

Two-Stroke Gas Exchanging Method for Controlling Auto-Ignition : Activated Radical Combustion

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ABSTRACT

The auto-ignition is focused as a combustion process, where improving efficiency and lowering NOx emission could be compatible. A current subject would be controlling combustion process. Honda demonstrated a method to control the auto-ignition, i.e. Activated Radical (AR) combustion, in several commercial motorcycle engines.

Through the experimental analyses of the AR combustion, this article shows that control of the gas exchange generates a temperature for the auto-ignition in the end of compression stroke, due to heat exchange between residual and new mixture. It is also revealed that its combustion speed is affected by the engine speed, scavenging efficiency, and etc.; however, 10% to 90% mass burn timing is settled on the same crank-angle duration as a result regardless of the engine operating conditions. This is owing to the dualism of the residual gas, i.e. combustion promotion action as heat energy potential and the inhibitory action as the inert gas. Both actions should balance for the two-stroke gas exchanging method for controlling auto-ignition.

1. Introduction

Simultaneously improving fuel economy and reducing emissions is a theme for engines. In particular, improving the thermal efficiency of automotive engines, which are said to account for 20% of the total CO₂ emitted by developed countries, is an important issue. Theoretically, thermal efficiency should be improved by a higher compression ratio and lean operation; however, practically, it has limits. A higher compression ratio is limited by the occurrence of knocking and the structure of the engine. And again, lean operation is also limited by the ignitability of the mixture and the NOx deoxidization by the catalyst. Even direct injection diesel engines, which are known as the most effective

internal combustion engines, due to their higher compression ratio, have not become mainstream in automotive engines due to the issue of reducing NOx and soot produced by stratified combustion. Gas direct injection engines, which aim to improve fuel economy owing to the super lean burn operation and its lower pump loss results from non-throttle operation, also basically face these similar issues.

Recently, auto-ignition is focused as a new combustion process that aims to simultaneously improve fuel economy and clean emissions^{(1), (2), (3)}. This type of combustion, which does not have a flame propagation process, should be extremely promising as a means of simultaneously improving fuel economy and reducing emissions. This is because it provides superior ignition stability with a lean mixture or

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large amounts of diluted inert gas, which expands the lean burn limit, reduces the combustion temperature, reduces NOx emission, avoids knocking, increases the compression ratio limit, and so on. However, in terms of combustion control this technology is still incomplete, because producing auto-ignition itself is a primary barrier, especially for four-stroke gas engines. Heating the intake air and using a higher compression ratio were tried in the initial research stages, and the prescribed effects were confirmed in specific operating states^{(4), (5)}. However, in this case the operation used to produce auto-ignition acted to simultaneously accelerate the combustion speed. This meant that the ignition timing and the combustion speed changed both together, which greatly limited the operating conditions under which ideal combustion could be achieved.

On the other hand, in cases where auto-ignition is applied to a two-stroke gas engine, the ignition timing is controlled by varying the gas exchange state using a throttle valve located in the exhaust system⁽⁶⁾. Honda has already applied a controlled auto-ignition system, named Activated Radical (AR) combustion to commercial motorcycle engines, and has demonstrated a significant fuel economy improvement and emissions reduction effects^{(7), (8)}. This AR combustion is generated by changing the percentage of inert gas in the cylinder by varying the opening area of the exhaust port with a valve, which is to heat or reform⁽⁹⁾ the fresh air by using the thermal energy of the inert gas. In this case, the dualism of the inert gas, that is to say its thermal energy and inertness, act differently on the generation of auto-ignition and the subsequent combustion. In other words, the thermal energy of the inert gas causes the gas inside the cylinder to reach the auto-ignition temperature at the end of the compression stroke, whereas the increase in the inert gas works as a factor to suppress the combustion speed and the combustion temperature.

This paper introduces an example of auto-ignition control in two-stroke engines, and clarifies the combustion control mechanism peculiar to these types of AR combustion engines through the experiments concerning the combustion speed. It aims to deepen understanding of the two-stroke engine gas exchange process^{(10), (11), (12)}, which is recently focused as a means of the auto-ignition also in the four-stroke gasoline engines, and to investigate further application possibility of gasoline auto-ignition combustion in the future.

