Thermal Efficiency Improvement of a Lean-Boosted Spark Ignition Engine by Multidimensional Simulation with Detailed Chemical Kinetics

Sok Ratnak(1) Jin Kusaka(2) Yasuhiro Daisho(3) Kei Yoshimura(4) Kenjiro Nakama(5)
1) 3-4-1 Okubo, Shinjuku, Tokyo, 169-8555, Japan
(Email: ratnak@ruri.waseda.jp)
2) 2-1, Sakuranamiki, Tsuzuki-ku, Yokohama-shi, Kanagawa
4)-5) Suzuki Motor Corporation

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ABSTRACT: This paper aims to improve thermal efficiency of spark ignition engine by numerical calculation with detailed chemistry. Experimental results from a four-stroke-single-cylinder engine are compared with that of simulations. It is experimentally found that peak efficiency is achieved at lean-limit combustion under excess air ratio $\lambda=1.6$. Due to engine output power loss, further investigations are conducted under lean-boost operations. The best condition of the lean-boost mode is at $\lambda=1.3$ and 150 kPa boosted pressure (abs). To further improve the efficiency without power loss, simulations are conducted under lean-boost combustion with dilution rate, high engine swirl, and high knock resistant fuel.

KEY WORDS: heat engine, spark ignition engine, numerical calculation, detailed chemical kinetics [A1]

1. Introduction

Due to shortage of fossil fuel supply and stringent emission regulations, it is necessary to search for methods to address these problems from internal combustion engine by lowering harmful exhaust gas emission and improving fuel economy. In spark ignition engine, lean-burnt combustion mode is one of techniques to improve fuel efficiency with lower harmful exhaust gas emissions. In this method, an excess air is introduced and mixed with port-injected gasoline to form a lean-burnt mixture inside combustion chamber. By leaning the fuel-air mixture, the fuel consumption improvement is achieved. Furthermore, the combustion temperature of the lean mixture is then reduced and simultaneously it lowers nitrogen oxide emissions compared to stoichiometric combustion. Under the lower temperature combustion mode, heat loss from combustion charge to cylinder wall is also reduced. At the same time, its high thermal efficiency can be obtained. The authors experimentally found that a single-four-stroke gasoline spark ignition engine can be operated under lean-burnt combustion mode at high excess air ratio of $\lambda=1.6$ to reach the peak thermal efficiency. However, there are certain consequences when engine is operated under such a lean mixture. One of which is a penalty of power output due to a reduction of injected fuel amount. In on-road engine application especially when engine load and speed are highly transient, it is not always possible to operate the engine at high excess air ratio. In order to keep high efficiency, lean-burnt combustion mode is still taken into account, while necessary high intake manifold pressure is needed in order to stabilize the on-road load changes. Supercharger is one of the measures to increase intake pressure and boost output power while maintaining lean combustion operation. It is believed that the peak thermal efficiency with stable load range can be improved by increasing intake pressure to its maximum limit with allowable advanced spark ignition timing. However, when operated under high load condition, spark ignition engine has a constraint to improve load range and the efficiency due to an undesirable phenomenon, knock, which engine manufacturers tend to avoid. This phenomenon is caused by auto-ignition of end-gas charge mixture before spark-ignited flame arrival. Under boosted condition, the temperature and pressure of fuel-air mixture is high enough to self-ignite the mixture and it is likely to damage engine components such as piston head and piston rings if the engine runs at the condition within certain number of cycles. In order to further advance spark ignition timing for peak mean effective pressure, certain measures accounted for knock suppression need to be considered by preventing end-gas mixture from auto-ignition. Firstly, the end-gas temperature requires to be cooled down to avoid auto-ignition so as to mitigate knock tendency. Mixing between fresh charge mixture and exhaust gas recirculation (EGR) has tendency to reduce end-gas mixture temperature as well as knock suppression. Another method is to increase turbulent combustion flame burning velocity as fast as possible to reach cylinder wall before occurrence of the self-ignition. This measure can be realized by inducing high engine swirl flow into combustion chamber. In term of fuel composition effect on knock resistance in spark ignition engine, octane number represents gasoline fuel quality to the anti-knock resistance. Commercial gasoline fuel has research octane number 90 (RON90), while typical gasoline of RON95 available in certain countries especially in Europe, has been used in previous experimental researches (1, 2). Then the RON95-fuel is also one of the candidates to mitigate combustion knock.

