

Study on Application of Engine Load Estimation Method Using Crank Angular Velocity Variation to Spark Advance Control

Hirotaka KAWATSU*

Ryosuke IBATA*

Tetsuya KANEKO*

Kenji NISHIDA*

ABSTRACT

The technology to estimate engine load using the amplitude of crankshaft angular velocity variation during a cycle, which is referred to as “ $\Delta\omega$ (delta omega)”, in a four-stroke single-cylinder gasoline engine has been established in our former studies. This study was aimed to apply this technology to the spark advance control system for small motorcycles. The cyclic variation of the $\Delta\omega$ signal, which affects engine load detection capability, was a crucial issue when developing the system. To solve this issue, filtering functions that can cope with various running conditions were incorporated into the computation process that estimates engine loads from $\Delta\omega$ signals. In addition, the system made it possible to classify engine load into two levels without a throttle sensor currently used. We have thus successfully developed the new spark advance system that is controlled in accordance with the engine speed and load.

1. Introduction

A large number of small motorcycles are used widely around the world, and the improvement of the fuel economy is one of the important issues to be solved from the global environmental viewpoint. One of the basic techniques for fuel economy improvement is the spark advance control that functions in accordance with the engine load. Generally, the engine load is detected by a throttle sensor. In small motorcycles, two types of throttle sensors are commonly used. One of the types is a potentiometer, which continuously detects throttle angle. Another is a throttle switch, which classifies throttle opening into two levels. The potentiometer is mainly applied to motorcycles equipped with a fuel injection system. In this type, the spark timing is controlled in accordance with the engine speed and throttle angle on the basis of three-dimensional control map stored in the ECU. The throttle switch is applied to some motorcycles equipped with a carburetor. In this type, the spark timing is controlled in accordance with the throttle opening by switching two kinds of two-dimensional control maps in which the relationship between the engine speed and the spark timing is stored. However, regarding small motorcycles with affordable prices using a carburetor, a large number of models are not equipped with a throttle sensor for spark advance control.

It has been known that the single-cylinder four-stroke engine has the typical pattern of variations in crankshaft angular velocity during a cycle consisting of intake, compression, expansion and exhaust strokes⁽¹⁾. In our former studies, the technology estimating engine load with this amplitude of variation has been already established (we call it “ $\Delta\omega$ method”)^{(2), (3)}. The amplitude ($\Delta\omega$) is calculated on the basis of signals from the crank position sensor which is commonly equipped in conventional engines.

In this project, we aimed to realize an alternative system to the conventional system using a throttle switch. In the alternative system, engine loads are classified into two levels using $\Delta\omega$ method. First of all, we worked out a basic logic using $\Delta\omega$ method. However, when the load-level classification using $\Delta\omega$ was tested on the marketed motorcycle, it was made clear as an issue that the spark timing was unstable due to hunting of engine load-level classification resulting from cyclic variations of $\Delta\omega$ signals. To cope with this issue, we theoretically studied the causes of cyclic variations of $\Delta\omega$ and clarified the conditions under which such variations occurred. Moreover, after conducting validation tests on the basis of the said presumption, new filtering processes were applied in the execution algorithm to classify the engine load-level, and the issue was resolved.

Consequently, the spark advance control system using $\Delta\omega$ method enabled to have the same level of load-level

* Motorcycle R&D Center

classification capability as the conventional system. We have thus successfully developed a simple spark advance control system without using a throttle switch.

2. Conventional Spark Advance Control

This section outlines a conventional spark advance control system. Such a system classifies engine load into two levels using output signals from the throttle switch, and controls the spark timing accordingly.

2.1. Experimental Motorcycle

Figure 1 shows the experimental motorcycle. Table 1 shows the major specifications of the experimental engine. The experimental motorcycle is a marketed small scooter equipped with a four-stroke, single-cylinder 110 cm³ engine. The fuel is supplied by a carburetor.

2.2. Load-level Classification by Throttle Switch

Figure 2 shows the cross-sectional view of the piston-valve-type carburetor with the throttle switch. The throttle switch is attached to the side of the carburetor cylinder that retains the piston valve. The push rod is depressed at the specific piston position and the switching signal is transmitted. The switching signal turns off when the throttle opening exceeds the prescribed level at which a certain engine load is presumed. Accordingly, the engine load is classified in two levels using the throttle switch without dependence on the engine speed. Figure 3 shows a motion of the load-classification using the throttle switch.

2.3. Basic Logic of Spark Advance Control

Figure 4 shows a flowchart of a conventional spark advance control using a throttle switch. In the control flow, the engine speed is calculated, the engine load is classified by the throttle switch, the load-level signal is transmitted, and the spark timing map is selected according to the

Table 1 Specifications of experimental motorcycle

Item	Specification
Engine type	Four-stroke single-cylinder
Cooling system	Air-cooled
Bore × Stroke	50.0 mm × 55.6 mm
Engine displacement	109.2 cm ³
Compression ratio	9.5
Spark supplying system	Full-transistor
Fuel supplying system	Carburetor
Transmission type	Continuously Variable Transmission (CVT)
Maximum brake power output	5.6 kW at 7500 rpm
Maximum brake torque	8.57 Nm at 5500 rpm
Curb weight	108 kg

classified load. When the load is classified as “Low”, the advancing map is selected correspondingly. On the other hand, when the load is classified as “High”, the retarding map is selected. Finally, the spark timing matching the engine speed is picked up from the selected spark-timing map.