2. Experimental Engine

In the case of conventional two-stroke engines, there is a unique relationship between the amount of inert gas and the amount of short-circuiting fresh air in the cylinder. AR combustion is controlled by varying the rate of inert gas in the cylinder, consequently the amount of short-circuiting fuel also changes with the combustion state, which complicates the analysis of individual phenomena. Therefore, these experiments used a pneumatic direct injection (PDI) engine in order to eliminate the effects of short-circuiting fuel as much as possible. The experimental engine shown in Fig. 1 has a rotary type mechanical injection valve in the cylinder wall. After the scavenging process using air only, the pre-measured fuel for one cycle is injected directly into the cylinder using the gas pressure in the accumulator chamber. Therefore, the fresh air short-circuited to the exhaust does not include the fuel. The principle of this PDI engine has already been reported in detail^{(13), (14)}, so these details are omitted here. A valve, which varies the opening area in order to change the

amount of inert gas for the AR combustion is installed in the exhaust port.

Two types of experimental engines were used in the experiments: engine A with a displacement of 250 cm³, and engine B with a displacement of 400 cm³ as shown in Table 1. The PDI assist gas supply method differs according to the operating speed range of each engine. The high-speed engine A (12,000 r/min) supplies the assist gas by using a special air pump, while the medium speed engine B (8,000 r/min) supplies it by using the in-cylinder gas accumulated in the chamber at the compression stroke. Therefore, engine A has a higher degree of freedom for

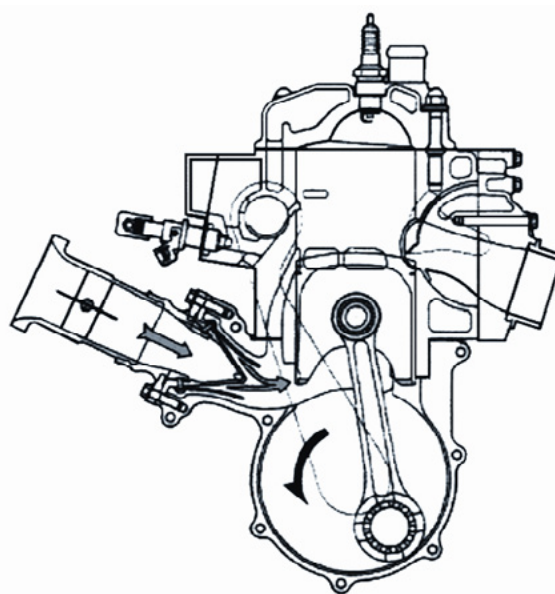


Fig. 1 Overview of the experimental engine B

Table 1 Principal specifications of the Experimental Engines

Specifications	Engine A	Engine B
Bore × Stroke	68mm × 68.8mm	80mm × 80mm
Displacement	249.9cm ³	402cm ³
Compression ratio (Trapped)	7.2	6.3
Compression ratio (Geometric)	12.4	10.5
Exhaust opening duration	194deg.C.A.	188deg.C.A.
Exhaust opening area	2 437cm ² deg	2 159cm ² deg
Scavenging opening duration	130deg.C.A.	113deg.C.A.
Scavenging opening area	1 712cm ² deg.	1 449cm ² deg.
Charge means for PDI	Air pump	In-cylinder gas
Exhaust opening rate	100%–5%	100%–5%

adjusting the amount of assist gas. Other basic structures and operation principles are exactly the same, so that there should be no significant difference in the auto-ignition combustion itself between these two engines.

3. AR Combustion Control Analysis from the Gas Temperature Aspect

3.1. Estimation of Auto-ignition Temperature

When the exhaust port is throttled in order to generate the auto-ignition by maintaining a constant indicated mean effective pressure, the exhaust recirculation increases, and a rise of in-cylinder pressure is observed at the exhaust port close timing. The scavenging efficiency (η_s) is assumed to deteriorate due to this increase in the relative charge (Crel). In order to predict a change in the in-cylinder gas temperature of this case, the enthalpic balance models of the working gas during the exhaust process and the scavenging process are constructed.

