An assessment of the ultra-lean combustion direct-injection LPG (liquefied petroleum gas) engine for passenger-car applications under the FTP-75 mode

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Korea Institute of Machinery and Materials, 156 Gajungbukro Daejeon, 34103, South Korea

**Abstract**

Fuel economy and emissions characteristics of the lean-burn combustion strategy were assessed in a direct-injection liquefied petroleum gas engine under urban driving conditions. This study investigated: (1) engine experiments under steady-state conditions that frequently occur during city driving and (2) zero-dimensional drive cycle simulation under the FTP-75 mode. From the engine experiments, two sets of operation maps were constructed for the lean and conventional combustion modes. The lean combustion set consisted of the lean operation between 20 and 80 Nm and the stoichiometric operation for the rest of the operating envelope. The selected lean operation range was found to be the largest possible to operate under the lean condition. The entire conventional combustion set was taken at the stoichiometric condition for comparison with the lean combustion set. Then, the steady-state performance maps of the fuel consumption, emissions, and exhaust temperature of the two combustion sets were used in a vehicle model of the cycle simulation. The drive cycle simulation results showed that the fuel economy of the lean combustion set was 10.65 km/l, which is lower than that of the conventional combustion set by 6.3%. Carbon dioxide emission was 146.25 g/km, down by 8 g/km from the conventional combustion set. The time-resolved emissions of the particle mass and particle number of the lean combustion set were 0.005 g/km and $3.2 \times 10^{11}$ #/km, respectively. The particle emissions of the lean combustion set were higher than those of the conventional combustion set, but considered as acceptable levels based on the upcoming emission standards.

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**1. Introduction**

Liquefied petroleum gas (LPG) has been a reliable alternative fuel in the passenger car applications. LPG adaptation into gasoline engines has been available in the EU for many years, and its characteristics have been well established [1,2]. LPG is composed of butane and propane, and the ratio varies depending on the season. Its faster vaporization and subsequent enhanced mixing helps increase its thermal efficiency and decrease particulate emission. Most LPG spark-ignition (SI) engines operate at the stoichiometric condition for NOx reduction with the use of a three-way catalyst. However, LPG SI engines suffer from lower fuel economy due to the lower energy density per volume of the fuel than that of petroleum fuels.

Many approaches have been undertaken to improve the fuel economy of LPG engines. It is well known that lean operations in which the air-to-fuel ratio divided by the stoichiometric air-to-fuel ratio (refer to as “lambda”) is $>1$ can achieve a thermal efficiency improvement. Lean conditions can be achieved by less throttling when subject to part-load conditions. Subsequent pumping reduction during decreased throttling leads to fuel economy improvement. Because the pumping reduction is most effective under low load conditions, it is adequate to focus on applying the lean combustion strategy to low load conditions. The lean limit of port-fuel injection (PFI) engines or direct-injection (DI) engines with fuel injection during the intake stroke is known to be around a lambda of 1.5 [3,4]. Lean SI combustion often deteriorates combustion stability when the fuel mixture is either over lean or too distant from the ignition source [4,5]. It has been suggested that additional means are necessary to achieve an acceptable level of combustion stability when subject to ultra-lean operation [6–14].

One of the most promising approaches in improving combustion stability of the lean combustion is a direct-injection system implementation. The direct-injection allows applying the stratified fuel injection strategy. Instead of early fuel injection during the intake stroke, the stratification strategy utilizes late injection close to the top-dead-center (TDC) to create a rich fuel cloud near the spark plug. It is preferable that the fuel injector locates at the center of the combustion chamber and in a proximity to the spark plug in order to achieve an adequate stratification. Reynolds et al. [14] concluded that partial stratification achieved a brake specific fuel consumption (BSFC) reduction of 15% with a coefficient of variation in terms of the indicated mean effective pressure (COV_{IMEP}) of $<5.0$ in a natural gas SI engine. The center-
mounted direct-injection is superior to other configurations for its intuitive combustion chamber design with respect to fuel spray. Efforts have been made in LPG fuel spray modeling to enhance the understanding of LPG fuel injection characteristics [15].

However, the potential of the stratified ultra-lean combustion strategy in a center-mounted direct-injection LPG engine has not been investigated yet. Most studies in lean combustion conducted with either port-fuel LPG engines [1–3, 6–7] or direct-injection gasoline engines [1, 4–5, 11]. LPG fuel in combination with a high-pressure direct-injection system has a great potential in the ultra-lean combustion with its faster vaporization. In addition to the performance of the combustion strategy over the operating envelope, it is of great interest to assess the fuel economy and emission levels of the combustion strategy under the urban drive cycle, since the combustion strategy is advantageous under the part load conditions at large.

