

# Combustion Characteristics of Syngas and Natural Gas in Micro-pilot Ignited Dual-fuel Engine

Ulugbek Azimov, Eiji Tomita, Nobuyuki Kawahara, and Sharul Sham Dol

**Abstract**—The objective of this study is to investigate the combustion in a pilot-ignited supercharged dual-fuel engine, fueled with different types of gaseous fuels under various equivalence ratios. It is found that if certain operating conditions are maintained, conventional dual-fuel engine combustion mode can be transformed to the combustion mode with the two-stage heat release. This mode of combustion was called the PREMIER (PREmixed Mixture Ignition in the End-gas Region) combustion. During PREMIER combustion, initially, the combustion progresses as the premixed flame propagation and then, due to the mixture autoignition in the end-gas region, ahead of the propagating flame front, the transition occurs with the rapid increase in the heat release rate.

**Keywords**—Combustion, dual-fuel engine, end-gas autoignition, PREMIER.

## I. INTRODUCTION

TO satisfy increasingly strict emissions regulations, engines with alternative gaseous fuels are now widely used. Renewable energy sources such as synthesis gas (syngas) appear to be greener alternatives for internal combustion engines [1-3]. The means of utilizing natural gas in SI engines are already well established and documented, whereas its use in CI engines is still under development. Some gas engines fueled with syngas have been developed recently [4-6]. Most of them have a spark-ignition (SI) combustion system. An SI engine is not suitable for this kind of fuel under high load conditions because of the difficulty in achieving stable combustion due to the fluctuation of the syngas components. In addition, the syngas is a low energy density fuel and the extent of power degrading is large when compared with high-energy density fuels like gasoline and natural gas.

Nevertheless, under present conditions, economic factors seem to provide the strongest argument of considering the use of syngas as fuel [3]. In many situations where the price of petroleum fuels is high or where supplies are unreliable, the syngas can provide an economically viable solution. Syngas is

produced by gasifying a solid fuel feedstock such as coal or biomass. For example, the biomass gasification means incomplete combustion of biomass resulting in production of combustible gases. Syngas consists of about 40% combustible gases, mainly carbone monoxide (CO), hydrogen (H<sub>2</sub>) and methane (CH<sub>4</sub>). The rest are non-combustible gases and consists mainly of nitrogen (N<sub>2</sub>) and carbon dioxide (CO<sub>2</sub>). Varying proportions of CO<sub>2</sub>, H<sub>2</sub>O, N<sub>2</sub>, and CH<sub>4</sub> may be present [7].

H<sub>2</sub> as a main component of a syngas has very clean burning characteristics, a high flame propagation speed and wide flammability limits. H<sub>2</sub> has a laminar combustion speed about eight times greater than that of natural gas, providing a reduction of combustion duration and as a result, an increase in the efficiency of internal combustion (IC) engines, if the H<sub>2</sub> content in the gaseous fuel increases. Main point of interest in increasing H<sub>2</sub> content in the gaseous fuel is that with the addition of H<sub>2</sub>, the lean limit of the gas operation can be extended, without going into the lean misfire region. Lean mixture combustion has a great potential to achieve higher thermal efficiency and lower emissions [8]. In particular, the lean mixture combustion will result in low and even extremely low NO<sub>x</sub> levels with only a slight increase in hydrocarbons [9, 10].

A number of researchers have performed experiments to determine engine performance and exhaust emissions in dual-fuel engines. Their results indicate that lower NO<sub>x</sub> and smoke can be achieved in dual-fuel engines compared with conventional diesel engines, while maintaining the same thermal efficiency as a diesel engine. McTaggart-Cowan et al. [11] investigated the effect of high-pressure injection on a pilot-ignited, directly injected natural gas engine. Su and Lin [12] studied the amount of pilot injection and the rich and lean boundaries of natural gas dual-fuel engines. Tomita et al. [13] investigated the combustion characteristics and performance of the supercharged syngas with micro-pilot ignition in a dual-fuel engine. Karim et al. [14, 15] examined the ignition characteristics in a dual-fuel engine and studied techniques for improving the performance of dual-fuel engines under light load. Liu and Karim [16] concluded that the observed values of the ignition delay in dual-fuel operation are strongly dependent on the type of gaseous fuels used and their concentrations in the cylinder charge.

The afore mentioned results suggest that if certain operating conditions are maintained, including the control of pilot fuel injection quantity, pressure and timing, gaseous fuel equivalence ratio, and EGR rate, a compromise between

Ulugbek Azimov is with Curtin University, Miri, Sarawak, Malaysia (phone: +60-085-443939; fax: +60-085-443837; e-mail: u.azimov@curtin.edu.my).

Eiji Tomita is with Okayama University, Japan (e-mail: tomita@mech.okayama-u.ac.jp).

Nobuyuki Kawahara is with Okayama University, Japan (e-mail: kawahara@mech.okayama-u.ac.jp).

Sharul Sham Dol is with Curtin University, Miri, Sarawak, Malaysia (e-mail: sharulsham@curtin.edu.my).

increased efficiency and low exhaust emissions can be achieved. In this paper, we document the range of operating conditions under which the new higher-efficiency PREMIER (PREmixed Mixture Ignition in the End-gas Region) combustion mode was experimentally tested. The objective of this work is to investigate PREMIER combustion and emission characteristics in a pilot ignited supercharged dual-fuel engine fueled with natural gas and syngas, and to study the effect of H<sub>2</sub> and CO<sub>2</sub> content in syngas on the combustion and emission formation over the broad range of equivalence ratios under lean conditions.

