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# COMBUSTION AND EMISSION CHARACTERISTICS OF A LEAN BURN NATURAL GAS ENGINE

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**ABSTRACT**−Lean burn is an effective way to improve spark ignition engine fuel economy. In this paper, the combustion and emission characteristics of a lean burn natural gas fuelled spark ignition engine were investigated at various throttle positions, fuel injection timings, spark timings and air fuel ratios. The results show that ignition timings, the combustion duration, the coefficient of variation (COV) of the indicated mean effective pressure (IMEP) and engine-out emissions are dependent on the overall air fuel ratio, spark timings, throttle positions and fuel injection timings. With the increase of the air fuel ratio, the ignition delays and combustion duration increases. Fuel injection timings affect ignition timings, combustion duration, IMEP, and the COV of the IMEP. Late fuel injection timings can decrease the COV of the IMEP. Moreover, the change in the fuel injection timings reduces the engine-out CO, total hydrocarbon (THC) emissions. Lean burn can significantly reduce NOx emissions, but it results in high cyclic variations.

**KEY WORDS :** Natural gas, Emission, Lean burn, Combustion, Engine

## 1. INTRODUCTION

Natural gas has a wide flammability range and a high research octane number. The knock resistance of lean mixtures is higher than that of stoichiometric mixtures, thus permitting the development of high compression ratio engines. Therefore, a lean burn natural gas engine with a high compression ratio can achieve a high thermal efficiency, due to the increased specific heat ratio, lower combustion temperature and reduced throttling losses (Corbo *et al*., 1995; Das and Watson, 1997; Manivannan *et al*., 2003). On the other hand, the H/C ratio of natural gas is approximately in the range 3.7 to 4.0 (Einewall *et al*., 2005). Accordingly, natural gas engines can achieve $CO_2$ levels below those of diesel engines through burning mixtures with the same air fuel ratio, while keeping almost the same thermal efficiency under very lean conditions (Mtui *et al*., 1996; Tilagone *et al*., 1996). $CO_2$ emissions from a natural gas engine can be reduced by more than 20% compared to a gasoline engine at equal power (Kato *et al*., 2001). By increasing boost pressure levels, lean burn natural gas engines with the same displacement can produce higher power (Borges *et al*., 1996), and their full-load thermal efficiencies can even be very close to those of diesel engines (Yamato *et al*., 2001). Therefore, natural gas can be used as a reliable, safe and efficient alternative fuel of internal combustion engines for transportation mainly due to its abundance and indigenous availability at attractive prices, and is expected to find widespread use as a means to reduce $CO_2$ emissions and toxic gases from vehicles.

Lean mixtures can be formed through either premixed or stratified charge strategies, but the variability in air or fuel or both could change the air fuel ratio from cylinder-to-cylinder and from cycle-to-cycle even when the fluctuations of the overall air fuel ratio are limited (Hassaneen *et al*., 1996). If an excessively lean mixture in the vicinity of the spark plug appears at the time of spark discharge, the risk of misfire increases (Hiltner *et al*., 1996), resulting in increased exhaust emissions and decreased thermal efficiency due to unstable combustion.

Various strategies have been put forward to prepare lean mixtures for spark ignition natural gas engines in order to extend their lean burn limits. On a four cylinder spark ignition natural gas engine with a multipoint fuel injection system and a single point fuel injection system, Puzinauskas *et al*. (2000) found that the coefficient of variation (COV) of the indicated mean effective pressure (IMEP) in the case of multipoint fuel injection is higher than that of the well mixed single point case at various air fuel ratio conditions. However, with the increase of engine speed and load, the COV of the IMEP sharply decreases in the former case, and the influence of the two fuel injection systems on the COV of the IMEP decreases. Kato *et al.* (2001) used an optimized swirl control valve in the intake port and tumble-enhancing piston crown geometry to generate gas flow fields with strong tumble. At an excess air ratio of approxi-

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mately 1.6, stable combustion can be achieved and the fuel economy is improved by approximately 10 percent compared to a stoichiometric condition at minimum ignition advance for the best torque. Chung *et al.* (2003) enlarged the equivalence ratio limit of lean combustible mixture from 0.7 to 0.5 through stratified charge by an injector-type spark plug in a constant volume chamber. Reynolds *et al.* (2005) developed a partially stratified charge spark-plug injector to extend the lean burn limit on a single cylinder natural gas engine. They found that partial stratification through pure fuel injection increases diffusion combustion and reduces the COV of the IMEP to below 5 percent at lambda=1.66.

