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Combustion and Emission Characteristics of a Natural Gas Engine under Different Operating Conditions

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Abstract

Natural gas is a promising alternative fuel of internal combustion engines. In this paper, the combustion and emission characteristics were investigated on a natural gas engine at two different fuel injection timings during the intake stroke. The results show that fuel injection timing affects combustion processes. The optimum spark timing (MBT) achieving the maximum indicated mean effective pressure (IMEP) is related to fuel injection timing and air fuel ratio.

At MBT spark timing, late fuel injection timing delays ignition timing and prolongs combustion duration in most cases. But fuel injection timing has little effect on IMEP at fixed lambdas. The coefficient of variation (COV) of IMEP is dependent on air fuel ratio, throttle positions and fuel injection timings at MBT spark timing. The COV of IMEP increases with lambda in most cases. Late fuel injection timings can reduce the COV of IMEP at part loads. Moreover, engine-out CO and total hydrocarbon (THC) emissions can be reduced at late fuel injection timing.

Keywords: Natural gas, Emission, Lean burn, Combustion, Engine

1. Introduction

To meet the increasing severity of vehicle emission limits, significant reduction in exhaust pollutants from internal combustion engines should be imposed. Marked improvement in engine fuel economy is needed to reduce vehicle CO_2 emissions. There are many technologies to promote global environmental protection and effectively utilize energy resources. Natural gas, as an alternative vehicle fuel, is the one to meet the goals above and is expected to find widespread use.

Natural gas can be used as engine fuel in two ways. One is to use it on a dual fuel engine. In this case, natural gas is the main fuel and is ignited by pilot diesel fuel. Therefore, soot and NO_x emissions from conventional diesel engines can be minimized through the replacement of diesel by natural gas.[1)] As compared to their diesel counterparts at light loads, however, dual fuel engines usually have a relatively low efficiency and power output, coupled with high unburned hydrocarbons and CO emissions. Injection timing and the quantity of pilot diesel fuel are the most important control variables related to the performance and exhaust emissions of a dual fuel engine. At light loads, advancing injection timing of pilot diesel fuel can improve thermal efficiency and reduce unburned hydrocarbons and CO emissions while it increases NOx emissions.[1,2)] The reduction in the quantity of pilot diesel fuel can decrease soot emissions.[3)] Usually, dual fuel engines are retrofitted from diesel engines. However, they need two fuel supply systems.

Another is to use natural gas on a spark ignition engine. Since methane is the primary constituent of natural gas, it has a high research octane number and a wide flammability range, permitting the development of high compression ratio engines. Lean mixtures have higher knock resistance than stoichiometric mixtures. Therefore, the thermal efficiency of a lean burn natural gas engine can be improved dramatically thanks to increased specific heat ratio, low combustion temperatures, high compression ratio and reduced throttling losses.[4-6)] By increasing boost pressure level, a lean burn natural gas engine can produce higher power[7)] and its full-load thermal efficiency can even be very close to that of a diesel engine.[8)] NOx emissions from a natural gas engine can also be reduced to below the corresponding levels of a gasoline engine due to lower combustion temperature of the natural gas engine as compared to that of a gasoline engine.[6)] Moreover, CO_2 emissions from a natural gas engine can be decreased by more than 20 percent in comparison with those of a

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gasoline engine operating at the same loads due to the high hydrogen content of natural gas fuel.[9,10] The CO_2 levels of a natural gas engine can even be lower than those of a diesel engine at the same air fuel ratio, while keeping almost the same thermal efficiency under very lean conditions.[11] Therefore, natural gas is a promising alternative fuel of spark ignition engines.