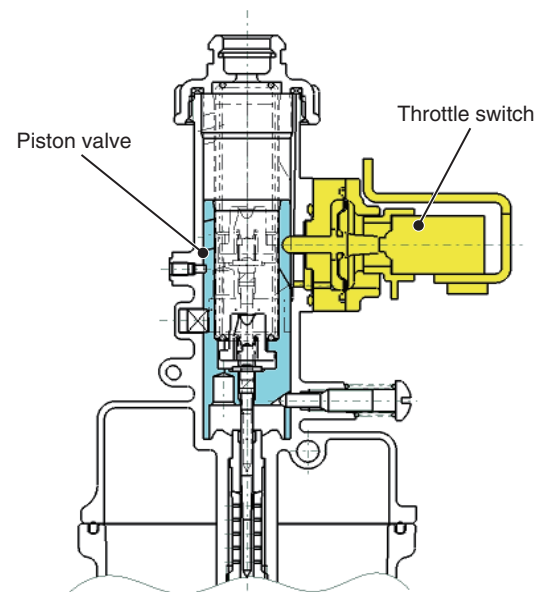


Fig. 2 Carburetor with throttle switch system



Fig. 1 Experimental motorcycle

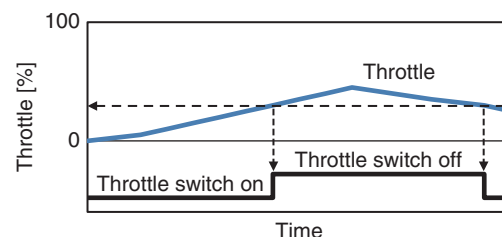


Fig. 3 Throttle switch motion

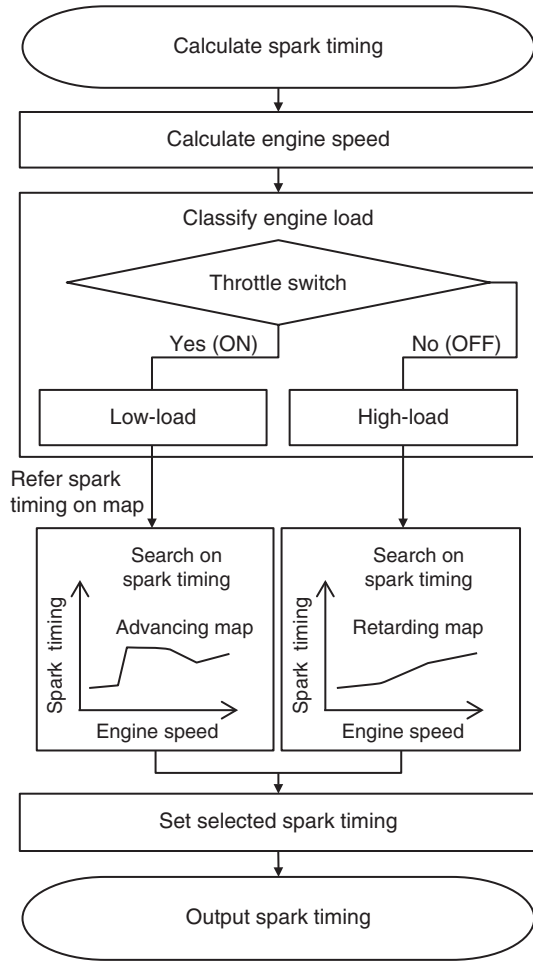


Fig. 4 Flowchart of conventional spark advance control

We investigated a new logic to replace the throttle switch method with $\Delta\omega$ method for the process of load-level classification.

3. Spark Advance Control Using $\Delta\omega$ Method

3.1. Basic Principles of Engine Load Estimation Using $\Delta\omega$

In this section, the system structure and the principles of $\Delta\omega$ method are explained, with reference to a previous papers^{(2), (3)}.

3.1.1. Measurement of $\Delta\omega$

In the $\Delta\omega$ method, the crankshaft angular velocity ω_i ($i = 1, 2$) is used. Figure 5 shows the structure of the ω_i measurement system⁽²⁾. The crank position sensor detects the front end and the rear end of the reluctor on the alternator rotor. The ω_i is determined by Eq. (1) using the time τ_i between the detected signals P_1 and P_2 , and the reluctor angle θ . Note that there are two suffixes (i) because P_1 and P_2 are detected two times during a cycle.

$$\omega_i = \frac{\theta}{\tau_i} \quad (1)$$

Figure 6 shows the waveform of crankshaft angular velocity (ω waveform), the measured ω_i , the reluctor pulse signals P_1 , P_2 , and the reluctor position⁽²⁾. These data were gotten under the constant load and engine speed (N_e) of approximately 3000 rpm. The continuous ω waveform is measured with the rotary encoder, which is integrated on an end of crankshaft and generates 360 pulses per revolution. When the ω_i in a cycle is defined as ω_1 in the compression stroke and ω_2 in the exhaust stroke, the $\Delta\omega$ is represented by Eq. (2).

$$\Delta\omega = \frac{\omega_2 - \omega_1}{2} \quad (2)$$

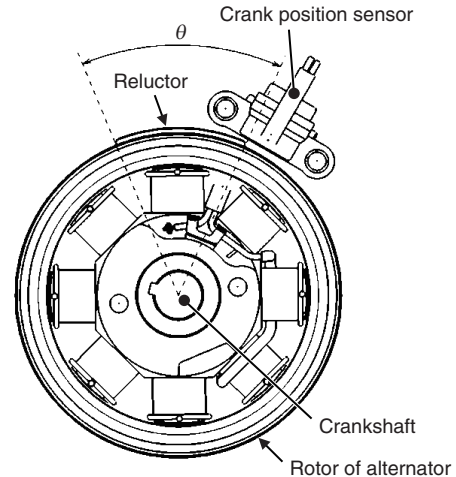
In this measurement method, $\Delta\omega$ is determined from the ω_i in the half of cycle, and a $\Delta\omega$ per cycles provided with a delay of a cycle (period: 4π).

3.1.2. Experimental studies on cyclic variations of $\Delta\omega$

The following two important features on $\Delta\omega$ under constant speed and load conditions have been proven by the previous papers^{(2), (3)}.

- (a) $\Delta\omega$ is proportional to the indicated mean effective pressure.
- (b) $\Delta\omega$ is counter-proportional to N_e .

The indicated mean effective pressure is represented by a value in that the cylinder work delivered to the piston over the entire four-stroke cycle is divided by the displacement, and denoted as $IMEP_{net}$ (Net Indicated Mean Effective Pressure). Figure 7 shows the relationship between the $\Delta\omega$ and the $IMEP_{net}/N_e$ at each engine speed, when operating with a wide open throttle. The plotted data are an average of approximately 500 cycles. From Fig. 7, it is proven that $\Delta\omega$ changes proportionally to $IMEP_{net}$ and counter-proportionally to N_e ⁽²⁾.

Fig. 5 Schematic of onboard system to detect $\Delta\omega$

Meanwhile, the relationship between $\Delta\omega$ and $IMEP_{net}$ at each cycle is also indicated in the previous papers^{(2), (3)}. Figure 8 shows the cycle-by-cycle data of N_e , $\Delta\omega$ and $IMEP_{net}$ at an idling. It has been confirmed from the figure that the cyclic variations of $\Delta\omega$ follow the variations of $IMEP_{net}$.

3.2. Basic Logic of Load-level Classification in $\Delta\omega$ Method

The following load-level classification logic was established to change the load-level classification process shown in the flowchart in Fig. 4 from a conventional system using the throttle switch to $\Delta\omega$ method.

3.2.1. Indexing of engine load conditions

$\Delta\omega$ proportionally changes to the engine load. However its proportionality constant changes inverse proportionally to the engine speed. We therefore defined the load ratio (hereafter referred to as LR), which is calculated from $\Delta\omega$, to standardize engine load conditions without dependence on N_e . LR is defined as the ratio of the value of $\Delta\omega$ when operating at a given throttle opening divided by the value of $\Delta\omega$ when operating at wide open throttle (defined as $\Delta\omega_{wot}$) as shown by Eq. (3). The relationship between $\Delta\omega_{wot}$ and engine speed is pre-written in the ECU in the form of two-dimensional data. The ratio of $\Delta\omega$ measured when operating, divided by the $\Delta\omega_{wot}$ at the same engine speed is calculated as the LR for each cycle.

$$LR = \frac{\Delta\omega}{\Delta\omega_{wot}} \times 100 [\%] \quad (3)$$

3.2.2. LR characteristics in relation to engine speed

Figure 9 shows the relationship between the engine speed and $\Delta\omega$ together with the LR . The data shown in blue are for wide open throttle (WOT). The data in red are

for partially open throttle (POT), where the throttle switch works. As the $\Delta\omega$ is counter-proportional to the engine speed, the higher the engine speed, the smaller the $\Delta\omega$ becomes. With LR introduced, $\Delta\omega$ at WOT is expressed as $LR = 100\%$. Every load can be thus indicated between 0% and 100% at each engine speed. Using LR , the load level is classified by comparing it with the reference LR .