The objective of the present study is to determine the potential fuel economy gain of the lean combustion strategy in automotive application. In the present study, ultra-lean combustion was achieved by a high-pressure direct-injection system to apply the stratification strategy. Steady-state engine experiments have been performed at engine speeds from 1000 to 3000 rev/min in a 500-rev/min increments and torques from 20 Nm to the maximum in 20-Nm increments. The maximum torque varied with the engine speed. Engine performance at idling was estimated based on two 10-Nm cases at engine speeds of 800 and 900 rev/min. The constructed operating envelope included all of the operating conditions used in the FTP-75 drive cycle.

Direct fuel injection was selected to apply various injection strategies. The engine head was modified to incorporate the gasoline direct injection (GDI) system. Each GDI injector was placed at the center of the cylinder. A spark plug was located 3 mm away from the injector at an angle of 30° toward the injector. The direct injection system was capable of multiple injections at any crank angles at any fuel pressure up to 20 MPa. The center-mounted injector allowed for the investigation of various injection strategies, including homogenous and stratification injection strategies. The homogeneous injection strategy was applied by early injection near the start of the intake stroke. Although it produces less fuel rich spots and a lower NOx emission, this strategy is prone to deteriorate combustion stability under lean conditions. Therefore, the strategy is more suitable for stoichiometric operation. The stratification strategy that makes use of near top-dead center injection is considered to be a more adequate lean-burn injection strategy, particularly for ultra-lean operation. In addition to those two injection strategies, a couple of multiple injection strategies were assessed under various load conditions.

The zero-dimensional (0-D) vehicle simulation was carried out with engine operation maps from the steady-state engine experiments. The FTP-75 mode was selected to investigate the urban-driving performance of the lean-burn LPG engine in terms of fuel economy and emissions. A commercial software package, GT-SUITE, was used for the vehicle simulation. Kim et al. [16] showed that the fuel consumption estimated using the cycle simulation with the fuel consumption map from steady-state operations exhibited a 1.9% discrepancy from the time-resolved fuel consumption measured with a mass flow meter. An oxidation catalyst model was implemented to oxidize carbon monoxide (CO) and hydrocarbon (HC) in the exhaust, which was necessary for carbon dioxide (CO₂) emission estimation. The test vehicle model was a mid-sized sedan equipped with a 6-speed automatic transmission.

Table 1

<table>
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<tr>
<th>Parameters</th>
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<tr>
<td>Bore</td>
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<tr>
<td>Stroke</td>
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<tr>
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<td>mm</td>
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<tr>
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<td>cm³</td>
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<tr>
<td>Squish height</td>
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<tr>
<td>Swirl ratio</td>
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</tr>
<tr>
<td>Compression ratio</td>
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<td>–</td>
</tr>
<tr>
<td>Intake valve closing</td>
<td>− 147</td>
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<tr>
<td>Exhaust valve opening</td>
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Table 2

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<tr>
<td>C3</td>
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<td>15–35</td>
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<td>C4</td>
<td>%</td>
<td>&gt;85</td>
<td>&gt;60</td>
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<td>Butadiene</td>
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<td>Sulfur</td>
<td>ppm</td>
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<td>Vapor pressure</td>
<td>MPa (40°C)</td>
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<tr>
<td>Density</td>
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<td>Lower heating value</td>
<td>kcal/kg</td>
<td>2.517</td>
<td>2.566</td>
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<tr>
<td>HC ratio</td>
<td></td>
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</table>

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fuel in Table 2 were interpolated based on the density calculated from the measured mass and volumetric flow rates.

An external electric motor operated the high-pressure fuel pump at 1750 rev/min independent of the engine speed. The low-pressure supply pump maintained the fuel at 800 kPa to sustain the liquid phase. The fuel supply system incorporated the flow measurement system that accommodated the recirculation system for the fuel returned from the injectors. The fuel flow rate was measured using a mass flowmeter (Onosokki, Japan). The principle of the flowmeter was the Coriolis effect. The flowmeter provided both the mass and volumetric flow rates while the engine was running. The flowmeter reading was averaged over 60 s.

A prototype engine control unit (ECU) (Motohawk, Mototron, U.S.A.) was utilized for fuel injection, sparking, and electric throttle controls. A separate GDI injector driver controlled the injectors using the trigger signals from the ECU.