## II. EXPERIMENTAL PROCEDURE AND CONDITIONS

This study used water-cooled four-stroke single-cylinder engine, with two intakes and two exhaust valves, shown in Fig. 1.

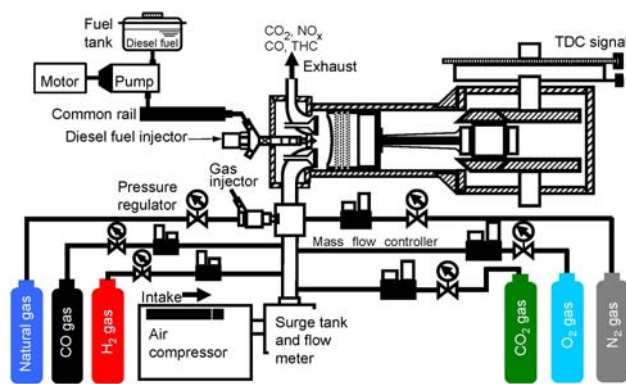


Fig. 1 Experimental engine layout

In this engine, the autoignition of a small quantity of diesel pilot fuel, injected into the combustion chamber before top dead center, initiates the combustion. The burning diesel fuel then ignites the gaseous fuel. The pilot fuel was ultra low-sulfur (<10ppm) diesel. Diesel fuel properties are given in JIS standards. A commercial solenoid-type injector that is typically used for diesel-only operations was modified to ensure a small quantity of injected fuel. A nozzle of the commercial injector with seven holes was replaced by the one with four holes of 0.1 mm in diameter, and with three holes of 0.08 and 0.1 mm in diameter. Diesel fuel injection timing and injection duration were controlled through the signals transferred to the injector from the injector driver. A common rail injection system (ECD U2-P, Denso Co.) was employed to supply the constant injection pressure to the injector. The common rail fuel injection pressure varied from 40 MPa to 150 MPa, and the injected pilot diesel fuel quantity was 2 mg/cycle and 3 mg/cycle. The experimental conditions and different types of primary gaseous fuel compositions used in this study are given in Table I and Table II, respectively. The reason for using several types of syngas is that the range of compositions found in syngas varies substantially. The different compositions of syngas depend on the different feedstock, like biomass, coal or refinery residues. Therefore, the evaluation of engine combustion under different syngas compositions is necessary

in order to verify efficient engine operational flexibility for this type of fuel.

The in-cylinder pressure history of combustion cycles was measured with KISTLER-6052C pressure transducer in conjunction with a 0.5° crank-angle encoder to identify the piston location, and the rate of heat release (ROHR) was calculated from these data. The combustion transition from the first stage (slow flame propagation) to the second stage (end-gas autoignition) is identified from the ROHR. CO and NO<sub>x</sub> emissions were measured with a four-component analyzer Horiba PG-240, smoke was measured with a smoke meter Horiba MEXA-600s and HC emissions were measured with Horiba MEXA-1170HFID.

TABLE I  
EXPERIMENTAL CONDITIONS

Engine type	4-stroke, single cylinder, water cooled
Bore x Stroke	96x108 mm
Swept volume	781.7 cm <sup>3</sup>
Compression ratio	16
Combustion system	Dual-fuel, direct injection
Combustion chamber	Shallow dish
Engine speed	1000 rpm
Intake pressure	101 kPa, 200 kPa
Injection system	Common-rail
Pilot fuel injection pressure	
- Natural gas case	40 MPa, 80 MPa, 120 MPa, 150 MPa
- Syngas case	80 MPa
Pilot fuel injection quantity	
- Natural gas case	2 mg/cycle, 3 mg/cycle
- Syngas case	3 mg/cycle
Nozzle hole x diameter	
- Natural gas case	3x0.08 mm, 3x0.10 mm, 4x0.10 mm
- Syngas case	4x0.10 mm
Equivalence ratio	
- Natural gas case	0.6
- Syngas case	Variable
EGR rate	
- Natural gas case	10%, 20%, 30%, 40%, 50%
- Syngas case	none
EGR composition	N <sub>2</sub> -86%, O <sub>2</sub> -10%, CO <sub>2</sub> -4%

TABLE II  
GAS COMPOSITION

Gas type	Composition					LHV (MJ/kg)	Source
	H <sub>2</sub> (%)	CO (%)	CH <sub>4</sub> (%)	CO <sub>2</sub> (%)	N <sub>2</sub> (%)		
Type 1	13.7	22.3	1.9	16.8	45.3	4.13	BMG
Type 2	20.0	22.3	1.9	16.8	39.0	4.99	BMG
Type 3	56.8	22.3	1.9	16.8	2.2	13.64	COG
Type 4	13.7	22.3	1.9	23.0	39.1	3.98	BMG
Type 5	13.7	22.3	1.9	34.0	28.1	3.74	BMG
Type 6	56.8	5.9	29.5	2.2	5.6	38.69	COG
Type 7	56.8	29.5	5.9	2.2	5.6	20.67	COG
Type 8	100.0	-	-	-	-	119.93	Hydrogen
	CH <sub>4</sub> (%)	C <sub>2</sub> H <sub>6</sub> (%)	C <sub>3</sub> H <sub>8</sub> (%)	n-C <sub>4</sub> H <sub>10</sub> (%)		LHV (MJ/kg)	
Natural gas	88.0	6.0	4.0	2.0		49.20	

