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ENGINE INDICATED TORQUE ESTIMATION OF A NATURALLY ASPIRED GASOLINE ENGINE

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ABSTRACT−The information of clutch transmitted torque is needed when controlling a clutch in a vehicle powertrain. However, engine indicated torque is a very important variable when estimating the clutch transmitted torque. The engine indicated torque is generally estimated as a function of engine rotational speed and the mass flow rate of air from an intake manifold into cylinders, which is referred as port air mass flow. The port air mass flow cannot be measured but should be estimated in a real vehicle. In previous studies, the port air mass flow was estimated using the information of the mass flow rate of air into the intake manifold through a throttle plate, which is referred as throttle air mass flow, and the information of the air pressure in the intake manifold together. However, this study proposes a new estimation method of the port air mass flow using only the information of the throttle air mass flow. In addition, the engine indicated torque is estimated using the estimated port air mass flow and compared with the measured value. The estimation method of the port air mass flow and engine indicated torque, presented in this study, is verified with production engine experiments.

KEY WORDS : Engine indicated torque, Engine torque, Naturally aspired engine, Gasoline engine, Mass flow rate

NOMENCLATURE

T_{ind} : engine indicated torque, N·m
H_u : fuel energy constant, N·m/kg
η_i : thermal efficiency multiplier, -
$\dot{m}_f$: fueling rate, kg/s
ω_e : engine crankshaft rotational speed, rad/s
$\dot{m}_{ao}$: port air mass flow, kg/s
λ : air fuel equivalent ratio, -
L_{th} : stoichiometric air fuel mass ratio, -
η_{vol} : volumetric efficiency, -
V_d : displacement volume of engine cylinders, m^3
P_{man} : pressure of the air in the intake manifold, N/m^2
T_{man} : temperature of the air in the intake manifold, K
V_{man} : volume of the intake manifold, m^3
R : ideal gas constant, N·m/kg·K
k : proportional constant, -
m_a : air mass in the intake manifold, kg
$\dot{m}_{ai}$: throttle air mass flow, kg/s
M_a : air mass in the intake manifold in the Laplace domain, -
M_{ai} : instantaneous sucked-air mass into the intake manifold through a throttle plate in the Laplace domain, -
M_{ao} : instantaneous sucked-air mass from the intake manifold into cylinders in the Laplace domain, -
x_{egr} : EGR rate, -
$\dot{m}_{egr}$: mass flow rate of air into the intake manifold through EGR passage, kg/s
$\dot{m}_{fresh}$: air mass flow rate fresh air from the intake manifold into cylinders, kg/s
T_{fric} : engine friction torque, N·m
i_{ISG} : gear ratio of the ISG motor, -
T_{ISG} : ISG motor torque, N·m
T_{dyno} : dynamometer torque, N·m
J_e : engine inertia, kg·m^2
J_{ISG} : ISG motor inertia, kg·m^2
J_{dyno} : dynamometer inertia, kg·m^2
J_{eq} : equivalent inertia of rotational parts, kg·m^2

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1. INTRODUCTION

Vehicle environmental regulations are strengthening all over the world, and vehicles are getting intelligent due to electronization. As a result, interest in improving vehicle efficiency and ride comfort has been increased.

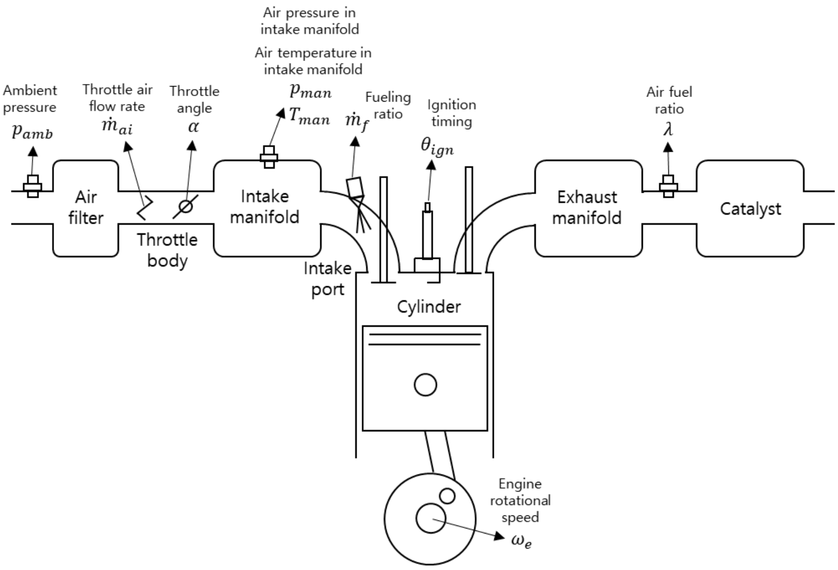

Figure 1. Schematic diagram of air flow line of a non-EGR, NA gasoline engine

Appropriate control of a clutch in a vehicle powertrain can play a major role in improving vehicle efficiency and ride comfort.

The clutch friction energy loss and vehicle jerk are influenced by clutch transmitted torque during slip engagement. Therefore, precise control of the clutch torque during the slip engagement is necessary to reduce the clutch friction energy loss and vehicle jerk(Minowa et al., 1999, van Berkel et al., 2014). However, in order to precisely control the clutch torque, measurement feedback of the clutch torque is required. However, since torque measuring sensors are expensive and bulky, production vehicles do not have torque sensors for measuring the clutch torque. Thus, the clutch torque is generally controlled based on an estimator(Kim et al., 2017, Oh et al., 2017b, Kim et al., 2018).

On the other hand, in order to estimate the clutch torque, the information of engine indicated torque generated by fuel combustion of an engine is necessarily required(Oh and Choi, 2015, Oh et al., 2017a, Kim and Choi, 2018). In this paper, the method of estimating the engine indicated torque of a non-Exhaust Gas Recirculation (non-EGR), naturally aspiration (NA) engine will be discussed.