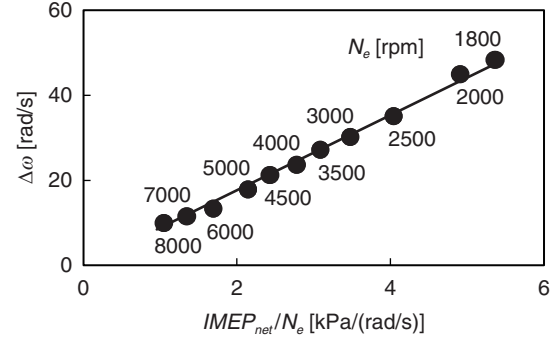


Fig. 7 Relationship between $\Delta\omega$ and $IMEP_{net}/N_e$

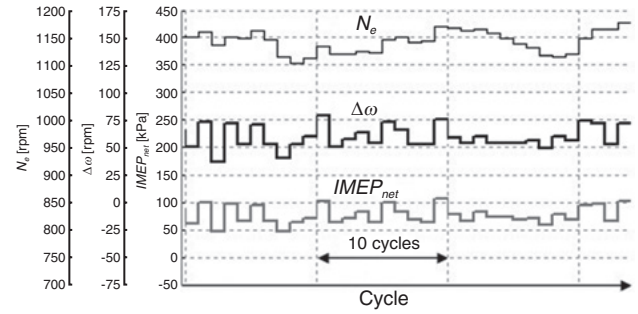


Fig. 8 Cyclic variation of $\Delta\omega$ and $IMEP_{net}$ at an idling

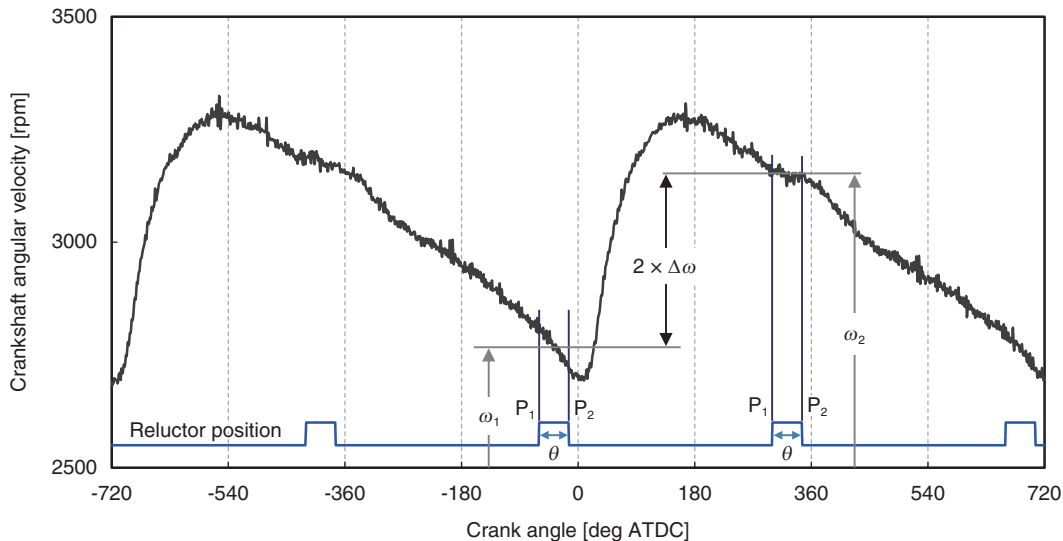


Fig. 6 Waveform of crankshaft angular velocity, reluctor position and definition of $\Delta\omega$

3.2.3. Load-level classification zone using LR

Figure 10 shows a schematic diagram of the load-level classification zone when the threshold level for load-level classification is set using LR . In this case, the threshold level was set referring to the LR at the throttle position where the throttle switch works. When the LR is larger than the threshold level, the load is classified as “High”, and “Low” when smaller.

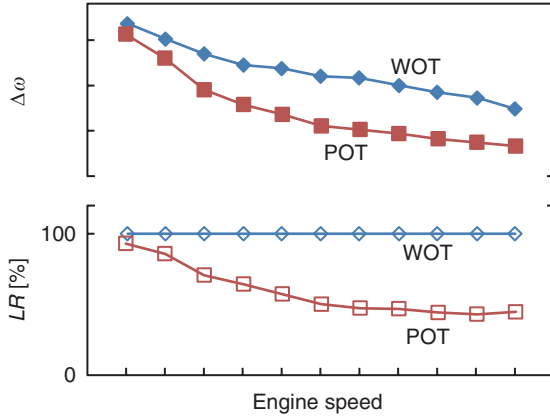


Fig. 9 $\Delta\omega$ and LR in relationship with engine speed

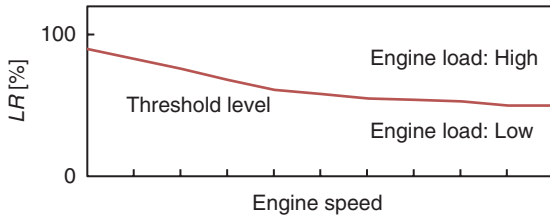


Fig. 10 Schematic diagram of load-level classification zone using LR

4. Theoretical Studies on How Cyclic Variations of $\Delta\omega$ Affect Load-level Classification

4.1. Issues in Load-level Classification

As mentioned in Chapter 2, the load-level classification using the throttle switch is performed referring to a specific throttle opening. Accordingly, when the throttle opening is fixed, the load-level classification and the spark timing map are fixed. Consequently, when the engine speed is constant, the spark timing becomes constant.

In the meantime, the load classification by the $\Delta\omega$ method is performed referring to a specific LR . In this method, when the $\Delta\omega$ cyclically varies even if the throttle opening is fixed, “hunting” of load-level classification occurs because the LR alternately moves across the threshold level. Consequently, it creates such a concern that the spark timing becomes unstable.

In order to obtain the same level of stability as the

conventional spark advance control system using a throttle switch, the load-level classification and the spark timing should be constant when the engine speed and the throttle opening are constant. We call this requirement “spark timing stability”.

Keeping the above in mind, we turned our attention to the cyclic variations of $\Delta\omega$ and clarified the conditions under which such variations occurred through the theoretical studies.

4.1.1. Governing equation of $\Delta\omega$

It is necessary to work out a governing equation of $\Delta\omega$ when studying cyclic variations of $\Delta\omega$. This chapter is therefore aimed at introduction of such a governing equation. The following preconditions are given to the motions of crankshaft system.

