

Development of a Simple Model for Simulation of Engine Gas Flow

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ABSTRACT

An engine model allowing real-time computation and having a high degree of reusability was developed for application as a system design tool and as hardware in the loop system (HILS) for electronic control unit (ECU) development.

A distinctive feature of this model is that it is composed of simple models of engine parts. These parts are units in which the gases flow through the engine, they are modeled by applying the law of energy and mass conservation.

The model was installed in a rapid prototype system and used to conduct a comparison with an actual vehicle while it was in operation. The results of the comparison confirmed that the engine air inlet mass rate, the pressure, and temperature in the inlet and exhaust pipes could be predicted with acceptable accuracy.

1. Introduction

In order to provide the customer with high-quality, timely, low-cost products, it is vital that automotive manufacturers raise the level of development efficiency.

To make this happen, the number of trials should be reduced, and the most common approach is, at the design stage, to predict performance, and fix important details of part models by using simulations. In recent years, there has been a movement towards even higher design quality, through integration of CAD and CAE, reduction in labor for model data creation, and inclusion of simulations in the design flow.⁽¹⁾ On the one hand, CAE utilization for single parts has become common, on the other hand, there is an ever-stronger demand for performance predictions for systems which consist of any number of parts.

In ECU development, a virtual system called HILS has

been created, and systems that develop control programs and carry out verifications have also been completed.

It has also been generally true that in the design of this system and development of the ECU, simulations have been implemented with different models and in different environments. However, if it were possible to pass over the system built at the development stage, to the ECU development center, as a virtual system, without having to wait for completion of a prototype vehicle, then it would be possible to improve development efficiency through parallel development.

This research project focuses on gas flow in the engine system, and proposes a simulation model that can be applied to both performance predictions, such as turbocharger matching, and HILS for ECU development. This paper reports on the concept of the proposed model, the results of a study of the algorithms used, and the results of a verification comparison between the model and an actual vehicle.

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2. Concept of the Model

For the purposes of system performance predictions and application to HILS, the model should meet the following requirements.

- (1) Real-time computation
- (2) A high degree of reusability

For the application to HILS, real-time computation is an essential condition. Even though computer performance has been improved, modeling a finely discretized system requires significant computing resources, the time taken for calculations is not practical, so generally it is handled as a coarse model using a lumped constant system. In examining the engine gas flow handled in this research project, while there are some phenomena such as pulsation or interference that can only be expressed by spatially finely resolved models, it became clear that even with a coarse model using a lumped constant system, adequate results can be obtained.⁽²⁾ In this research project, use of the lumped constant system model, giving priority to calculation speed, was assumed, and phenomena expressed with a distributed constant system were expressed as a compensation value.

Because the created model has a high degree of reusability, the time to create new models can be reduced when studying other systems. Therefore, development efficiency is improved. In order to have a desired high degree of reusability, it was very important to fully consider input and output parameters that do not limit single part modeling and connections between models. In this research project, the law of energy and mass conservation was applied to the flow of gas between parts, and as the input/output parameters common between models used to calculate energy, gas mass flow, gas composition ratio, and gas pressure relating to temperature and mass flow were selected.

As explained by Takahashi⁽³⁾, gas composition needs to be calculated in order to account for the effect of EGR gas on combustion. In particular, with lean-burn combustion that is applied in direct injection gasoline engines and diesel engines, exhaust gases, including excess air, is once again sucked into the cylinder via the EGR valve, and so its effect is even higher. In this research project, flow in each component is calculated, and at the same time, changes in gas specific heat according to component mass proportions are consecutively calculated. The result of the calculation is reflected in calculations of pressure and temperature affected by the specific heat.

In the sections below, details of the model and results of a verification comparison with an actual vehicle are explained.

