1 Introduction

Combustion of very lean premixed hydrocarbon–air mixtures exhibits low flame propagation speeds leading to a loss in power output, and an increase in fuel consumption and hydrocarbon emissions [1]. Due to these restrictions and also increasing the catalyst efficiency, spark-ignition engines always operate close to the stoichiometric mixture. Traditionally, to improve the lean-burn capability of the hydrocarbon–air premixed charge, an increase of turbulence intensity in the cylinder is needed. However, these measures are always accompanied by increases in energy loss to the cylinder wall and fuel consumption. One effective method to solve the problem is to enrich the region near the spark for initiating the flame. This can be accomplished by high pressure fuel direct injection to create a stratified mixture just before the spark.

In order to achieve better fuel economy and meet the requirements of stringent emission regulations, the development of four-stroke, spark-ignition engines to inject CNG directly into the combustion chamber is an important development in the automotive and motorcycle industries [2–6]. Compressed natural gas is regarded as one of the most promising alternative fuels and it is composed primarily of methane (CH₄) [7]. The benefits of CNG as an alternative fuel are that it emits lower amounts of air pollutants and it is very economical compared to conventional fuels [8]. Compressed natural gas has a high research octane number (RON = 110–130) and therefore can be easily employed in spark-ignited (SI) internal combustion engines. Due to the high RON of CNG, engines could be operated with a higher compression ratio for better thermal efficiency [9]. Furthermore, since CNG has a low carbon/hydrogen (C/H) ratio, it produces less CO₂ per unit of energy released. Therefore, CNG appears to be an excellent fuel for SI engines [10].

In recent years, a DI gasoline engine has been developed for automobile engines to improve fuel economy [2]. Direct-injection technology strongly increases the engine’s volumetric efficiency, which permits the engine to run at higher speed and produce more overall power. Direct-injection technology also reduces the need for throttling for control purposes, thus reducing the cycle pumping loss. During low loads and low engine speeds, the DI engines operate with a stratified charge. The charge stratification in the combustion chamber permits extremely lean combustion without high cycle-to-cycle variations and with high combustion efficiency, although the problem of high nitrogen oxides (NOₓ) and particulate matter (PM) emissions remains [11–14].

The spark-ignited direct-injection (SIDI) CNG engine adopts DI technology in an SI engine, and uses alternative fuel. Up until now, studies of SIDI CNG engines have concentrated on the CNG homogeneous charge, and few reports can be found related to SIDI CNG engines with stratified charge [15–20]. For the design and optimization of an SIDI CNG engine with stratified charge,
an examination of the essential features is needed. For the design and optimization of an SI engine adopting DI technology with CNG fuel, it is necessary to investigate the spray development process in order to develop more precise control of the overall equivalence ratio and the combustion propagation process.

In this study, a visualization experimental system consisting of a combustion chamber, methane fuel supply system representing CNG fuel, air supply system, electronic control unit (ECU), and data acquisition system was designed and built. Spray development and investigation of fuel injection characteristics including spray tip penetration, spray cone angle and overall equivalence ratio have been conducted under 30–90 bar fuel pressures, 1–5 bar chamber pressure. Flame propagation images and combustion characteristics via pressure-derived parameters have been analyzed at a fuel pressure of 90 bar and a chamber pressure of 1 bar at different stratification ratios (from 0% to 100%) at overall equivalence ratios of 0.6, 0.8, and 1.0.

2 Experimental Facilities

A Hitachi gasoline direct-injection (GDI) single hole injector, model number E7T05091, with 0.9 mm hole diameter was installed on the top of the combustion chamber. Methane was injected into the combustion chamber and then ignited by the extended spark plug placed in the center of the chamber as shown in Fig. 1. This arrangement of the injector and spark plugs provided a stratified charge of methane around the spark discharge position. A visualization experiment was designed and set up to investigate methane spray and combustion characteristics. Pressure of the vessel during the combustion process is recorded by a miniature piezoelectric Kistler absolute dynamic pressure transducer. Figure 2 shows the experimental setup that is consisted of five main parts [21,22]: a combustion chamber, an initial mixture supply system (filling system), a fuel supply system, an ECU, and an optical system (Z-type Schlieren setup [23]).

The fuel supply system can provide a constant 10–150 bar pressure to the injector. The fuel supply system consists of a high pressure injector, a methane tank with 99.5% purity, a high pressure regulator, a stainless steel connecting tube with an inside diameter of 4 mm that can tolerate maximum pressure of 300 bar, and a check valve and three ball valves for safety purposes. The injection pressure was adjusted by using a high pressure regulator. Method of partial pressure [24] was used to make the desired mixture.