- (1) Target is four-stroke, single cylinder engine
- (2) The crankshaft system makes two-dimensional motions
- (3) All components are assumed as a rigid body
- (4) No change of conditions occurs in each contacting part
- (5) Influence of gravity is ignored

The crankshaft system model is composed of the three modules, the assembly of crankshaft, the connecting rod and the assembly of piston as Fig. 11 shows. The geometric conditions of the crankshaft system are expressed by the general coordinate $\mathbf{q}$ and the geometric restriction function $\mathbf{h}(\mathbf{q})$ of the system. Vector $\mathbf{q}$ is defined by Eq. (4), and consisted of the coordinates (x_{ig} , y_{ig}) of each component's center of gravity and the rotational angle θ_i ($i = c$: crankshaft, l : connecting rod, p : piston) around that.

$$\mathbf{q} = \begin{pmatrix} x_{cg} \\ y_{cg} \\ \theta_c \\ x_{lg} \\ y_{lg} \\ \theta_l \\ x_{pg} \\ y_{pg} \\ \theta_p \end{pmatrix}, \mathbf{h}(\mathbf{q}) = \begin{pmatrix} x_{cg} - r_{cg} \cos(\theta_c + \varphi_{cg}) \\ y_{cg} - r_{cg} \sin(\theta_c + \varphi_{cg}) \\ x_{lg} - r_{cp} \cos \theta_c - r_{lg} \cos(\theta_l + \varphi_{lg}) \\ y_{lg} - r_{cp} \sin \theta_c - r_{lg} \sin(\theta_l + \varphi_{lg}) \\ x_{pg} - r_{cp} \cos \theta_c - r_{lp} \cos \theta_l - r_{pg} \cos \varphi_{pg} \\ y_{pg} - r_{cp} \sin \theta_c - r_{lp} \sin \theta_l - r_{pg} \sin \varphi_{pg} \\ y_{pg} - e_a - r_{pg} \sin \varphi_{pg} \\ \theta_p \end{pmatrix} = 0 \quad (4)$$

Next, think about the distribution of energy in the crankshaft system. The work W_{cyl} that the working fluid provides on the piston is redistributed in three ways, the kinetic energy K of the crankshaft system, the work W_{out} given out to the lower stream of crankshaft (hereafter referred to as resistance load) and the dissipation energy D . These energies satisfy Eq. (5).

$$W_{cyl} = K + D + W_{out} \quad (5)$$

Here, following the definition of $\Delta\omega$ seen in Section 3.1.1 and paying attention to the motions in a half cycle, the kinetic energy can be represented by Eq. (6) as a function of $\Delta\omega$. It should be noted that while Γ is a function with a half cycle periodicity of the dimension, mass characteristics and θ_c of each component, it can be treated as a constant in this case.

$$K|_0^{2\pi} = I' \frac{\omega_2 + \omega_1}{2} (\omega_2 - \omega_1) = I' N_e' \Delta\omega \quad (6)$$

Note that with respect to the N_e' , which is an average of measured ω_1 and ω_2 ; there is an error δ_{err} from the true N_e (one-cycle average of ω). Although the δ_{err} can be made zero at an N_e depending on the position of reluctor seen in Section 3.1.1, as the ω waveform also changes in accordance with changes of N_e , it is impossible to make δ_{err} zero at a given N_e . Accordingly, when the N_e and the N_e' are approximately equal, the $\Delta\omega$ is counter-proportional to the N_e .

Based on the aforementioned, Eq. (7) that governs $\Delta\omega$ is derived from Eq. (5) and Eq. (6).

$$\Delta\omega = \frac{1}{I' N_e'} \left\{ W_{cyl}|_0^{2\pi} - W_{out}|_0^{2\pi} - D|_0^{2\pi} \right\} \quad (7)$$

Moreover, due to the facts that the Lagrangian in the crankshaft system depends only on K because the influence of gravity is ignored, and that the restricting conditions of each coordinate is acquired from $h(q)$ of Eq. (4), the equation of motion around the crankshaft (8) is introduced from the Lagrange equation, thus determining ω_1 and ω_2 .

$$(I_{cg} + \eta_2) \ddot{\theta}_c - \eta_1 \dot{\theta}_c^2 = T_{cyl} - T_f - T_{out} \quad (8)$$

Note that the coefficients η_1, η_2 are a function composed of θ_c , geometric dimension, and mass characteristics, having a period of 2π .

The left side of Eq. (8) indicates the inertia. The right side is the torque around the crankshaft with the first term T_{cyl} representing the torque acquired from the working fluid, the second term T_f representing the friction torque, and the third term T_{out} representing the resistance load torque.

From the above findings, the parameters decisive for the crankshaft rotation characteristics and the $\Delta\omega$, and their features can be clarified from Eq. (7) and Eq. (8).

4.1.2. Cyclic variations of $\Delta\omega$

When ω_1 and ω_2 , which are determined by Eq. (8), do not have a one-cycle periodicity, variations of $\Delta\omega$ occur. For this reason, cyclic variations of $\Delta\omega$ will occur if the vibrations without a one-cycle periodicity act on the crankshaft system. Figure 12 shows the characteristic diagram of parameters decisive for the $\Delta\omega$ based on Eq. (7) and Eq. (8).

In Fig. 12, geometrical and physical parameters decisive for (a) moment of inertia of the crankshaft system I' depend on the shape and density of each component. Unusual phenomena such as a large deformation, extensive wear, fracture, etc. are required to cause a change of shape or density during operation. Since such cases can be omitted as irregularities, the geometrical and physical parameters do not affect $\Delta\omega$ during operation.

In Fig. 12, a change of (b) engine speed N_e is counter-proportionally linked with $\Delta\omega$ by Eq. (7). Accordingly, $\Delta\omega$ also changes along with a change of N_e .

In Fig. 12, when a change occurs in (c) cylinder work W_{cyl} performed by the working fluid, the IMEP also changes. The cause is in the inconsistent combustion. The linked cyclic changes of IMEP and $\Delta\omega$ are evident in Fig. 8 in Section 3.1.2.

We consider the reason of such linked changes as follows.

The combustion characteristics depend on the characteristics of the working fluid. The air mass (O_2 quantity) in the cylinder depends on the volumetric efficiency (η_v), and the fuel mass depends on the air-to-fuel ratio (A/F). Moreover, the flow of working fluid creates mixture and determines concentration distribution. Such phenomena depend on the in-cylinder pressure (P_{cyl}) and the piston speed, in result of the ω . If a fixed ignition timing (θ_{ig}) and ignition energy (E_{ig}) are provided to the working fluid, the combustion corresponding to the given conditions should occur. Consequently, along with the heat generation pattern, the in-cylinder pressure $P_{cyl}(\theta_c)$ is determined uniquely. The $P_{cyl}(\theta_c)$ determines ω waveform $\omega(\theta_c)$ through the equation