Spark ignition natural gas engines have been investigated worldwide. On a single cylinder natural gas engine with a swirl generator, Goto et al found that to achieve stable combustion at 1000 rpm, a low main swirl flow with strong turbulence strength near the spark plug is needed and the cycle variation of mixture in the intake port should be suppressed.[12] Evan et al compared the performance and emissions of a single cylinder engine fueled with natural gas and gasoline at various engine operating conditions. They found that the maximum brake torque spark timing for natural gas was more advanced between 2º and 10º crank angle degree than that for gasoline at air fuel ratio close to stoichiometry while the difference between the two fuels remained unchanged regardless of air-fuel ratios at low loads. Total hydrocarbon emissions were about 50 percent lower than those for gasoline at wide open throttle (WOT).[13] Puzinauskas et al.[14] conducted combustion tests on a four cylinder spark ignition natural gas engine with different injection systems. They found that the coefficient of variation (COV) of indicated mean effective pressure (IMEP) in the case of multipoint injection is higher than that for the well mixed single point case at low engine speed. But in dependent of injection systems, the COV of IMEP sharply decreases and their differences between the two injection systems decrease with increased engine speed and load. Moreover, the injection systems have little effect on ignition delay. On a modified single cylinder diesel engine with one swirl intake port and one straight intake port, Yamato et al.[8] carried out an experiment on the mixture distribution and ignitability when natural gas was injected into the different regions in the straight intake port. They found that stable ignition and extended lean limits could be achieved if the injection position could create a rich mixture region beneath an intake valve and the rich mixture region were transported to the vicinity of the spark plug at the spark timing. Improved ignition stability through adjusting the fuel injection position could increase engine thermal efficiency.

To meet stringent emission regulations, sequential fuel injection systems will be widely used on modern spark ignition natural gas engines. However, to my knowledge, few literatures are related to the combustion and emission characteristics of a port fuel injection engine under different fuel injection timings. The objective of present contribution is to extend the understanding of the relationship between fuel injection timing, combustion and emission characteristics of a port fuel injection natural gas engine under various air fuel ratios.

2. Experimental Apparatus and Test Procedures

2.1. Experimental Apparatus

An experimental investigation was conducted on a four-stroke four cylinder, four valve port fuel injection spark ignition research engine with pent-roof combustion chamber. Fig. 1 shows the experimental apparatus. Only No. 1 cylinder was fueled with natural gas and the other three cylinders were fueled with gasoline. The engine had 78.7 mm bore, 69 mm stroke and 9.3 compression ratio.