The increase of spark energy can extend lean burn limits of natural gas engines. Unfortunately, it negatively impacts spark plug durability and effectiveness in transmitting adequate energy as an ignition source. Laser ignition offers the potential to improve ignition system durability and engine combustion performance, as well as to reduce maintenance. Richardson *et al.* (2004) found that a laser can significantly extend the misfire limit and the total operating envelope can be increased by 46 percent. Half the minimum NOx emissions from the engine using spark plug ignition systems can be achieved with no appreciable degradation in thermal efficiency while the hydrocarbon emissions are comparable.

In addition, hydrogen-enriched natural gas can extend lean burn limits and reduce NOx emissions. On an 11 liter heavy duty engine, Collier *et al.* (2005) found that equivalence ratios of 0.53 have been achieved over the operating range for a mixture with high hydrogen content. In steady state, tailpipe NOx emissions below 0.15 g/bhp can be obtained.

For a sequential injection natural gas engine, fuel injection timing during the intake stroke affects the mixture formation in the cylinder and combustion processes. The objective of this study is to investigate the combustion and emission characteristics of a port fuel injection lean burn natural gas engine at different operating conditions.

## 2. EXPERIMENTAL APPARATUS AND TEST PROCEDURE

The engine used in this study was a modified four-stroke four cylinder, four valve port fuel injection spark ignition engine with three cylinders that use gasoline and one cylinder fuelled with natural gas (due to the inconsistencies of natural gas injectors). The details of the test engine and test conditions are given in Table 1.

The amount of natural gas injected into the intake port could be changed by an electronically controlled unit. To measure the air fuel ratio of the mixture, an ETAS linear oxygen sensor with an accuracy of ±1.5% was mounted in a separate exhaust pipe connected to the cylinder fueled with natural gas.

The engine was coupled to an eddy dynamometer.

Table 1. Engine specifications and test conditions.

| Engine | Port fuel injection |
|---|---|
| Bore×Stroke (mm) | 78.7×69 |
| Compression ratio | 9.3 |
| Engine speed (rpm) | 2000 |
| Coolant temperature (ºC) | 80±1 |
| Oil temperature (ºC) | 80±1 |

Through adjusting the fuel injection timing and the injection pulse width of a natural gas injector, the relative air fuel ratio (lambda) of the mixture could be altered under different operating conditions. The pressure of the cylinder burning natural gas was acquired with a piezoelectric transducer, a YE5850 type charge amplifier, and an encoder with an angular resolution of 0.5º crank angle. The cylinder pressure used to analyze the combustion processes was ensemble averaged over 100 consecutive cycles.

Through a Horiba MEXA-7100DEGR exhaust gas analyzer, the total hydrocarbon (THC), CO, and NOx emissions were analyzed with a flame ionization detector (FID), a non-dispersive infrared analyzer (NDIR) and a chemiluminescent detector (CLD) respectively. The THC, CO and NOx emissions were the average values of the acquired data in one minute at each steady state operating condition.

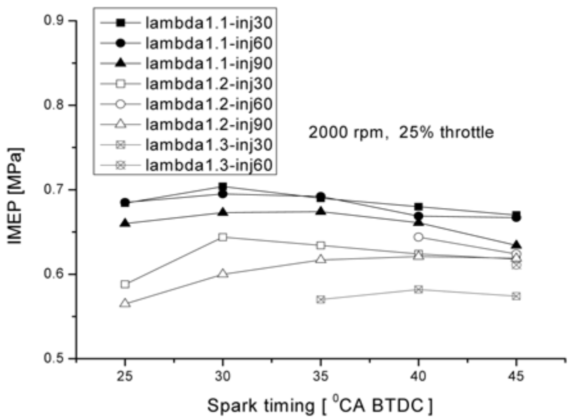
The natural gas used was composed of 92.6% methane, 5.8% ethane, 1.2% carbon dioxide and 0.4% other components by volume.

## 3. RESULTS AND DISCUSSION

Based on the cylinder pressure and the first law of thermodynamics, the IMEP and the rate of heat release can be calculated (Heywood, 1988). The ignition timing (CA10) is indicated by the crank angle at which 10 percent of the fuel is burned. The main combustion period (CA90) is defined as the 10~90 percent mass fraction fuel burn duration.

Figure 1 shows the effect of spark timing on IMEP under different operating conditions. The numbers following "lambda" and "inj" in Figure 1 are the overall excess air ratio in the cylinder and the start of fuel injection after top dead center (ATDC) on the intake stroke respectively. For example, "lambda1.1-inj30" means that the overall lambda of the mixture in the cylinder is 1.1 and fuel injection begins from 30ºCA ATDC on the intake stroke. Since the injector could not inject enough natural gas when the intake valves were opened during the intake stroke, only 30ºCA ATDC and 60ºCA ATDC fuel injection timings were investigated at full throttle.

