink, and is linked with GT-Power. This enables verification of control algorithm operation using a controlled object having similar dynamic characteristics to those of an actual engine.

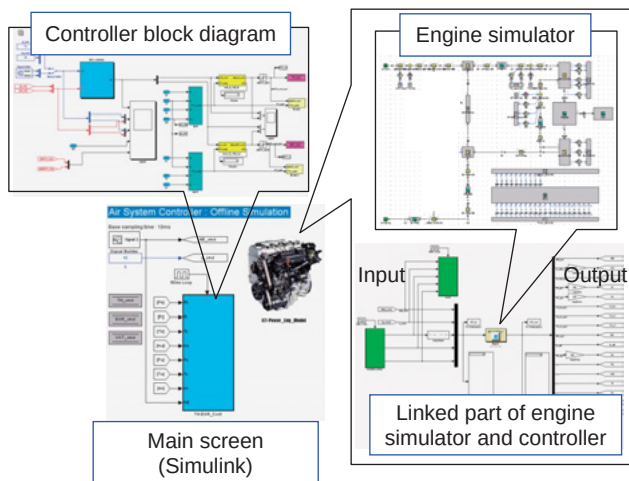


Fig. 12 Simulation environment

5.2. Results

Target value response simulations were conducted under fixed engine speed and fuel injection amount conditions.

MPC includes multiple calibration parameters. These calibration parameters are selected by the generally known MPC tuning index⁽¹¹⁾, and ultimately determined through simulations so that the output characteristics have fast