For simplification, models stand on the following assumptions;

- (1) The temperature of the fresh air, inert gas, exhaust and the in-cylinder gas after scavenging are typified by the average temperatures T_f , T_r , T_e and T_i , respectively.
- (2) Labeling the isobaric specific heat of each gas as cp_f , cp_r , cp_e and cp_i , respectively, $cp_f = cp_r = cp_e = cp_i$.
- (3) Heat exchange between each gas is performed adiabatically and instantly.
- (4) The compression stroke is polytropic change.

Labeling the mass of the fresh air and inert gas in the cylinder as M_f and M_r , respectively, the mass of the fresh air and inert gas in the exhaust at a trapping efficiency of η_r are $M_f(1/\eta_r - 1)$ and M_r , respectively.

From the enthalpic balance of the exhaust process:

$$M_f cp_r T_r + M_f(1/\eta_r - 1) cp_f T_f = M_f/\eta_r cp_e T_e$$

From the enthalpic balance of the scavenging process:

$$M_f cp_f T_f + M_r cp_r T_r = (M_f + M_r) cp_i T_i$$

The scavenging efficiency $\eta_s = M_f/(M_f + M_r)$, thus,

$$T_r = (T_e - T_i)/\eta_r + T_i \quad (1)$$

$$T_i = T_r - (T_r - T_i) \eta_s \quad (2)$$

$$T^* = T_i (\varepsilon^*)^{n-1} \quad (3)$$

Here, T^* and ε^* are the in-cylinder gas temperature and specific volume at the ignition point, respectively and n is the mean polytropic index of the compression stroke.

3.2. In-cylinder Gas Temperature in the AR Combustion Engine

The experimental PDI-engines control the exhaust valve opening by using a computer so that auto-ignition always occurs at the point several degrees before the top dead center (TDC), where the combustion efficiency is the best⁽⁴⁾. That is to say, the exhaust valve opening for the AR combustion is pre-programmed in the ECU, according to each engine speed and throttle opening.

This AR combustion control system was analyzed from the in-cylinder gas temperature aspect, using experimental engine B. Figure 2 shows an example of varying the load at a speed of 3,000 r/min, and Fig. 3 shows the gas exchange state at that time.

The residual gas temperature and the in-cylinder gas temperature at the end of the compression stroke were obtained successively, according to the method introduced in the previous section, by using the direct measurement results

such as the average exhaust gas temperature, average scavenging gas temperature, intake air amount, exhaust O_2 concentration, indicator diagram, power output and the model test results of Blair's single cycle gas exchange rig⁽¹⁵⁾ for the scavenging efficiency characteristic of the experimental engine.

As gradually decreasing the load from the full load, the exhaust temperature peaks at a brake mean effective pressure (BMEP) of 400 kPa and decreases thereafter. This phenomenon occurs because the inert gas is diluted by the short-circuiting fresh, while the estimated residual gas temperature peaks at the full load and then decreases according to the load decrease as shown in Fig. 2. This drop in the residual gas temperature is understood that the combustion temperature decreases as the inert gas increases, and also it is due to a decrease of the total heat release to the heat loss.

On the other hand, the in-cylinder gas temperature after the gas exchange depends on the heat exchange between the residual gas and the fresh air. As shown in Fig. 3, decreasing the load, the scavenging efficiency decreases, the rate of the residual gas increases, thus the in-cylinder gas temperature rises. When the gas temperature reaches $1,000 \text{ K} + \alpha$ the AR combustion occurs. After that, the exhaust valve controls the gas exchange state so as to maintain this temperature constant at $1,000 \text{ K} + \alpha$, as seen in the dotted line enclosed area in the

