

Study on Controlled Auto-ignition Using Sequence of Low-temperature Combustion States

- Macroscopic Understanding of Controlled Auto-ignition and Approach from Gas Exchanging States -

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ABSTRACT

A possibility to control the auto-ignition timing and the heat release process was experimentally examined. Through a macroscopic analysis of the controlled auto-ignition (CAI), two physical parameters were introduced to explain the CAI condition. These are T_{AI} , which is the in-cylinder gas temperature at the end of compression stroke, and ΔT , which indicates the heat capacity of the working gas. The experimental analysis revealed that T_{AI} dominates the onset of auto-ignition and ΔT dominates its heat rising process. If these parameters were maintained in certain ranges, desirable CAI should be generated. The both parameters are thermo-dynamically calculated with the exhaust gas recirculation (EGR) ratio and the equivalence ratio only.

In order to examine the gas exchange state that allows of the above-mentioned condition, a typical gas exchanging chart that represents all the possible gas-exchange states in a naturally aspirated four-stroke engine was introduced. On this map, the CAI area and the surroundings were carefully plotted. As the result, it was revealed that there was the approach to the CAI from the SI combustion with a sequence of low-temperature combustion due to the dilution on the map.

The experimental results finally suggest that BSFC and BSNOx of a 1300 cm³ motorcycle could be decreased by 20% and 90% respectively in the LA-4 mode, by utilizing the sequence of the low-temperature combustion including the CAI.

1. Introduction

Controlled auto-ignition (CAI)⁽¹⁾ has attracted attention as a method of combustion that simultaneously reduces both fuel consumption and NOx emissions. The issues to be resolved for practical use of CAI are (1) the realization of methods of controlling ignition timing and heat release following ignition, (2) expansion of the zone of operation of CAI, and (3) appropriate transition between auto-ignition (AI) and spark ignition (SI).

In-cylinder working gas temperature is the key to ignition timing. 1000 K + α is known to be the auto-ignition temperature (T_{AI}) in the case of gasoline engines⁽²⁾. With regard to a method of controlling heat release, the concept of ΔT as a parameter for the evaluation of the speed of heat release has been introduced⁽³⁾. ΔT can be theoretically derived by treating in-cylinder combustion phenomena as zero-dimension phenomena. Amid a great deal of detailed research concerning CAI⁽⁴⁾ from the perspective of chemical reactions, this paper will attempt an experimental

investigation of the potential for the realization of adequate CAI by achieving appropriate values for T_{AI} and ΔT . It is believed that adopting a macroscopic understanding on CAI and taking a simpler approach to combustion control in this way will be effective when CAI is applied practically.

The major characteristic of AI combustion is the potential for high thermal efficiency and low NOx emissions offered by low-temperature combustion. Because of this, the expansion of the zone of operation of CAI is one area of focus⁽⁵⁾. Supercharging may be considered as an effective technological method of expanding the upper load limit for CAI⁽⁶⁾. However, because vehicle engines have a wide variety of load ranges, even if the zone of operation of CAI is expanded, the employment of AI-SI hybrid combustion may be considered as a practical solution^{(7), (8)}.

This paper will propose a method for the realization of a smooth transition between AI and SI from the perspectives of in-cylinder gas exchange and combustion control. In order to do so, the paper will first attempt to clarify all the possible gas exchange characteristics of a naturally

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aspirated 4-stroke engine. The paper will then propose a control method making use of the benefits of a sequence of LTC states achieved via continuous control of gas exchange, and will further verify this method.

2. Introduction of ΔT Concept

ΔT has been introduced as a parameter that expresses heat generation per unit of heat capacity of the working gas. This functions as a parameter for the evaluation of the speed of heat release. Assuming isochoric combustion with no heat loss, dissociation, or non-uniformity of the mixture, from the first law of thermodynamics, heat capacity Q is given by

$$Q = \Delta T (C_{vf} \cdot G_f + C_{vr} \cdot G_r) \quad (1)$$

where C_{vf} is the constant volume specific heat of the mixture, C_{vr} is the constant volume specific heat of the residual gases, G_f is the mass of the mixture, and G_r is the mass of the residual gases.