In previous studies, there are a few methods to estimate the engine indicated torque: a mothod using a Mean Value Engine Model (MVEM)(Cho and Hedrick, 1989, Hendricks and Sorenson, 1990, Hendricks et al., 1996, Smith et al., 1999, Chevalier et al., 2000), a method using the information of engine block acceleration such as knock sensor information(Azzoni, 1997, Azzoni et al., 1998, Gao and Randall, 1999, Siano and D'Agostino, 2014), a method using engine driveline dynamics and engine output torque information(Rizzoni and Zhang, 1994, Rizzoni et al., 1995, Kim et al., 1999, Potenza et al., 2007, Na et al., 2018, Yar et al., 2018), and so on(Maass et al., 2011, Shamekhi and Shamekhi, 2015). Among the methods mentioned, the method using MVEM is a good way to analyze average engine behaviors such as the engine indicated torque during multiple cycles of an engine, and the other two methods are a good way to analyze instantaneous engine behaviors such as the ignition state within one cycle of an engine. In this study, MVEM was used to estimate the engine indicated torque.

Fig. 1 shows the schematic diagram of air flow line of a non-EGR, NA gasoline engine. In MVEM of a non-EGR, NA gasoline engine, the engine indicated torque is generally estimated as a function of engine rotational speed and mass flow rate of air from an intake manifold into cylinders, which is shortly referred as port air mass flow. But, in a non-EGR, NA gasoline engine, the port air mass flow is not measured, but estimated. In this time, the mass flow rate of air into the intake manifold through a throttle plate, which is measured, is generally utilized to estimate the port air mass flow. However, since there is the intake manifold between a throttle body and intake port, the mass flow rate of air into the intake manifold through a throttle plate, which is referred shortly as throttle air mass flow, and the port air mass flow may differ from each other depending on the variation of the air mass in the intake manifold(Taylor, 1985, Rajamani, 2011, Eriksson and Nielsen, 2014).

Here, the method of estimating the port air mass flow and engine indicated torque, by assuming that the port air mass flow and throttle air mass flow are equal to each other, ignoring the variation of the air mass in the intake manifold, is referred to as a steady state estimation method. On the other hand, the method of estimating the port air mass flow and engine indicated torque, considering the variation of the air mass in the intake manifold, is referred to as a transient state estimation method.

In the transient state method of previous studies, the port air mass flow was estimated using the information of the throttle air mass flow and the information of the air pressure in the intake manifold together(Cho and Hedrick, 1989, Hendricks and Sorenson, 1990, Hendricks et al., 1996, Smith et al., 1999, Chevalier et al., 2000). On the other hand, it is not necessary to use both the throttle air mass flow sensor and the air pressure sensor in the intake manifold as an engine load measurement sensor when controlling the engine air-fuel ratio. Furthermore, it is not cost effective to use both sensors to estimate the engine indicated torque.

Therefore, this paper presents a new model of air flow dynamics in the intake manifold of a non-EGR, NA gasoline engine and a new estimation method of the port air mass flow using only the information of the throttle air mass flow. Finally, this paper presents a new transient state estimation method of the engine indicated torque using the estimated port air mass flow. A new transient estimation method is verified by comparing the estimated engine indicated torque with the measured one.

This paper is organized as follows: Section 2 explains existing MVEM, a new model of air flow dynamics in the intake manifold, a new estimation method of the port air mass flow and engine indicated torque. Section 3 analyzes the estimation result of the port air mass flow and the engine indicated torque by existing methods and the new method. And, section 4 concludes this paper.

2. MEAN VALUE ENGINE MODEL

This section introduces MVEM of non-EGR, NA engine(Taylor, 1985, Rajamani, 2011, Eriksson and Nielsen, 2014), and presents a new model of air flow dynamics in the intake manifold. In addition, this section explains a new estimation method of the mass flow rate of air from the intake manifold into cylinders, which is shortly referred as port air mass flow, and engine indicated torque.

2.1. Engine indicated torque

Since the power generated by an engine is proportional to the amount of fuel consumed in cylinders, the following torque model can be derived.

$$T_{ind} = \frac{H_u \eta_i \dot{m}_f}{\omega_e} \tag{1}$$

Where T_{ind}, H_u, η_i, $\dot{m}_f$, and ω_e represent engine indicated torque, fuel energy constant, thermal efficiency multiplier, fueling rate, and engine crankshaft rotational speed, respectively.

Since the amount of fuel consumed in cylinders is proportional to the amount of air sucked into cylinders, the following equation can be derived.

$$\dot{m}_f = \frac{\dot{m}_{ao}}{\lambda(\omega_e, \dot{m}_{ao}) L_{th}} \tag{2}$$

Where $\dot{m}_{ao}$, λ, and L_{th} represent port air mass flow, air/fuel equivalence ratio, and stoichiometric air fuel mass ratio, respectively.

Combining equation (1), and (2), the engine indicated torque of a NA gasoline engine can be expressed as a function of engine speed and port air mass flow.

$$T_{ind}(\omega_e, \dot{m}_{ao}) = \frac{H_u \eta_i \dot{m}_{ao}}{L_{th} \omega_e \lambda(\omega_e, \dot{m}_{ao})} \tag{3}$$

The method of estimating the engine indicated torque differs according to the method of estimating the port air mass flow. So, the methods of estimating the port air mass flow are introduced in the next subsections.

2.2. Port air mass flow

The port air mass flow can be modeled as follows using the concept that an engine sucks the air in the amount of engine displacement per two engine revolutions.

$$\dot{m}_{ao} = \eta_{vol} \frac{\omega_e}{4\pi} V_d \frac{P_{man}}{RT_{man}} \tag{4}$$

Where η_{vol}, V_d, P_{man}, T_{man}, and R represent volumetric efficiency, displacement volume of engine cylinders, pressure of the air in the intake manifold, temperature of the air in the intake manifold, constant used in ideal gas law for the intake manifold, respectively.

Here, a new simple model of the port air mass flow is presented, and this is utilized to induce a new method estimating the port air mass flow using only the information of the mass flow rate of air into the intake manifold through a throttle plate, which is short referred as throttle air mass flow, in the later subsection. A new model of the port air mass flow is derived from the relationship like below.

$$V_d = \frac{V_{man}}{k} \tag{5}$$

Where V_{man}, and k represent volume of the intake manifold, and proportional constant, respectively.