## III. CONCEPT OF PREMIER COMBUSTION

Before giving a description to PREMIER (PREmixed Mixture Ignition in the End-gas Region) combustion, it is necessary to explain the differences of phenomenological outline between conventional combustion and knocking combustion in the dual-fuel engine. Conventional combustion

is a combustion process which is initiated by a timed pilot ignited fuel and in which the flame front moves completely across the combustion chamber in a uniform manner at a normal velocity. Knocking combustion is a combustion process in which some part or all of the charge may be consumed (auto-ignited) in the end-gas region at extremely high rates. Much evidence of end-gas mixture auto-ignition followed by knocking combustion can be obtained from high-speed laser shadowgraphs [17], high-speed Schlieren photography [18], chemiluminescent emission [19], and laser-induced fluorescence [20]. In addition, Stiebels et al. [21] and Pan and Sheppard [22] showed that multiple auto-ignition sites occur during knocking combustion. The combustion mode we have monitored differs from knocking combustion in terms of the size, gradients, and spatial distribution of the exothermic centers in the end-gas. This combustion concept was given a name PREMIER combustion.

A conceptual outline of PREMIER combustion is presented in Fig. 2. In the first stage of this combustion mode, the pilot diesel fuel is injected, evaporated, and auto-ignited prior to top dead center (TDC). The energy released by the diesel fuel auto-ignition initiates the gaseous flame development and outward propagation from the ignition centers toward the cylinder wall. Once the end-gas region is sufficiently heated and the temperature of the fuel mixture has reached the auto-ignition temperature of the gaseous fuel/air mixture after TDC, the second-stage combustion begins and is completed as the gas expands and cools, producing work. The second-stage heat release occurs over a chemical reaction timescale and is faster than heat release by turbulent flame propagation. Thus, the combustion transition from the first stage to the second stage takes place when the overall heat release rate changes from the slower first-stage flame rate to the faster second-stage rate, and that transition is here measured as the point where the second derivative of the heat release rate is maximized, as shown in Fig. 2.

PREMIER combustion in a dual-fuel engine is comparable to combining SI and CI combustion, which is being investigated by several researchers [23-25]. One disadvantage of these combustion strategies is that they are difficult to control under lean mixture conditions. The spark discharge is very short, and under light load and lean mixture conditions, the flame is too weak to propagate strongly and may be extinguished. Therefore, the combustion chamber must be specially designed to facilitate a stratified mixture charge around the spark plug electrodes. In a dual-fuel engine, on the other hand, combustion of the injected diesel fuel proceeds concurrently with that of the gaseous fuel mixture. This slow combustion of the diesel fuel helps to maintain the natural gas flame propagation and prevents the misfires that may occur under lean mixture conditions. The lean limit for the gaseous fuel/air mixture is of practical importance, as lean operation can result in both higher efficiency and reduced emissions.

The major benefit of lean operation is the accompanying reduction in combustion temperature, which leads directly to a significant reduction in NO<sub>x</sub> emissions. The lean limit is the point where misfire becomes noticeable, and it is usually described in terms of the limiting equivalence ratio  $\phi_{lim}$  that supports complete combustion of the mixture.

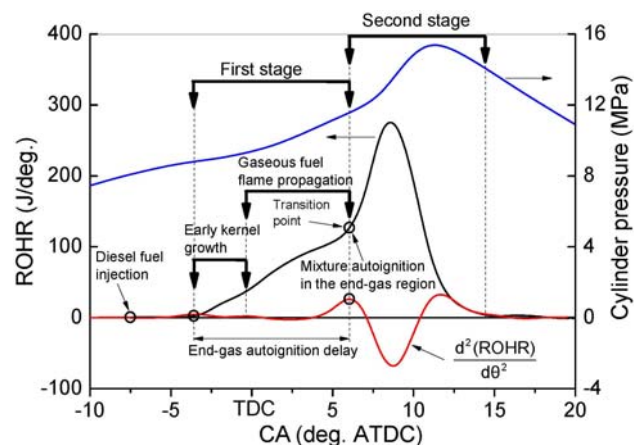


Fig. 2 PREMIER combustion concept

For example, with 13A natural gas, if we operate slightly above the limiting equivalence ratio (for methane,  $\phi_{lim} \sim 0.5$  [26]), the mixture reactivity becomes very sensitive to even very small variations in the air-fuel ratio. This high sensitivity is due to the presence of n-butane in 13A natural gas. It is known that small changes in the volume fraction of n-butane strongly affect the ignition properties of natural gas [27-30]. During hydrocarbon fuel oxidation, an H-atom is more easily abstracted from an n-butane molecule (with two secondary carbon atoms) than from other hydrocarbons such as methane, ethane, or propane [31]. As the equivalence ratio increases, the n-butane mass fraction in the natural gas/air mixture increases proportionally. During a fuel oxidation reaction, the in-cylinder gas temperature rises, and more of the radicals that initiate methane oxidation are created by increasing the ratio of n-butane. Similar results were documented by other researchers who investigated the auto-ignition and combustion of 13A natural gas in an HCCI engine [32]. They found that very small increases in the equivalence ratio of the methane/n-butane/air mixture produced significant changes in the profiles of the in-cylinder pressure traces.