When V =displacement, η_c =charging efficiency, ρ =density of air, ϕ =equivalence ratio, α =stoichiometric A/F ratio, ξ =EGR rate, and H_u =lower heating value of fuel,

Mass of intake air is: $G_a = V \cdot \eta_c \cdot \rho$

Mass of supplied fuel is: $M_f = G_a \cdot \phi / \alpha$

Mass of mixture is: $G_f = G_a + M_f = G_a (1 + \phi / \alpha)$

Mass of residual gas is: $G_r = G_f \{ \xi / (1 - \xi) \}$
 $= G_a (1 + \phi / \alpha) \{ \xi / (1 - \xi) \}$

Calorific value of fuel is: $Q = H_u \cdot M_f = H_u \cdot G_a \cdot \phi / \alpha$

Dividing Eq. (1) by G_a gives

$$\Delta T = H_u (\phi / \alpha) / [C_{vf} \{ 1 + (\phi / \alpha) \} + C_{vr} \{ 1 + (\phi / \alpha) \} \{ \xi / (1 - \xi) \}] \quad (2)$$

Here, ΔT is given as an equation between ϕ and ξ , irrespective of displacement or the load conditions of the engine. In physical terms, ΔT refers to an increase in the temperature of the in-cylinder working gas following auto-ignition.

3. Test Apparatus and Test Method

A 4-stroke single-cylinder water-cooled gasoline engine with a bore of 100 mm by a stroke of 82.6 mm and a displacement of 650 cm³ was employed as the test engine. Table 1 shows the specifications of the test engine. An electrohydraulic valve actuator system (Lotus AVT)⁽⁹⁾ was employed as a variable valve mechanism, and the EGR rate was varied with negative overlap⁽¹⁰⁾. A compression ratio of 10 was used. Port injection fuel supply was employed to form a uniform pre-mixture in the cylinder. Regular gasoline (RON91) was used as the test fuel.

Figure 1 shows an outline of the test apparatus. Because high-temperature residual gases are essential to auto-ignition operation, the transition from SI combustion via the spark plug to AI combustion was induced by gradually increasing

Table 1 Specifications of experimental engine

Engine type	4-stroke single-cylinder liquid-cooled
Bore x Stroke (mm)	100 x 82.6
Displacement (cm ³)	648.7
Compression ratio	10.0
Valve diameter (mm x number)	Intake : $\phi 36 \times 2$ Exhaust : $\phi 31 \times 2$
Valve drive	Hydraulic-electric (Lotus AVT)
Valve timing (1mm lift)	Exhaust opening : 35 deg BBDC
	Exhaust closure : -120 ~ 0 deg ATDC
	Intake opening : -120 ~ 0 deg ATDC
	Intake closure : 35 deg BBDC
Valve lift (mm)	6
Fuel system	Port injection
Injection pressure	350 kPa (gauge)
Fuel	Gasoline (RON 91)

the EGR rate. The EGR rate was controlled within a range from 10 to 80% by varying the degree of negative overlap. Symmetrical negative overlap (SNO), in which the exhaust valve closing timing and the inlet valve opening timing are symmetrical, was employed to simplify phenomena in the test. Figure 2 shows an example of the valve lift curve. Pressure sensors and thermocouples were positioned in the exhaust and inlet pipes to help enable study of the gas exchange process. In-cylinder pressure was measured using a piezoelectric pressure sensor fitted to the cylinder head. In order to help enable correction of in-cylinder pressure, the absolute in-cylinder pressure close to bottom dead center was measured using a semiconductor pressure sensor fitted to the side-wall of the cylinder. Combustion phenomena in the transition from SI to AI combustion were analyzed by means of analysis of pressure and exhaust gases.