Numerical simulations are being used extensively by engine manufacturers and academia to reduce cost of experiment and improve time consuming of engine design. In this paper, methods to improve thermal efficiency of spark ignition engine are numerically investigated and compared with experimental results from a gasoline four-stroke spark ignition engine. Simulations are conducted by multi-dimensional computational fluid dynamics with detailed chemical kinetics of primary reference fuel (PRF) of octane number 90 (PRF90) which is a blend of iso-octane and normal heptane. A level-set method or G-equation is used for turbulent combustion flame propagation model. Experiments are conducted up to lean-limit operation of excess air ratio $\lambda=1.6$ without boosting. Due to penalty of output power under this lean-no-boost condition, further experiments are conducted with...
high intake pressure by maintaining the highest output power as possible. Results from numerical simulation are then compared with experiments under lean-no-boost and lean-boost conditions as baseline. Further investigations to improve the efficiency are numerically studied using simulated cooled EGR, high engine swirl ratio, and high knock resistant fuel of PRF95.

2. Methodology

2.1. Experimental setup and methodology

Various experimental data were taken from a gasoline four-stroke single cylinder engine fueled by port-injected primary reference fuel of octane number 90 (PRF90) under no-boost conditions. Engine was operated at 4000 RPM and was previously studied elsewhere (1). Effect of ambient air humidity and temperature are considered due to relative effects on maximum efficiency. Intake and exhaust pressures were obtained by measurement while intake and exhaust pressures were measured by three water-cooled pressure transducers mounted on intake port, exhaust port, and cylinder head, respectively. In-cylinder pressure was recorded every 0.5 degree crank angle and is averaged from 300-cycle measurement while intake and exhaust pressures were obtained by one crank angle degree. Detail of engine specification is shown in table 1.

Table 1. Engine specifications

| Engine type | Four-stroke single cylinder |
|-------------|-----------------------------|
| Number of valve | 4 |
| Bore mm | 73 |
| Stroke mm | 82 |
| Compression ratio | 13.1 |
| Fuel preparation | Port-fuel injection |
| Experimental fuel | PRF90 |

Firstly, experiments were conducted from stoichiometric to lean-burnt limit, \( \lambda = 1.0 - 1.6 \), by adjusting intake throttle valve angle. Details are described in table 2. The peak indicated thermal efficiency saturates at \( \lambda = 1.6 \) without boosting. However, indicated mean effective pressure (IMEP) drops from 891 kPa (abs) at \( \lambda = 1.0 \) to 716 kPa (abs) at \( \lambda = 1.6 \). Increasing further excess air ratio causes misfire due to insufficient amount of fuel to initiate combustion.

Therefore, in later experiments (table 3), lean-boost combustions were investigated at 4000 rpm and full load because it is common that the engine thermal efficiency at high load is higher than that of the low load which fractions of cooling loss, mechanical loss, and pumping loss under high load are lower than under low load. To achieve high IMEP under lean conditions, intake manifold pressure is boosted by the variation of electrical frequency of the electrical supercharger under wide-open throttle (WOT) conditions therefore a reduction of fueling rate is obtained to achieve desired excess air ratio. Increasing intake pressure also boosts up intake temperature and it limits the performance of the engine due to retarded spark ignition timing to avoid excessive maximum pressure rise rate (MPRR) of the combustion. It is then necessary to use intercooler to control intake temperature. Under all lean-boost operations, intake and exhaust manifold pressures were controlled equally and spark ignition timings were advanced until knock limit points. Table 3 shows the lean-boost experimental conditions of \( \lambda = 1.0, 1.2, \) and \( 1.3 \) (case1-3) without dilution rate. IMEPs increase from 891-978 kPa (abs) combined with absolute boosted pressure from 95-117 kPa. IMEP drops to 962 kPa due to lean mixture of \( \lambda = 1.4 \) which is thought to be caused by a further reduction of injected fuel mass. Therefore, mixture of \( \lambda = 1.3 \) is selected as a baseline (case 3) to find maximum intake boosted pressure and \( P = 150 \) kPa (abs) is the optimum boosted intake pressure (case 5) with an acceptable maximum pressure rise rate.