3. Model Calculation Algorithms

3.1. Overall Outline

A model was created, focusing on gas flow, for the engine component parts of the turbocharger, intercooler, exhaust and inlet pipes, EGR valve, throttle valve, and cylinders. That is to say, no consideration was given to the EGR mechanical parts of the EGR valve, etc., at this time. Because, in terms of gas flow function, inlet and exhaust pipes are equivalent, and EGR valve, throttle valve, and turbocharger nozzle functions are equivalent, basically the same model was used. In order to reduce the amount of data and the calculation load, physical formulae were used as much as possible, and calculation steps were based on time.

An explanation of each model follows.

3.2. Gas Model

3.2.1. Application of the law of energy and mass conservation

Gas pressure, temperature, and composition in the system vary due to compression and expansion due to the turbocharger, cooling in the intercooler, combustion in the cylinder, and gas mixing in the EGR. These thermodynamic parameter changes are governed by the equation of state for ideal gas (Equation 1).

$$pv = mRT \quad (1)$$

p : pressure
 v : volume
 m : mass
 R : gas constant
 T : temperature

In this way, the various states of gas, changing according to engine part function, can be modeled by applying the law of energy and mass conservation — for example cooling in the intercooler cannot be converted physically, so it can be captured as thermal energy reduction; and gas mixing in the EGR is a physical conversion, so it can be modeled as addition of the energy and mass of the air and the EGR gas. In this way, the various phenomena in gas flow can be expressed in simple arithmetic calculations of energy and mass, and measurable parameters such as temperature and pressure can be easily calculated.

3.2.2. Gas components

Specific heat, which changes according to gas components, is determined by the components and their mass proportions. In particular, because exhaust gas recirculation amount is continually adjusted by the EGR valve, the air and exhaust gas mix proportion continually changes, so does specific heat. Therefore, in model calculations, too, it is necessary to continually calculate gas specific heat inside the inlet pipe.

Among the exhaust gases generated by combustion, only CO_2 (carbon dioxide) and H_2O (water) are considered, while NO_x (nitrous oxide), CO (carbon oxide) and HC (hydrocarbon) are in such small quantities in normal combustion that they have been excluded. In addition, the CO_2 , N_2 (nitrogen) and H_2O , which are generated by reaction of air (N_2 , O_2) and fuel (C_nH_m) at the stoichiometric air-fuel ratio, are together defined as combustion gases, and the gases handled by the model are of two types — combustion gas and air. Combustion gas components are assumed to be fixed, and that specific heat is calculated as follows (Equation 2).

$$C_{cmb} = K_{CO_2} \cdot CO_2 + K_{H_2O} \cdot H_2O + K_{N_2} \cdot N_2 \quad (2)$$

C_{cmb} : combustion gas specific heat
 CO_2 : Carbon dioxide's molecular weight
 H_2O : Water's molecular weight
 N_2 : Nitrogen's molecular weight
 K_{CO_2} , K_{H_2O} , K_{N_2} : constants

In this way, by handling the gases flowing in the engine as the two components of air and combustion gas, it is possible to reduce the load of computing load. The specific heat in any part of the system can be calculated in Equation 3.

$$c_{mix} = c_{air} \cdot (1-\lambda) + c_{cmb} \cdot \lambda \quad (3)$$

c_{mix} : Specific heat of mixture of air and combustion gases

c_{air} : Specific heat of air

c_{cmb} : Specific heat of combustion gases

λ : gas component ratio

3.3. Turbocharger

The turbocharger consists of two parts – the compressor that compresses the air, and the turbine that recovers the exhaust gas energy and converts it into rotation energy. In addition, the turbocharger revs are determined from the balance of turbine shaft torque received from the compressor's and turbine's expansion and compression work.