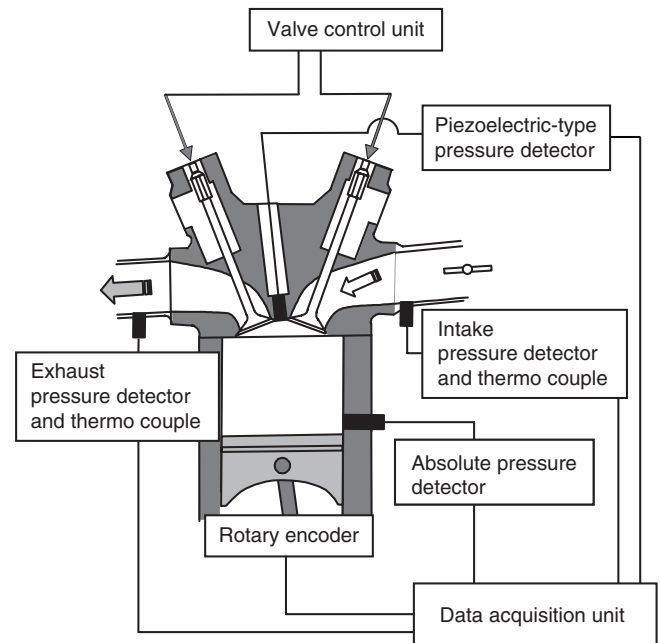


Fig. 1 Layout of detectors

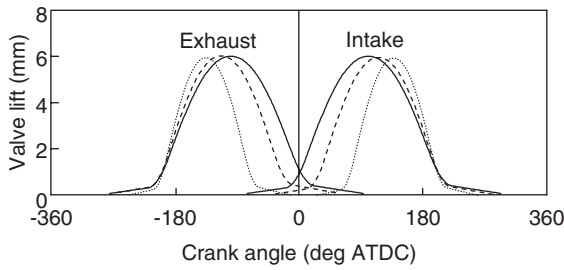


Fig. 2 Variations in valve events

4. Application of ΔT Concept to AI Combustion Control

As indicated in Chapter 2, the physical meaning of ΔT is an increase in the temperature of the in-cylinder working gas following ignition. Given this, it is possible to determine the difference between the maximum combustion temperature and the temperature at ignition timing in tests. Figure 3 shows the correlation between theoretical ΔT , as calculated using Eq. (2) and ΔT as obtained from tests. ΔT was determined in tests from an indicator diagram analysis when the indicated mean effective pressure (IMEP), equivalence ratio ϕ , and EGR rate ξ were varied at an engine speed of 2000 rpm. Theoretical ΔT was unambiguously defined based on ϕ and ξ alone, and was calculated by using figures for ϕ and ξ corresponding to test figures in Eq. (2). The solid circles in the figure show results for SI combustion, while the clear circles show those for AI combustion.

When ΔT is high, figures for test ΔT fall below figures for theoretical ΔT . This is believed to be due to the fact that the heat capacity of the in-cylinder working gas declines in proportion to the magnitude of ΔT , and the influences of dissociation and heat loss become conspicuous as the

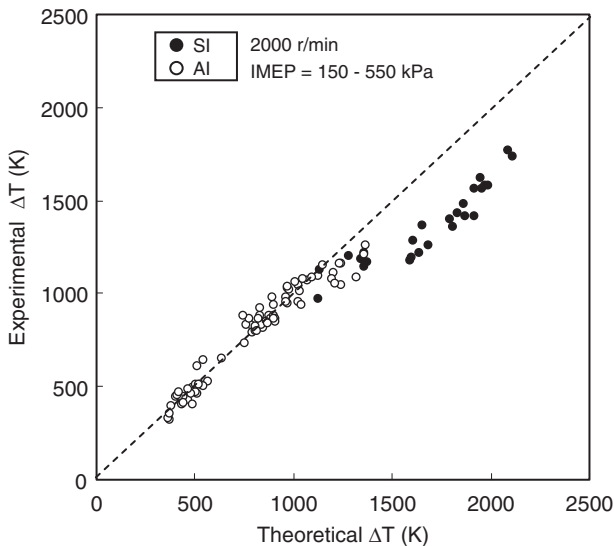


Fig. 3 Correlation between experimental and theoretical ΔT

maximum combustion temperature increases. By contrast, when ΔT is low, there is a good correlation between theoretical ΔT and test ΔT . The correlation is particularly good in the AI combustion zone. This indicates that ΔT is effective as an index for the evaluation of heat release following ignition in AI combustion.

Figure 4 shows the relationship between theoretical ΔT and the pressure rise rate, $dP/d\theta$, during AI combustion at IMEP 140-450 kPa. When the heat capacity of the working gas declines as ΔT increases, combustion becomes rapid and $dP/d\theta$ increases, resulting in an increase in engine noise, which eventually becomes knocking. If 300 kPa/deg is defined as the knocking limit, ΔT must be 1000 K or less. Meanwhile, when ΔT is 500 K or less, it is not possible to maintain combustion due to misfires. This is believed to be because the rate of heat release slows due to an excessive increase in the heat capacity of the working gas. It would therefore be possible to control proper auto-ignition without misfires or knocking by maintaining ΔT within the range of 500-1000 K.