As expected, the IMEP decreases with increasing lambda at a given fuel injection timing, spark timing and throttle position since the available energy released from the leaner

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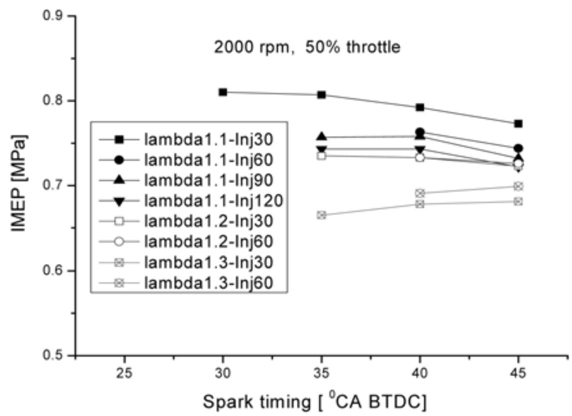


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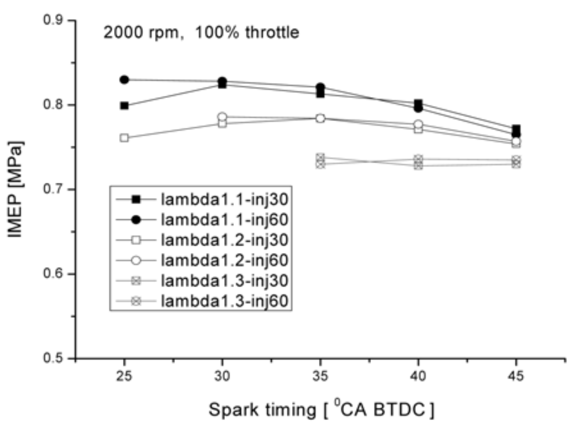



(c)

Figure 1. The effect of spark timing on the IMEP at different operating conditions.

mixture to produce power decreases. On the other hand, late ignition with a slow burning rate results in a long main combustion duration, and a significant fraction of heat is released well after top dead center. Hence, the available energy released from the leaner mixture decreases and the IMEP falls.

Moreover, IMEP is dependent on throttle positions and injection timings at a given lambda and spark timing. The IMEP distinctly decreases with a delay of the fuel injection timing at lambda=1.1 and 1.2 at 25% and 50% throttle in most cases. It is interesting that when fuel injection timing is 60ºCA ATDC during the intake stroke at lambda=1.2 and 25% throttle, the IMEP is higher than that at the two other fuel injection timings. However, at lambda=1.3, the IMEP at 60ºCA ATDC fuel injection timing is higher than that at 30ºCA ATDC fuel injection timing in most cases. Since a late fuel injection timing causes early ignition and a short combustion duration at partial throttles, the available energy released from the mixture increases and the IMEP is enhanced. This indicates that injection timing can be a way to change the IMEP through controlling the ignition phase and combustion duration.

In addition, the IMEP increases sharply from 25% throttle to 50% throttle while it only increases slightly when the throttle open position is further increased. Natural gas is a gaseous fuel, it occupies some cylinder volume when it is drawn into the cylinder, and the amount of fresh air in the cylinder is reduced. Accordingly, the increment rate of air fuel mixture from 50% throttle to 100% throttle is less than that from 25% throttle to 50% throttle, and the IMEP decreases. How to maintain a port fuel injection naturally aspirated natural gas engine power retrofitted from a gasoline engine is an issue that remains to be solved. Furthermore, fuel injection timing has a slight effect on the IMEP at full throttle. The primary reason for this is that, as the amount of fuel drawn into the cylinder is higher and the time interval between the fuel injection end and spark discharge decreases, the combined effect of the ignition timing and combustion duration on the IMEP decreases.

Independent of throttle positions, the optimum spark timing (MBT) for the maximum IMEP advances with the increase of lambda, which is related to the slow burn rate and late ignition timing under lean burn conditions.

CA10 depends on the temperature, pressure, and the composition of the air fuel mixture in the cylinder (Stone, 1999) and has a close relation to the laminar flame speed of the mixture (Marforio *et al.*, 1994).

Figure 2 shows the effect of spark timing on CA10 at different operating conditions. Independent of fuel injection timings and throttle positions at fixed lambdas, CA10 initiates early at early spark timings, while it occurs late with an increased lambda at fixed spark timings and injection timings. When the air fuel mixture becomes lean, the chemical reaction in the early stage of flame development slows down. As a result, the time required for the initial flame development increases and ignition delays.