Substituting equation (5) into (4), and using the ideal gas equation of the air in the intake manifold, the following new model of the port air mass flow can be derived. The following equation is utilized to derive a new method estimating the port air mass flow.

$$\dot{m}_{ao} = \eta_{vol} \frac{\omega_e}{4\pi} \frac{V_{man}}{k} \frac{P_{man}}{RT_{man}} = \frac{\eta_{vol}\omega_e}{4k\pi} m_a \tag{6}$$

Where m_a represents air mass in the intake manifold.

2.3. Air pressure in the intake manifold

The rate of change of the air mass in the intake manifold can be expressed as follows using the ideal gas equation. At this time, it is assumed that the temperature of the air in the intake manifold hardly changes.

$$\dot{m}_a = \frac{\dot{P}_{man} V_{man}}{RT_{man}} \tag{7}$$

Using the law of conservation of the air mass in the intake manifold, the rate of change of the air mass in the intake manifold can again be expressed as follows.

$$\dot{m}_a = \dot{m}_{ai} - \dot{m}_{ao} \tag{8}$$

Where $\dot{m}_{ai}$ represents throttle air mass flow.

In the introduction section, the steady state and transient state estimation method of the engine indicated torque are explained. In the steady state estimation method of the engine indicated torque, it is assumed that the rate of change of the air mass in the intake manifold is 0 as follows.

$$\dot{m}_{ao} = \dot{m}_{ai} \tag{9}$$

2.4. Estimation method of port air mass flow

In previous studies(Cho and Hedrick, 1989, Hendricks and Sorenson, 1990, Hendricks et al., 1996, Smith et al., 1999, Chevalier et al., 2000), the port air mass flow was estimated using the following equation by combining equation (7) and (8).

$$\dot{m}_{ao} = \dot{m}_{ai} - \frac{\dot{P}_{man} V_{man}}{RT_{man}} \tag{10}$$

This uses two information to estimate the port air mass flow: the throttle air mass flow and the air pressure in the intake manifold. As described in the introduction section, it is not necessary to use the information of the throttle air mass flow and the information of the air pressure in the intake manifold together when controlling the engine air-fuel ratio. Moreover, it is not cost effective to install both sensors in an engine in order to estimate the engine indicated torque.

Therefore, in this paper, a method estimating the port air mass flow, considering the variation of the air mass in the intake manifold, using only the information of the throttle air mass flow, is proposed. The following equation can be derived, combining equation (6) and (8).

$$\dot{m}_a + \frac{\eta_{vol}\omega_e}{4k\pi} m_a = \dot{m}_{ai} \tag{11}$$

The following equation can be obtained, transforming the equations (8) and (11) into the Laplace domain.

$$\mathrm{sM}_a(\mathrm{s}) = \mathrm{sM}_{ai}(\mathrm{s}) - \mathrm{sM}_{ao}(\mathrm{s}) \tag{12}$$

$$M_a(s) = \frac{s}{s + \frac{\eta_{vol}\omega_e}{4k\pi}} M_{ai}(s) \tag{13}$$

Where M_a, M_{ai}, and M_{ao} represent air mass in the intake manifold in the Laplace domain, instantaneous sucked-air mass into the intake manifold through a throttle plate in the Laplace domain, and instantaneous sucked-air mass from the intake manifold into cylinders in the Laplace domain, respectively.

Combining the above two expressions, it is possible to derive a new estimation equation of the port air mass flow as follows in which the variation of the air mass in the intake manifold is considered. In the new method, only the information of the throttle air mass flow is utilized to estimate the port air mass flow.

$$\dot{m}_{ao} = \frac{\frac{\eta_{vol}\omega_e}{4k\pi}}{s + \frac{\eta_{vol}\omega_e}{4k\pi}} \dot{m}_{ai} \tag{14}$$

In the above equation, it can be seen that the port air mass flow of a non-EGR, NA gasoline engine is a low pass filtered value of the throttle air mass flow and the cut-off frequency of the low pass filter is proportional to engine speed.

Here, there are three methods mentioned above for estimating the port air mass flow of a non-EGR, NA gasoline engine: equation (9), (10), and (14). And, the engine indicated torque can be estimated by a torque map which is referred in equation (3) using each estimated port air mass flow. Hereinafter, the method of estimating the port air mass flow and engine indicated torque using equation (9) is referred to as the steady state estimation method, the method of estimating the port air mass flow and the engine indicated torque using equation (10) is referred to as the transient state estimation method1, and the method estimating the port air mass flow and engine indicated torque using equation (14) is referred to as the transient state estimation method2.

The mentioned estimation methods of the port air mass flow and engine indicated torque are compared with each other in the later result section.

2.4. EGR case

In the previous subsections, the new estimation method of the port air mass flow and engine indicated torque, of a non-EGR, NA gasoline engine, were discussed. On the other hand, by a slight modification of the estimation method of the port air mass flow and engine indicated torque, of a non-EGR, NA gasoline engine, it can be shown mathematically that the estimation method can be extended to an EGR, NA gasoline engine.

The EGR engine has an EGR passage connected to the intake manifold from exhaust manifold. At this time, the air sucked from the EGR passage does not affect the generation of the engine indicated torque, and only the new air sucked from throttle influences the generation of the engine indicated torque. In general, the EGR rate and the mass flow rate of the fresh air from the intake manifold into cylinders can be expressed as follows(Azzoni et al., 1997, Fons et al., 1999, Eriksson and Nielsen, 2014).

$$x_{egr} = \frac{\dot{m}_{egr}}{\dot{m}_{ai} + \dot{m}_{egr}} \quad (15)$$

$$\dot{m}_{fresh} = (1 - x_{egr})\dot{m}_{ao} \quad (16)$$

Where x_{egr}, $\dot{m}_{egr}$, and $\dot{m}_{fresh}$ represent EGR rate, mass flow rate of air into the intake manifold though EGR passage, and mass flow rate of fresh air from the intake manifold into cylinders, respectively. The mass flow rate of fresh air from the intake manifold into cylinders is shortly referred as port fresh air mass flow.