Table 2. Lean-burnt experimental conditions (non-boost)

| Case | 1 | 2 | 3 | 4 | 5 | 6 |
|------|---|---|---|---|---|---|
| Engine speed rpm | 4000 | | | | | |
| Fuelling mg/cyc | 18.8 | 15.7 | 14.7 | 13.9 | 13.2 | 12.9 |
| Excess air ratio \( \lambda \) | 1.0 | 1.2 | 1.3 | 1.4 | 1.5 | 1.6 |
| Int. Pres. kPa (abs) | 95 | | | | | |
| IMEP kPa | 891 | 846 | 805 | 758 | 730 | 716 |
| Igt. timing bTDC (knock limit) | 10 | 17.5 | 21 | 24 | 31 | 33 |
| EGR rate % | 0 | | | | | |
| Int. air temp. deg.C | 25 | | | | | |
| Int. air humidity % | 50 | | | | | |
| Coolant temp. deg.C | 85 | | | | | |
| Oil temp. deg.C | 85 | | | | | |

Table 3. Lean-boost experimental conditions

| Case | 1 | 2 | 3 | 4 | 5 |
|------|---|---|---|---|---|
| Engine speed rpm | 4000/WOT | | | | | |
| Fuelling rate mg/cyc | 18.87 | 18.6 | 18.19 | 18.17 | 23.94 |
| Excess air ratio \( \lambda \) | 1.0 | 1.2 | 1.3 | 1.4 | 1.3 |
| Int. Pres kPa (abs) | 95 | 107 | 117 | 123 | 150 |
| IMEP kPa (abs) | 891 | 942 | 978 | 962 | 1112 |
| Ignition timing bTDC (knock limit) | 10 | 12 | 14 | 16 | 5 |
| EGR % | 0.0 | | | | | |

Therefore, in later experiments (table 3), lean-boost combustions were investigated at 4000 rpm and full load because it is common that the engine thermal efficiency at high load is higher than that of the low load which fractions of cooling loss, mechanical loss, and pumping loss under high load are lower than under low load. To achieve high IMEP under lean conditions, intake manifold pressure is boosted by the variation of electrical frequency of the electrical supercharger under wide-open throttle (WOT) conditions therefore a reduction of fueling rate is obtained to achieve desired excess air ratio. Increasing intake pressure also boosts up intake temperature and it limits the performance of the engine due to retarded spark ignition timing to avoid excessive maximum pressure rise rate (MPRR) of the combustion. It is then necessary to use intercooler to control intake temperature. Under all lean-boost operations, intake and exhaust manifold pressures were controlled equally and spark ignition timings were advanced until knock limit points. Table 3 shows the lean-boost experimental conditions of \( \lambda = 1.0, 1.2, \) and \( 1.3 \) (case1-3) without dilution rate. IMEPs increase from 891-978 kPa (abs) combined with absolute boosted pressure from 95-117 kPa. IMEP drops to 962 kPa due to lean mixture of \( \lambda = 1.4 \) which is thought to be caused by a further reduction of injected fuel mass. Therefore, mixture of \( \lambda = 1.3 \) is selected as a baseline (case 3) to find maximum intake boosted pressure and P=150 kPa (abs) is the optimum boosted intake pressure (case 5) with an acceptable maximum pressure rise rate.
2.2. Calculation method

Multidimensional simulations combined with detailed kinetics are conducted by using a commercial RANS-based computational fluid dynamics (CFD) code, FORTE by Reaction Design (46). A reaction mechanism of primary reference fuel, a two-component-blend of iso-octane (iC\textsubscript{8}H\textsubscript{18}) and n-heptane (n-C\textsubscript{7}H\textsubscript{16}), is used in detailed chemical kinetics calculations which contains 178 chemical species and 1271 reactions. The mechanism is capable to capture low and high temperature oxidations (63). Figure 1 shows STL (STereoLitography) geometry when piston is at intake valve closure (IVC) position and its numerical domain at top-dead center (TDC) for the CFD calculations. The calculation starts from IVC to exhaust valve closure (EVO) considering the transition criterion from ignition kernel to combustion model, affects the flame development process, so that the effect of flame development coefficient \(C_m\) is a crucial tuning constant. The transition criterion from ignition kernel to combustion model, from equation (5), is activated when the flame kernel radius grows larger than the turbulent integral length scale multiplied by a model constant \(C_{nl}\) and it is set at 2.0 for all simulation conditions.