If not extinguished during the early combustion stage, the pilot diesel flame may continue to a later stage, and pilot flame energy contributes to the stability of the combustion process [33]. The remaining unburned in-cylinder mixture from the first stage, located beyond the boundary of the flame front, is then subjected to a combination of heat and pressure for certain duration. As the flame front propagates away from the primary ignition points, end-gas compression raises the end-gas temperature and pressure. When the mixture is preheated throughout the combustion chamber volume and the end-gas mixture reaches the auto-ignition point, simultaneous auto-ignition occurs in several limited locations (known as exothermic centers), with a sharp increase in heat release. This part of the combustion appears as a rapid energy release in the second stage of the heat release curve shown in Fig. 2.

A prime requirement for maintaining PREMIER combustion mode in a dual-fuel engine is that the mixture must not auto-ignite spontaneously during or following the rapid release of pilot energy. Failure to meet this requirement can lead to the onset of knock, which manifests itself in excessively sharp

pressure increases and overheating of the walls, resulting in significant loss of efficiency with increased cyclic variations. When much smaller pilots are used, the energy release during the initial stages of ignition and the resulting turbulent flame propagation can (under certain conditions) lead to auto-ignition of the charge well away from the initial ignition centers, in the end-gas regions ahead of the propagating flames. This can occur in a manner that resembles the occurrence of knock in spark-ignition engines, but with controlled heat release. For the sake of convenience, the total energy release rate during PREMIER combustion can be divided into three sequential components. The first of these is due to the ignition of the pilot fuel. The second is due to the combustion of the gaseous fuel in the immediate vicinity of the ignition centers of the pilot, with consequent flame propagation. The third is due to auto-ignition in the end-gas region.

#### IV. PREMIER COMBUSTION CHARACTERISTICS

##### A. Natural Gas

###### 1. Effect of Injection Timing

To maintain PREMIER combustion in a dual-fuel natural gas engine, the effects of several operating parameters must be identified. Our experimental results show that the major parameters that may significantly influence the energy release pattern during dual-fuel PREMIER combustion are pilot diesel fuel injection timing and the EGR rate, which can affect the total equivalence ratio based on oxygen content. Other parameters such as pilot fuel injection pressure, injected pilot fuel amount, nozzle hole diameter, and hole number have minor effects on PREMIER combustion. For example, as shown in Fig. 3, PREMIER combustion can be maintained within a wide range of pilot fuel injection pressures. However, it can be maintained within only a very narrow range of fuel injection timings. At a fixed total equivalence ratio, advancing the injection timing resulted in the earlier occurrence of combustion during the cycle, increasing the peak cylinder pressure during first-stage combustion. With the burned gas of the first-stage combustion, the in-cylinder pressure and temperature continued to rise after TDC, as shown in Fig. 3. Although the piston began to move downward after TDC, and the volume thus expanded, the heat release from first-stage combustion induced local temperature and pressure increases during second-stage combustion. Higher peak cylinder pressures resulted in higher peak charge temperatures. Retarding the injection timing decreased the peak cylinder pressure during first-stage combustion, as more of the fuel burned after TDC. Advancing the injection timing resulted in better diesel fuel evaporation and mixing with the in-cylinder gas. Therefore, diesel fuel auto-ignition occurred more quickly and with more complete diesel fuel combustion and natural gas flame propagation during the first stage, resulting in rapid combustion and high heat release rate during the second stage due to the rapid heating of the end-gas region mixture.

###### 2. Effect of EGR

PREMIER combustion becomes clearly recognizable if the EGR rate remains below a certain level. When the EGR rate surpasses this level, the unburned mixture temperature decreases, retarding the combustion of the natural gas and affecting the reactivity of the mixture to auto-ignite in the end-gas region. As Fig. 4 shows, at 200 kPa of intake pressure and moderate EGR rates, the first-stage combustion rate increased, and second-stage heat release was able to occur. A similar trend was also observed at 101 kPa of intake pressure, although not shown here. These results suggest that the use of EGR may not be advantageous for achieving PREMIER combustion. However, it should be noted that for engine operation close to the knock-limit conditions, the high combustion rate of natural gas may be markedly decreased by using a certain limited EGR rate and maintaining PREMIER combustion mode, as the knocking effect is suppressed.

##### B. Syngas

Fig. 5 indicates the relationship between the rate of maximum pressure rise and the second stage autoignition delay. The second stage autoignition delay time was estimated as time between two peaks of the second derivative of the ROHR as shown in Fig. 2. It was found that for all types of gases investigated in this paper the autoignition delay of the second stage decreases with the increase of the rate of maximum pressure rise.

###### 1. Effect of H<sub>2</sub> Content on PREMIER Combustion

An increase in hydrogen content of syngas results in an increase of ignitability and a corresponding reduction in ignition delay of the first stage. To investigate the effect of mass fraction burned in the second stage and the effect of hydrogen content on the rate of maximum pressure rise, the ratio  $MFB_{2stHR}/MFB_{total}$  was evaluated.  $MFB_{2stHR}$  is the integral of the mass fraction burned during the second stage, computed from the transition crank angle degree, where the peak of  $d^2(ROHR)/d\theta^2$  is maximized, to the crank angle degree where MFB reaches 80%.  $MFB_{total}$  is the integral of the total mass fraction burned, computed from the crank angle degree at the first peak of  $d^2(ROHR)/d\theta^2$  to the crank angle degree where MFB reaches 80%. Fig. 6 shows that as the mass fraction burned increases the rate of maximum pressure rise also increases. The same trend was monitored at various equivalence ratios.