In the case of an EGR engine, by modifying equation (3), the engine indicated torque can be expressed as follows.

$$T_{ind} = T_{ind}(\omega_e, \dot{m}_{fresh}) \quad (17)$$

In addition, the air in the intake manifold of an EGR engine can be divided into two types of the air: the fresh air sucked from throttle, and the EGR air sucked from the EGR passage. Modifying equation (7) and (8), the following equation can be derived, considering only the fresh air flow in the intake manifold.

$$(1 - x_{egr})\dot{m}_a = \frac{(1 - x_{egr})\dot{P}_{man}V_{man}}{RT_{man}} \quad (18)$$

$$(1 - x_{egr})\dot{m}_a = \dot{m}_{ai} - (1 - x_{egr})\dot{m}_{ao} \quad (19)$$

Also, the following equation can be obtained by substituting equation (6) into (19).

$$\dot{m}_a + \frac{\eta_{vol}\omega_e}{4k\pi} m_a = \frac{1}{(1 - x_{egr})}\dot{m}_{ai} \quad (20)$$

As in the previous non-EGR case, transforming equation (19) and (20) into the Laplace domain and combining the two results, a new estimation equation of the port fresh air mass flow can be derived as follows.

$$\dot{m}_{fresh} = (1 - x_{egr})\dot{m}_{ao} = \frac{\frac{\eta_{vol}\omega_e}{4k\pi}}{s + \frac{\eta_{vol}\omega_e}{4k\pi}}\dot{m}_{ai} \quad (21)$$

Here, it can be shown that equation (21) is the same as equation (14) and it can be seen mathematically that the port fresh air mass flow can be estimated by the low pass filtering method even in an EGR case. The cut-off frequency of the low pass filter is proportional to engine speed.

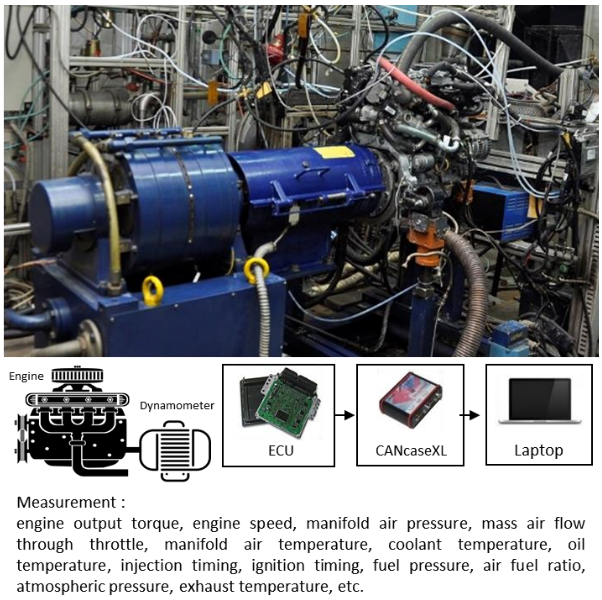

Figure 2. Experimental engine and scematic diagram of the data flow during engine experiments

3. EXPERIMENTAL RESULTS

In this section, the estimation results of the port air mass flow, by the proposed method in this paper are compared with the results by the conventional methods. Also, the engine indicated torque is again estimated using each estimated port air mass flow and the results are compared with the measured engine indicated torque.

3.1. Experimental environment

Fig. 2 shows an experimental engine and schematic diagram of the data flow during engine experiments. The displacement volume of the experimental engine was 2400cc and the engine was non-EGR, NA, and Multi-Point Injected engine. A motor for idle stop and go(ISG) was connected to the crankshaft in parallel and utilized to start and stop the engine. ISG motor torque was estimated accurately using motor current. Engine output torque was measured by a dynamometer which is connected in the crankshaft in series. And, engine speed, fuel injection timing, and ignition timing were measured by a crankshaft position sensor which is installed in the crankshaft. The air pressure in the intake manifold was measured by a manifold air pressure sensor which is installed in the intake manifold, and the throttle air mass

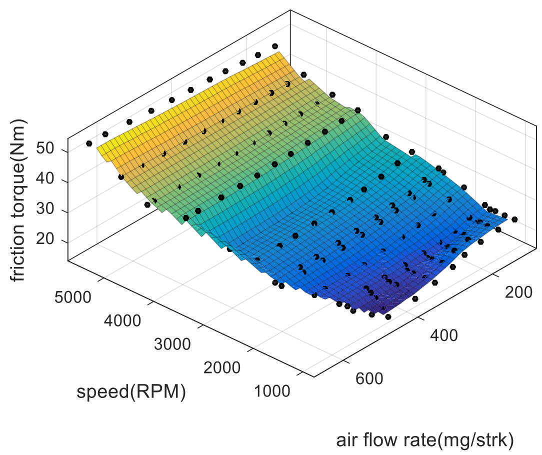

Figure 3. Engine friction torque map

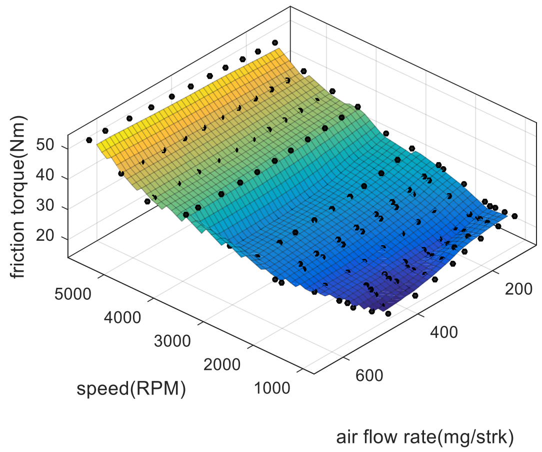

Figure 4. Engine indicated torque map

flow was measured by a mass air flow sensor, which is installed in the position between an air filter and throttle plate. The air temperature in the intake manifold, coolant temperature, oil temperature, fuel pressure were measured by temperature and pressure sensors which were installed in related positions. All measurements were logged to a laptop computer via CAN communication.

3.2. Steady state experiment

First, steady state engine experiments were conducted to make an engine friction torque map and engine indicated torque map which are a function of engine speed and port air mass flow. In the steady state, it is considered that the port air mass flow is equal to the throttle air mass flow.