3.3.1. Compressor model

The compressor model calculates compressor upstream and downstream gas flows, component ratio, as well as downstream gas temperature and shaft torque necessary for compression, from the compressor upstream and downstream pressures, upstream gas temperature, and shaft rev can be calculated. Upstream gas flow rate is calculated from a 3D Characteristics Table according to the upstream and downstream compression ratio, as shown in Fig. 1-(a), and turbine revolution speed. Because the gas flow calculated from the characteristics table is volume flow rate on the upstream side, it is converted to mass flow rate according to upstream gas temperature and pressure. Leak is negligible for the compressor, thus gas flow both upstream and downstream, and component ratios, are the same (Equations 4, 5).

$$G_{C_in} = G_{C_out} = f(N_{tb}^*, \lambda_{C_in}, T_{C_in}, P_{C_in}, P_{C_out}) \quad (4)$$

$$\lambda_{C_out} = \lambda_{C_in} \quad (5)$$

G_{C_in} : compressor upstream gas flow

G_{C_out} : compressor downstream gas flow

N_{tb}^* : turbine revs

λ_{C_in} : compressor upstream gas component ratio

λ_{C_out} : compressor downstream gas component ratio

T_{C_in} : compressor upstream gas temperature

P_{C_in} : compressor upstream pressure

P_{C_out} : compressor downstream pressure

The asterisk (*) shown in Equation 4 indicates a parameter handled inside the model, and is used separately from the externally input and output parameters.

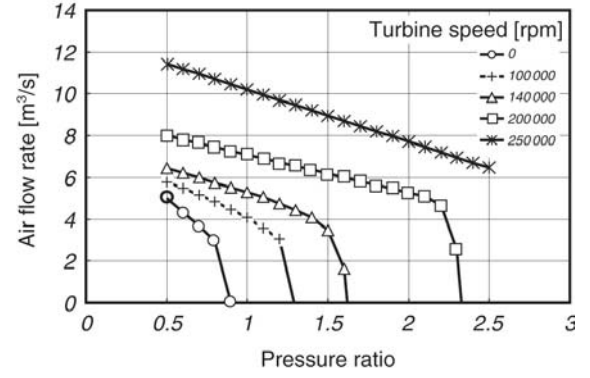
Downstream gas temperature rises due to compression, in reality there is a tendency for the temperature to be higher than of ideal adiabatic compression. As a non-dimensional value expressing this, the compressor adiabatic efficiency characteristics determined, as shown in Fig. 1-(b), by upstream and downstream compression ratio and gas flow calculated in (a) is used, and compressor downstream temperature is calculated (Equation 6).

$$T_{C_out} = f(\eta_{tb}^*, \lambda_{C_in}, T_{C_in}, P_{C_in}, P_{C_out}) \quad (6)$$

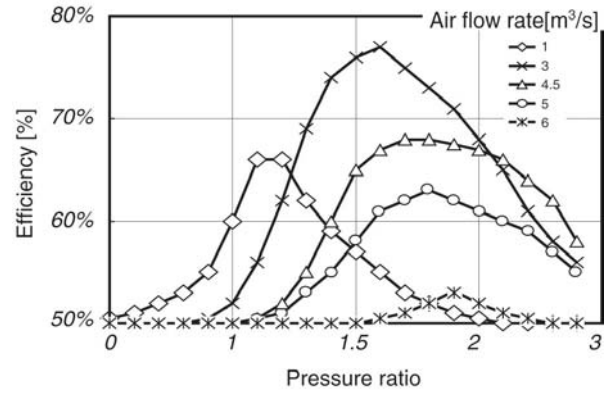
T_{C_out} : compressor downstream gas temperature

η_{tb}^* : compressor adiabatic efficiency

λ_{C_in} : compressor in-flow gas component ratio



(a) Air flow rate



(b) Efficiency

Fig. 1 Turbine Characteristic 3D Table

Considering mechanical energy used to compress the gas, based on the law of energy conservation to be equivalent to the difference of gas energy between compressor upstream and downstream, the shaft torque necessary for future compression is calculated (Equation 7). The upstream and downstream energies are determined from temperature, gas components, and the amount of gas passing through, used in Equations 4, 5, and 6.

$$M_C^* = f(N_{tb}^*, h_C^*) \quad (7)$$

M_C^* : shaft torque necessary for compression

h_C^* : compressor upstream and downstream gas energy difference

3.3.2. Turbine

It was assumed that the turbine is fitted with a variable nozzle, and calculates shaft torque obtained through turbine upstream gas flow, downstream gas temperature and expansion, from turbine upstream and downstream pressures, upstream gas temperature, shaft revs and nozzle opening.