\[
r_{k2} \geq C_{nl} \frac{l}{k} \geq C_{nl} \frac{1.6}{k} \geq C_{nl} \frac{0.16}{k} \geq 15\% (5)
\]

\[t_{\tau} \geq C_{nl} \frac{l}{k} \geq C_{nl} \frac{1.6}{k} \geq C_{nl} \frac{0.16}{k} \geq 15\% (5)
\]

\[
\frac{dr_v}{dt} = \frac{\rho_0}{\rho_k} (S_i + S_{\text{plasma}})
\]

\[
\rho_0, \rho_k \text{ are unburnt gas density and density of kernel particle, respectively. } \quad S_{\text{plasma}} \text{ is the kernel growth rate by plasma obtained from energy balance as following.}
\]

\[
S_{\text{plasma}} = \frac{\eta_{\text{eff}} \dot{Q}}{4\pi l^2 (\rho_0 u_k - \rho_k u_h) + p \frac{\rho_0}{\rho_k}}
\]

\[
\eta_{\text{eff}} \text{ is spark discharge energy chosen at 30\% in these calculation.} (65) \text{ Energy heat release } \dot{Q} \text{ equals 20 J/s. Typical spark gap is about 1 mm so the initial spark kernel radius is chosen at 0.3 mm. } u_k, h_k \text{ are specific internal energy of burnt gas inside kernel and enthalpy of unburnt mixture, respectively. } S_i \text{ is turbulent flame speed and is computed from correlation (7).}
\]

\[
\frac{S_i}{S_j} = 1 + I_p \left[ -a_i b_i^2 l + \left( a_i b_i^2 l \right)^2 + a_i b_i^2 l - S_j \right]^{0.5}
\]

\[
I_p = \left[ 1 - \exp \left( -\frac{t}{C_{\tau}} \right) \right]^{0.5}
\]

\[
\text{where } \alpha_i, b_i, b_i \text{ are tunable constants in the laminar/turbulent flame correlation. } l, u', \text{ and } l_F \text{ are turbulent integral length scale, turbulent intensity and laminar flame thickness, respectively. Laminar flame speed } S_l \text{ is calculated from power-law method (55).}
\]

\[\text{The effect of surround eddies on the flame kernel growth is taken into account due to the high transient flow from piston movement. It affects the flame kernel from laminar to fully developed turbulent flame speed and it is interpreted as progress variable } I_p, \text{ which is given as}
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\[
I_p = \left[ 1 - \exp \left( -\frac{t}{C_{\tau}} \right) \right]^{0.5}
\]

\[
\text{Where } \tau = k / \varepsilon \text{ is timescale calculated from turbulent kinetic energy } k \text{ and its dissipation rate } \varepsilon. \text{ Boosted intake pressure affects the flame development process, so that the effect of flame development coefficient } C_m \text{ is a crucial tuning constant. The transition criterion from ignition kernel to combustion model, from equation (5), is activated when the flame kernel radius grows larger than the turbulent integral length scale multiplied by a model constant } C_{nl} \text{ and it is set at 2.0 for all simulation conditions.}
\]

\[
r_{k2} \geq C_{nl} \frac{l}{k} \geq C_{nl} \frac{1.6}{k} \geq C_{nl} \frac{0.16}{k} \geq 15\% (5)
\]

\[t_{\tau} \geq C_{nl} \frac{l}{k} \geq C_{nl} \frac{1.6}{k} \geq C_{nl} \frac{0.16}{k} \geq 15\% (5)
\]

3. Results and Discussions

3.1. Results of in-cylinder pressure and rate of heat release from lean-no-boost experiments and simulations

Figure 2 presents comparison of results from experiments and simulations based on thermodynamic analysis of in-cylinder pressure and rate of heat release as function of crank angle degree. Comparison conditions details are listed in table 2. The numerical results of different lean-burnt combustions, which stoichiometric mixture (\(\lambda = 1.0\)) to maximum operable lean mixture (\(\lambda = 1.6\)) under natural aspirated condition, match reasonably well with the experiments.