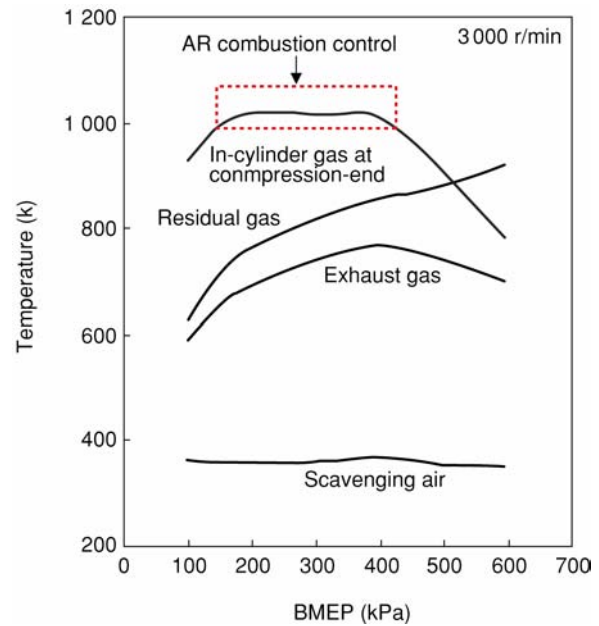


Fig. 2 In-cylinder gas temperature in the AR engine

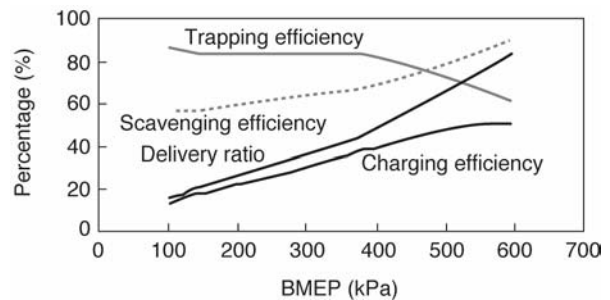


Fig. 3 Gas exchange state in the AR engine

figure. This area is the AR combustion zone for the experimental engine at 3,000 r/min. Note that the value of α is a constant depending on the engine speed, which tended to rise on the order of 10-odd K in accordance with each 1,000 r/min engine speed rise. Decreasing the load further from this AR combustion zone, the residual gas temperature drops excessively so that the in-cylinder gas temperature can no longer be maintained. However, the exhaust valve controls the scavenging efficiency, which is the AR combustion lower limit. Irregular combustion occurs in the load area below this lower limit.

3.3. AR Combustion Zone

The exhaust valve controls the mixing state of the residual gas and fresh air, i.e. scavenging efficiency, so as to reach an in-cylinder gas temperature which induces the AR combustion. Such gas exchange state for the AR combustion is obtained in the range of the scavenging efficiency approximately from 30% to 70%⁽⁴⁾. There occurs the irregular combustion in the scavenging efficiency range of approximately 30% or less. Figure 4 compares the AR combustion operating zones for experimental engines A and B. The upper limit load for AR combustion operation is a BMEP of less than 400 kPa, regardless of differences in the engine output characteristics, and the lower limit load characteristics exhibit similar tendencies. In addition, AR combustion was observed up to the extremely high-speed range.

The AR combustion lower limit mechanism was explained in the former section as a drop in the residual gas temperature. When the coolant temperature and other adiabatic conditions vary, the lower limit load fluctuates 10-odd kPa order due to the change of the residual gas temperature. In addition, the effect of change in the compression ratio on the residual gas temperature is canceled by the change in the expansion ratio, it only affects the fresh air temperature rise. Thus, there is a little effect on the AR lower limit load, as far as the normal compression ratio range, which allows the full load operation. However, an exceedingly higher compression ratio would enlarge the lower limit load, and the upper limit load. The auto-ignition zone can be expanded in this manner, however it would be limited by the occurrence of the knocking in the higher load side.