The fuel injection timing has a different impact on the phasing of CA10 at various lambdas and throttle positions at fixed spark timings. At 25% and 50% throttle, CA10 often takes place early at late fuel injection timings, especially in the cases of the spark timings that are earlier than 35ºCA BTDC. Since late fuel injection decreases the time available to prepare a homogeneous mixture in the cylinder, a relatively rich mixture may form near the spark plug at the

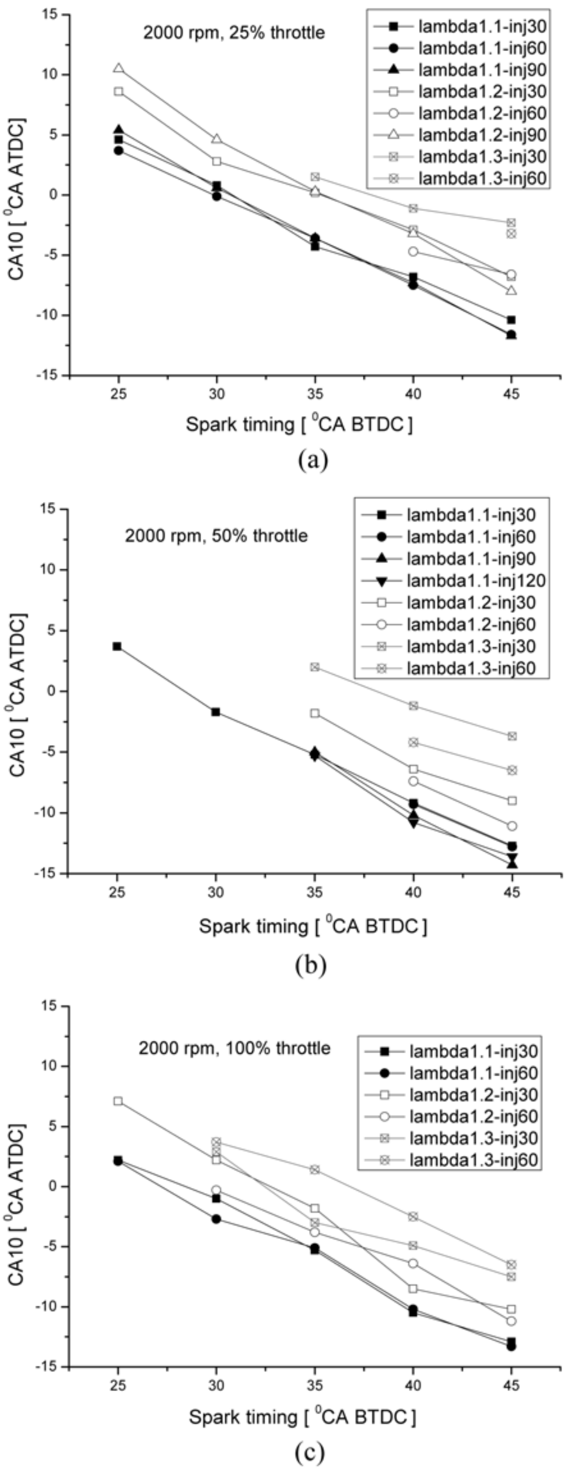



Figure 2. The effect of spark timing on CA10 at different operating conditions.

spark time, which is beneficial to flame kernel development. It indicates that the change of fuel injection timings may be a way to control the ignition phase and IMEP. However, the relationships between ignition timings and fuel injection timings are more complicated at full throttle.

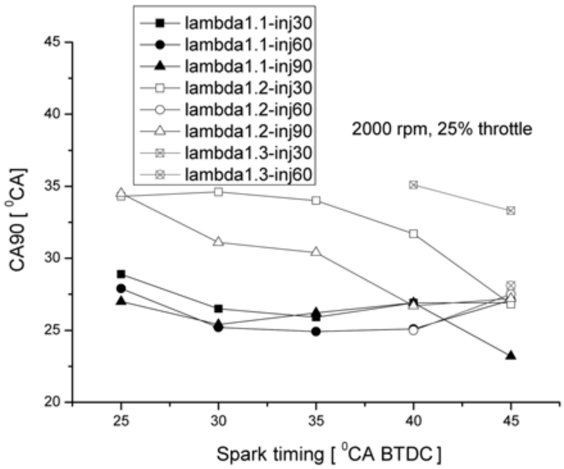
At late fuel injection timings, CA10 takes place early in most cases at lambda=1.1 and 1.2, while it occurs late at lambda=1.3. Since the amount of natural gas in the cylinder is less at lambda=1.3, early fuel injection indicates that the mixture has a long time to exchange heat and accelerate low temperature oxidation before spark discharge. This may improve ignition.