Calculated indicated thermal efficiency can be found from Figure 3. The graphs show indicated thermal efficiency, output power of the engine in term of indicated mean effective pressure IMEP, advanced spark ignition timing at knock limit, and combustion phasing which CA50 (crank angle at 50% fuel burn), CA90 (crank angle position at 90% fuel burn), are plotted as function of excess air ratio \(\lambda = 1.0-1.6\).
1) Indicated thermal efficiency and IMEP vs $\lambda$
2) Indicated thermal efficiency and ignition timing vs $\lambda$
3) Combustion phasing (CA50, CA90) vs $\lambda$

Fig. 3. Indicated thermal efficiency, ignition timing, IMEP, and combustion phase CA50 as a function of excess air ratio

Another benefit of lean combustion is on the reduction of NOx emission as the lean combustion reduces peak combustion temperature and therefore NOx emissions. Comparisons of engine-out nitrogen oxide emission (NOx) are presented in Figure 4. A good tendency of NOx reduction is obtained from stoichiometric to the leanest mixtures. Simulated NOx is slightly higher than experiments under $\lambda = 1.0-1.4$. This is due to slightly higher peak cylinder pressure (and therefore peak temperature) compared to experimental pressures in Figure 2. Under conditions of $\lambda = 1.5$ and 1.6, predicted NOx is little lower than experiments because of slightly lower peak simulated pressures (and therefore peak temperature) compared to the experiments. However, overall tendency in both experiments and simulations is in a good agreement. To illustrate the NOx emission reduction under leaner conditions, in-cylinder temperature distributions of three conditions are post-processed in accordance with chemical specie (Figure 5) that dominates the high temperate region of hydroxyl radical OH under $\lambda = 1.0, 1.3$, and 1.6.
Figure 5 presents the in-cylinder distributions of temperature and OH chemical species at fixed position at 14 deg.aTDC. The temperature distribution of $\lambda = 1.0$ is higher than that of $\lambda = 1.3$ and 1.6. The highest OH magnitude can be seen from the mixture of $\lambda = 1.0$. The peak OH radical region is behind the flame brush and that is the reason where highest temperature region and highest NOx magnitude can also be noticed under stoichiometric compared to leanest mixtures.

### 3.2. Results of in-cylinder pressure and rate of heat release from lean-boosted experiments and simulations

As noticed earlier, when the amount of excess air ratio is increased in order to lean the combustion mixture, there is penalty of output power IMEP. Then the further experiments are conducted by increasing intake manifold pressure by mean of supercharger. Results from simulations are compared with experiments which details can be seen in table 3 for five cases. The validations of experiment and simulation of five investigated cases are presented in Figure 6 by comparing in-cylinder pressure and rate of heat release generated at each crank angle position. Thermodynamic comparisons of in-cylinder pressure and rate of heat release in simulation and experiment are in good agreement.

![In-cylinder pressure and rate of heat release generated at each crank angle position. Validation of experiment and simulation from case 1 to 5 in table 3](image1)

Figure 7 summarizes the indicated thermal efficiency and engine load IMEP as a function of excess air ratio $\lambda$ starting from $\lambda=1.0$, 1.2, 1.3, and 1.4 shown in table 3 in which it is noticed that the peak IMEP (978 kPa) is at the mixture of $\lambda=1.3$ with intake boosted pressure 117 kPa (abs). Under the combustion of $\lambda=1.4$ in case 4, with boost pressure 123 kPa, highest indicated thermal efficiency can be reached at 46.7%, however its IMEP drops to 962 kPa due to high excess air ratio and lower injected fuel mass. Considering that the engine drivability for the on-road application is the main objective in these investigations, losing output power is unacceptable. Therefore, experimental condition of case 3 with excess air ratio $\lambda=1.3$ is chosen for further study to find optimum intake boosted pressure without knock.

![Comparisons of simulated IMEPs and experimental IMEPs for four cases in table 3](image2)

Figure 8 presents the engine-out NOx emission generation history of case 1 to 5 for each crank angle degree.

![Engine-out NOx emission generation history of case 1 to 5](image3)
Comparison of nitrogen oxide NOx engine-out emission generation history as a function of crank angle position of five cases in table 3 are graphed in Figure 8 and NOx emissions from experiments and simulations are plotted in summary in Figure 9. The predicted pressure curves match with experimental pressure curves reasonably well and therefore the NOx emissions. Mole fraction of fuel in the mixture is reduced under lean combustion, thus, the reaction rate which leads to heat release is also reduced. As expected, the trend of engine-out NOx emissions decrease gradually as the excess-air ratio \( \lambda \) increases due to the lower peak temperature in leaner compared to the stoichiometric conditions.