Gaseous emissions were measured by an emission bench (AMA-i60, AVL, Austria). The bench was equipped with 2 infrared detectors for CO and CO\(_2\) emissions, 1 flame ionization detector for HC emissions, 1 chemiluminescence detector for NOx emissions, and 1 paramagnetic detector for O\(_2\) measurements. The particle mass (PM) measurements were conducted using an opacimeter (439, AVL, Austria), whereas the particle number (PN) measurements were conducted using a condensation particle counter (TSI, U.S.A.) via a dilution tunnel (TSI, U.S.A.). The dilution ratio was set to 20:1.

### Table 3

The specifications of the vehicle model.

<table>
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<tr>
<th>Parameter</th>
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<tr>
<td>Gear ratio 2nd</td>
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<tr>
<td>Gear ratio 3rd</td>
<td>-</td>
<td>1.841</td>
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<tr>
<td>Gear ratio 4th</td>
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<td>1.386</td>
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<tr>
<td>Gear ratio 5th</td>
<td>-</td>
<td>1.000</td>
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<tr>
<td>Gear ratio 6th</td>
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<td>0.772</td>
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2.2. Drive cycle simulation

The 0-D vehicle simulation for the FTP-75 cycle mode was carried out in a commercial 0-D engine and vehicle simulation package (GT-SUITE, Gamma Technologies, U.S.A.). The vehicle simulation based on the steady-state engine operation maps is well established when assessing the fuel economy and emissions of the engine of interest [18]. The vehicle model was composed of an engine, oxidation catalyst, and vehicle. A group of maps of the flow rates of the fuel, air, HC, CO, CO\(_2\), PM, and PN from the steady-state engine operations was used as an input to the engine model. The selected analysis mode in the engine model, VehKinemAnalysis, read the engine states in the maps at the engine speed and load at every second given in the regulatory driving cycle [19]. Kim et al. [16] found that this steady-state map-based simulation exhibited 1.9% lower total fuel consumption than the real-time flow rate measurement. It was noted that the greater fuel consumption of the real-time measurement method might be attributed to the slower response of the diesel fuel flow measurement system due to the recirculating flow circuit for the fuel returned from the high-pressure pump, fuel rail, and injectors in the diesel common-rail fuel injection system [16].

![Fig. 2. Shifting schedule with respect to pedal position and vehicle speed.](image)

![Fig. 3. COV\(_{\text{IMEP}}\) of the lean combustion set with respect to brake torque and engine speed.](image)

![Fig. 4. Lambda values of the lean combustion set with respect to brake torque and engine speed.](image)
The oxidation catalyst was placed in the exhaust model for HC and CO oxidation. The purpose of the catalyst is to oxidize all carbon containing species so as to achieve an accurate total CO\textsubscript{2} emission estimation from the drive cycle simulation. The resulting HC and CO emissions were below the current emission standard levels. The details of the catalyst model can be found in ref. [19].

The vehicle model was designed to represent a popular medium-size sedan registered in South Korea in 2013. The vehicle has a 1998-cm\textsuperscript{3} LPG engine. The specifications of the vehicle are listed in Table 3. The vehicle was equipped with a 6-speed automatic transmission. The shifting schedule shown in Fig. 2 was employed in the present simulation. The shift strategy was rather simplified compared to the strategies used in regular vehicles, but was sufficient for comparing two combustion modes in terms of fuel economy.

The FTP-75 driving cycle, which is known as the representative city driving cycle for the federal emission regulations in the U.S.A., consists of 3 phases: cold start, transient, and hot start. The total duration of the cycle was 1874 s for a 17.77-km travel distance. The average and maximum speeds of the drive mode were 34.2 and 91.2 km/h, respectively [20].

3. Results and discussion

3.1. Engine experiments

The steady-state engine experiment was carried out at all operating points to determine the optimal cases. At each operating point, the injection timings and spark advance were carefully adjusted for the lowest BSFC. Then, the operating point repeated at a greater lambda until combustion was no longer stable at any injection or spark timing. It was critical to increase the lambda value, i.e., reduce the pumping loss, to improve the fuel economy. After the search was completed, the optimal case at each operating point was identified based on the BSFC, COV\textsubscript{IMEP}, and soot emission. The COV\textsubscript{IMEP} limit was set at 5.0 for all operating points except for a few light load conditions, as shown in Fig. 3.

Two sets of experiments were performed in parallel for the lean and conventional combustion modes. The conventional combustion set was taken under the stoichiometric condition throughout the entire operating envelope. On the other hand, the lean combustion set was taken under the lean condition at low-medium load conditions. The rest of the points in the lean-burn combustion set were taken under stoichiometric conditions, as shown in Fig. 4. Fig. 5 shows all of the operating points in both the lean and conventional combustion sets. The resulting lambda was as high as 4.0 in the 2000 rev/min and 34 Nm case. Subsequently, the lean combustion set exhibited substantial reduction in
pumping power (PP) in the lean operating range, as shown in Fig. 6. PP reached to 0 kW in the lean operation region. Note that no fuel enrichment, injecting fuel over the stoichiometric condition to reduce the exhaust temperature, was applied, even at the greatest load conditions. The maximum engine torque in the present study was 160 Nm in both combustion sets.