Fig. 3 shows the engine friction torque map which is a function of engine speed and port air mass flow. The engine friction torque map was made by measuring the

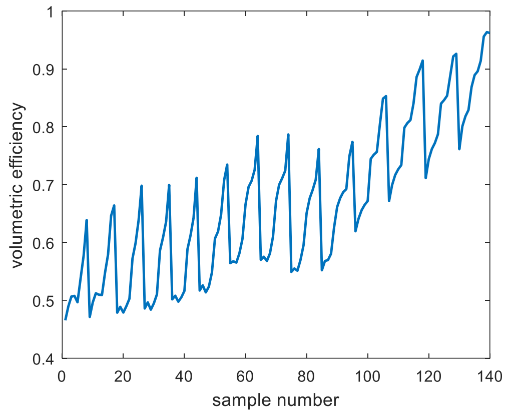

Figure 5. Volumetric efficiency

throttle air mass flow and dynamometer torque at a specific engine speed without burning the fuel.

Fig. 4 shows the engine indicated torque map which is a function of engine speed and port air mass flow, described in equation (3). The engine indicated torque map was made by measuring the throttle air mass flow and dynamometer torque at specific engine speed while burning the fuel, and by adding the engine friction torque to the measured dynamometer torque.

The method of estimating the engine indicated torque differs according to the estimation method of the port air mass flow, which is used in the mentioned relationship map.

Hereinafter, to compare each method, the method of estimating the port air mass flow and engine indicated torque using equation (9) is referred to as the steady state estimation method, the method of estimating the port air mass flow and engine indicated torque using equation (10) is referred to as the transient state estimation method1, and the method of estimating the port air mass flow and engine indicated torque using equation (14) is referred to as the transient state estimation method2.

The method proposed in the previous study using equation (10) uses the information of the throttle air mass flow and the information of the air pressure in the intake manifold together to estimate the port air mass flow. But, the method proposed in this study using equation (14) uses only the information of the throttle air mass flow.

3.3. Transient state experiment

This section analyzes the results of estimating the port air mass flow by the steady state and transient state method, and the results of estimating the engine indicated torque using each estimated port air mass flow. In transient state engine experiments, the process of

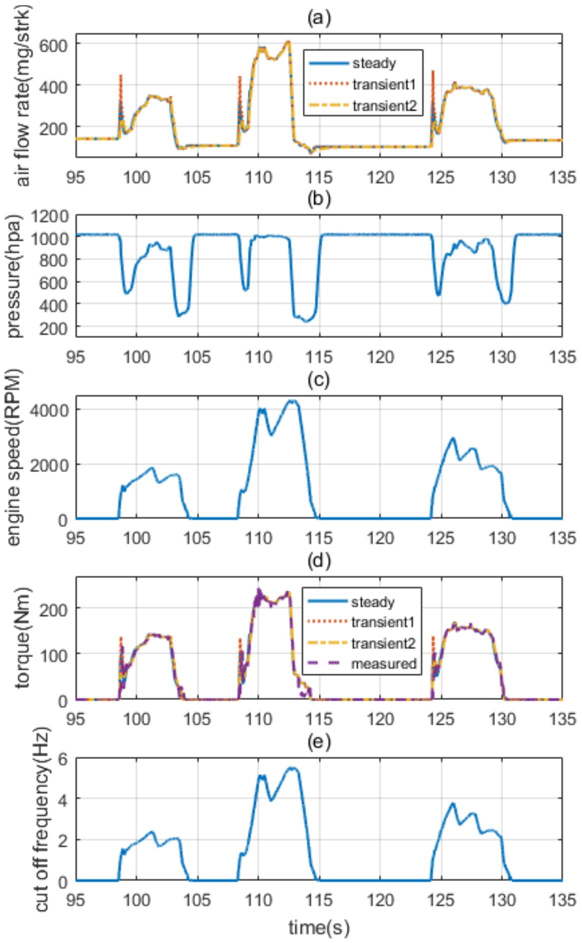

Figure 6. Measurements and estimates during the transient engine experiments: (a) the estimated port air mass flow, (b) the air pressure in the intake manifold, (c) the engine speed, (d) the estimated engine indicated torque, (e) the cut off frequency of the low pass filter in the transient state method2

increasing and decreasing engine speed was repeated several times.

Volumetric efficiency used in the transient state method2 for estimating the port air mass flow can be calculated using equation (4) in the steady state engine experiments. In the steady state, the throttle air mass flow can be regarded as the port air mass flow, so all variables in the left side and the right side of equation (4) except for the volumetric efficiency can be measured.

Fig. 5 shows the volumetric efficiency for different engine speed and port air mass flow. The volumetric efficiency in the transient state method2 should be calculated as a function of engine speed and port air mass flow. However, in this study, the value of 0.9634 was utilized as the volumetric efficiency in order to consider the phase lag during signal measurement of the throttle air mass flow. This value is the maximum value of the volumetric efficiency obtained from the steady state engine experiments. The larger the volumetric efficiency is, the larger the cut-off frequency of the low pass filter in the transient state estimation method2 is. This decreases the phase lag of the port air mass flow.

The mass air flow sensor used in this study was a hot wire type, and a hot wire type sensor measures the amount of mass flow rate of air by measuring the amount of temperature change of the hot wire as the air passes through the hot wire. Therefore, the phase lag occurs when the mass flow rate signal is read due to the measurement principle of the sensor.

Also, the intake manifold volume is used in the transient state estimation method1. The value of 1200cc was used as the intake manifold volume, referring to the intake manifold volume in the design process.

The measured engine indicated torque was calculated using the equation like below.

$$T_{ind} = -i_{ISG}T_{ISG} + T_{fric} + T_{dyno} + J_{eq}\dot{\omega}_e \quad (22)$$

$$J_{eq} = J_e + i_{ISG}^2 J_{ISG} + J_{dyno} \quad (23)$$

Where, T_{fric}, i_{ISG}, T_{ISG}, T_{dyno}, J_e, J_{ISG}, J_{dyno}, and J_{eq} represent the engine friction torque which is calculated using the engine friction torque map in Fig. 3, gear ratio of the ISG motor, ISG motor torque, measured dynamometer torque, engine inertia, ISG motor inertia, dynamometer inertia, and equivalent inertia of rotational parts. The value of 0.09 was used as the equivalent inertia in this study.

The noise on the signal of the derivative of the engine speed was eliminated using a zero-phase digital filter in the data post-processing. The cut-off frequency of the zero-phase digital filter was 15 Hz.