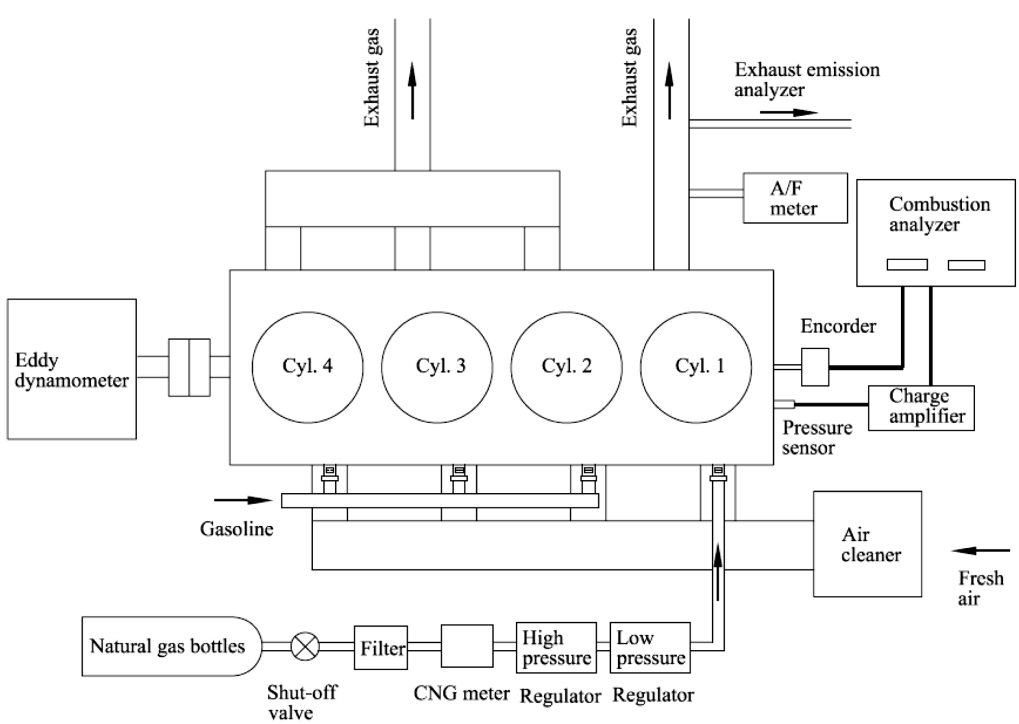


Fig. 1. Schematic diagram of the experimental setup.

Table 1 The compositions of natural gas

Component	Volumetric fraction (%)
Methane	92.6
Ethane	5.8
Carbon dioxide	1.2
Others	0.4

The gas fueling system consisted of high pressure natural gas bottles, a filter, a natural gas flow meter, a coolant heated high pressure regulator and a low pressure regulator. After the two regulators, natural gas was decompressed from 18 MPa to 0.45 MPa and was injected into the intake port of No. 1 cylinder. The amount of natural gas was changed through controlling the fuel injection duration of a SP010A CNG injector by an electronic control unit and the air fuel ratio of air-natural gas mixture was measured by an ETAS LA4-4.9 lambda meter with an accuracy of ±1.5% mounted in a separate exhaust pipe connecting to No.1 cylinder. The mass flow rate of natural gas was measured by a DMF-1 natural gas flow meter with an accuracy of ±0.2%. The compositions of natural gas used were shown in Table 1.

Exhaust gases from No.1 cylinder were sampled. In a Horiba MEXA-7100DEGR exhaust gas analyzer, total hydrocarbon (THC) was analyzed with a flame ionization detector (FID), CO was analyzed with a non-dispersive infrared analyzer (NDIR), and NOx was analyzed with a chemiluminescent detector (CLD). THC, CO and NOx emissions were the average values of the acquired data on line at each steady state operating condition.

With the aid of a fast data acquisition system, a piezoelectric transducer, a YE5850 type charge amplifier and an encoder with an angular resolution of 0.5º crank angle, the in-cylinder pressure of No.1 cylinder was acquired. Cylinder pressure used to analyze combustion processes was ensemble averaged over 100 consecutive cycles. Base on the cylinder pressure and the first law of thermodynamics, the rate of heat release can be calculated,[15] IMEP, ignition timing and combustion duration are obtained.

2.2. Test Procedures

The air inlet temperature is 18ºC and the humidity is 31% during experiments. Before test, the engine was warmed up until the temperatures of coolant and oil reached 80±1°C in order to eliminate their effect on combustion and emission characteristics. Through adjusting fuel injection timing and fuel injection pulse width, the relative air fuel ratio (lambda) of air-natural gas mixture could be altered under different engine operating conditions.

Since the natural gas injector has small nozzle orifice area, it could not inject enough fuel during intake stroke to prepare ignitable mixture at much late injection timings and 2000 rpm, only 30ºCA and 60ºCA fuel injection timings after top dead center during the intake stroke were investigated.

3. Results and Discussion

To find the optimum spark timing (MBT) for the maximum IMEP at the same lambdas and engine speed, the relationship between IMEP and spark timing in the range of 25º to 45º CA before top dead center (BTDC) was investigated with increment of 5º CA. Fig. 2 shows the variation in MBT with lambda at WOT conditions. “Inj30” and “Inj60” in Figs. 2 to 10 mean that fuel was injected into the intake port at the timing of 30ºCA and 60ºCA after top dead center (ATDC) during the intake stroke respectively. The fuel injection duration is depended on the air fuel ratios in a test condition. It is shown that, independent of fuel injection timing, MBT spark timing increases with lambda due to the decreasing flame speeds[13] and increased combustion duration at lean mixture conditions.[16] Fuel injection timing also affects MBT spark timing, which means that fuel injection timing influences early flame development. As compared to the MBT spark timings in reference,[13] the later had advanced MBT spark timings at 2000 rpm and WOT when lambda was richer than 1.2, which is attributed to its relatively low compression ratio (8.25:1) and ‘bath-tube’ combustion chamber. This indicates that MBT is affected by compression ratio and combustion chamber configuration. In the following, only combustion and emission characteristics at MBT spark timing were discussed.