Compared to the CA10 at different throttle positions, it is found that, at the same lambda, spark timing and fuel injection timing, CA10 advances with increasing throttle. One reason for this is that the amount of the mixture increases at wider throttles and the mixture temperature in the cylinder at spark timing is higher due to compression, which facilitates the subsequent flame kernel development. Another reason is that the residual fraction in the cylinder decreases with increasing throttle, and enhances the combustion stability, which is helpful for early flame development.

The effect of spark timings on CA90 is shown in Figure 3. Independent of the throttle positions, in most cases, CA90 increases with lambda at given fuel injection timings due to slow burn rate. Late ignition timings result in a long combustion duration. The shortest combustion duration corresponds to early spark timing in most cases. Early spark timings means an early occurrence of ignition, combustion proceeds in the increasing cylinder temperature. Therefore, the combustion duration decreases.

It is found that, at lambda=1.3, a late spark timing leads to a long combustion duration at 25% and 100% throttle. Since CA10 takes place late and even occurs after top dead center on the expansion stroke, the combustion velocity slows down and, finally, the combustion duration increases. However, the effect of the fuel injection timing on the combustion duration is more complicated. At 25% throttle, the combustion duration decreases with delayed fuel injection timings at fixed spark timings in most cases. Although the amount of residual gas is high, the late fuel injection timing helps to form stratified mixture in the cylinder. As a result, the residual gas has a slightly negative effect on the occurrence of the ignition and combustion velocity. In the meantime, early ignition takes place in a decreasing cylinder volume, and burning rate increases with an increasing cylinder temperature. Eventually, the main combustion period is shortened. At 50% throttle, the combustion duration increases when the injection timing retards at lambda=1.1, although the ignition timing occurs early at late fuel injection timing. Earlier ignition indicates a more heterogeneous mixture formed in the cylinder, and the combustion duration increases. At full throttle, the effect of fuel injection timing on the combustion duration decreases. The primary reason is that the amount of fuel in the cylinder is much higher than that at the other two throttle positions. Hence, the fuel injection pulse width is longer. The effect of fuel injection timing on mixture formation in the cylinder has a slight effect and the combustion duration changes somewhat.

Figure 4 illustrates the relationship between spark timing

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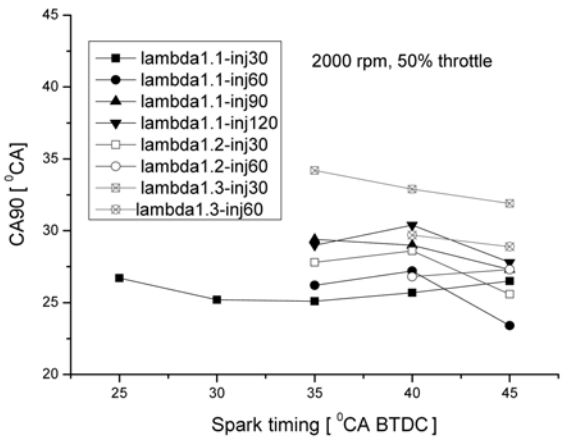


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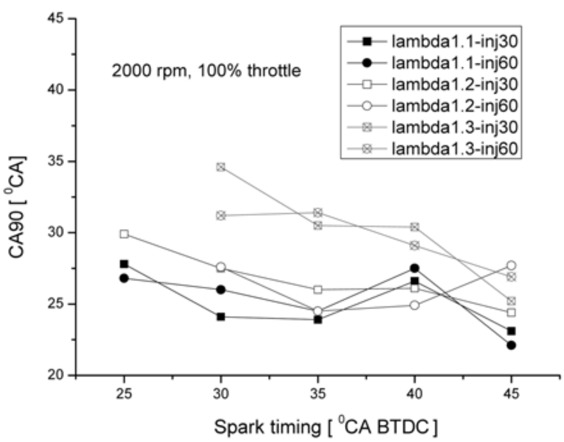


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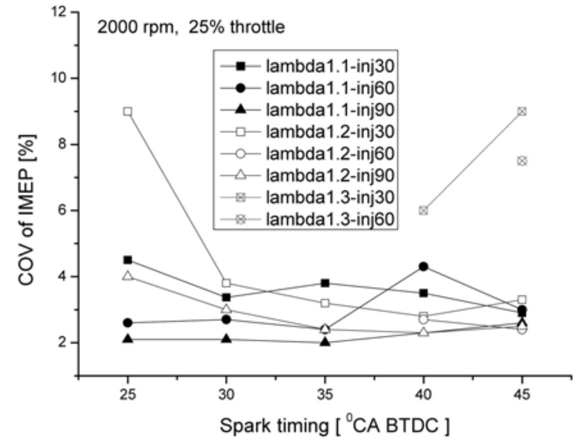
Figure 3. The effect of spark timing on CA90 at different operating conditions.