Fig. 6 shows the measurements and estimates during the transient engine experiments. Fig. 6(a), (b), (c), (d), and (e) show the estimated port air mass flow, air pressure in the intake manifold, engine speed, estimated engine indicated torque, and cut off frequency of the low pass filter in the transient state method2. In the legends, steady means the result using the steady state estimation method, and transient1 and transient2 mean the result using the transient state estimation method1 and transient state estimation method2, respectively. Also, measured means the measurement.

Enlarged figures of Fig. 6(a), and Fig. 6(d) will be shown later. As can be seen in equation (14), the cut off frequency of the low-pass filter in the transient state estimation method2 is proportional to the engine speed. This is well shown in Fig. 6(c), and Fig. 6(e).

Fig. 7(a), (b), and (c) are enlarged figures of Fig. 6(a) for a specific time, Fig. 7(d) is an enlarged figure of Fig. 6(b) for a specific time, and Fig. 7(e) is an enlarged figure of Fig. 6(c) for a specific time. In the legend of Fig. 7 and the following figures, phase means the period when the engine speed is over 1000 RPM.

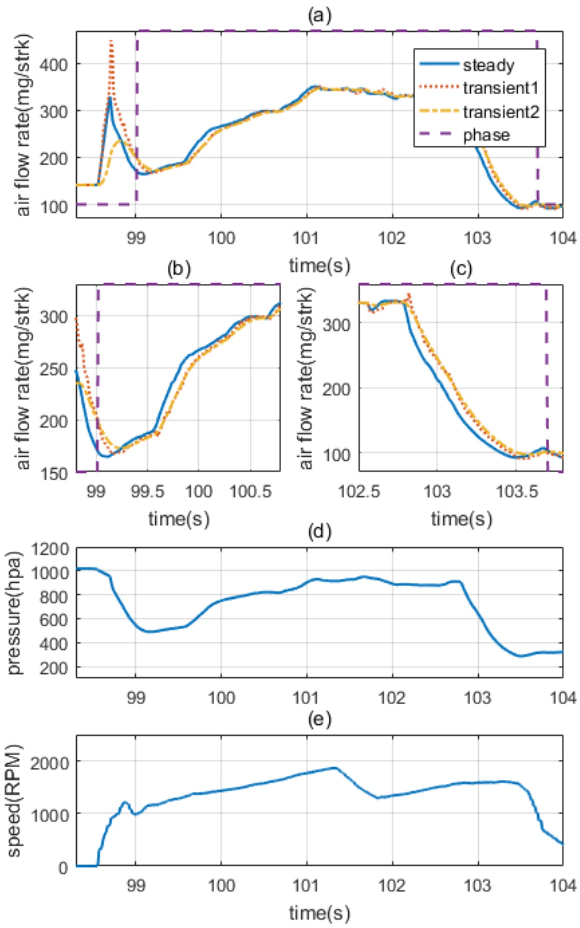

Figure 7. Enlarged figures#1 of measurements and estimatesduring the transient engine experiments: (a),(b),(c) theestimated port air mass flow, (d) the air pressure in the intakemanifold, (e) the engine speed

In this study, the port air mass flow was not measured. Therefore, the engine indicated torque was firstly estimated using the estimated port air mass flow by each method, and the estimation result of the port air mass flow was verified indirectly by comparing the estimated engine indicated torque and the measured engine indicated torque. The measured engine indicated torque was calculated by adding the engine friction torque to the measured dynamometer torque.

Referring to the estimated port air mass flow by each method in Fig. 7(a), (b), and (c), it can be seen that the estimated port air mass flow by the transient state method1 and transient state method2 is phase lagged compared with the estimated port air mass flow by the steady state method. This result can be explained with the common sense that a reservoir tank with a volume in the middle of the pipe in the hydraulic and pneumatic flow serves as a low pass filter.

However, in Fig, 7(a), (b), and (c), it can be seen that the estimated result of the port air mass flow using the

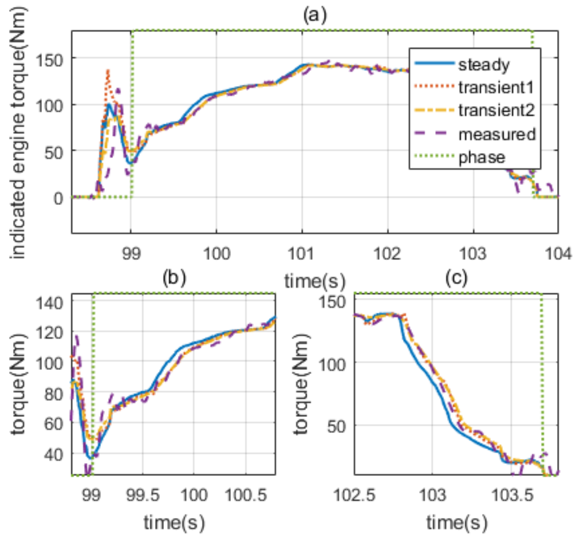

Figure 8. Enlarged figures#2 of measurements and estimates during the transient engine experiments: (a),(b), and (c) the estimated engine indicated torque

transient state method2 is almost the same as the estimated result using the transient state method1 although the transient state method2 uses only the information of the throttle air mass flow but the transient state method1 uses the information of the throttle air mass flow and the information of the air pressure in the intake manifold together.

This means that the information of the air pressure in the intake manifold is not needed to estimate the port air mass flow and engine indicated torque. The development of this method is a major contribution of this study.

Fig. 8(a), (b), and (c) are enlarged figures of Fig. 6(d) for a specific time and show the estimated engine indicated torque using the estimated port air mass flow by each method. The time of x-axis in Fig. 7 and Fig. 8 is the same.

Referring to the period when the engine speed is larger than 1000 RPM in Fig. 8, it can be seen that the engine indicated torque estimated by the transient state method1 and transient method2 was phase lagged compared with the engine indicated torque estimated by the steady state method. Also, the engine indicated torque estimated by the transient state method1 and transient method2 was closer than the engine indicated torque estimated by the steady state method to the measured engine indicated torque. As a result, it can be indirectly proved that the port air mass flow estimated by the transient state method1 and transient state method2 is closer to the actual value than the port air mass flow estimated by the steady state method1.