In the lean combustion set, a variety of injection and ignition strategies were applied, including MCI. Late single or double injection, approximately 40° ATDC, was found to be most effective in terms of extending the lean limit. The ignition delay contours in Figs. 7 and 8 show the consequent distinction of the injection and ignition strategies between the lean and conventional combustion sets. The ignition delay was defined as duration in CA between the crank angle of spark ignition and crank angle of 10% heat release. The ignition delay was shortened in the lean operation region where late injection led to stratified combustion.

The BSFC contours in Figs. 9 and 10 show that the lean operation reduced BSFC to 250 g/kWh between 40 and 80 Nm. A large portion of the decreased BSFC was attributed to pumping reduction. The pumping power reduction reached 90% between 40 and 80 Nm in the lean operation region. The gross IMEP was reduced by 2% in most regions in the lean combustion set, which indicated combustion deterioration.

Gaseous and PM emissions over the entire operating envelope in the lean combustion set are presented in Figs. 11 to 14. The lean operation reduced CO emission under light load and low speed conditions. However, HC and NOx were increased at all lean operation points. This discrepant result might be attributed to the late-injection, stratified-combustion strategy, of which combustion tended to occur in proximity to the spark plug, which is located in the center of the rich fuel and air mixture. Further investigation is required to interpret the discrepancies in the emissions results. Fig. 14 reveals the gradual PM elevation as the increased engine speed in the lean operation region. Figs. 15 to 16 show the PN emissions of the two combustion sets. The PN emission showed the similar trend as the PM emission shown in Fig. 14. The PN emission increased at the high-speed lean operation region in the lean combustion set. Although it was elevated at higher speed condition in the conventional combustion set as well, the highest PN emission of the conventional combustion set, $0.04 \times 10^{14}$ #/hr., was substantially lower than that of the lean combustion set.

3.2. Drive cycle simulation

Fig. 17 shows the cycle simulation results. The vehicle speed in the cycle simulation was identical to the target speed defined in the FTP-75 regulation. The two fuel consumption rate traces show that the majority of the discrepancy between the two combustion sets was under low load conditions. The time-resolved fuel economy and emissions of the FTP-75 cycle simulation are listed in Table 4. The fuel economy is reported in units of km per liter, which is a widely used and standard fuel economy unit. The post-catalyst CO and HC emissions were at
acceptable levels. As previously mentioned, the oxidation catalyst was placed in the model for CO\(_2\) emission estimation. The CO\(_2\) reduction and fuel economy increase were 5.5% and 6.3%, respectively. The difference between the two parameters might indicate the discrepancies in the measurements among the gas emission analyzers and fuel flowmeter. In addition, the air flowrates, which were estimated from the measured air-fuel ratios and fuel flowrates, could contribute to the disagreement.

The fuel economy of the conventional combustion set agreed well with the registered fuel economy value of the same vehicle, which was 10.0 km/l under the same cycle mode [21]. Myung et al. [22] found that the vehicle equipped with the similar 2000-cm\(^3\) LPG DI engine exhibited a fuel economy of 9.2 km/l in the actual vehicle testing under the FTP-75 mode, which was 25% lower than that of a GDI-engine vehicle.

The PN emission of the conventional combustion set was 2.8 × 10\(^{10}\) #/km, which was one order of magnitude lower than the EURO 5–6 emission standard for spark-ignition engines [20]. Myung et al. [22] found that the PN emission was 1.1 × 10\(^{10}\) #/km from a 2000-cm\(^3\) LPG DI engine equipped vehicle in the chassis dynamometer vehicle test. It has been reported that LPG DI engines exhibit substantial particulate emission reduction in comparison with gasoline DI engines [23].

Fig. 18 shows the BSFC reduction of the lean combustion set and operating points visited during the FTP drive cycle. It was evident that the engine was frequently operated at low load and medium speed conditions during the drive cycle. The greatest BSFC reduction of 18% was found under the most frequently visited operating conditions. The BSFC reduction was marginal at 0–9% in the brake torque range of 0 to 20 Nm.