Fig. 7 shows the in-cylinder pressure and the ROHR for Type 1 (A) and Type 2 (B) of syngas combustion at different equivalence ratios for various injection timings.

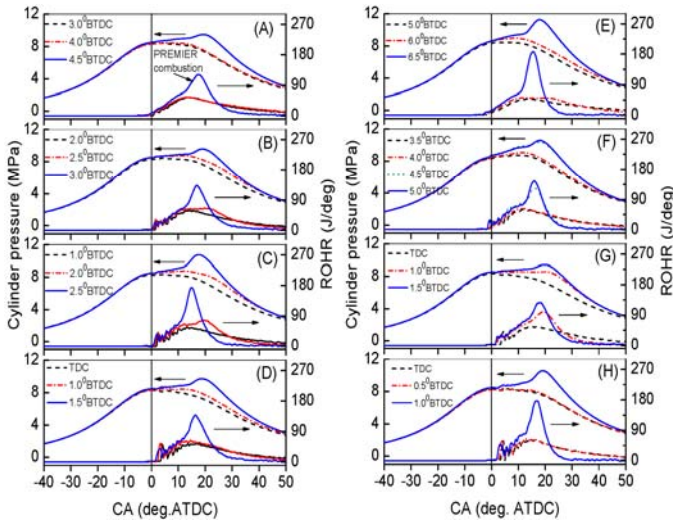


Fig. 3 Effect of pilot fuel injection timing on cylinder pressure and the rate of heat release. See table below.

	Injection pressure $P_{inj}$	Intake pressure $P_{in}$	Nozzle hole dia. $D_{hole}$	Nozzle hole num $N_{hole}$	Injected pilot fuel amount $m_{df}$
A	40 MPa	200 kPa	0.1 mm	3	3 mg/cycle
B	80 MPa	200 kPa	0.1 mm	3	3 mg/cycle
C	120 MPa	200 kPa	0.1 mm	3	3 mg/cycle
D	150 MPa	200 kPa	0.1 mm	3	3 mg/cycle
E	40 MPa	200 kPa	0.1 mm	3	2 mg/cycle
F	80 MPa	200 kPa	0.1 mm	3	2 mg/cycle
G	150 MPa	200 kPa	0.08 mm	3	3 mg/cycle
H	150 MPa	200 kPa	0.1 mm	4	3 mg/cycle

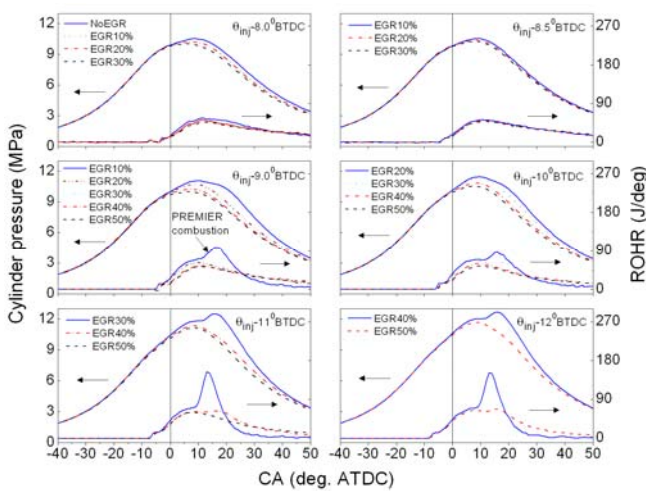


Fig. 4 Effect of EGR on cylinder pressure and the rate of heat release.  $P_{inj}=40$  MPa,  $P_{in}=200$  kPa,  $D_{hole}=0.1$  mm,  $N_{hole}=3$ ,  $m_{df}=2$  mg/cycle

The results show that maximum pressures and heat release rates reach higher values for the gas with the higher  $H_2$  content. As the injection timing is gradually advanced the second stage of heat release occurs. The two-stage heat release rate was

observed at various equivalence ratios at the certain injection timings.

For instance, for Type 1 at  $\phi=0.4$  two-stage heat release appeared starting from  $23^\circ BTDC$ , at  $\phi=0.52$  from  $15^\circ BTDC$ , at  $\phi=0.68$  from  $9^\circ BTDC$  and at  $\phi=0.85$  from  $7^\circ BTDC$ . The same trend was observed for Type 2, however, the maximum peak of the heat release rate of Type 2 syngas is higher than that of Type 1 due to the higher  $H_2$  content.

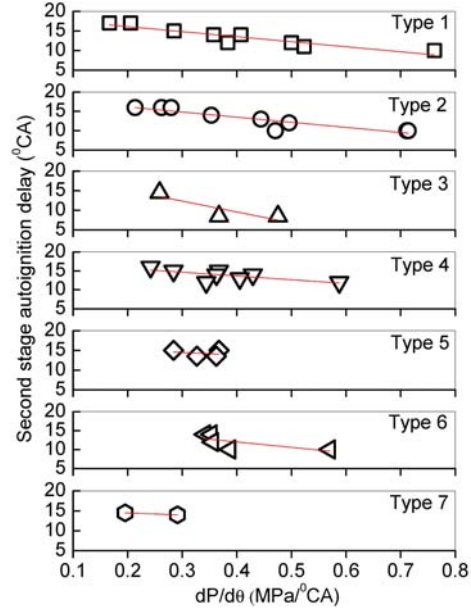


Fig. 5 Second stage autoignition delay.  $P_{inj}=80$  MPa,  $P_{in}=200$  kPa,  $m_{df}=3$  mg/cycle