Fig. 9(a), (b), and (c) are enlarged figures of Fig. 6(a) for a specific time, Fig. 9(d) is an enlarged figure of Fig. 6(b) for a specific time, and Fig. 9(e) is an enlarged

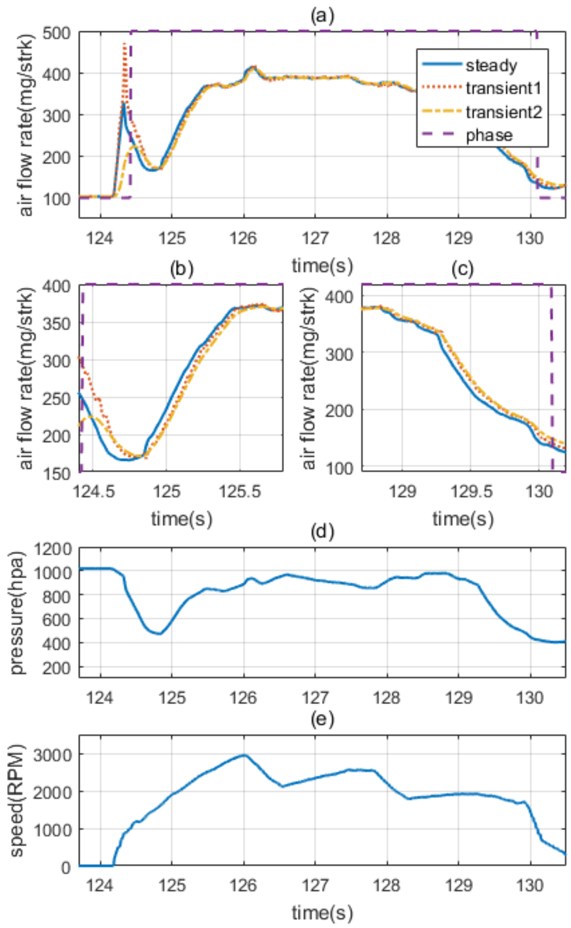

Figure 9. Enlarged figures#3 of measurements and estimates during the transient engine experiments: (a),(b),(c) the estimated port air mass flow, (d) the air pressure in the intakemanifold, (e) the engine speed

figure of Fig. 6(c) for a specific time. The time of x-axis in Fig. 7 and Fig. 9 is different.

Referring to the estimated port air mass flow by each method in Fig. 9(a), (b), and (c), it can be also seen that the estimated port air mass flow by the transient state method1 and transient state method2 is phase lagged compared with the estimated port air mass flow by the steady state method.

However, in Fig, 9(a), (b), and (c), it can be also seen that the estimated result of the port air mass flow using the transient state method2 is almost the same as the

Table 1. RMS error between the estimated engine indicated torque and the measured torque

Method	RMS error (Nm)
Steady state method	3.1701
Transient state method1	2.7673
Transient state method2	2.7698

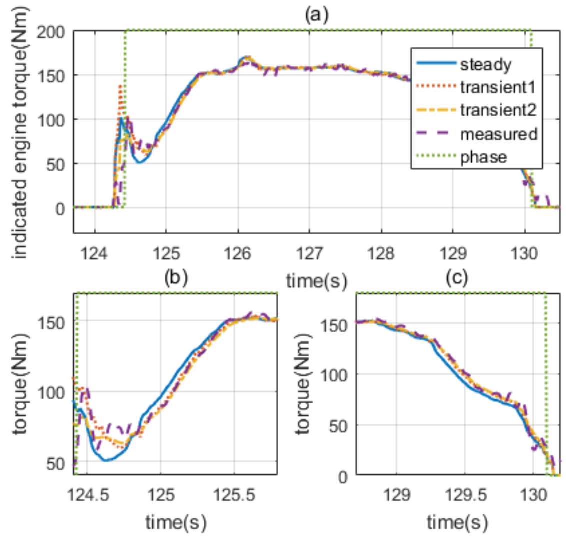

Figure 10. Enlarged figures#4 of measurements and estimates during the transient engine experiments: (a),(b), and (c) the estimated engine indicated torque

estimated result using the transient state method1 although the transient state method2 uses only the information of the throttle air mass flow but the transient state method1 uses the information of the throttle air mass flow and the information of the air pressure in the intake manifold together.

Fig. 10(a), (b), and (c) are enlarged figures of Fig. 6(d) for a specific time and show the estimated engine indicated torque using the estimated port air mass flow by each method. The time of x-axis in Fig. 9 and Fig. 10 is the same.

Referring to the period when the engine speed is larger than 1000 RPM in Fig. 8, it can be also seen that the engine indicated torque estimated by the transient state method1 and transient method2 was phase lagged compared with the engine indicated torque estimated by the steady state method. Also, the engine indicated torque estimated by the transient state method1 and transient method2 was closer than the engine indicated torque estimated by the steady state method to the measured engine indicated torque. As a result, it can be also indirectly verified that the port air mass flow estimated by the transient state method1 and transient state method2 is closer to the actual value than the port air mass flow estimated by the steady state method1.

Table 1 shows the RMS error between the measured engine indicated torque and the engine indicated torque estimated by each method during the transient state experiment. In Table 1, it can be known that the RMS error by the transient state method1 decreased by around 12 percent compared with the RMS error by the steady state method, also the RMS error by the transient state method2 decreased by around 12 percent compared with the RMS error by the steady state method. The amount of RMS error decrease by the transient state method1 and transient state method2 was almost close to each other.

As a result, it can be seen that the variation of the air mass in the intake manifold can be considered without using the information of the air pressure in the intake manifold when the port air mass flow and engine indicated torque are estimated.

3. CONCLUSION

This paper presented a new model of air flow dynamics in the intake manifold of a non-EGR, NA gasoline engine and a new estimation method of the mass flow rate of air from the intake manifold into cylinders, which is referred as port air mass flow, using only the information of the mass flow rate of air from the intake manifold through a throttle plate, which is referred throttle air mass flow. Also, this paper presented a new transient state estimation method of the engine indicated torque using the estimated port air mass flow. In previous studies, the information of the throttle air mass flow and air pressure in the intake manifold were utilized together to estimate the port air mass flow. However, in the proposed method, only the information of the throttle air mass flow was utilized. The main contribution of this study is to suggest a method to reduce the number of sensors when estimating the port air mass flow and engine indicated torque. In the new method, the port air mass flow is the low pass filtered result of the throttle air mass flow, and the cut off frequency of the low pass filter is proportional to engine speed. In production engine experiments, the proposed estimation method of the port air mass flow was used to estimate the engine indicated torque, and indirectly verified by comparing the estimated engine indicated torque and measured one.