3.3 Knock simulations under based conditions

It is found experimentally from the lean-boost operation without exhaust gas recirculation, combustion of the spark ignition engine can be operated at the maximum allowable intake boosted pressure of 150 kPa (abs) under the mixture of excess air ratio \( \lambda = 1.3 \) (case 5, table 3). This condition is a stable and repeatable combustion without knock. Increasing further intake pressure of 150 kPa (abs) increases engine load IMEP but the repeatable combustion without knock. Increasing further intake pressure of 150 kPa (abs) under the mixture of excess air ratio \( \lambda = 1.3 \) is chosen as a baseline for lean-boost simulations to find alternative means to improve thermal efficiency. To simulate spark ignition combustion under knock condition, six simulated pressure probes are modeled to detect combustion pressure which each output pressure is extracted every 0.1 degree crank angle. The maximum simulation timestep is one microsecond. Three ignition timing sweeps are investigated for these knock combustion simulations. Knock simulation conditions and pressure probe positions are shown in table 4 and Figure 10, respectively.

Table 4: Base condition for knock simulations

| Engine Speed rpm | 4000 |
|------------------|------|
| Excess air ratio \( \lambda \) | 1.3 |
| Boost pressure kPa (abs.) | 150 |
| EGR rate % | 0 |
| Ignition timing deg.ATDC | -25, -15, -10 |
| Simulated pressure probes | 6 |
| Wall temp. K | 550 |

Fig. 9. Engine-out NOx emission of five cases (table 3) (Blank bar: Simulations, Crossed bar: Experiments)

Fig. 10. Six simulated pressure transducer positions on cylinder head for knock pressure detections

Simulated in-cylinder pressures during knock combustion are shown in Figure 11. The graph plots parameter studies of the effect of three spark ignition timing on knock combustion tendency for lean-boost mixture \( \lambda = 1.3 \) and boosted pressure \( \Pi = 150 \) kPa (abs) without the effect of dilution rate. It is shown that the most intense knock magnitude is generated under the most advanced spark ignition timing of 25 deg.bTDC. As the ignition timings are retarded at 15 and 10 deg.bTDC, the knock pressure are reduced to medium and light intensities, respectively.

Raw and band-pass filtered pressures of three spark ignition timings at position 2 and at the exhaust valve side are plotted separately as shown in Figure 12. It is logical that the highest temperature region is at the exhaust valve side due to the direct contact between its surfaces with the post-combustion flow during the exhaust stroke. Temperature and dominant chemical species distributions of in-cylinder combustion are also illustrated to provide information of knock regions. 3D contours of temperature, OH, H2O, and HO2 radical are shown in Figure 13 for the severe knock condition (\( \lambda = 1.3 \), \( \Pi = 150 \) kPa, no EGR). For the high boosted spark ignition engine, it shows that the auto-ignition of premixed charge occurs earlier before the spark-ignited flame arrival. At high temperature oxidation region, the chemical species are dominant which can be described by chain branching reaction as following (36).

\[
\begin{align*}
H^+ + O_2 &\rightarrow O^+ + OH^+ \quad (R.1) \\
HO_2^+ + RH &\rightarrow H_2O_2 + R^+ \quad (R.2) \\
H_2O_2 + M &\rightarrow 2OH^+ + M \quad (R.3) \\
OH^+ + H_2 &\rightarrow H_2O + H^+ \quad (R.4) \\
H^+ + O_2 + M &\rightarrow HO_2^+ + M \quad (R.5)
\end{align*}
\]

(R^+ is radical of hydrocarbon)
Fig. 12. Simulated raw and band-pass filtered pressures at 25, 15, 10 deg.bTDC at position 2 of exhaust valve side

It is clearly seen from Figure 13 for the server knock simulation that the auto-ignition temperature zone, in reaction R.1 which dominates combustion at high temperature, occurs visibly from crank angle degree = 2 (CAD=2) where the OH radical also appears at the same crank angle (see Figure 13). The OH-radical then reproduces HO from reaction R.4 and R.5 which can be used to explain knock in this simulation.

3.4 Simulation results for further thermal efficiency improvement

The further parametric studies of combustion simulation are investigated to improve thermal efficiency without output lower loss of the lean-boost spark ignition engine. The parametric studies are based on reduction of peak combustion temperature so that the ignition timing can be advanced to avoid knock. This can be realized by using cooled exhaust gas dilution rate which compositions the diluted gas such as CO$_2$ and H$_2$O reduce reaction rate of the combustion. Another method to suppress knock is to increase turbulent flame burning velocity to reach walls before the auto-ignition of end-gas occurrence by inducing high in-cylinder flow. Higher burning velocity leads to shorter combustion duration and therefore provides less time for the auto-ignition of the end-gas mixture.