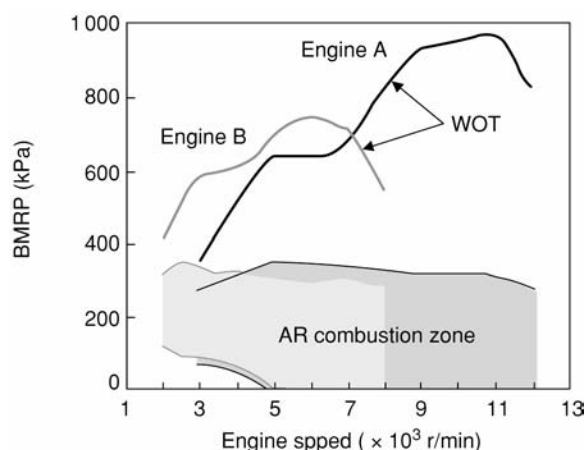


Fig. 4 AR combustion zones of the experimental engines

3.4. Ignition Timing Control

In the AR combustion zone, the ignition timing can be controlled in a stable manner by changing the scavenging efficiency. The spark ignition does not affect the AR combustion timing in this zone. Exhaust valve operation for experimental engine B was switched to manual, and the scavenging efficiency was changed by both the throttle and the exhaust valve opening, while maintaining a constant load of BMEP = 300 kPa and a constant speed of 3,000 r/min. Lowering the scavenging efficiency, that is to say increasing the inert gas, the mass fraction burnt curve against the crank angle advances, as shown in Fig. 5. This advance in the combustion timing is limited by the onset of detonation or misfiring. Conversely, increasing the scavenging efficiency retards the combustion timing, and it is limited by the onset of irregular combustion. Incidentally, Under the Fig. 5 conditions, the maximum thermal efficiency was marked at a scavenging efficiency of 0.54. The figure also represents that, in a scavenging efficiency more than 0.59, auto-ignition cannot be maintained and irregular combustion results. The adjustment width of the auto-ignition timing gradually narrows toward the retardation side around the surrounding AR combustion zone. In addition, a typical auto-ignition combustion, namely “semi-AR combustion”, which occurs with the assistance of the spark ignition is seen in a narrow range of the outer surroundings of the AR combustion zone.

4. Auto-ignition Combustion Speed

The previous section explained the mechanism of the auto-ignition timing control, however combustion speed control is another important factor of the combustion control. For example, in spark ignition (SI) engines the flame propagation speed increases in accordance with the engine speed, therefore combustion always takes place approximately at the same crank angle. This is an extremely important characteristic for the OTTO cycle, and thanks to this, constant volume combustion is performed from the several hundred r/min of idling state up to the racing engine's speeds as high as 20,000 r/min. It is explained that as the engine speed rises, due to the increase of the in-cylinder turbulence, the flame propagation also speeds up. It would naturally come to raise a question in the case of auto-ignition, which does not involve the flame propagation process.

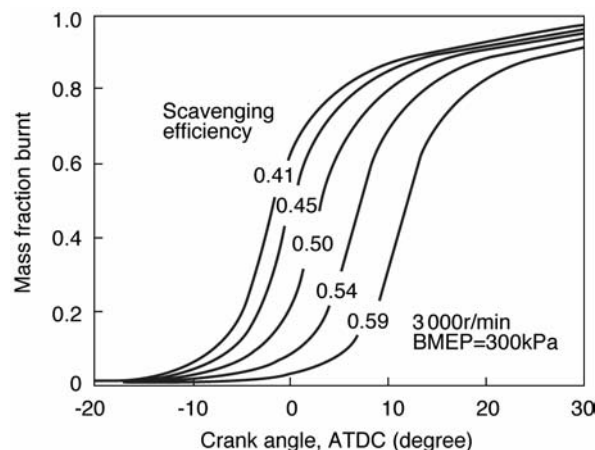


Fig. 5 Effect of scavenging efficiency on mass fraction burnt curve

4.1. Relationship between Load and Combustion Speed

Figure 6 shows the heat release curve variations against the load variations of the 250 cm³ experimental engine A. The combustion form shifts from SI combustion to AR combustion as the load decreases. The pressures noted on the heat release curves represent the respective indicated mean effective pressures (IMEP). The stoichiometric in-cylinder air to fuel ratio is given for all loads and the ignition timing for the SI combustion is the minimum advance for best torque (MBT). Expressing the amount of PDI assist gas as a mass-rate, which symbol is Las, of the assist gas against the air required to fill the swept volume under the reference condition, Las of 5% is given at all examined loads. The upper two curves, dotted lines, are SI combustion. The third dotted curve of 420 kPa is semi-AR combustion, which requires assistance of the spark ignition, and the lower four curves by the solid lines are AR combustion. According to the slopes of these heat release curves, which indicate the combustion speed, Fig. 6 shows a general tendency that the combustion speed decreases as the load decreases for both SI and AR combustion. This could be explained that deterioration in the scavenging efficiency, i.e. increase in the in-cylinder inert gas, controls the combustion speed. However the graph also shows that combustion speeds up in accordance with the onset of the AR combustion.