Turbine upstream and downstream gas flow can be calculated according to the same characteristics table method used for the compressor, but as a consideration to reduce the amount of data, a physical formula was applied, based on Bernoulli's principle. In addition, in the same way as with the compressor, upstream and downstream gas flow and component ratios are assumed to be the same (Equations 7, 8).

$$G_{tb_in} = G_{tb_out} = f(A_{tb}, \lambda_{tb_in}, T_{tb_in}, P_{tb_in}, P_{tb_out}) \quad (7)$$

$$\lambda_{tb_in} = \lambda_{tb_out} \quad (8)$$

G_{tb_in} : turbine upstream gas flow

G_{tb_out} : turbine downstream gas flow

A_{tb} : turbine nozzle opening surface area

λ_{tb_in} : turbine upstream gas component ratio

λ_{tb_out} : turbine downstream gas component ratio

T_{tb_in} : turbine upstream gas temperature

P_{tb_in} : turbine upstream pressure

P_{tb_out} : turbine downstream pressure

In the same way as with the compressor, downstream gas temperature is calculated using the turbine adiabatic efficiency (Equation 9).

$$T_{tb_out} = f(\eta_{tb}^*, \lambda_{tb_in}, T_{tb_in}, P_{tb_in}, P_{tb_out}) \quad (9)$$

T_{tb_out} : turbine downstream gas temperature

η_{tb}^* : turbine adiabatic efficiency

Considering mechanical energy used to turn the turbine to be equivalent to the turbine upstream and downstream gas energy difference, the shaft torque necessary to turn the turbine is calculated (Equation 10).

$$M_{tb}^* = f(N_{tb}^*, h_{tb}^*) \quad (10)$$

M_{tb}^* : shaft torque obtained according to expansion

h_{tb}^* : turbine upstream and downstream gas energy difference

3.3.3. Rev calculation

Turbine revs are calculated from shaft torque difference determined by compressor and turbine, and from friction torque of the rotating shaft (Equation 11).

$$N_{tb}^* = K_N^* \cdot \int (M_{tb}^* - M_c^* - M_f^*) dt \quad (11)$$

M_f^* : turbine shaft friction torque

K_N^* : constant

3.4. Intercooler

As for the intercooler, downstream gas temperature is calculated from cooling efficiency, upstream gas flow, upstream gas temperature and external temperature (Equation 12).

$$T_{IC_out} = f(\eta_{IC}^*, G_{IC_in}, T_{IC_in}, T_a) \quad (12)$$

T_{IC_out} : intercooler downstream gas temperature

η_{IC}^* : cooling efficiency

G_{IC_in} : intercooler upstream gas flow

T_{IC_in} : intercooler upstream gas temperature

T_a : external temperature

Because there is no mass conversion in the intercooler, downstream gas flow and component ratio is equal to that upstream. In addition, ignoring pressure loss, there is no change between upstream and downstream pressures.

3.5. Throttle Valve, EGR

For both the throttle valve and EGR models, in the same way as the turbocharger's variable nozzle, upstream and downstream gas flows are calculated using Bernoulli's principle.