The ignition timing (CA10) is indicated by the crank angle at which 10 percent mass fraction fuel is burned. In Fig. 3, CA10 is plotted against lambda at MBT spark timing under different fuel injection timing conditions. It is shown that except at lambda = 1, the earlier the fuel injection timing, the earlier the phasing of CA10 takes place at a fixed lambda. Since the MBT spark timings at 60º CA ATDC fuel injection timing are advanced or equal to those at 30º CA ATDC fuel injection timing in most cases, this reveals that flame kernel development are much slower in the former case. On the other hand, ignition and subsequent flame kernel development are essentially laminar processes occurring at a very small scale relative to fully-developed flame propagation, they are highly sensitive to reductions in laminar-flame speed and local variations in mixture preparation.[16] This indicates that the change of fuel injection timing affects the flame kernel development and alters combustion processes.

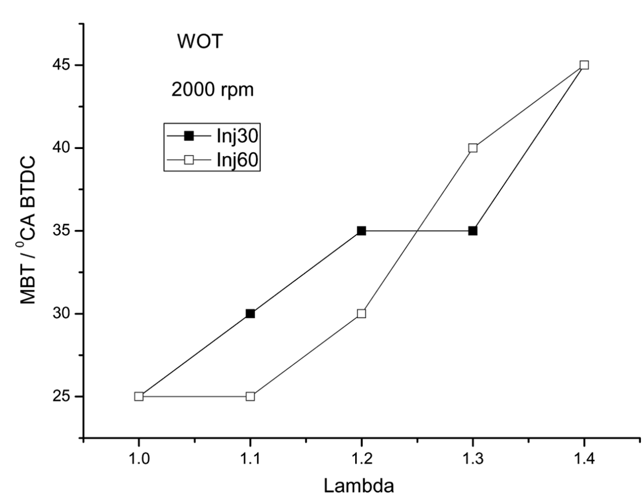


Fig. 2. MBT spark timing versus lambda under different operating conditions.

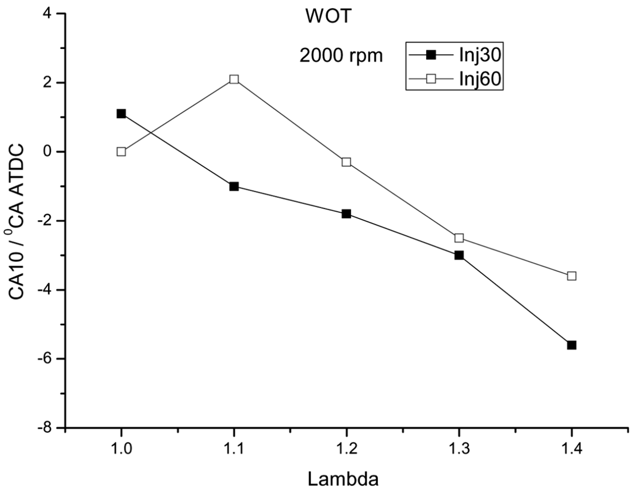


Fig. 3. CA10 versus lambda at MBT spark timing under different operating conditions.