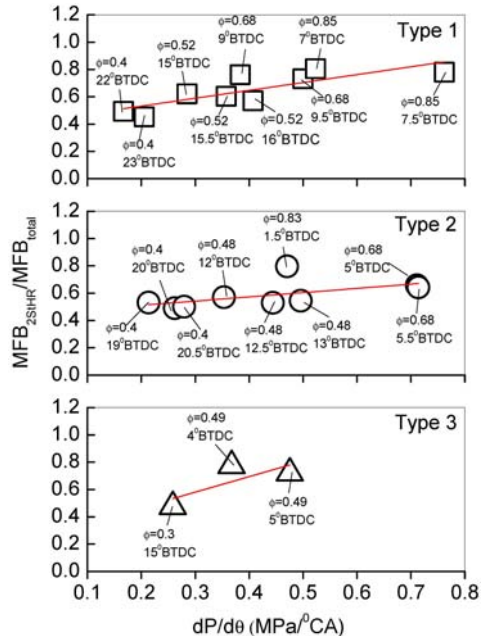


Fig. 6 Fuel mass fraction burned with the change of  $H_2$  content

The maximum cylinder pressure decreased at an equivalence ratio of 0.83 of Type 2 because in that case injection timing needed to be retarded to around TDC to avoid knocking. On the

other hand, for Type 1 at an equivalence ratio of 0.85 and injection timings before TDC, two-stage heat release rate was clearly observed without any occurrence of knocking combustion. These results may suggest that increased H<sub>2</sub> content in Type 2 gas affected the ignitability and corresponding progress of combustion leading to the engine knock.

Fig. 8 shows the effect of H<sub>2</sub> content on the mean combustion temperature, IMEP, indicated thermal efficiency and NO<sub>x</sub> for conventional and PREMIER combustion at the same input energy  $Q_{in}=2300$  J/cycle and injection timing at the minimum advance for best torque (MBT). As this figure shows, the increase in H<sub>2</sub> amount affects the engine combustion characteristics. For PREMIER combustion, the mean combustion temperature and consequently the NO<sub>x</sub> significantly increase when compared with those of conventional combustion. IMEP and indicated thermal efficiency increase about 10%. Therefore, in dual-fuel combustion of low-energy density syngas, PREMIER combustion is an important combustion mode that tends to increase the engine efficiency.

## 2. Effect of CO<sub>2</sub> Content on PREMIER Combustion

An increase in CO<sub>2</sub> content in syngas results in a dilution of the mixture with the corresponding reduction in the rate of fuel oxidation reactions and consequent combustion. To investigate the effect of mass fraction burned in the second stage and the effect of CO<sub>2</sub> content on the rate of maximum pressure rise, the same procedure was applied as explained in the previous section.

Fig. 9 shows that for Type 1 and Type 4, as the mass fraction burned during the second stage increases, the rate of maximum pressure rise also increases. However, for Type 5 with 34% of CO<sub>2</sub> content in syngas, the rate of maximum pressure rise decreases with the increase of the mass of fuel burned in the second stage.

This can be explained by the fact that although the total mass of syngas burned during the second stage increases, the CO<sub>2</sub> fraction in the gas also proportionally increases. Eventually, the certain threshold can be reached when the effect of CO<sub>2</sub> mass fraction in the gas on combustion overweighs the effect of H<sub>2</sub>.

Fig. 10 shows the effect of CO<sub>2</sub> content on the mean combustion temperature, IMEP, indicated thermal efficiency and NO<sub>x</sub> for conventional and PREMIER combustion at the same input energy  $Q_{in}=2300$  J/cycle and injection timing at the minimum advance for best torque (MBT). In this figure the trend mentioned earlier in Fig. 9 is clearly observed.

The mean combustion temperature, IMEP, the indicated thermal efficiency and the NO<sub>x</sub> shows the increase when CO<sub>2</sub> content in the gas increases from 16.8% to 23%. However, as CO<sub>2</sub> concentration reaches 34%, above mentioned combustion characteristics decrease. This trend was observed for both conventional and PREMIER combustion.

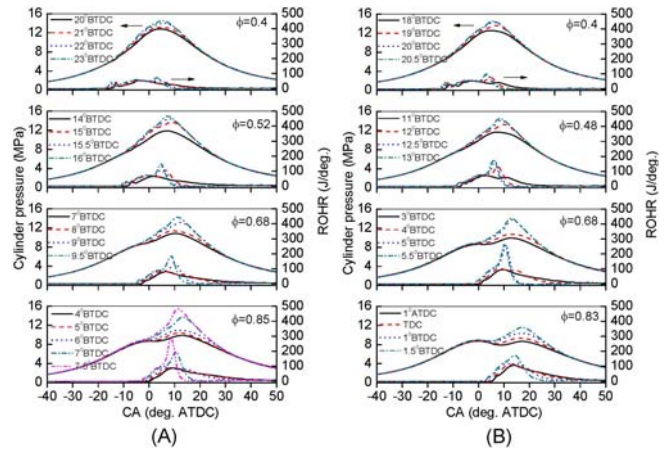


Fig. 7 Comparison of in-cylinder pressure and ROHR. Syngas, (A) Type 1, (B) Type 2