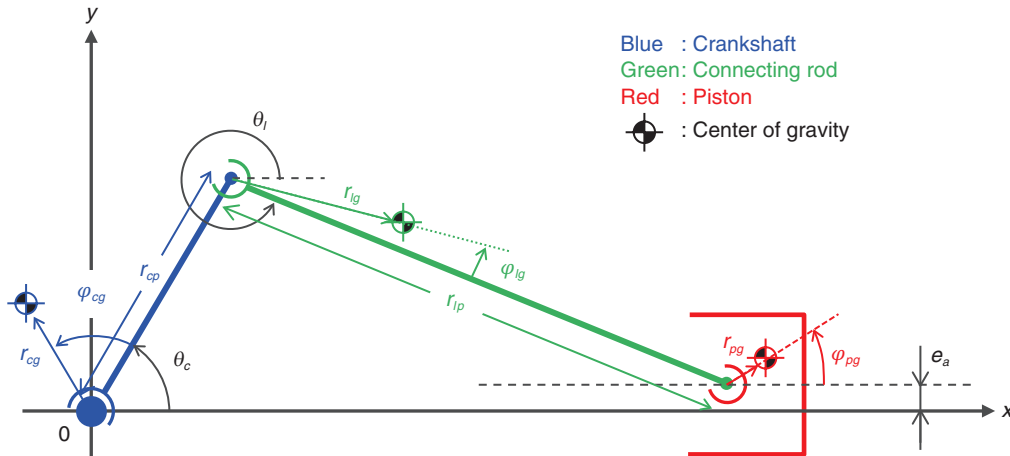
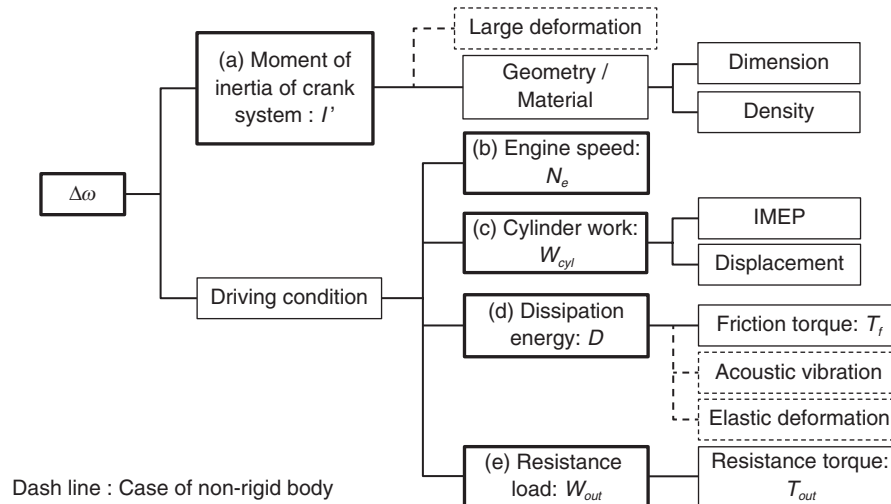


Fig. 11 Mathematical model of crank system

Fig. 12 Characteristic diagram of $\Delta\omega$

of motion (Eq. (8)) along with T_{out} and T_f .

In Fig. 12, (d) dissipation energy D depends only on the friction force under the presumption of a rigid body, which is a precondition. Among the friction forces, some shows self-exciting behavior such as the stick slip. In such a case, vibrations irrelevant to the one-cycle periodicity occur due to the friction force, causing cyclic variations of $\Delta\omega$. It should however be noted that such a phenomenon is deemed as occurring at the time of irregularity such as a piston seizure, and not subjected to the validation.

In Fig. 12, (e) resistance load W_{out} is applied freely regardless of the engine. Such a condition corresponds to the running resistance including the one from the road roughness and the air resistance depending on the randomly-changing wind direction, etc. The vibrations irrelevant to the engine and occurring in the subsequent system from the

crankshaft system (e.g., influence of the damper in the drive train, etc.) also affect as an external force to the crankshaft system. As mentioned above, regarding the resistance load, a phenomenon containing vibrations without a one-cycle periodicity exists. Cyclic variations of $\Delta\omega$ therefore occur due to the resistance load without a one-cycle periodicity.

Finally, if the presumption that the component is a rigid body is freed, the vibration frequencies should occur due to the deformation as shown by the dotted lines in Fig. 12. In this case, as the natural frequency of each component is irrelevant to the one-cycle periodicity, the vibration frequencies provides cyclic variations to $\Delta\omega$.

From the above considerations, the cause of cyclic variations of $\Delta\omega$ is in the vibrations irrelevant to the one-cycle periodicity in addition to variations of IMEP. The specific example is shown in Table 2.

Table 2 Factors of cyclic variation of $\Delta\omega$

Classification	Factors	Parameters in Eq. (7) and (8)	Examples of specific phenomenon
Internal factors	(a) Change in moment of inertia of crank system	I'	Change deformation, Large wear, Breakage (Unusual phenomenon)
	(c) Variation of IMEP	T_{cyl}	Variation in combustion state, Occurrence of misfiring
	(d) Variation of friction	T_f	Repetition of sticking and slipping between piston skirt and cylinder sleeve (Unusual phenomenon)
	Vibration derived from deformation of components in crank system	Internal force	Torsional vibration, Out-of-plane vibration and In-plane vibration of components
External factors	(b) Variation of engine speed	N_e	Acceleration and deceleration by throttle control
	(e) Variation of load	T_{out}	Unevenness of the road surface, Changes of air resistance, Changes in posture of motorcycle and rider
	Vibration derived from deformation of components outside the crank system	T_{out}	Vibrations of drive system, vehicle body and crankcase, etc.

5. Examination and Considerations on $\Delta\omega$ Method

According to the studies in the chapter 4, the factors of cyclic variations of $\Delta\omega$ can be classified to two categories; (i) those affecting to the engine as internal factors, (ii) those affecting to the engine as external factors. To satisfy “spark timing stability” required for a spark advance control system using the $\Delta\omega$ method, we firstly analyzed how the internal factors affected cyclic variations of $\Delta\omega$ under the constant load and speed conditions. Then, we analyzed how the cyclic variations of $\Delta\omega$ from external factors affected “spark timing stability” under the non-constant load and speed conditions. On the basis of the findings from the analysis, we developed the system that ensures “spark timing stability”.

5.1. Running with Constant Speed and Load

5.1.1. Test results

Keeping in mind the aforementioned requirements, tests were conducted on a chassis dynamometer under constant speed and load conditions to verify the spark timing control by the newly developed system.

Figure 13 shows cycle-based data for $\Delta\omega$ when the engine is running at 4500 rpm at 40% constant throttle opening. The load is classified as “High” by the throttle switch under this condition. It has been clarified that $\Delta\omega$ is not constant, but has variations up to approximately 50 rpm. It can be considered due to variations of $IMEP_{net}$.

Next, Fig. 14 shows the control operations based on basic logic corresponding to the $\Delta\omega$ variations shown in Fig. 13. The upper graph shows cyclic variations of LR and threshold level for load-level classification, the middle graph shows load-level classification signals based on LR , and the lower graph indicates the determined spark timing. The threshold level is set referring to the LR value at the throttle position where the throttle switch works.

Along with the fluctuations of $\Delta\omega$, the LR alternately moves across the threshold level, consequently causing hunting of the load-level classification signals and the spark timing. Accordingly, it was confirmed that the requirement of “spark timing stability” cannot be satisfied using this classification method.

We then analyzed how each engine speed affects the cyclic variations of $\Delta\omega$. Figure 15 shows the $\Delta\omega$ at each

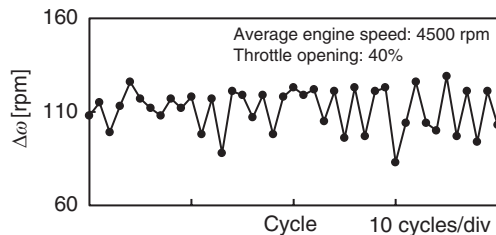


Fig. 13 Cyclic variations of $\Delta\omega$

engine speed with WOT, its maximum variation $\Delta\omega_{p-p}$, and its variation ratio. The black dot in the upper graph shows $\Delta\omega_{ave}$, which is an average of $\Delta\omega$ over approximately 500 cycles, and the error bar indicates $\Delta\omega_{p-p}$. The lower graph shows the variation ratio ($\Delta\omega_{p-p}/\Delta\omega_{ave} \times 100\%$). The $\Delta\omega_{p-p}$ stays almost a constant value irrespective of engine speed. However $\Delta\omega_{ave}$ decreases as engine speed increases. Accordingly, the variation ratio increases along with an increase of engine speed, and reaches the maximum of 20%. We consider that the variations are affected not only by the variations of $IMEP_{net}$, but also by the vibrations in the crankshaft system and the power train components including the reciprocating system listed in Table 2 in Section 4.1.2.