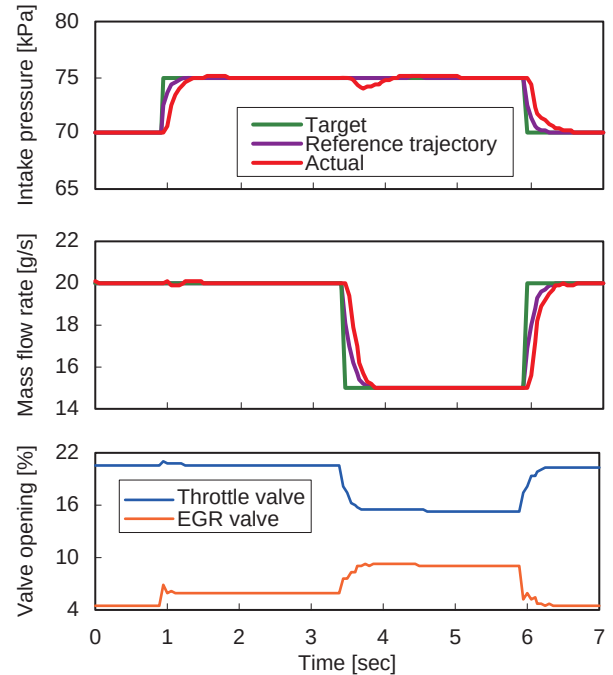


Fig. 13 Simulation results for variable coincidence point

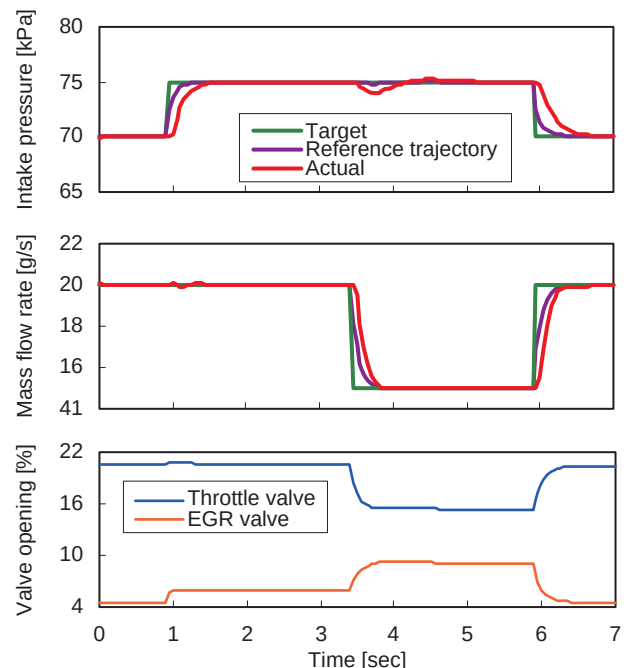


Fig. 14 Simulation results with variation of target value

responsiveness with little overshoot. Here, the effects of the coincidence point parameter are described as an example. The coincidence point parameter is a calibration parameter that stipulates in which steps of the prediction horizon period to match the prediction value to the reference trajectory obtained from the output target value, within the optimization calculations used to determine the control input.

Figures 13 and 14 show the target value response simulation results. Figure 13 shows the results when coincidence points are set over the entire prediction horizon period. This obtained a sufficiently fast output response, but unstable, oscillating behavior tended to appear in the input variables of the throttle valve opening and EGR valve opening, particularly during the transient period in which the output changes. In contrast, Fig. 14 shows the results when coincidence points are set at only the two points of the start and end of the prediction horizon period. Here, relatively well-balanced results were obtained with good response and stability for both inputs and outputs. In addition, setting fewer coincidence points also enables the calculation load on the ECU to be reduced.

The simulation results show that the proposed MPC

operates as intended, and that the two output values converge rapidly, without oscillation due to coupling, relative to the changes in two independent target values.

6. Verification of Operation Using Actual Vehicles

Next, target value response tests were conducted using MPC and the conventional control method mounted in actual vehicles. The test conditions were steady state operation with a constant engine speed and fuel injection amount.

Figure 15 shows the test results for the intake chamber pressure target value step response using a constant air mass flow target value. Here, comparison was performed using the time to reach 90% of the target value (90% response time, hereafter, “T90”) as the index of the response speed to the target value. For the conventional control, T90 needed approximately 500 ms in the decrease direction (a in the figure) and approximately 2 s or more in the increase direction (b in the figure). In contrast, while MPC exhibited some overshoot, it had a faster response speed than the conventional control, with a T90 of only approximately