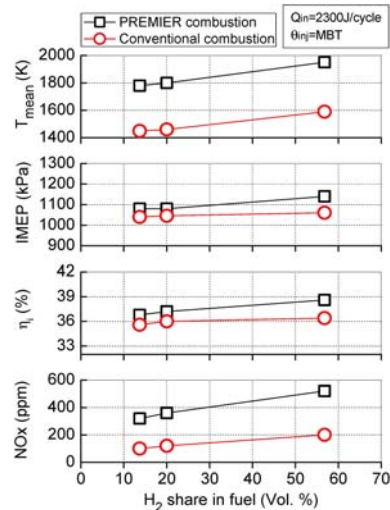


Fig. 8 Effect of H<sub>2</sub> content on engine performance characteristics at  $\theta_{inj}=MBT$

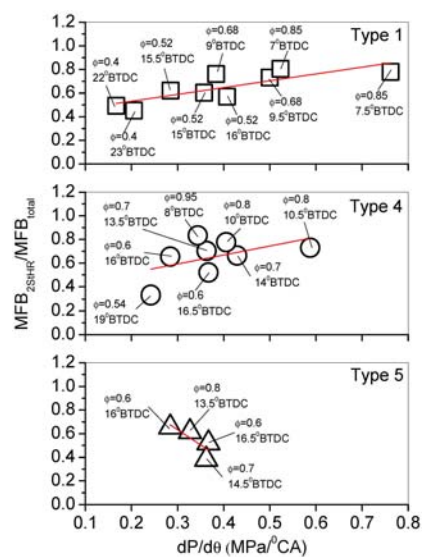


Fig. 9 Fuel mass fraction burned with the change of CO<sub>2</sub> content

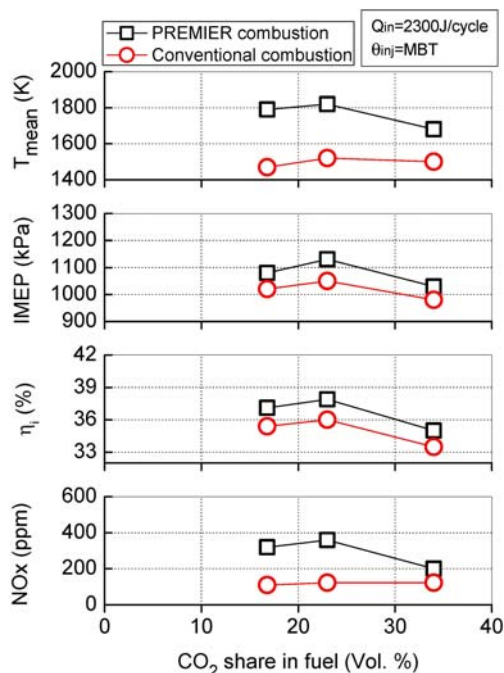


Fig. 10 Effect of CO<sub>2</sub> content on engine performance characteristics at  $\theta_{inj}$ =MBT

Therefore, in order to achieve high combustion efficiencies in dual-fuel engines fuelled with low-energy density syngas, CO<sub>2</sub> fraction in syngas needs to be controlled.

## V. CONCLUSION

The new PREMIER combustion mode in a dual-fuel engine fuelled with natural gas, syngas and hydrogen was investigated via engine experiments. The following conclusions can be drawn from this research:

PREMIER combustion combines two main stages of heat release, the first is gaseous fuel flame propagation and the second is end-gas mixture auto-ignition. The second stage can be mainly controlled by the pilot fuel injection timing, gaseous fuel equivalence ratio, and EGR rate. The delay time for mixture autoignition in the end-gas region is defined as the time from early kernel development to the transition point where slower combustion rate (flame propagation) is changed to faster combustion rate (autoignition). It was found that the rate of maximum pressure rise increases as the second stage ignition delay decreases. PREMIER combustion was observed for all gas types investigated in this paper. This mode of combustion can enhance the engine performance and increase the efficiency.

An increase in the fuel mass fraction burned in the second stage of heat release affects the rate of maximum pressure rise. When hydrogen content in syngas is increased the same rate of maximum pressure rise can be achieved with lower amount of fuel mass fraction burned during the second stage, meaning that the increased amount of hydrogen in syngas induces an increase in the mean combustion temperature, IMEP and efficiency, but also a significant increase in NO<sub>x</sub> emissions.

The results also show that when CO<sub>2</sub> content in the gas reaches 34%, the rate of maximum pressure rise, as well as, the mean combustion temperature, IMEP, thermal efficiency and NO<sub>x</sub> decrease despite the increase in fuel mass fraction burned during the second stage of heat release rate.

## ACKNOWLEDGMENT

The authors are grateful to the Japan Society for the Promotion of Science (JSPS) for providing partial funding for this work.