3 Experimental Procedures

Two types of experiments were performed. The first experiment investigated the injection characteristics and the second experiment studied the combustion process of a partial premixed mixture under different turbulent intensities. In both cases, the chamber was first evacuated and then air was introduced into the vessel via the inlet valve at an initial temperature of 298 K. Mixture in the vessel was allowed to settle for 1 min to become quiescent. The first part of the experiment was to measure the injector characteristics such as overall equivalence ratio, spray tip penetration, and spray cone angle. In this step, the range of pressures of the injected fuel and chamber...
pressure were varied from 30 to 90 bar and 1 to 5 bar, respectively. At each condition, various injection durations were used and spray development images were recorded. The electronically controlled injector is the key component in the high pressure fuel injection system. Therefore, characterizing the dynamic performance of the high pressure injector was essential to the implementation of CNG–DI engine air–fuel ratio [25,26]. The injector needed to be controlled in order to achieve the precise air–fuel ratio control for the combustion experiments. The overall equivalence ratio of the mixture was determined by exact information on the partial pressures of the components.

The second part of the experiment dealt with the combustion characteristics of the partially premixed mixtures. Fuel with a specific overall equivalence ratio was injected into the vessel. The injection and chamber pressure was maintained constant at 90 bars and 1 bar, respectively. Methane jet penetrated and air was entrained into the jet, leading to the expansion of the jet. The fuel jet with high momentum collides with the opposite wall 2.1 ms from the beginning of the fuel injection and diffuses rapidly in the constant volume vessel. The injected fuel generates turbulence in the vessel and forms a turbulent heterogeneous fuel–air mixture in the vessel, similar to that in a gas direct-injection engine [13].

Fuel injection was divided into two parts in order to obtain the partially premixed direct-injection combustion. First, a portion of fuel was injected into the vessel. The setup was allowed to stabilize for 5 min to ensure the fuel air mixtures were homogeneous and remained quiescent in the vessel. Then, the rest of fuel with a specific S.R. was injected into the vessel. The stratification ratio is defined as the ratio of the amount of fuel injected in the second part to the total amount of injected fuel. The second part of the injected fuel generates the turbulence in the vessel while mixing with the homogeneous ultralean fuel–air mixtures formed by first injected fuel. The fuel–air mixture in the vessel will be relatively richer in the bottom part and leaner in the upper part before turbulence dies out. The fuel–air mixture was ignited by centrally located electrodes at a given spark delay timing of 1, 40, 75, and 110 ms after fuel injection to reflect different turbulence intensity. In addition, homogeneous premixed mixture was studied having 5 min spark delay time to provide information on laminar homogeneous mixture combustion. At least three runs at each initial condition were made to provide a good statistical sample. Based on statistical analysis, it was found that three runs are sufficient to achieve a 95% confidence level [27].

4 Results and Discussion

4.1 Fuel Injection Quantities Investigation. An injecting pulse signal from the ECU was fed into the DRIVVEN’S DI driver module kit to generate the three-stage current (10/5.5/2.5 A) required by the injector. Then, the actuator coils of the high pressure injector were charged by the dc 65 V supply voltage. The first section of current command (10 A peak current) was generated by high voltage dc 65 V to induce the electromagnetic force to draw back the nozzle needle of the high pressure injector. The overall equivalence ratio was evaluated experimentally using partial pressure coefficient of the components.

The overall equivalence ratio at various fuel pressures ranging from 30 to 90 bars at the chamber pressure of 1 bar is shown in Fig. 3. As shown, increasing the injector duration time increases the overall equivalence ratio. It is clear in this figure that we can achieve a greater equivalence ratio in constant pulse duration by increasing the fuel injection pressure.

4.1.1 Effect of Fuel Pressure on Equivalence Ratio. The overall equivalence ratio between various fuel pressures ranging from 30 to 90 bars at the chamber pressure of 1 bar is shown in Table 1. The second part of the experiment dealt with the combustion characteristics of the partially premixed mixtures. Fuel with a specific overall equivalence ratio was injected into the vessel. The injection and chamber pressure was maintained constant at 90 bars and 1 bar, respectively. Methane jet penetrated and air was entrained into the jet, leading to the expansion of the jet. The fuel jet with high momentum collides with the opposite wall 2.1 ms from the beginning of the fuel injection and diffuses rapidly in the constant volume vessel. The injected fuel generates turbulence in the vessel and forms a turbulent heterogeneous fuel–air mixture in the vessel, similar to that in a gas direct-injection engine [13].