Final parametric study of the combustion simulation without knock is investigated by using high knock resistant fuel. As it is known that the higher octane number fuel prolongs ignition delay period of the end-gas mixture and reduces knock. A primary reference fuel with octane number 95 is numerically investigated. Selected experimental conditions from case 5 in table 3 are extended for further numerical investigations to improve thermal efficiency while maintaining high IMEP as possible. Parameter studies of simulation conditions are listed in table 4. To prevent knock in the calculations, the same approach to avoid knock mentioned earlier is applied which six simulated pressure probes are used in cases 4, 5, and 6. Next, the parameter studies of spark ignition timing sweep are simulated.
Simulated and experimental results of conditions in Table 5 are shown in Figure 14. Results of experimental and simulated pressures are compared for cases 1, 2, and 3 while simulated pressures are for cases 4, 5, and 6. Cylinder pressure in case 4 is in wide and high tendency due to high inlet boost pressure of 150 kPa (abs) combined with 10% of simulated external cooled dilution rate. With 10% of the dilution rate, ignition delay is prolonged and the peak gas temperature of the combustion is minimized, therefore, knock tendency is reduced. A simple introduction of EGR effect is that the compositions of diluted gases such as CO₂ and H₂O reduce mole fraction of the gasoline in the mixture and therefore reduce reaction rate of the end-gas. Thus, the reductions of the temperature, reaction rate, and ignition delay prolonging allow the combustion simulations to run at more advanced spark ignition timing. The same tendency of in-cylinder pressure curves in cases 5 and 6 continue to increase as the ignition timings are advanced.

**Table 5. Simulated conditions for further efficiency improvement**

| Case | Engine speed rpm | 1 | 2 | 3 | 4 | 5 | 6 |
|------|-----------------|---|---|---|---|---|---|
|      | Engine speed rpm | 4000/Wide-open throttle |  |  |  |  |  |
|      | Excess air ratio λ | 1.0 | 1.3 | 1.3 | 1.3 | 1.3 | 1.3 |
|      | Int. press. kPa (abs) | 95 | 117 | 150 | 150 | 150 | 150 |
|      | EGR % | 0 | 0 | 0 | 10 | 10 | 10 |
|      | Swirl ratio | 1.4 | 1.4 | 1.4 | 1.4 | 1.6 | 1.6 |
|      | Octane number | 90 | 90 | 90 | 90 | 90 | 95 |

Fig. 14. Case 1-2-3: Simulation and experimental pressure comparisons as based points, Case 4-5-6: Simulated pressures

This paper investigates the thermal efficiency improvement of SI engine. Experiment of stoichiometric conditions up to the lean-boost limit while maintaining high output power are conducted. It is found that the lean-boost combustion has potential to improve engine efficiency without losing its drivability. Peak indicated thermal efficiency (ITE) is achieved at 43.56 % at λ = 1.3 with absolute intake pressure 117 kPa. Increasing intake pressure to 150 kPa allows the engine to generate higher output power with penalty of efficiency due to retarded ignition timing to mitigate knock. However, when cooled EGR is used, higher ITE is obtained at 45.9%. It is found from simulations that, by using 3D CFD simulation with detailed chemistry, the highest ITE can be achieved at 48.2% by combining lean-boost with cooled dilution rate, high swirl ratio, and high knock resistant fuel of PRF95.

4. Conclusion

This paper presents the efficiency of cases 1-6 from Table 5 which the drop of thermal efficiency at lean-high boost mixture is noticed due to the retarded spark ignition timing to mitigate combustion knock. In overall conditions, the ignition timings can be further advanced and the combustion phase where 50% of fuel is consumed (CA50) is closed to top-dead center. These lead to increase the thermal efficiency of the engine. The highest indicated thermal efficiencies as a function of IMEP are illustrated in Figure 15.3 in which the spark ignition timings are at the knock limit. At based condition (case 1), the peak indicated thermal efficiency is obtained at 41.6%. A great potential of high efficiency, high IMEP combustion mode is achieved at 48.2% (case 6) which cooled dilution, high engine swirl ratio, and high knock resistant fuel (PRF95) are used in combination.

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