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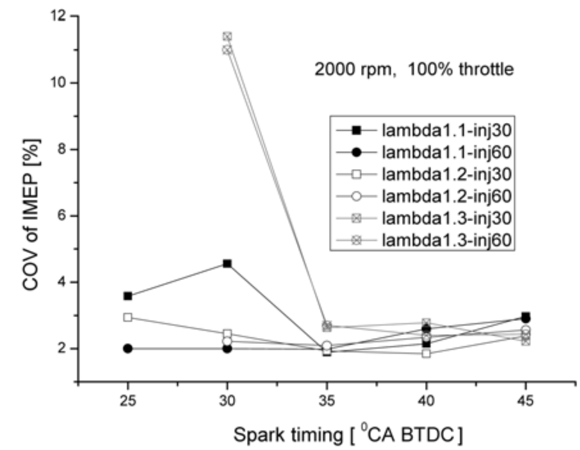


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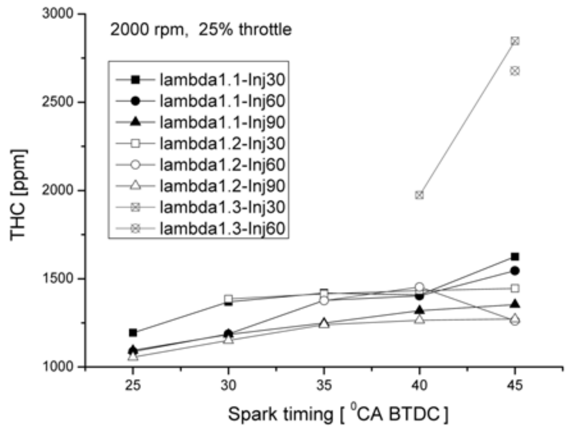
Figure 4. The effect of spark timing on the COV of the IMEP at different operating conditions.

and the COV of the IMEP under different operating conditions. It is shown that the COV of the IMEP is dependent on the spark timing, lambda, and fuel injection timing. At 25% throttle, the COV of the IMEP decreases with increasingly delayed fuel injection timing at fixed lambdas, in most cases. The amount of fuel in the cylinder is low under lean burn conditions, the later the fuel injection timing, the shorter the time it has to prepare a homogeneous mixture at a given spark timing. Changes in injection timings may alter the direction or the intensity of the bulk flow (Ting *et al.*, 1994; Lee *et al.*, 1996). The mixture is more heterogeneous in the cylinder at late fuel injection timings and a relatively rich mixture is probably formed around the spark plug at the time of spark discharge. This enhances the early flame development and improves the combustion stability. Moreover, the COV of the IMEP changes little with spark timing except when ignition occurs too late.

However, at full throttle, the effect of fuel injection timing on the COV of the IMEP is dependent on spark timings. In the case of spark timings later than 35ºCA BTDC, the COV of the IMEP decreases with delayed fuel injection timing, while it increases when spark timings are earlier than 35ºCA BTDC. This is related to the mixture formation in the cylinder at the time of spark discharge.

Therefore, changing the fuel injection timing is a way to improve the combustion stability of a stratified natural gas

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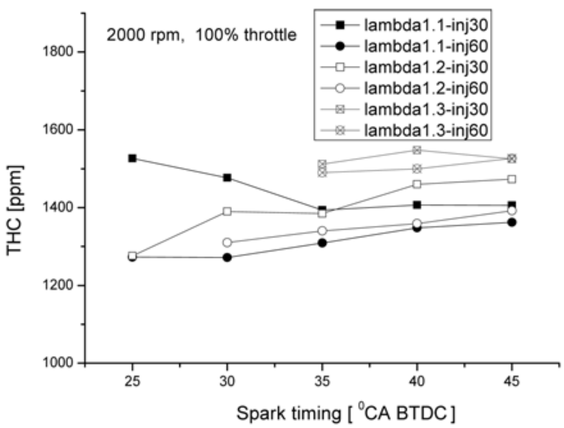


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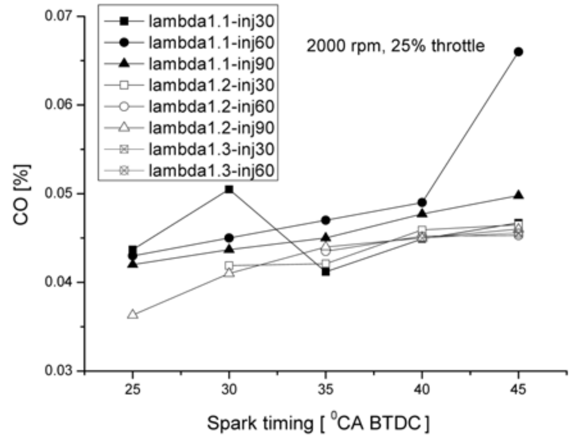
Figure 5. The effect of spark timing on THC emissions at different operating conditions.