Reading the crank angles for the mass burnt fraction of 10% and 90% of each curve, the same data are rearranged on the IMEP – crank angle axes in Fig. 7. The 10% mass burnt fraction timing for AR combustion is controlled, so that it is the same as that of the SI combustion applying MBT. According to this graph, when the combustion shifts from SI to AR, the 90% mass burnt fraction timing advances. Which means that a degree of constant volume combustion increases in the AR combustion. The combustion speeds down as the load decreases, however at the same time the mass to be burnt also decreases, consequently, combustion always takes place at the same crank angle regardless of the load. In the case of SI combustion, the initial combustion time from the ignition timing up to the 10% mass burnt increases as the load decreases, which would be also related to the increase of inert gas concentration.

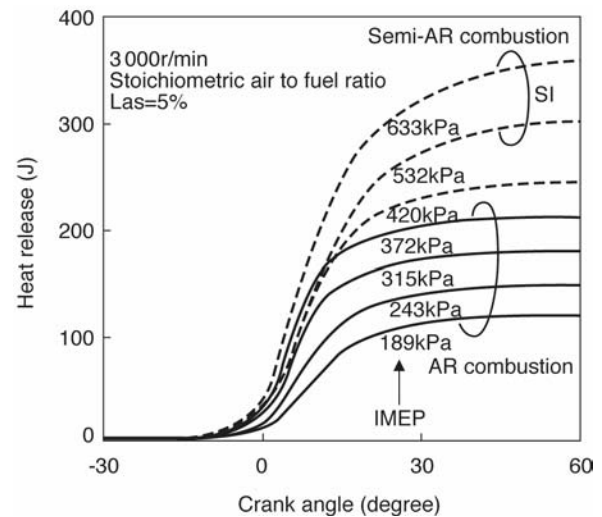


Fig. 6 Heat release variations against the load

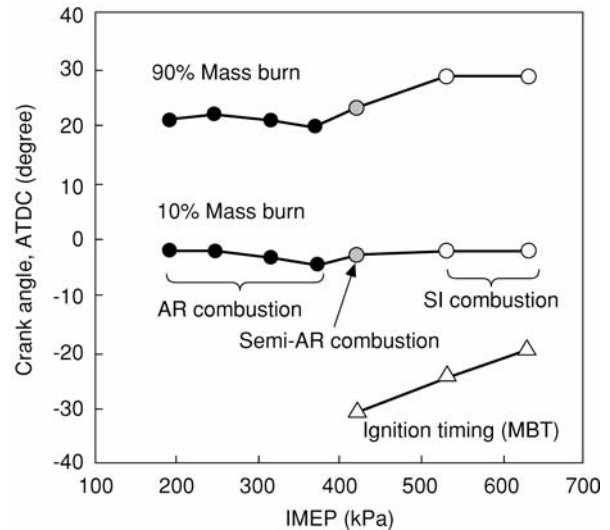


Fig. 7 Combustion duration on the crank angle against IMEP

4.2. Relationship between Engine Speed and Combustion Speed

The next issue is the influence of the engine speed on the AR combustion speed. Experimental results thus far indicate ;

- (1) The scavenging efficiency is adjusted to induce the AR combustion, i.e. the exhaust enthalpy performs the auto-ignition at the end of the compression stroke.
- (2) The scavenging efficiency itself affects the AR combustion speed.

As it had been empirically known that in the case of the constant IMEP, the scavenging efficiency for inducing the AR combustion was also constant in rough. Therefore, the relationship between the engine speed and the AR combustion rate was tested at a constant IMEP. Figure 8 shows the heat release curve variations of the engine speed from 3,000 r/min to 6,000 r/min at a constant IMEP of 300 kPa, using experimental engine A. The horizontal axis is the time from TDC. The in-cylinder air to fuel ratio is the stoichiometric condition for all engine speeds, and the amount of assist gas is held constant at Las = 5%. This data reveals that the AR combustion speeds up as the engine

speeds up.