5.1.2. Application of two thresholds

A hysteresis is applied to the threshold level shown in Fig. 10 to cope with the hunting of load-level classification caused by cyclic variations of $\Delta\omega$.

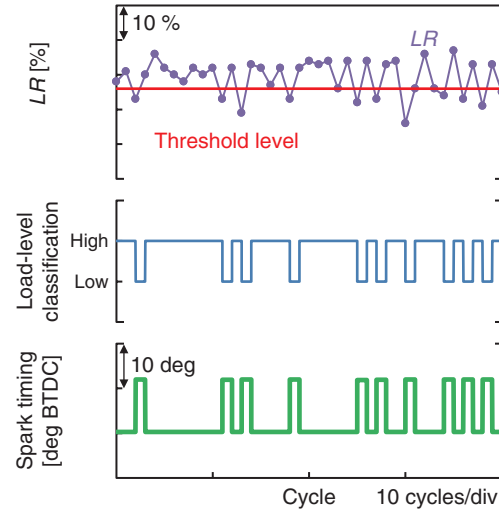


Fig. 14 Variations of LR , load-level classification signal and spark timing

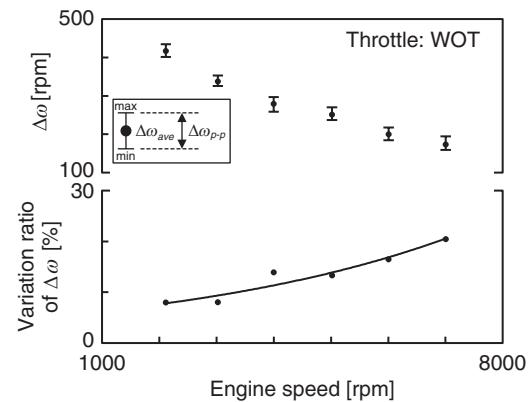


Fig. 15 Relationship between engine speeds and $\Delta\omega$, variation ratio of $\Delta\omega$

Figure 16 shows the schematic diagram of load-level classification having a hysteresis composed of two thresholds. The engine load-level classification turns from “Low” to “High” when the LR goes up above the high-threshold (red line), and turns from “High” to “Low” when the LR comes down under the low-threshold (black line). Here, the difference of LR between the two thresholds is defined as the hysteresis width. The yellow LR zone in the graph indicates the region with possibilities of knocking occurrence when the spark timing of the advancing map is applied (knock zone). The retarding map, therefore, must be selected. The green LR zone indicates the zone frequently used in city driving experienced in daily life. The blue line shows the load when cruising on a flat road. In these zones, it is required to apply the spark timing in the advancing map to improve fuel economy.

Each threshold, therefore, has been designated considering the following.

(1) High-threshold: To prevent knocking

The LR for the high-threshold is set at a level that corresponds to the throttle position where the throttle switch works. Thus, knocking can be prevented by selecting the retarding map when the load is above the high-threshold.

(2) Low-threshold: To stabilize spark timing and to improve fuel economy

To achieve spark timing stability, the hysteresis width must be larger than the cyclic variation width of $\Delta\omega$. At the same time, from the standpoint of fuel economy, a wide LR zone, in which spark timing is advanced, should be maintained. For that purpose, a small hysteresis width is preferable.

Taking the aforementioned into account, the hysteresis width was set at 25 points of LR . Consequently, this technique enabled the prevention of hunting of spark timing while suppressing knocking and, at the same time, improving the fuel economy when driving in city mode.

We have tested the above-mentioned load classification having hysteresis on the chassis dynamometer to clarify effectiveness in prevention of spark timing hunting. The operating conditions are the same as those in Section 5.1.1. The “High” threshold is set at the same level as in Fig. 12.

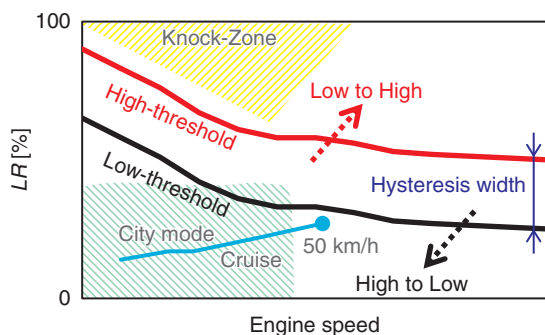


Fig. 16 Schematic diagram of load-level classification having hysteresis composed of two thresholds

Figure 17 shows the control operations when the load classification having hysteresis is applied. Although the LR alternately moves above and below the high-threshold, it never falls below the low-threshold, and the load-level classification remains continuously at “High”. Thus, the requirement of “spark timing stability” has been satisfied.

5.2. Rough Road Running

5.2.1. Test results

As shown in Table 2, a rapid change of road load is a conceivable cause of cyclic variations of $\Delta\omega$. As the changes of resistance load corresponds to T_{out} in Eq. (8), if the variance between the T_{out} at the moment the ω_1 is measured and T_{out} at the moment the ω_2 is measured increases, the gap between ω_1 and ω_2 widens, causing the $\Delta\omega$ to increase. Conversely, if the gap narrows, the $\Delta\omega$ decreases. When such a variation does not have a one-cycle periodicity, cyclic variations occur in the $\Delta\omega$.

Based upon the above-mentioned considerations, it is required to satisfy “spark timing stability” under conditions where the resistant load changes most rapidly in order to make the system feasible. What is conceivable as such operating conditions is such a running in which the rear wheel repeats spinning and gripping from jumping and landing, i.e., running on off roads. We therefore measured cyclic variations of $\Delta\omega$ while running on a rough road in which the motorcycle was repeatedly jumping off and landing on the ground. The rough road consisted of trapezoid blocks, approximately 5 cm in height, laid on a flat surface at a constant interval.

Figure 18 shows $\Delta\omega$ and the control operations when running on a flat road with the throttle held at a constant opening. Figure 19 shows $\Delta\omega$ and the control operations when running on the rough road. Although no hunting

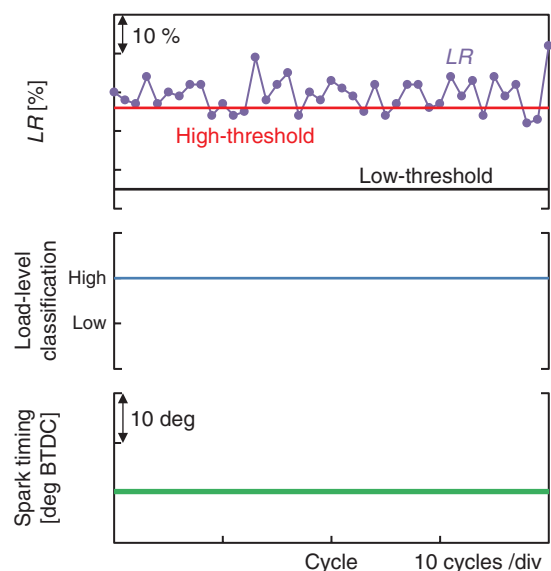


Fig. 17 Control operation of load-level classification having hysteresis

occurred in the load-level classification or in the spark timing when tested on the flat road, hunting occurred in the load-level classification and in the spark timing when tested on the rough road. It has been clarified from the comparison of Fig. 18 and Fig. 19 that an increase of cyclic variations of $\Delta\omega$ due to variations of N_e is a predominant cause of the hunting.