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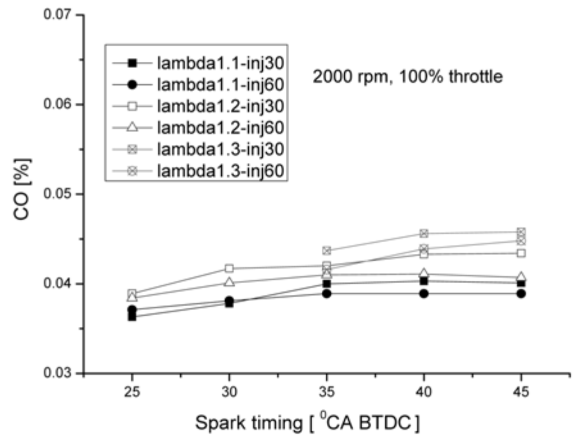


(b)

Figure 6. The effect of spark timing on CO emissions at different operating conditions.

fuelled engine.

As shown in Figure 4, the COV of the IMEP increases sharply at lambda=1.3 at full throttle when the spark timing is later than 35ºCA BTDC. CA10, shown in Figure 2, occurs after top dead center, which worsens the initial flame development and combustion stability. However, at early spark timings, the occurrence of ignition for the lean mixture is before top dead center, and the effect of spark ignition timings on the COV of the IMEP decreases.

Figure 5 shows the engine-out THC emissions at different operating conditions. THC emissions gradually increase with advanced spark timing in most cases. Early spark timings give rise to an early onset of ignition, and combustion finished early. Accordingly, the burned gas temperature is low in the expansion stroke. This impedes the oxidation of unburned fuel in the cylinder. As a result, engine-out THC emissions are relatively high. THC emissions at lambda =1.3 and 25% throttle are much higher than those at the other operating conditions due to the relatively higher COV of the IMEP shown in Figure 4. Bulk quenching of the flame is also attributed to high THC emissions at lambda =1.3 due to stratified mixture in the cylinder.

It is also found that late fuel injection timing decreases THC emissions at given lambdas for fixed spark timings. Late injection indicates that fuel has less time to mix with air and to contact the cylinder walls. This decreases the bulk quenching and boundary layer quenching in the cylinder, and lowers THC emissions. On the other hand, less unburned fuel trapped in the crevices in the cylinder is released late in the exhaust stroke, which is helpful to the reduction of THC emissions.

Moreover, THC emissions at lambda=1.2 can be reduced by changing the fuel injection timings at given spark timings. Late ignition and a long combustion duration increase the burned gas temperature late in the expansion stroke, which facilitates the complete oxidation of unburned fuel and lowers THC emissions. A lean mixture, on the other hand, can reduce the unburned mixture trapped in the crevices, and is beneficial to the reduction of THC emissions.

Figure 6 shows the trend of engine-out CO emissions under different operating conditions. Regardless of the throttle positions, CO emissions increase steadily, except for the case at lambda=1.1 and 30ºCA ATDC fuel injection timing at 25% throttle. Early spark timing at a given lambda indicates that the time to prepare the mixture is decreased. The fuel distribution is more heterogeneous in the cylinder, and incomplete combustion may occur in the fuel-rich regions or fuel-lean regions, thus increasing the engine-out CO emissions. On the other hand, early spark timing results in early ignition and short combustion duration, which lowers the burned gas temperature during the expansion stroke and decreases the amount of CO oxidation in the cylinder. Accordingly, CO emissions rise.

Furthermore, the general trend of CO emissions is different with increasing values of lambda at the two different throttle positions. At 25% throttle, CO emissions decrease with lambda, except for the case at lambda=1.1 and 30ºCA ATDC fuel injection timing although it is not the case at full throttle. Since the occurrence of ignition is later and the combustion duration is relatively longer at 25% throttle than that at full throttle at the same lambda and spark timing, this can improve CO oxidation late in the expansion stroke. On the other hand, the low combustion temperature also diminishes the dissociation of $CO_2$ in the cylinder, and decreases CO emissions (Heywood, 1988). Besides, CO emissions at full throttle are lower than those at 25% throttle at the same operating conditions. The higher combustion temperature in the former case facilitates complete oxidation of fuel and decreases CO emissions.