Figure 5 shows indicated specific NO_x emissions (ISNO_x) against theoretical ΔT during auto-ignition combustion. ISNO_x remains at a low level when ΔT is less than 1000 K, but increases rapidly when ΔT increases above 1000 K. The AI temperature, T_{AI} , is approximately 1000

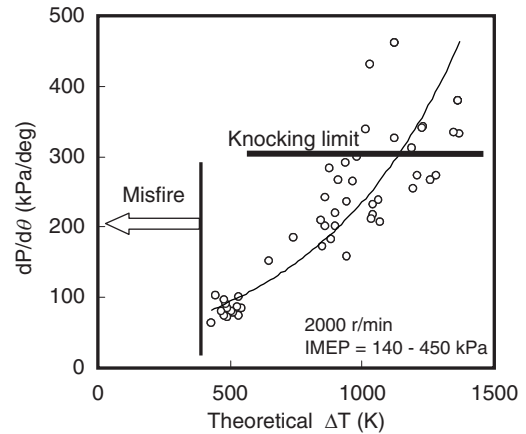


Fig. 4 Effect of ΔT on pressure rise rate

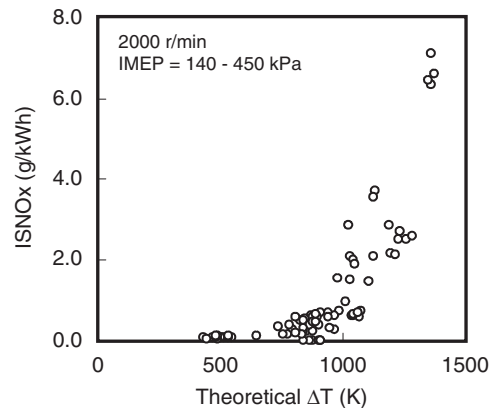


Fig. 5 Effect of ΔT on NO_x emissions

K. And $T_{AI} + \Delta T$ is the maximum combustion temperature, T_{max} . Because T_{AI} is fixed regardless of load or engine speed⁽²⁾, ΔT corresponds to the maximum combustion temperature. The increase in ISNOx is believed to be due to the fact that the maximum combustion temperature exceeds the thermal NOx production temperature (approximately 1800 K) when the heat capacity of the working gas declines and ΔT increases. It is therefore necessary to employ low-temperature combustion ($\Delta T < 1000$ K) in order to control NOx emissions to low levels during AI combustion.

5. An Approach from the Perspective of Gas Exchange

Figure 6 shows the relationship between IMEP and the in-cylinder gas exchange rate (gas charging efficiency, η_g). η_g is defined as the ratio of the mass of total gas (fresh gas and EGR gas) to the mass of fresh gas that fills the stroke volume at the reference conditions. IMEP in the figure ranges from the idle to the wide open throttle (WOT).

The solid line in the figure shows the ratio of fresh gas in the cylinder (η_c) when the equivalence ratio, ϕ , is 1. η_c is proportional to IMEP. The η_c line corresponds to zero EGR, and the IMEP in a conventional 4-cylinder engine is adjusted along this line. The curve in the upper section of the graph is the upper limit of η_g during natural aspiration, i.e., the EGR limit curve. The EGR limit curve is determined by the in-cylinder working gas temperature. Because the mass of fresh gas declines as the IMEP is reduced, the rate of EGR that can be supplied increases in proportion with the decline in IMEP. Because the in-cylinder gas temperature increases with the amount of EGR, the EGR limit curve tends to become lower as the IMEP is reduced. However, because the EGR gas temperature declines as the EGR rate increases, the EGR limit curve ceases to decline close to an IMEP of 300 kPa. The broken lines in the figure show equal EGR rates. Similar to the η_c line, the equal EGR rate lines are straight lines extending from a common point of origin. The gas exchange states between fresh gas and EGR gas that can be realized are

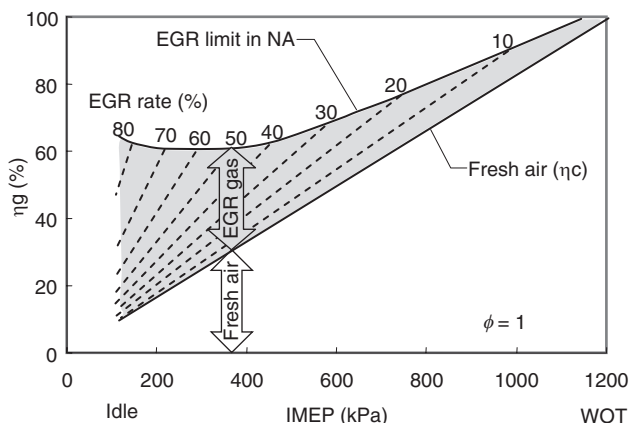


Fig. 6 Gas exchange chart

positioned between the η_c line and the EGR limit curve.