3.6. Inlet and Exhaust Pipes

Both the inlet pipe and the exhaust pipe can be considered as constant volume chambers where gases flow in and out. Temperature and pressure inside the pipes are calculated from gas energy in and out, and from mass balance (Equations 13-1 and 13-2).

$$P_{ch} = f(V_{ch}, \lambda_{ch_in}, \lambda_{ch_out}, G_{ch_in}, G_{ch_out}, T_{ch_in}, T_{ch}) \quad (13-1)$$

$$T_{ch} = f(V_{ch}, \lambda_{ch_in}, \lambda_{ch_out}, G_{ch_in}, G_{ch_out}, P_{ch}) \quad (13-2)$$

P_{ch} : pressure inside pipes

V_{ch} : volume inside pipes

λ_{ch_in} : component ratio of gas flowing into the pipe

λ_{ch_out} : component ratio of gas flowing out of the pipe

G_{ch_in} : gas flow into the pipe

G_{ch_out} : gas flow out of the pipe

T_{ch_in} : temperature of gas flowing into the pipe

T_{ch} : temperature in the pipes

Component ratio of gas flowing out of pipes is a complete mixture of air and combustion gas held in the constant volume chamber.

3.7. Cylinder

The cylinder model calculates cylinder in-flow gas flow, exhaust gas flow, temperature and component ratio, from cylinder in-flow gas pressure, temperature, and component ratio, and from fuel flow, and engine speed.

3.7.1. In-flow gas

Cylinder model is expressed as time-averaged constant flow without considering intermittent gas flow due to number of cylinders and piston reciprocating motion. In-flow gas flow is calculated from intake volume efficiency, intake gas pressure, temperature, and cylinder exhaust amount (Equation 14).

$$G_{cyl_in} = f(V_{cyl}^*, \eta_{v_cyl}^*, Ne, \lambda_{cyl_in}, T_{cyl_in}, P_{cyl_in}) \quad (14)$$

G_{cyl_in} : cylinder intake gas flow

V_{cyl}^* : engine displacement

$\eta_{v_cyl}^*$: intake volumetric efficiency

Ne : engine speed

λ_{cyl_in} : cylinder in-flow gas component ratio

T_{cyl_in} : cylinder in-flow gas temperature

P_{cyl_in} : cylinder in-flow gas pressure

3.7.2. Exhaust gas

Exhaust gas is calculated as the total of cylinder in-flow gas and fuel flow amount (Equation 15).

$$G_{cyl_out} = G_{cyl_in} + G_{fuel} \quad (15)$$

G_{cyl_out} : mass flow rate of exhaust gas

G_{fuel} : mass flow rate of injected fuel

Exhaust gas energy consists of intake gas energy and fuel energy being exhausted. Concerning thermal energy due to combustion, assuming that the water generated remains as vapor, a lower heat of combustion is used as the fuel generated energy for each unit weight. The exhaust temperature is calculated from the proportion of flow-in energy and exhaust gas energy, which is defined as the rate of exhaust loss (Equation 16).

$$T_{cyl_out} = f(\eta_{ex_cyl}^*, \lambda_{cyl_in}, G_{cyl_in}, G_{fuel}, T_{cyl_in}) \quad (16)$$

T_{cyl_out} : exhaust gas temperature

$\eta_{ex_cyl}^*$: exhaust loss rate

λ_{cyl_in} : cylinder in-flow gas component ratio

G_{cyl_in} : cylinder intake gas flow

G_{fuel} : fuel flow

T_{cyl_in} : cylinder in-flow gas temperature

If the amount of air in the intake gas is less than the minimum amount of required air for complete combustion, which is determined from fuel flow and ideal air-fuel ratio, excessive fuel is discharged as unburned fuel.

4. Verification Comparison with Actual Vehicle

4.1. Vehicle Engine System used for the Comparison

The vehicle engine system used at this time was one fitted with a turbocharger and EGR. Fig. 2 connects the models created as part units, and is a block diagram comprising the same system as the actual engine.

4.2. Environment for Verification Comparison with the Actual Vehicle

The verification comparison with an actual vehicle used a real-time method that compared the model calculations and the vehicle measurements, while the vehicle with the model installed is driven. Verification while driving has the advantage of carrying out an on-the-spot comparison under a variety of driving conditions.