5.2.2. Application of wait counter for shifting classification

As mentioned before, the hunting can be avoided even when the cyclic variation amplitude of $\Delta\omega$ increases while running on a rough road if the hysteresis width is large enough. However, that enlargement of hysteresis makes it impossible to satisfy both requirements for knock suppression and for fuel economy improvement in driving under partial loads. For these reasons, we had to investigate different measures to prevent the hunting without changing the hysteresis width.

After this investigation, we decided to adopt a computation logic that shifts the classification of load levels only when detecting a sequence of LR signals crossing the threshold but does not shift for only a single signal. We call this “shift-wait counter”, and we call the number

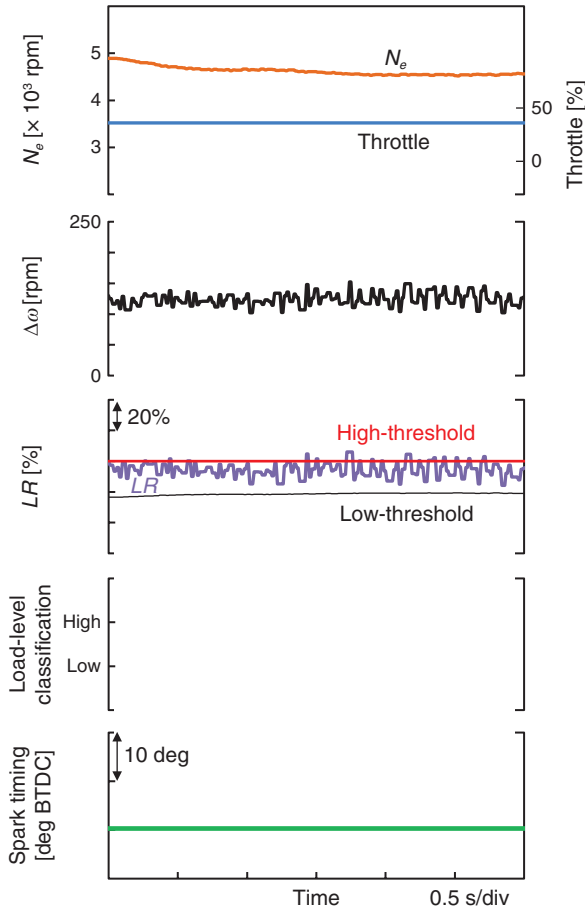


Fig. 18 $\Delta\omega$ variations and control on flat road

of continuous signals crossing the threshold “triggering counts”.

Figure 20 shows a schematic diagram of the shifting of load-level classification using a shift-wait counter. The upper graph indicates the variation of LR , the middle graph

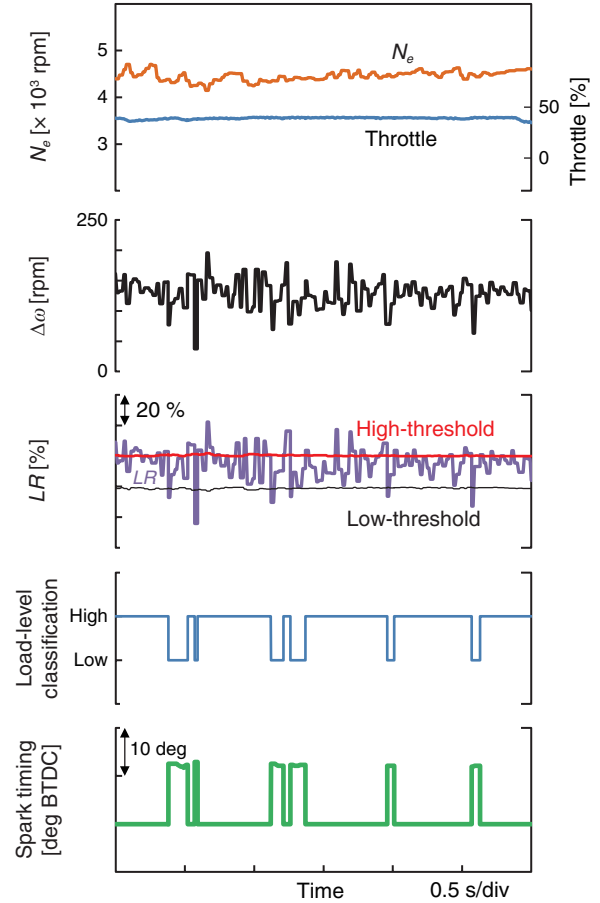


Fig. 19 $\Delta\omega$ variations and control on rough road

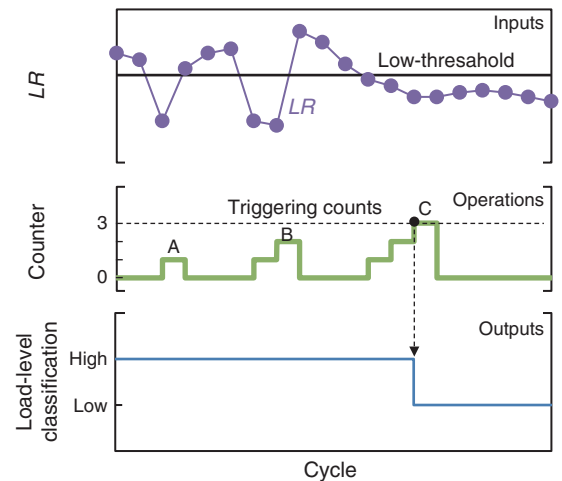


Fig. 20 Shifting of load-level classification by shift wait counter logic

shows the functional operations of the shift-wait counter, and the lower graph shows the load-level classification at that time. Here, the triggering count is set at three. As you see at Point A and Point B in the figure, the counts are not reaching the triggering counts. Consequently the load-level classification remains un-shifted. The load-level classification is first shifted at Point C where the requirement for shifting is satisfied. That is to say, the shifting occurs at the point where LR falls to the value under the low-threshold three times in a row. Although Fig. 20 shows only how the shift to the low load-level classification is determined, the same logic is applied to the shift to the high load-level classification.

We then verified the effectiveness of the shift-wait counter logic for the spark-timing hunting when running on rough roads. Figure 21 shows the control operation when the shift-wait counter is employed for the load-level classification. The operating conditions and the preset threshold are the same as those shown in Fig. 19. Although LR sometimes varies extensively along with the variations of N_e , and falls under the low-threshold, a shift to “Low level” is prevented by the application of the shift-wait counter. Thus, the “High level” classification is maintained, and the retarding map is continuously selected, satisfying the requirement of “spark timing stability”.