This gas exchange chart represents all the possible gas exchange states in a naturally aspirated 4-cylinder engine. Based on this chart, this paper will consider the in-cylinder gas exchange and the combustion control.

As indicated in the previous chapter, in order to realize desirable CAI, the following conditions for T_{AI} and ΔT must be satisfied: (1) $T_{AI} > 1000$ K; (2) $500 \text{ K} < \Delta T < 1000$ K. Because T_{AI} and ΔT are functions of the EGR rate and the equivalence ratio, conditions (1) and (2) can be expressed on the gas exchange chart. Figure 7 shows the CAI zone when the ΔT concept is applied to the gas exchange chart for $\phi=1$. This indicates that the upper limit of IMEP for CAI is 450 kPa, and the EGR rate is approximately 40%.

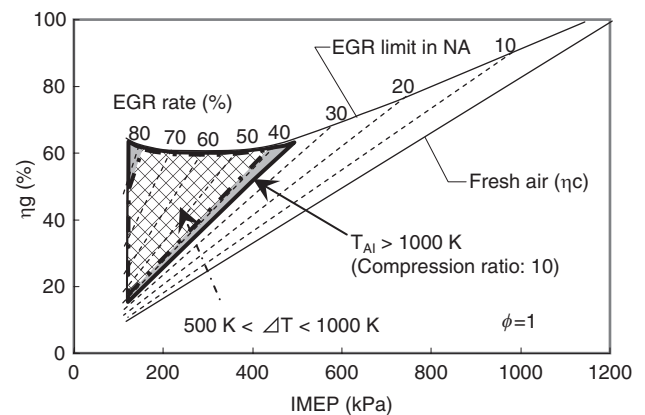


Fig. 7 Conditions for CAI obtained by applying ΔT concept to gas exchange chart

6. Positioning of Combustion Modes on Gas Exchange Chart

Figure 8 shows a plot, by combustion mode, on the gas exchange chart of the results of a firing test in which the EGR rate was varied by varying the SNO. The test was conducted at an engine speed of 2000 rpm

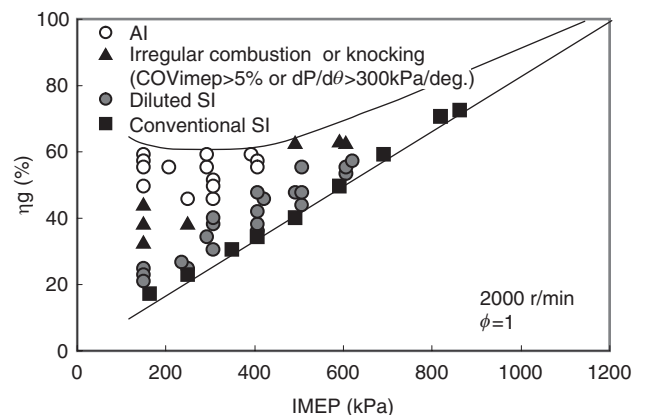


Fig. 8 Results of plotting combustion on chart

and a ϕ of 1. Clear circles show AI, solid circles show diluted SI, squares show conventional SI combustion, and triangles show irregular combustion. AI and diluted SI were distinguished by whether ignition was ON or OFF. When ignition was OFF and combustion continued, AI combustion was assumed, but when combustion could not continue and misfires occurred, diluted SI was assumed to be the combustion mode. Results for conventional SI combustion were drawn from test results at a valve timing of SNO=0 deg. Irregular combustion was defined as a combustion mode in which COVimep (Coefficient of Variation of indicated mean effective pressure) is higher than 5% or $dP/d\theta$ is higher than 300 kPa/deg. AI combustion is positioned in the low-IMEP, high- η_g zone. The possible gas exchange states in a vehicle engine should be controlled within a zone excluding the zone of irregular combustion.

7. Indicator diagrams of LTC Cycle

Figure 9 compares P-V diagrams and in-cylinder average gas temperatures for SI, diluted SI, and AI combustion at an engine speed of 2000 rpm, a ϕ of 1, and an IMEP of 300 kPa. The conventional SI results were obtained from a test conducted at a valve timing of SNO (amount of overlap) =0 deg; the EGR rate at this condition was 10%. The AI results were obtained at a valve timing of SNO=-90 deg. The EGR rate at this condition was 51%. The following characteristics are shown: (1) Rapid combustion; (2) High-pressure

combustion; (3) Low pumping loss; and (4) Low-temperature combustion (LTC). The maximum combustion temperature declines probably because of the increase in the heat capacity of the in-cylinder working gas in accordance with the increase in the EGR rate. The fact that η_g becomes high indicates that the level of LTC becomes high. The reduction in combustion temperature achieved by using a gas exchange state that promotes a high level of LTC makes it possible to reduce NOx emissions, increase thermal efficiency by reducing heat loss, and reduce pumping loss. These are the benefits of LTC.