Fig. 3 introduces the system used in the verification. The engine model described in the block diagram is created in a

personal computer (PC hereafter), and after converting it to a rapid prototyping system (RPS hereafter) executable file, it is downloaded to the RPS and executed. From the engine ECU, model input information such as the engine revs, EGR valve opening, etc., and data for actual vehicle comparison such as the intake air amount, etc., is given to the RPS. Furthermore, data that is not measured by the ECU, such as exhaust pressure and temperature, is input into the AD converter in the RPS, via the signal-processing device. The vehicle comparison data handled by the RPS can be monitored on the PC, along with the calculated model values.

Motorola's Power PC (100 MHz) was fitted to the RPS used, and it is possible to make calculations with floating point. The model's calculation load with the above system was such that model calculation time was approximately 2 msec, and calculation cycle 10 msec.

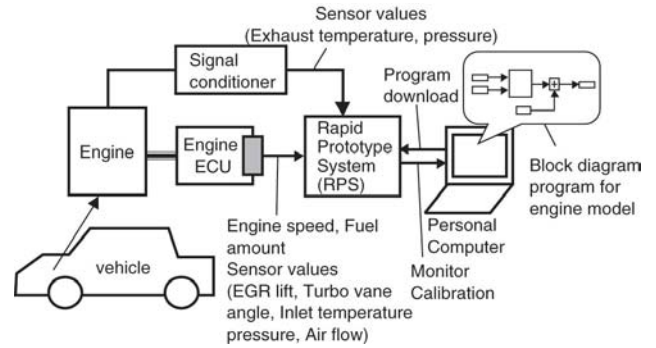


Fig. 3 Model verification system

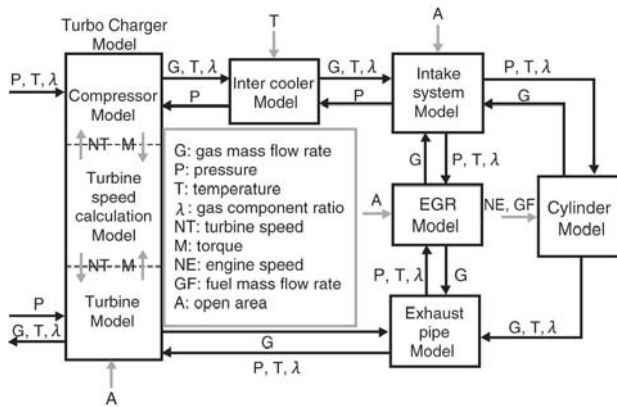


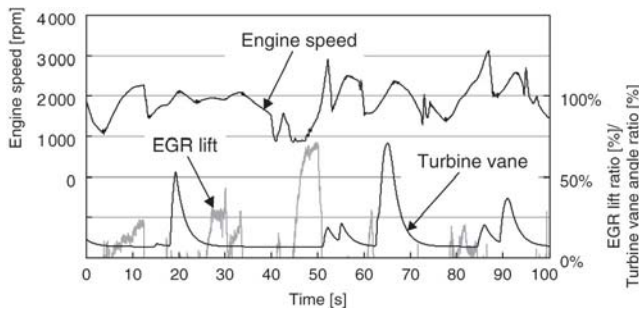
Fig. 2 Engine model block diagram

4.3. Results of Comparison Verification

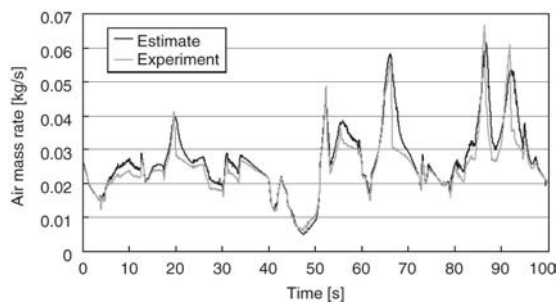
Fig. 4 shows an example of verification results. Vehicle driving was repetitions of arbitrary acceleration and deceleration. For model accuracy verification, a comparison was made of intake air amount, as well as pressure behavior inside inlet and exhaust pipes and temperature behavior.