The lean combustion set exhibited increased PM emission by an order of magnitude. Similarly, increased PN emission in the lean combustion engine truck was found in [24]. Although increased in the lean combustion set, the PM emission of 0.005 g/km in the FTP-75 mode was at an acceptable level considering the current emission regulations. The lean combustion set also exhibited an elevated NO\(_x\) emission by 6% compared with the conventional combustion set. The increment may not be substantial, but the lean combustion strategy would require an additional aftertreatment system due to the lean operation. The use of a conventional catalyst for engines operating at the stoichiometric point, which is known as a three-way catalyst (TWC), is likely to be insufficient for NO\(_x\) removal under lean conditions. Wang et al. [25] showed that a combination of selective catalytic reduction (SCR) could remove NO\(_x\), CO, and HC simultaneously from the lean combustion engine exhaust gas.

4. Conclusions

Investigations were performed to estimate the fuel economy and emission levels of the lean LPG combustion strategy and to compare its performance with the conventional stoichiometric combustion...
strategy under the urban drive cycle. First, engine experiments were conducted in a 4-cylinder direct-injection LPG engine coupled with an alternating-current (AC) dynamometer. The experiments were performed at a steady state condition to construct operation maps of fuel consumption and emission rates. The lean operation was applied as wide as possible. Then, drive cycle simulation was carried out using those engine maps in FTP-75 mode using the GT-SUITE simulation model. Several conclusions are drawn as listed below.

The lean combustion strategy was applicable in the brake torque range of 20 to 80 Nm for the tested engine speed range. The lean operation was extended to a lambda of 4.0 at a torque and speed of 40 Nm and 2000 rev/min where the fuel economy improvement was also maximized. At 20 Nm or under lower load conditions, the combustion deterioration compromised the fuel economy improvement that was achieved by pumping loss reduction. At 90 Nm or under greater load conditions, the lean operation was limited by the air supply approaching the stoichiometric condition.

The drive cycle simulation estimated the fuel economy improvement of the lean combustion strategy in FTP-75 mode to be 5.5–6.3%, depending on the measure of the fuel economy. The CO₂ and fuel consumption reductions were 5.5% and 6.3%, respectively. The simulation result showed that the frequent engine operation in the lean operation region resulted in the fuel economy improvement.

![Graphs and figures](image-url)

**Table 4**

<table>
<thead>
<tr>
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<th>Lean Set</th>
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<tr>
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<tr>
<td>CO₂</td>
<td>g/km</td>
<td>154.27</td>
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<tr>
<td>NOₓ</td>
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<tr>
<td>PN</td>
<td>#/km</td>
<td>2.8 × 10¹⁰</td>
<td>3.2 × 10¹¹</td>
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**Fig. 18.** BSFC reduction in the lean combustion set and operating points in the FTP75 drive cycle.
The lean combustion set exhibited particle emissions of 0.005 g/km and 3.2 × 10^11 kg/km, which were greater than those of the conventional combustion set. These values are considered to be acceptable levels as far as the current emission standards are concerned. Substantial increases in both PM and PN emissions were found at an engine speed of 2500 rev/min or higher. Therefore, the particulate emission increase can be alleviated by reducing the lean operation region at higher speed conditions. As shown in Fig. 18, switching to the stoichiometric operation at 2500 or higher engine speed would not deteriorate the fuel economy improvement in FTP-75 mode. Further investigation is required to determine whether a particulate filter is necessary for this combustion strategy. It was suggested that lean operation in combination with advanced spark timings could minimize PM emission during the cold start [26].

Additional fuel economy improvements could be achieved by two means: (1) intake boosting that widens the lean operation band toward greater load conditions. To maximize the potential of the intake boosting, the compression ratio and other critical in-cylinder geometries must be optimized to enhance the lean combustion regime [27]; (2) extending the lean operation under lower load conditions. The FTP-75 drive cycle makes frequent visits to a brake torque of 60 Nm or lower at an engine speed between 1000 and 2000 rev/min. The lean limit extension under this region likely requires the delivery of greater ignition energy in association with an increased compression ratio.

Nomenclature

- LPG: liquefied petroleum gas
- SI: spark ignition
- PFI: port-fuel injection
- DI: direct injection
- EGR: exhaust gas recirculation
- NOx: nitric oxides
- BSFC: brake specific fuel consumption
- COV_{IMEP}: coefficient of variation in indicated mean effective pressure
- GDI: gasoline direct injection
- 0-D: zero dimension
- CO: carbon monoxide
- HC: hydrocarbon
- CO₂: carbon dioxide
- MCI: multiple charge injection
- ECU: engine control unit
- PM: particle mass
- PN: particle number
- PP: pumping power
- TWC: three-way catalyst
- AC: alternating current

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