5.3. Load-level Classification Logic Using $\Delta\omega$ Method

The load-level classification logic was created in which the previously mentioned filtering methods are applied. It is shown in Fig. 22. The flowchart is composed of the acquisition of the crank position signal, calculation of $\Delta\omega$, calculation of LR , load-level classification using the

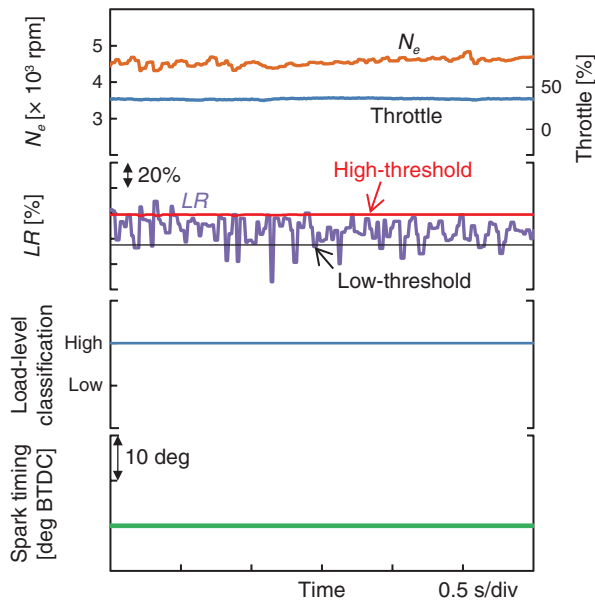


Fig. 21 Classification results of load-levels when shift wait counter was applied while running on rough roads

threshold-with-hysteresis and the shift-wait counter system, and transmission of the classification signal. This flowchart replaces the load-level classification using a throttle switch in the spark advance control flowchart shown in Fig. 4.

5.4. Validation of Modified System

The load-level classification logic using “thresholds with a hysteresis” and “shift-wait counter” was established to satisfy the requirement of “spark timing stability”. To determine whether or not this logic can be applied as an alternative to load-level classification using the throttle switch, we ran validation tests to verify the operations of load-level classification. In addition, we check how they affect drivability. The tests were conducted to compare the current motorcycle using a throttle switch and the modified motorcycle using the $\Delta\omega$ method.

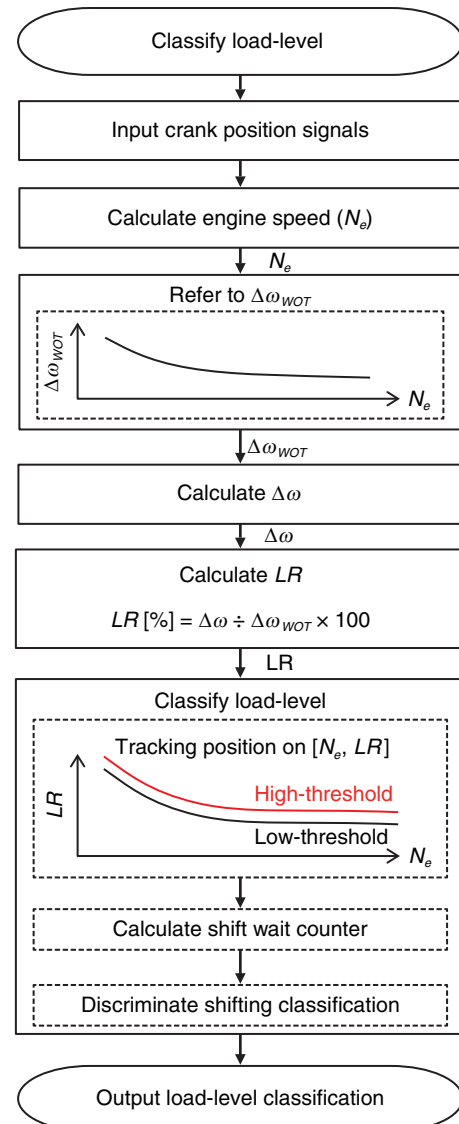


Fig. 22 Flowchart of load-level classification

Figure 23 shows the shifting of load-level classification when opening and closing the throttle. The driving condition of the motorcycle changes from acceleration to cruising. As the throttle is opened, N_e and LR increase, and a “High level” classification is determined when LR becomes higher than the high-threshold. The shift timing of classification is almost the same as for a throttle switch signal.

Then, as the throttle is closed, LR decreases and becomes lower than the low-threshold, and a “Low level” classification is determined before the change to cruising. Due to the application of “hysteresis” and “shift-wait counter”, there is a slight delay in classification shift timing compared with a throttle switch signal. Nevertheless, the same level of drivability as the conventional system was verified through subjective evaluation by test riders.

Accordingly, the developed spark advance control system makes it possible to replace a conventional system using a throttle switch and to improve the fuel economy of the models with no throttle sensor.

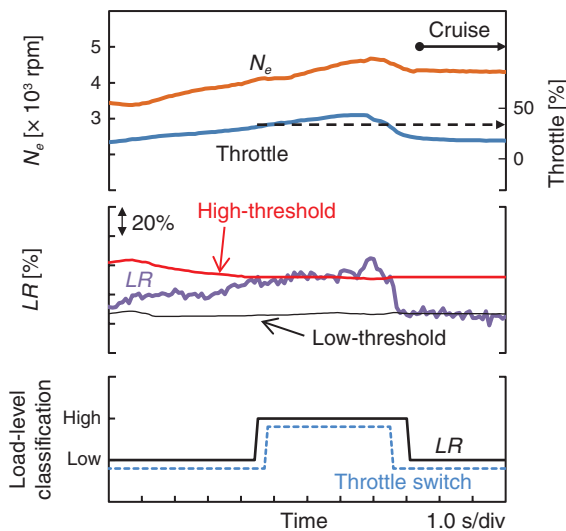


Fig. 23 Load-level classification when opening and closing throttle

6. Conclusions

We have established a logic that classifies engine load conditions into two levels using the amplitude of the crank angular velocity variation ($\Delta\omega$). Our conclusions follow:

- (1) Theoretical studies were conducted in advance on how the cyclic variations of $\Delta\omega$ affected the spark advance control system which is applied the engine load estimation using $\Delta\omega$. Consequently, it has been clarified that the cyclic variations of $\Delta\omega$ is caused by the vibrations without one-cycle periodicity acting on the crankshaft system in addition to the variations of

$IMEP_{net}$. Thus it is clarified that the cyclic variations of $\Delta\omega$ impairs “spark timing stability”.

- (2) Stable load-level classifications were made possible by applying a filtering procedure that adopts the two methods described below to cope with hunting of classification caused by cyclic variations of $\Delta\omega$.

- Hysteresis using two load-level classification thresholds
- “Shift-wait counter” to determine shifting after detecting a sequence of load signals that pass the load-level classification threshold

These methods enabled the same level of load-level classification functionality as the conventional system using a throttle switch and realized a simplified spark advance control system without a throttle switch.

The developed technology enables wider application of a spark advance control system that makes a significant contribution to the improvement of fuel economy.

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■ Author ■



Hiroataka KAWATSU



Ryosuke IBATA



Tetsuya KANEKO



Kenji NISHIDA