Fig. 4(a) shows the behavior of engine revs, EGR valve opening, and turbocharger nozzle opening. These values are measured by the ECU, and become model calculation inputs. Fig. 4(b) is on the same time axis as Fig. 4(a), and concerning intake air amount, a comparison is made of actual vehicle and model-calculated behaviors. In addition, Fig. 4(c) shows a comparison of pressures in the inlet and exhaust pipes, and Fig. 4(d) the temperatures.

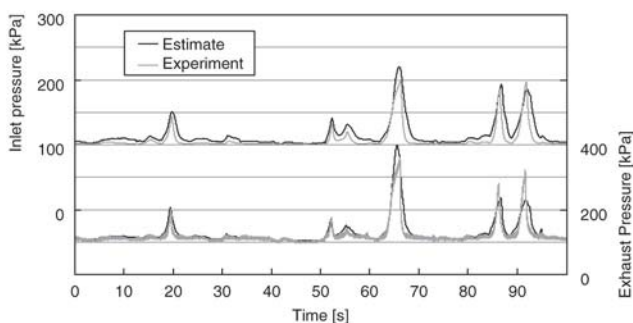
As shown in the comparison results, although when the load drops from a high load, the discrepancy of exhaust temperature is noticeable, as a simple model, gas behavior in various engine components can be roughly simulated. One cause of the discrepancy may be the fact that turbocharger characteristics accuracy, inlet and exhaust pipe radiation heat and their dynamic characteristics are not considered. Another potential cause of the discrepancy may result from measurement accuracy. Future improvements are planned.



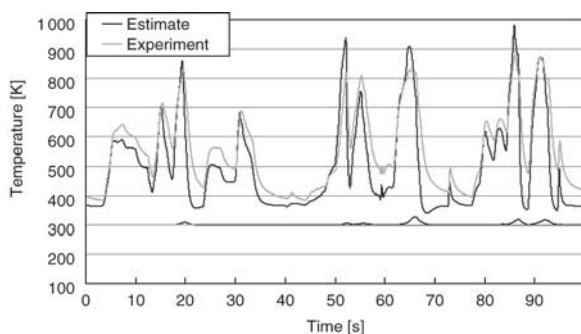
(a) Dynamic behavior of actual engine and actuator



(b) Air mass rate of engine intake



(c) Dynamic behavior of inlet and exhaust pressure



(d) Dynamic behavior of inlet and exhaust temperature

5. Summary

A model was created concerning gas flow in the engine, with the following main considerations: real-time calculation, a high degree of reusability, system performance predictions, and application to HILS. As for real-time calculation, the model was created on the assumption of lumped constant system modeling. For a high degree of reusability, in addition to part unit modeling, common input and output parameters were established for each part model, thereby eliminating the constraints in connections between models, and heightening the degree of reusability of the model for other systems. This was achieved through the application of the law of energy and mass conservation.

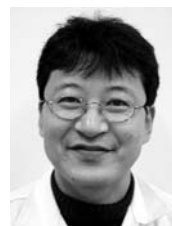
Model accuracy based on these concepts was verified through a real-time comparison between vehicle measurement and model calculation. The results showed that although there is still room for improvement in vehicle and calculation methods and accuracy, as a simple model it is possible to roughly simulate gas behavior in each part. The outlook is good for its use in outline system studies that do not require high precision, and for its application to HILS.

In order to apply this model as a study tool for a system requiring high accuracy, further model accuracy improvements are required. At the same time, it is possible to develop this model beyond gas flow and engine parts, therefore to include more parts for development work.

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Fig. 4 Test comparison