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Development of model-based control system for a low pressure loop EGR with a negative pressure control valve

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To improve the fuel economy, we developed a turbo-charged spark ignition engine combined with a low pressure loop EGR system. (Fig.1) In this system, a negative pressure control valve called by the admission valve (i.e. ADM valve) is applied to achieve high EGR ratio in wide engine operation condition. For a spark ignition engine where torque control is depended on air control, there is an issue concerning the accuracy of air control because the ADM valve is located in intake pipe. So ADM valve must be controlled coordinately with throttle valve and turbo wastegate valve which are conventional air control actuators.
In addition, LP-EGR system has transportation delay of EGR gas because of intake parts' volume. Therefore, on transient scene, it is necessary both to control EGR gas flow on required accuracy coordinately with ADM valve and to estimate transportation delay of EGR.

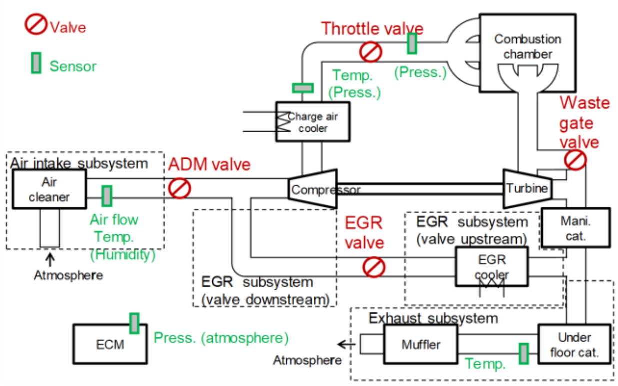

Fig.1 The schematic diagram of air path and EGR

To solve these issues, a new model-based control system for low pressure loop EGR with a negative pressure control valve was developed. This new control system has five main functions as followings, and the function 2) and 3) are essential to control ADM valve.

1) Calculation of required air and pressure corresponding to required torque 2) Estimation of temperature and pressure
3) Calculation of required negative pressure at ADM valve downstream 4) Calculation of required actuator motion,
5) Estimation of transportation delay of EGR gas

For the function 4), the following orifice valve formula is applied. The effective area introduced by this formula determine required valve motion.

$$\dot{m} = A * \frac{P_{us}}{\sqrt{T}} * \frac{f_{1(\kappa,R)}}{f_{2(\kappa,R,P_{ds}/P_{us})}}, \quad f_{1(\kappa,R)} = \sqrt{\frac{\kappa}{R}\left(\frac{2}{\kappa+1}\right)^{\frac{\kappa+1}{\kappa-1}}}, \quad f_{2\left(\kappa,R,\frac{P_{ds}}{P_{ds}}\right)} = \left(\frac{2}{\kappa+1}\right)^{\frac{1}{\kappa-1}}\left(\frac{P_{ds}}{P_{us}}\right)^{-\frac{1}{\kappa}}\sqrt{\frac{\kappa+1}{\kappa-1}\left(1-\left(\frac{P_{ds}}{P_{us}}\right)^{\frac{\kappa+1}{\kappa-1}}\right)}$$

$\dot{m}$: mass flow [kg/s] A: effective area [m^2] κ: specific heat [−] R: gas constant [J/(kg・K)]
P_{us}: upstream pressure [Pa] P_{ds}: downstream pressure [Pa] T: gas temperature [K]

For the function 2), a simple and practical model was developed. Especially, pressure estimation is characterized and uses the same orifice valve formula. Temperature estimation model is described by only 2 main parameters, which are EGR mass flow, and temperature of the coolant.

On a transient scene, accuracy of EGR control and estimation are validated (2nd drive pattern on LA4 mode, Fig.2). Estimated EGR rate includes transportation delay from EGR activation signal, and estimated profile is correspond to the measured one with a high accuracy. This means that not only the accuracy of estimation of transportation delay but also the accuracy of control on EGR gas flow are realized.

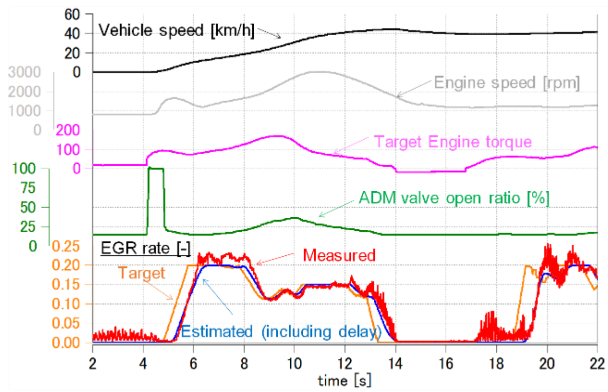

Fig.2 EGR control and estimation on transient scene

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