The same data is rearranged in Fig. 9, showing the crank angles for the mass burnt fractions of 10% and 90%. AR combustion occurs at approximately the same duration as the crank angle, regardless of the engine speed.

This result shows that the AR combustion speeds up with the engine speed, in the same manner as the SI combustion, however it does not have a flame propagation process. This remarkable feature allows the auto-ignition to maintain the ideal constant volume combustion even in the high-speed range.

4.3. Relationship between Mixture Formation and Combustion Speed

In order to investigate the influence of homogeneousness of the in-cylinder mixture on the AR combustion, the amount of PDI assist gas were varied. PDI of the experimental engine shows a tendency that with a larger amount of assist gas it promotes diffusion of the spray and promotes the homogeneousness of the mixture, and inversely with a smaller amount of assist gas, it becomes biased spray.

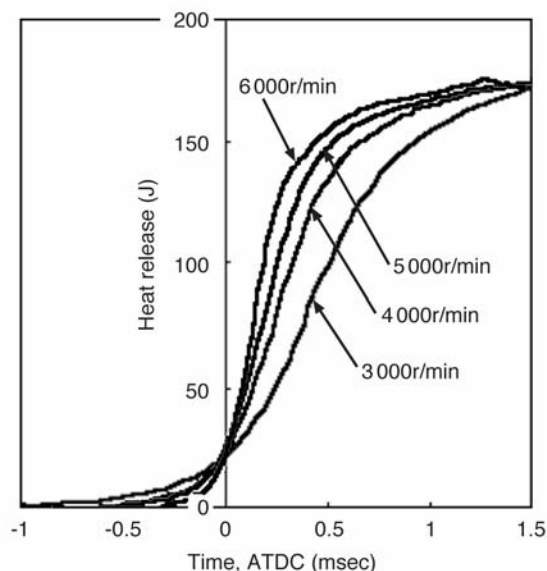


Fig. 8 Heat release speed variations against r/min

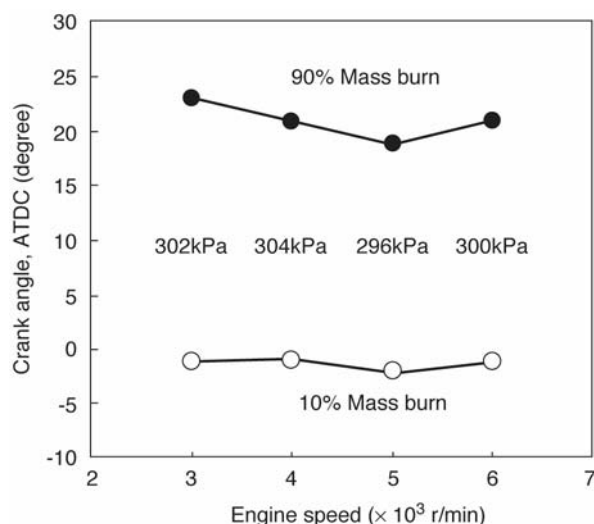


Fig. 9 AR combustion duration against the engine speed

The amount of PDI assist gas was varied between 1%, 8% and 16%, while maintaining a constant IMEP of 300 kPa at an engine speed of 4,000 r/min using experimental engine A. The stoichiometric in-cylinder air to fuel ratio was applied. There is no significant change in the exhaust valve opening required for the AR combustion in all cases of assist gas amounts, consequently, it is estimated that the scavenging efficiency would also be approximately the same in all cases. As increasing Las, BMEP decreases due to increase of the pump loss, and the brake mean specific fuel consumption ratio (BSFC) also increases accordingly as shown in Fig. 10. In this case, the indicated mean specific fuel consumption ratio (ISFC) remains roughly constant. Which means that the AR combustion efficiency remains unchanged and the mechanical loss simply increases. However, as shown in Fig. 11, the emission composition between HC and NOx is affected markedly according to the difference in the assist gas amount.

From the indicated pressure data aspect, a remarkable difference in the heat release between the cases of Las = 1% and Las = 16% was observed, as shown in Fig. 12. In case of Las = 16%, the heat release reaches the maximum value at

20° after top dead center (ATDC). In contrast, in case of Las = 1%, it reaches the maximum value around 40° ATDC. From this result, the difference in the amount of PDI assist gas, that is to say heterogeneity of the mixture formation is also a factor, which affects the AR combustion speed, like other factors such as the scavenging efficiency and engine speed.