8. Control Method using LTC

The choice of possible control of IMEP corresponds to achieving a gas exchange state in which the undesired combustion zone (irregular combustion and knocking) is avoided. Figure 10 shows three examples of IMEP control methods. The broad line shows control in conventional SI combustion. The narrow line shows a control method that passes through the diluted SI zone across the IMEP range. This can be realized at a fixed valve timing of SNO=-47 deg, but maximum IMEP is reduced by 10-20% due to the fact that EGR is trapped at a constant rate. In addition, the effect in reducing fuel consumption and NOx emissions is minimal because the level of LTC is low. The broken line shows the optimum LTC control proposed in this paper. This control method provides continuous control, offering the fuel

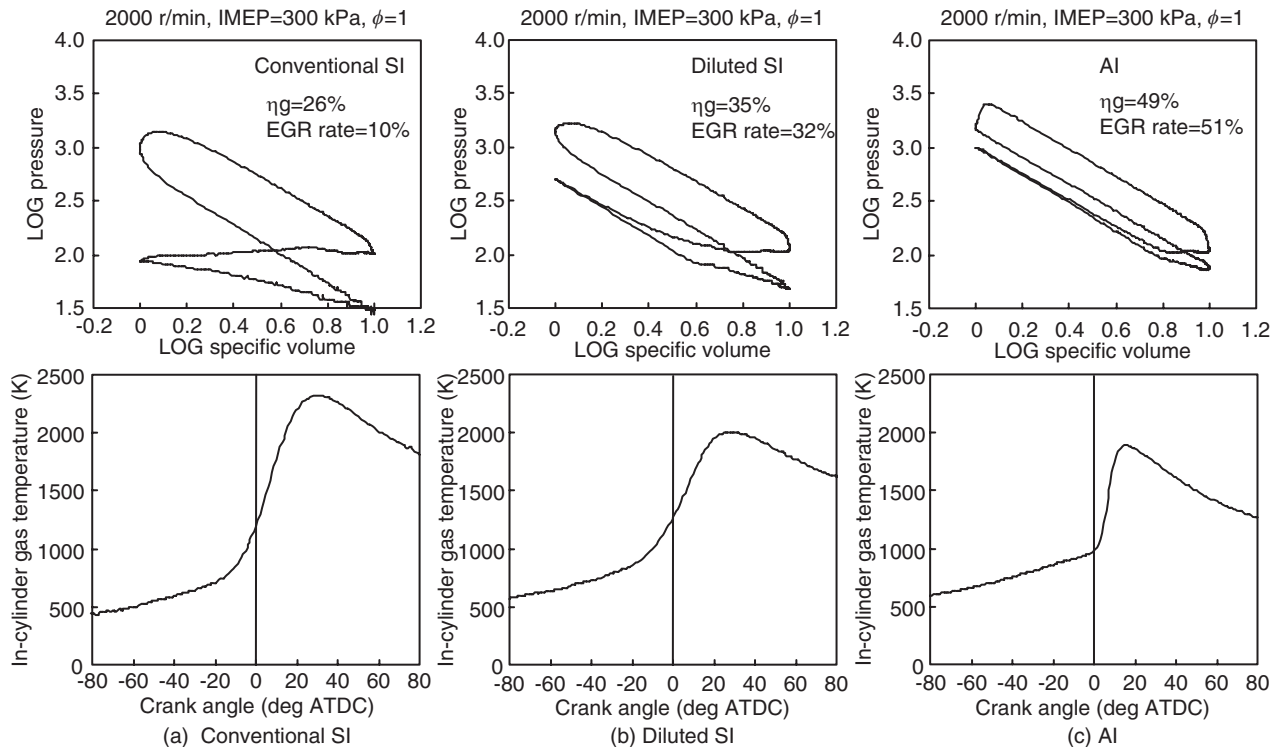


Fig. 9 Indicator diagrams for LTC cycles

consumption and emission benefits of a sequence of LTC cycles from SI combustion at the WOT to AI combustion at the idle (SI → Diluted SI → AI) throughout the IMEP range, without applying combustion mode switching. This optimum LTC control can be realized by creating the suitable gas exchange state for a certain IMEP, and does not absolutely necessitate the use of a high-speed variable valve mechanism.