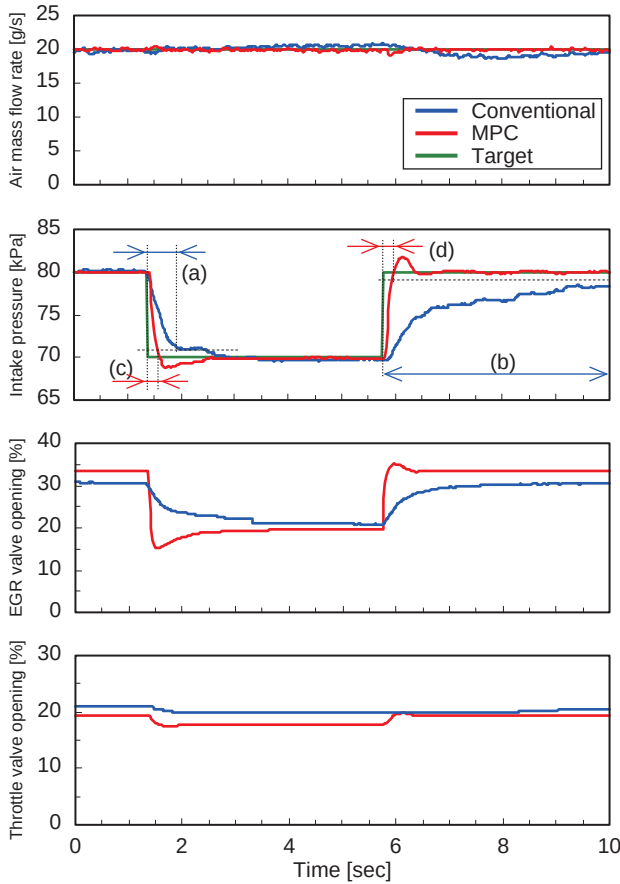


Fig. 15 Test results with variation of target pressure controlled by MPC

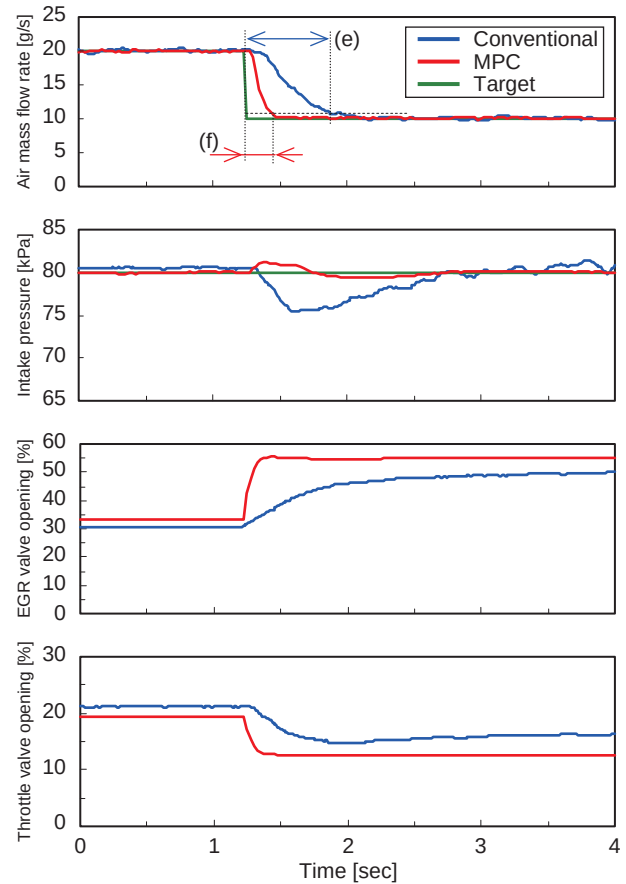


Fig. 16 Test results with variation of target mass flow controlled by MPC

200 ms in both the decrease and increase directions (c and d in the figure). In addition, the conventional control exhibited some fluctuation in the actual output, and did not converge completely with the air mass flow target value.

Figure 16 shows the test results for the air mass flow target value step response using a constant intake chamber pressure target value. T90 for the conventional control was approximately 800 ms (e in the figure). In addition, fluctuating behavior was seen in the actual pressure with respect to a constant intake chamber pressure target value due to the effects of coupling. In contrast, MPC had a T90 of approximately 200 ms (f in the figure), exhibited no pressure fluctuation due to coupling, and achieved a faster response speed than the conventional control.

The above results showed that applying MPC enabled two outputs to rapidly converge, independently, to two target values in the same manner as the engine simulator results.

Under these test conditions, MPC control converged to the target values at least 2.5 times faster than the conventional control. In addition, these enhancement effects further increased when the intake chamber pressure target value was increased and the air mass flow target value was varied.

7. Conclusion

Intake control that uses MPC was applied to a diesel engine intake system, which is a multi-input, multi-output coupled system, and the following items were verified.

- (1) An MPC internal model was constructed using a multivariable ARX model obtained through system identification, and non-linear compensation of the actuator mass flow characteristics based on the equation of convergent nozzle. This enabled an accurate internal model to be obtained.
- (2) MPC was applied to the intake control of a diesel engine. The results showed that two outputs could be rapidly converged, independently, to the respective target values, and a convergence speed of at least 2.5 times faster than that of conventional control was achieved.

As mentioned in the Introduction, intake state deviation is a factor that can adversely affect engine performance such as emissions and combustion noise. Therefore, application of intake control that uses MPC, which increases the intake state responsiveness, can be expected to enhance emissions, combustion noise, and other engine performance features in transient states.

In the future, the authors intend to further refine algorithms to enable operation over the entire engine range, including transient states during which the engine speed and fuel injection amount vary, and to practically apply highly accurate intake control systems to realize various advanced combustion technologies.

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