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REFERENCES

AZZONI, P. 1997. Reconstruction of indicated pressure waveform in a spark-ignition engine from block vibration measurements. *Journal of dynamic systems, measurement, and control,* 119**,** 614-619.

AZZONI, P., MINELLI, G., MORO, D. & SERRA, G. 1997. A model for EGR mass flow rate estimation. SAE Technical Paper.

AZZONI, P., MORO, D., PONTI, F. & RIZZONI, G. 1998. Engine and load torque estimation with application to electronic throttle control. SAE technical paper.

CHEVALIER, A., M LLER, M. & HENDRICKS, E. 2000. On the validity of mean value engine models during transient operation. *SAE transactions***,** 1571-1592.

CHO, D. & HEDRICK, J. K. 1989. Automotive powertrain modeling for control. *Journal of dynamic systems, measurement, and control,* 111**,** 568-576.

ERIKSSON, L. & NIELSEN, L. 2014. *Modeling and control of engines and drivelines*, John Wiley & Sons.

FONS, M., MULLER, M., CHEVALIER, A., VIGILD, C., HENDRICKS, E. & SORENSON, S. C. 1999. Mean value engine modelling of an SI engine with EGR. *SAE transactions***,** 1339-1346.

GAO, Y. & RANDALL, R. 1999. Reconstruction of diesel engine cylinder pressure using a time domain smoothing technique. *Mechanical systems and signal processing,* 13**,** 709-722.

HENDRICKS, E., CHEVALIER, A., JENSEN, M., SORENSON, S. C., TRUMPY, D. & ASIK, J. 1996. Modelling of the intake manifold filling dynamics. *SAE transactions***,** 122-146.

HENDRICKS, E. & SORENSON, S. C. 1990. Mean value modelling of spark ignition engines. *SAE transactions***,** 1359-1373.

KIM, J., CHOI, S. B. & OH, J. 2018. Adaptive Engagement Control of a Self-Energizing Clutch Actuator System Based on Robust Position Tracking. *IEEE/ASME Transactions on Mechatronics*.

KIM, S. & CHOI, S. 2018. Control-oriented modeling and torque estimations for vehicle driveline with dual-clutch transmission. *Mechanism and Machine Theory,* 121**,** 633-649.

KIM, S., OH, J. & CHOI, S. 2017. Gear shift control of a dual-clutch transmission using optimal control allocation. *Mechanism and Machine Theory,* 113**,** 109-125.

KIM, Y. W., RIZZONI, G. & WANG, Y.-Y. 1999. Design of an IC engine torque estimator using unknown input observer. *Journal of dynamic systems, measurement, and control,* 121**,** 487-495.

MAASS, B., DENG, J. & STOBART, R. 2011. In-Cylinder Pressure Modelling with Artificial Neural Networks. SAE Technical Paper.

MINOWA, T., OCHI, T., KUROIWA, H. & LIU, K.-Z. 1999. Smooth gear shift control technology for clutch-to-clutch shifting. SAE Technical Paper.

NA, J., CHEN, A. S., HERRMANN, G., BURKE, R. & BRACE, C. 2018. Vehicle engine torque estimation via unknown input observer and adaptive parameter estimation. *IEEE Transactions on Vehicular Technology,* 67**,** 409-422.

OH, J., CHOI, S., CHANG, Y. & EO, J. 2017a. Engine clutch torque estimation for parallel-type hybrid electric vehicles. *International Journal of Automotive Technology*, 18**,** 125-135.

OH, J. J. & CHOI, S. B. 2015. Real-time estimation of transmitted torque on each clutch for ground vehicles with dual clutch transmission. *IEEE/ASME Transactions on Mechatronics,* 20**,** 24-36.

OH, J. J., EO, J. S. & CHOI, S. B. 2017b. Torque Observer-Based Control of Self-Energizing Clutch Actuator for Dual Clutch Transmission. *IEEE Transactions on Control Systems Technology,* 25**,** 1856-1864.

POTENZA, R., DUNNE, J., VULLI, S. & RICHARDSON, D. 2007. A model for simulating the instantaneous crank kinematics and total mechanical losses in a multicylinder in-line engine. *International Journal of Engine Research,* 8**,** 379-397.

RAJAMANI, R. 2011. *Vehicle dynamics and control*, Springer Science & Business Media.

RIZZONI, G., DRAKUNOV, S. & WANG, Y.-Y. On-line estimation of indicated torque in IC engines via sliding mode observers. Proceedings of 1995 American Control Conference-ACC'95, 1995. IEEE, 2123-2127.

RIZZONI, G. & ZHANG, Y. 1994. Identification of a non-linear internal combustion engine model for on-line indicated torque estimation. *Mechanical Systems and Signal Processing,* 8**,** 275-287.

SHAMEKHI, A.-M. & SHAMEKHI, A. H. 2015. A new approach in improvement of mean value models for spark ignition engines using neural networks. *Expert Systems with Applications,* 42**,** 5192-5218.

SIANO, D. & D'AGOSTINO, D. 2014. Vibrational Signals Processing for In-Cylinder Pressure Reconstruction of a Four Cylinder Spark Ignition Engine. *Modelling and Simulations***,** 510.

SMITH, L., FICKENSCHER, T. & OSBORNE, R. 1999. Engine breathing-steady speed volumetric efficiency and its validity under transient engine operation. *SAE transactions***,** 268-282.

TAYLOR, C. F. 1985. *The Internal-combustion Engine in Theory and Practice: Combustion, fuels, materials, design*, MIT press.

VAN BERKEL, K., VELDPAUS, F., HOFMAN, T., VROEMEN, B. & STEINBUCH, M. 2014. Fast and smooth clutch engagement control for a mechanical hybrid powertrain. *IEEE Transactions on Control Systems Technology,* 22**,** 1241-1254.

YAR, A., BHATTI, A. I. & AHMED, Q. 2018. First principle based control oriented model of a gasoline engine including multi-cylinder dynamics. *Control Engineering Practice,* 70**,** 63-76.