Figure 11 shows SNO and degree of throttle opening against BMEP (Brake Mean Effective Pressure). At an engine speed of 2000 rpm, the optimum LTC control can be maintained by varying the valve timing so as to make SNO ranging across 0 to -90 deg. At this time CAI occurs at BMEP 300 kPa and below. The benefits of LTC, including in the diluted SI zone, can be maximized through the employment of smooth and continuous gas exchange control from maximum load (SI) to idle (AI).

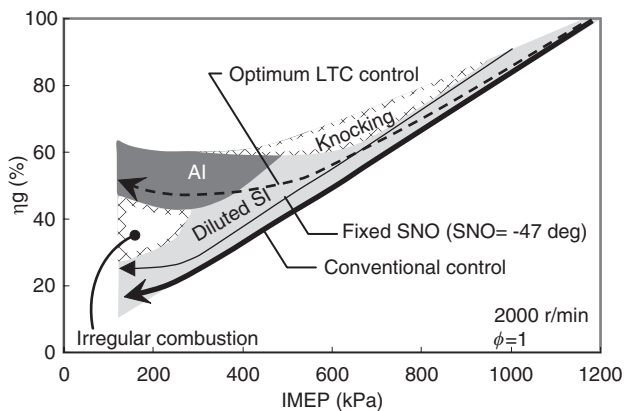


Fig. 10 Three examples of possible IMEP control

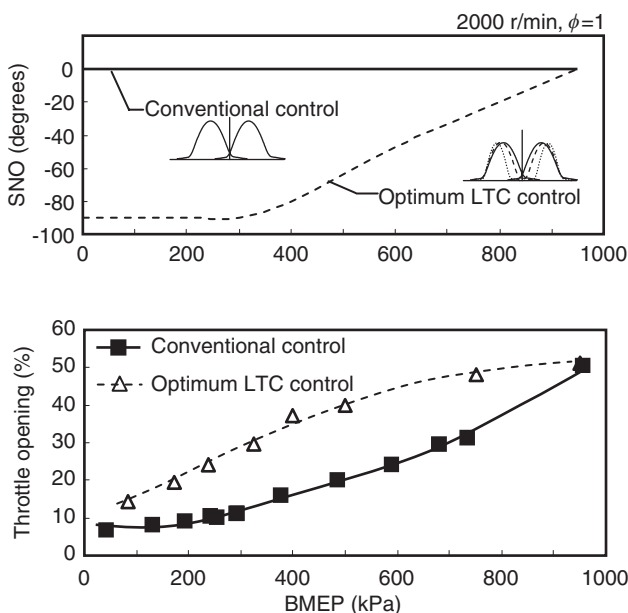


Fig. 11 SNO and throttle opening against BMEP

9. Benefits of Optimum LTC Control

Figure 12 compares BSFC (Brake Specific Fuel Consumption) and BSNO_x (Brake Specific NO_x) characteristics against BMEP for conventional SI and optimum LTC control. The bar graph at the top of the figure shows the distribution of BMEP frequency in LA-4 mode for a 1300 cm³ V-twin motorcycle. If the fuel consumption and NO_x emissions of the V-twin motorcycle are estimated from this data, reductions of approximately 20% in fuel consumption and approximately 90% in NO_x emissions can be expected in LA-4 mode.

Figure 13 shows ISFC (Indicated Specific Fuel Consumption) and PMEP (Pumping Mean Effective Pressure) for conventional SI and optimum LTC control. The use of optimum LTC control reduces ISFC in the low