Fuel injection timing also affects the CO emission characteristics. As shown in Figure 6, late injection timing at fixed spark timings decreases CO emissions in most cases. Fuel injection timings can change the fuel distribution in the cylinder and influence the combustion process. Late injection offers less time to mix fuel and air in the cylinder and reduces the likelihood that fuel contacts the cylinder walls. It can decrease the amount of fuel burned late in the expansion stroke and exhaust stroke. Consequently, the engine-out CO emissions can be decreased.

Figure 7 shows the effect of spark timings on NOx emissions under different operating conditions. As expected, NOx emissions decrease with increasing lambda. Low combustion temperatures at lean conditions contribute to the reduction of engine-out NOx emissions. However, NOx emissions at full throttle are higher than those at 25% throttle at the same lambda and spark timing, since more mixtures are burned and a higher combustion temperature can be reached in the cylinder. However, early spark timing causes high NOx emissions, since early ignition causes a high combustion temperature in the decreasing cylinder volume.

Fuel injection timing affects the time it takes to prepare the mixture at different lambdas, which influences the fuel distribution in the cylinder, local air fuel ratio, ignition timing, and combustion processes. At the same spark timing and lambda, late fuel injection leads to a more inhomogeneous mixture in the cylinder. As a result, higher combustion temperatures are present in some regions, and the

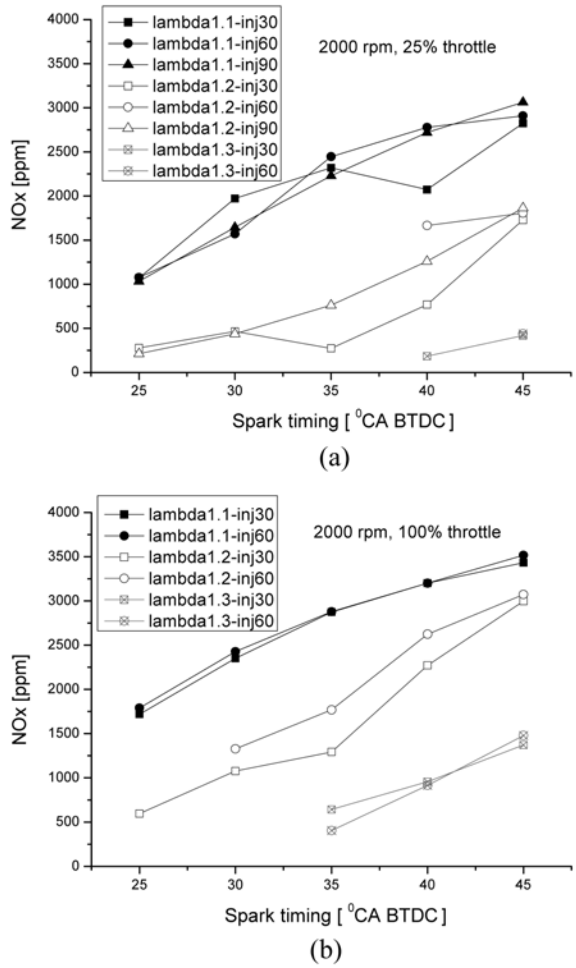


Figure 7. The effect of spark timing on NOx emissions at different operating conditions.

NOx emissions even increase. Therefore, engine-out NOx emissions are dependent on ignition timings, combustion duration and the local air fuel ratio. In the case of different fuel injection timings, engine-out NOx emissions are higher when ignition occurs earlier and the combustion duration is shorter.

## 4. CONCLUSION

The combustion and emission characteristics of a lean burn natural gas engine under various operating conditions were investigated and the main conclusions were drawn as follows:

By increasing the throttle open position, the IMEP is enhanced. However, the rate of the increase of the IMEP is slight after 50% throttle.

With the increase of lambda, the ignition timings occur late and the combustion duration and the COV of the IMEP increase. Lean burn can significantly decrease NOx emissions. However, engine-out CO and THC emissions under lean burn conditions are dependent on the fuel injection timing, throttle position and spark timing.

The fuel injection timing has different effects on the IMEP at various throttle positions. At part throttle, late fuel injection timing evidently decreases the IMEP, while it only slightly affects the IMEP at full throttle. Fuel injection timing also affects ignition timing at given spark timings. Late fuel injection timings reduce CO and THC emissions at 25% and 100% throttle.

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