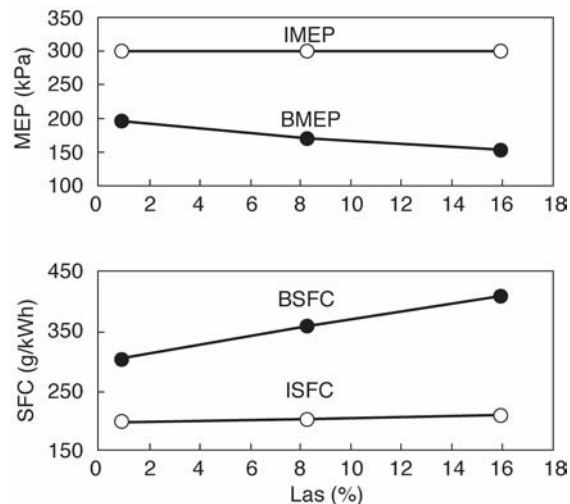


Fig. 10 Effect of assist gas of the PDI on BMEP and SFC in the AR combustion

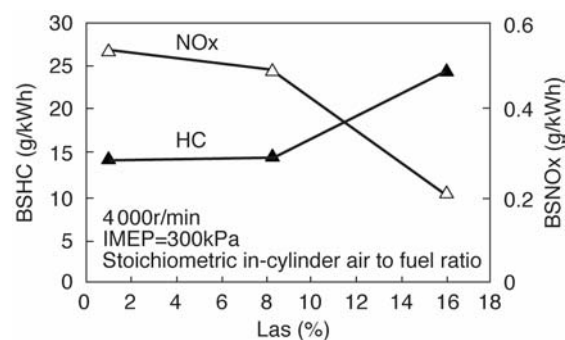


Fig. 11 Effect of assist gas of the PDI on HC NOx emissions in the AR combustion

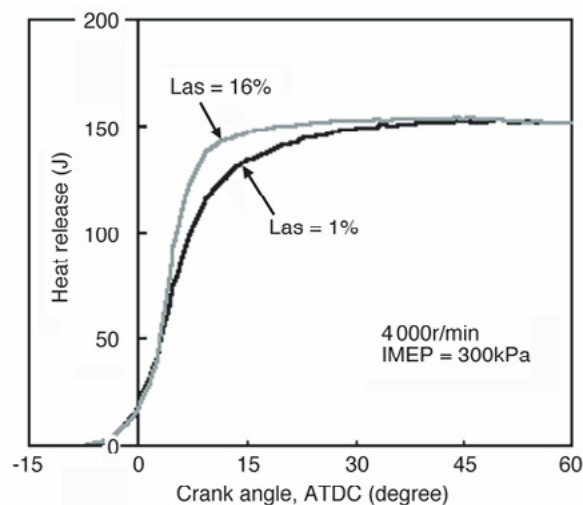


Fig. 12 Effect of assist gas of the PDI on the combustion speed of the AR combustion

5. Conclusion

- (1) The AR combustion ignition timing is controlled according to the in-cylinder gas temperature at the end of the compression stroke.
- (2) This gas temperature is maintained at a constant degree during the AR combustion zone, by varying the gas exchange with an exhaust valve and utilizing the enthalpy of the residual gas.
- (3) The auto-ignition combustion speed depends on the scavenging efficiency, engine speed and other factors, but the combustion duration always comes about a constant crank timing, regardless of the operating condition.
- (4) A heterogeneity of the mixture formation also affects the heat release characteristic and the exhaust gas composition in a PDI engine.

These experimental data thus reveal that the controlled auto-ignition applying the two-stroke gas exchanging method is a result of a well balance in the dualism of the inert gas, i.e. combustion promotion due to its thermal energy and combustion control due to its inertness. An unbalanced state of it results in irregular combustion. As a methodology for the controlled auto-ignition, it is becoming a trend to apply the two-stroke engines' gas exchanging process to four-stroke gasoline engines.

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