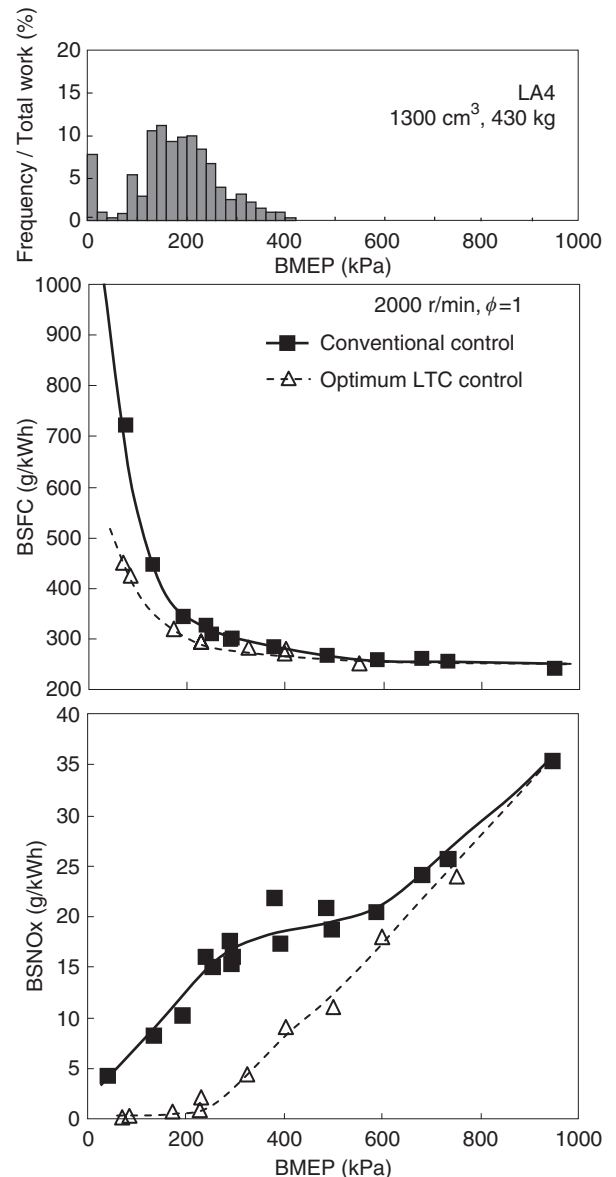


Fig. 12 BSFC and BSNO_x characteristics against BMEP

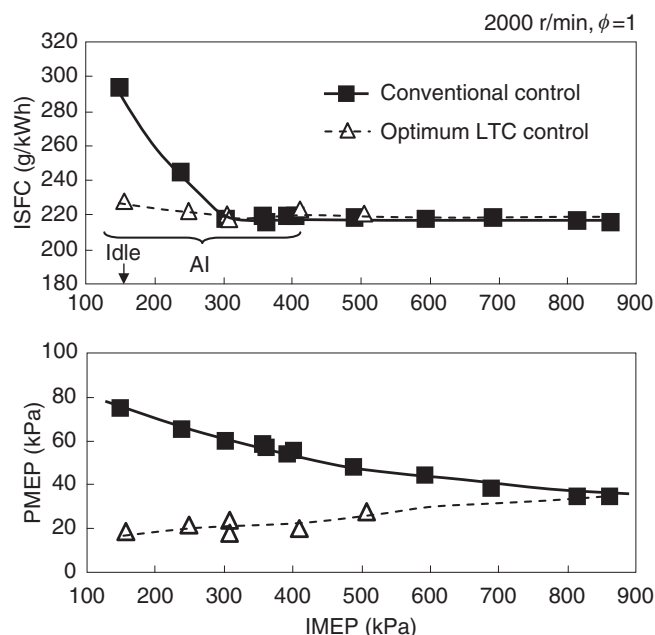


Fig. 13 ISFC and PMEP characteristics against IMEP

zone at IMEP 300 kPa and below. The combustion mode in this zone is AI. The control also reduces PMEP across the BMEP range.

10. Conclusion

A macroscopic understanding of physical parameters for the control of CAI ignition timing and heat release has produced the following experimental findings:

- (1) The auto-ignition temperature, T_{AI} , and ΔT are parameters for the realization of desirable CAI. T_{AI} expresses ignition timing, and ΔT expresses heat release following ignition.
- (2) Maintaining ΔT within the range from 500-1000 K makes it possible to control proper AI without misfires or knocking.

The adoption of an approach from the perspective of gas exchange in relation to the application of AI combustion to vehicle engines produced the following findings:

- (3) A gas exchange chart that shows all the possible gas exchange states in a 4-stroke naturally aspirated engine was proposed.
- (4) Combustion maps were formulated by plotting combustion modes including AI combustion, diluted SI combustion, and irregular combustion on this gas exchange chart.
- (5) A control method (optimum LTC control) was proposed that “continuously” selects the appropriate LTC state for each load on the combustion maps discussed in (4). This control method makes it possible to smoothly employ combustion modes from SI combustion to AI combustion without applying switching, and to maximize the fuel consumption and emission benefits of a sequence of LTC

states, including in the diluted SI zone.

- (6) It is estimated that the application of the optimum LTC control to a 1300 cm³ V-twin motorcycle would reduce fuel consumption by approximately 20% and NOx emissions by approximately 90% in LA-4 mode compared to conventional combustion methods.

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