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Table 1 Injector characteristics parameter, a, and fuel mass flow rate for fuel pressure of 90 bars as a function of chamber pressure

<table>
<thead>
<tr>
<th>P_{inj} (bar)</th>
<th>P_{ch} (bar)</th>
<th>a</th>
</tr>
</thead>
<tbody>
<tr>
<td>90</td>
<td>1</td>
<td>2.265 × 10^{-2}</td>
</tr>
<tr>
<td>90</td>
<td>2</td>
<td>1.1285 × 10^{-2}</td>
</tr>
<tr>
<td>90</td>
<td>3</td>
<td>7.4467 × 10^{-3}</td>
</tr>
<tr>
<td>90</td>
<td>5</td>
<td>4.3895 × 10^{-3}</td>
</tr>
</tbody>
</table>

4.2 Spray Development Process. Figure 5 shows methane STP, defined as the distance from the injector exit plane to the tip
of the spray. The SCA is the angle at which the methane spray expands in the radial direction. These two parameters, STP and SCA, are found by MATLAB image processing code. Tip of the spray is its head and it is a foremost position that spray reaches. The speed and extent to which the methane spray penetrates across the combustion chamber strongly affects the air utilization and fuel–air mixing rates.

Figure 6 shows a sequence of Schlieren images of the methane spray development process for injection pressure of 90 bars and an injection duration of 5 ms. The chamber temperature and pressure are 298 K and 1 bar, respectively. As shown in Fig. 6, methane spray rapidly penetrates axially and also expands in the radial direction just after the injection begins.

4.2.1 Injection Pressure Effects. Figure 7 shows the effect of injection pressure on the methane spray process. The figure shows the spray tip penetration and spray cone angle for four injection pressures of 30, 50, 70, and 90 bar. It is found that spray tip penetration was significantly affected by the injection pressure.

As the injection pressure increased, the spray tip penetration increased. The spray cone angle is influenced by the injection pressure for only the first 1.5 ms after the start of the injection. In these cases, the spray cone angle reaches 28 deg at 1.5 ms after injection.

4.2.2 Chamber Pressure Effects. The effect of chamber pressure on the methane spray process is shown in Fig. 8. These data were acquired for chamber pressures of 1, 2, 3, and 5 bars. It is shown that as chamber pressure increases, the injected methane spray penetration decreases in both the axial and radial directions. As shown in these figures, the chamber pressure has a significant effect on spray tip penetration and the spray cone angle.
The injected methane under the 5 bars chamber pressure condition penetrates quite slowly, but the penetration under the 1 bar was much faster. It took 2, 2.4, 3.2, and 4.4 ms for the methane spray tip under chamber pressures of 1, 2, 3, and 5 bars, respectively, to penetrate 131 mm from the injector exit (top of chamber) in the axial direction to bottom of chamber. In the early part of the injection process, the spray cone angle increases and then becomes constant. After approximately 1.6, 2, 2.8, and 3.8 ms from the start of injection, the spray cone angle under the 1, 2, 3, and 5 bars chamber pressure conditions increased to 28, 29.7, 31.2, and 34.1 deg, respectively, and then remained constant.

4.3 Flame Propagation Process. The momentum of the fuel jet reduces due to collision with walls and interaction between the fuel jet and the charge inside the vessel. Schlieren photographs show that the fuel–air mixtures in the vessel become almost quiescent after 5 min from the end of fuel injection. The time interval between the end of fuel injection and start of ignition, the spark delay timing ($T_{sd}$), reflects different turbulence intensities in the vessel at ignition time. Short spark delay timing creates a high...
turbulence intensity environment while long spark delay timing creates a low turbulence intensity environment.

Figure 9 shows snapshots of methane–air combustion at the stratification ratio of 100% and the overall equivalence ratio of 0.8 for different spark delay timings. The snapshots show the burned and unburned gases at 1, 4, 7, and 10 ms after spark ignition. The injection pressure, chamber pressure and temperature were 90 bars, 1 bar, and 298 K, respectively. Following the spark discharge, an electrical arc expands between the electrodes. Then, a high temperature plasma followed by a flame kernel is formed. For the homogeneous case (\(T_{sd}=5\) min), the flame becomes spherical and propagates outwardly toward the surfaces of the vessel.

For nonhomogenous cases, flame shape is a function of turbulence intensity which is inversely related to the spark delay times. Flame is smooth and laminar at the beginning but becomes cellular as pressure increases reducing flame thickness and increasing wrinkles. Turbulent flames at spark delay timing of 40, 75, and 110 ms propagate outwardly from the center of the vessel with wrinkled flame surfaces compared to the smooth flame front of homogeneous mixture combustion.

The effect of turbulence on the enhancement of combustion can be clearly observed from these images, and the wrinkled flame front is the typical indication of turbulent combustion. The images also show that the flame propagation speed of direct-injection turbulent combustion is much higher than that in the case of homogeneous mixture combustion. The increase of spark delay timing decreases the turbulent flame propagation speed, and this is due to the reduction of turbulence intensity generated by the fuel spray.