## REFERENCES

- [1] Weaver CS. Natural gas vehicles – a review of the state of the art, SAE Paper, 892133.
- [2] Nichols RJ. The challenges of change in the auto industry: Why alternative fuels? J Eng Gas Turb Power 1994; 116:727-32.
- [3] Lieuwen T, Yang V, Yetter R. Synthesis gas combustion: Fundamentals and applications. Taylor & Francis Group 2010.
- [4] Pushp M, Mande S. development of 100% producer gas engine and field testing with pid governor mechanism for variable load operation. SAE Paper 2008-28-0035.
- [5] Yamasaki Y, Tomatsu G, Nagata Y, kaneko S. Development of a small size gas engine system with biomass gas (combustion characteristics of the wood chip pyrolysis gas), SAE Paper 2007-01-3612.
- [6] Ando Y, Yoshikawa K, Beck M, Endo H. Research and development of a low-BTU gas-driven engine for waste gasification and power generation. Energy 2005;30:2206-18.
- [7] Shilling NZ, Lee DT. IGCC-Clean power generation alternative for solid fuels: GE Power Systems. PowerGenAsia 2003.
- [8] Bade Shrestha SO, Karim GA. Hydrogen as an additive to methane for spark ignition engine applications. Int J Hydrogen Energy 1999;24:577-86.
- [9] Stebar RF, Parks FB. Emission control with lean operation using hydrogen-supplimented fuel. SAE Paper 740187.
- [10] Jingding L, Linsong G, Tianshen D. Formation and restraint of toxic emissions in hydrogen-gasoline mixture fueled engines. Int J Hydrogen Energy 1998;23:971-75.
- [11] McTaggart-Cowan GP, Jones HL, Rogak SN, Bushe WK, Hill PG, Munshi SR. The effect of high-pressure injection on a compression-ignition, direct injection of natural gas engine. J Eng Gas Turb Power 2007;129: 579-88.
- [12] Su W, Lin Z. A study on the determination of the amount of pilot injection and rich and lean boundaries of the pre-mixed CNG/Air mixture for a CNG/Diesel dual-fuel engine. SAE paper, 2003-01-0765.
- [13] Tomita E, Fukatani N, Kawahara N, Maruyama K, Komoda T. Combustion characteristics and performance of supercharged pyrolysis gas engine with micro-pilot ignition. CIMAC congress 2007. Paper No. 178.
- [14] Karim GA. An examination of some measures for improving the performance of gas fuelled diesel engines at light load. SAE Paper, 912366.
- [15] Karim GA, Ito K, Abraham M, Jensen L. An examination of the role of formaldehyde in the ignition processes of duel fuel engine. SAE Paper, 912367.
- [16] Liu Z, Karim GA. The ignition delay period in dual-fuel engines. SAE Paper, 950466.
- [17] Nakagawa Y., Takagi Y., Itoh T., and Iijima T. Laser shadowgraphic analysis of knocking in SI engine. SAE Paper, 845001.
- [18] Pan J., Sheppard C. G. W., Tindall A., Berzins M., Pennington S. V., and Ware J. M. End-gas inhomogeneity, autoignition and knock. SAE Paper, 982616.
- [19] Kawahara N., Tomita E., and Sakata Y. Auto-ignited kernels during knocking combustion in a spark-ignition engine. Proc. Combust. Inst., 2007, 31, 2999-3006.
- [20] Bauerle B., Hoffman F., Behrendt F., and Warnatz J. Detection of hot spots in the end gas of an internal combustion engine using two-dimensional LIF of formaldehyde. In: 25th Symposium (Int.) on Combustion 1994, 25, 135-141.

- [21] Stiebels B., Schreiber M., and Sadat Sakak A. Development of a new measurement technique for the investigation of end-gas autoignition and engine knock. SAE Paper, 960827.
- [22] Pan J., and Sheppard C. G. W. A theoretical and experimental study of the modes of end gas autoignition leading to knock in SI engines. SAE Paper, 942060.
- [23] Li J, Zhao H, Ladommatos N. Research and development of controlled auto-ignition (CAI) combustion in a 4-stroke multi cylinder gasoline engine. SAE Paper, 2001-01-3608.
- [24] Santoso H, Matthews J, Cheng WK. Managing SI/HCCI dual-mode engine operation. SAE Paper, 2005-01-0162.
- [25] Persson H, Hultqvist A, Johansson B, Remon A. Investigation of the early flame development in spark assisted HCCI combustion using high-speed chemiluminescence imaging. SAE Paper, 2007-01-0212.
- [26] Ma F, Wang J, Wang Yu, Wang Y, Li Y, Liu H, Ding S. Influence of different volume percent hydrogen/natural gas mixtures on idle performance of a CNG engine. *Energy & Fuels* 2008;22:1880-87.
- [27] Higgin RMR, Williams A. A shock-tube investigation of the ignition of lean methane and n-butane mixtures with oxygen. In: *Symposium (Int.) on Combustion* 1969;12:579-90.
- [28] Zellner R, Niemitz KJ, Warnatz J, Gardiner Jr WC, Eubank CS, Simmie JM. Hydrocarbon induced acceleration of methane-air ignition. *Prog Astronaut Aeronaut* 1981;88:252-72.
- [29] Eubank CS, Rabinovitz MJ, Gardiner Jr WC, Zellner RE. Shock-initiated ignition of natural gas-air mixtures. In: *18th Symposium (Int.) on Combustion* 1981;18:1767-74.
- [30] Crossley RW, Dorko EA, Scheller K, Burcat A. The effect of higher alkanes on the ignition of methane-oxygen-argon mixtures in shock waves. *Comb. Flame* 1972;19:373-78.
- [31] Westbrook CK, Pitz WJ, Leppard WR. The autoignition chemistry of paraffinic fuels and pro-knock and anti-knock additives: a detailed chemical kinetic study. SAE Paper, 912314.
- [32] Jun D, Ishii K, Iida N. Autoignition and combustion of natural gas in a 4-stroke HCCI engine. *JSME International J* 2003;46:60-67.
- [33] Saito H, Sakurai T, Sakonji T, Hirashima T, Kanno K. Study on lean-burn gas engine using pilot oil as the ignition source. SAE Paper, 2001-01-0143.