Flame images of the partially premixed direct-injection combustion of methane at the equivalence ratio of 0.8, spark delay timing of 1 ms and different S.R. are shown in Fig. 10. It can be seen that the flame kernel grows faster as stratification ratio increases. This is due to the increase in turbulence intensity as the stratification ratio increases. It indicates that the burning rate is increasing with the increase of turbulence intensity within the experimental range.

Figure 10 also shows that the flame propagation process in the 25% stratification condition is much different than that of 100% stratification condition, where the latter is similar to that of turbulent premixed flame [28]. This indicates that the spark ignition

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becomes more stable with the decreasing of stratification ratio due to the compromise of relatively lower turbulence intensity and richer mixture near the spark position at the spark timing [29].

In these experiments, the rate of pressure rise is proportional to the rate of energy release, which is an important characteristic of the combustion process. Figure 11 shows the pressure (a) and the rate of pressure rise (b) of methane/air combustion at different spark delay timings and the overall equivalence ratio of 0.8. The pressure–time curve of the homogenous mixture combustion shows the slow increase compared to the turbulent flame, leading to the peak pressure of the homogenous mixture combustion being lower than those of turbulent flame. In the case of turbulent combustion, only a slight decrease in peak pressure is observed at different spark delay timings.

Meanwhile, turbulent combustion reaches its peak pressure earlier compared to homogeneous mixture combustion, and turbulent combustion shortens main combustion duration compared to homogenous mixture combustion. Main combustion duration, that is defined as the time interval from 10% of the pressure rise to 90% of the pressure rise, decreases with increasing turbulence intensity. Peak pressures versus stratification ratio at different spark delay timings and overall equivalence ratio of 0.6 and 1.0 are given in Fig. 12. The high burning speed of the mixtures at the stoichiometric condition (Φ = 1.0) leads to a minimal influence on the peak pressure among various stratification ratios in the case of both homogeneous mixture combustion and direct-injection turbulent combustion.

Peak pressure is increased with the increase of turbulence intensity in the vessel at all equivalence ratios. The peak pressure value of all mixtures gives the larger difference in the case of lean mixture combustion (Φ = 0.6), and this indicates that the influence of turbulence on combustion is larger in lean mixture combustion than in rich mixture combustion. The turbulence intensity at the same spark delay timing will decrease with the decrease of stratification ratio which would lead to the decrease of combustion rate, peak pressure and charge stratification of the fuel–air mixture. Peak pressure is increased with the decrease of spark delay timing at all stratification ratios.

This shows that the combustion of lean methane–air mixtures inside the CVCC can be enhanced by advancing the spark delay...
The initial combustion duration (the time interval from the ignition timing to 10% of the pressure rise) and the main combustion duration (the time interval from 10% of the pressure rise to 90% of the pressure rise) versus stratification ratio at different spark delay timings for the overall fuel air equivalence ratio of 0.6 and 1.0 are given in Figs. 14 and 15, respectively. The results show that the initial and main combustion durations increase with decreasing stratification ratio at an equivalence ratio of 0.6 and this is due to reduction of turbulence intensity. Both the initial and main combustion durations decrease as spark delay timing decreases regardless of the stratification ratio due to the combustion enhancement with high turbulence. The effect of stratification ratio on combustion duration is more obvious at lean mixture condition ($\Phi = 0.6$) than in stoichiometric mixture. Initial combustion duration and main combustion duration are slightly influenced by the stratification ratio. The turbulence generated by fuel direct injection can remarkably decrease the initial combustion duration and main combustion duration compared to homogeneous mixture combustion. This effect is more obvious at lean mixture combustion, which suggests that turbulence is an effective method to improve the lean mixture combustion.

5 Conclusions

In this study, a visualization experimental system was designed and constructed to investigate spray and lean-burn methane direct-injection combustion by using a constant volume vessel. The main conclusions are summarized as follows:

(1) It was found that spray tip penetration was significantly affected by the injection pressure. Under relatively higher chamber pressure conditions, the injected methane spray penetrated at a very low rate in both the axial and radial directions.

(2) A smooth flame front propagates in the case of homogeneous mixture combustion while a wrinkled flame front is demonstrated in the case of direct-injection turbulent combustion.

(3) The rate of pressure rise of lean-burn methane decreases with increasing of spark delay timing while it increases with increasing of stratification ratio.

(4) Fuel direct-injection turbulent combustion results in higher peak pressure and maximum rate of pressure rise, shorter initial combustion duration, and shorter main combustion duration compared with homogeneous mixture combustion.

Nomenclature

- $a =$ coefficient of overall equivalence ratio
- $n =$ fuel mass flow rate
- $P =$ pressure
- $T =$ time (ms)
- $\Delta t =$ injection duration (ms)
- $\Phi =$ overall equivalence ratio

Subscripts

- ch =$=$ chamber
- inj =$=$ injection
- sd =$=$ spark delay

References