It is also shown that, independent of fuel injection timing, CA10 advances with lambda in most cases. CA10 depends on the temperature, pressure and the composition of air fuel mixture in the cylinder.[17] As air fuel mixture becomes leaner, the spark energy transmitted to the air fuel mixture around spark plug is less and the chemical reaction in the early stage of flame development slows down although there is the presence of excess oxygen in the mixture. Hence, the time required for the initial flame development increases and ignition delays. However, when MBT spark timing is much advanced, it offsets the ignition delay caused by slow combustion velocity under lean burn conditions. As a result, CA10 occurs early at lean burn conditions.

The main combustion period (CA90) is defined as the 10-90 percent mass fraction fuel burn duration. Fig. 4 shows the relationship between CA90 and lambda under two different fuel injection timings at MBT spark timing. Independent of fuel injection timing, CA90 increases with lambda since flame propagation slows down and the time required for rapid flame development increases at lean conditions.[18] Furthermore, the earlier CA10 shown in Fig. 3 decreases burning velocity due to low temperature in the cylinder, which prolongs the combustion duration. Therefore, how to shorten combustion duration under lean conditions is important to improve the overall engine thermal efficiency.

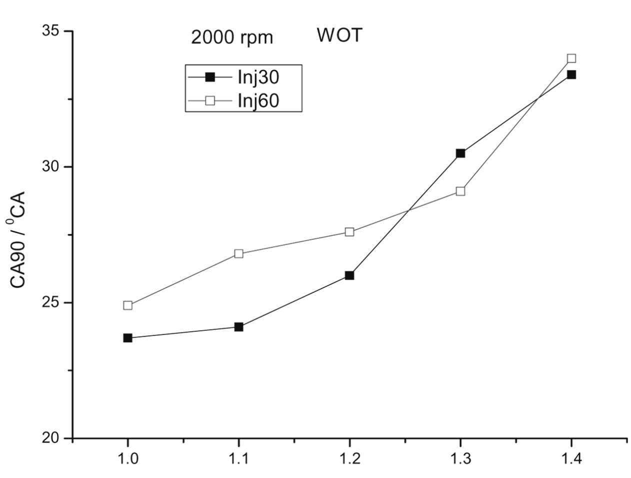


Fig. 4. CA90 versus lambda at MBT spark timing under different operating conditions.

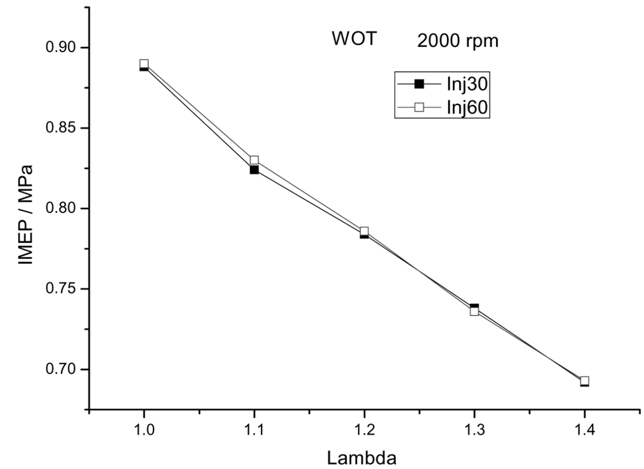


Fig. 5. IMEP versus lambda at MBT spark timing under different operating conditions.

As compared to the CA90 at the two fuel injection timings, it is found that in most cases, late fuel injection results in long combustion duration at MBT spark timings. Fuel injection timings influence the fuel distribution in the cylinder and turbulence intensity. As a result, it affects early flame development and flame propagation period.[19,20] This indicates that the change in fuel injection timing also affects combustion process and the main combustion periods at the same lambdas.

Fig. 5 shows the effect of lambda on IMEP at MBT spark timings. With the increase of lambda, the heat energy stored in the lean mixture decreases and IMEP falls. On the other hand, slow burning rate at lean conditions results in long main combustion duration and a significant fraction of energy releases well after top dead center. This increases the loss of work, the availability of heat release from the mixture worsens. As a result, IMEP also decreases.

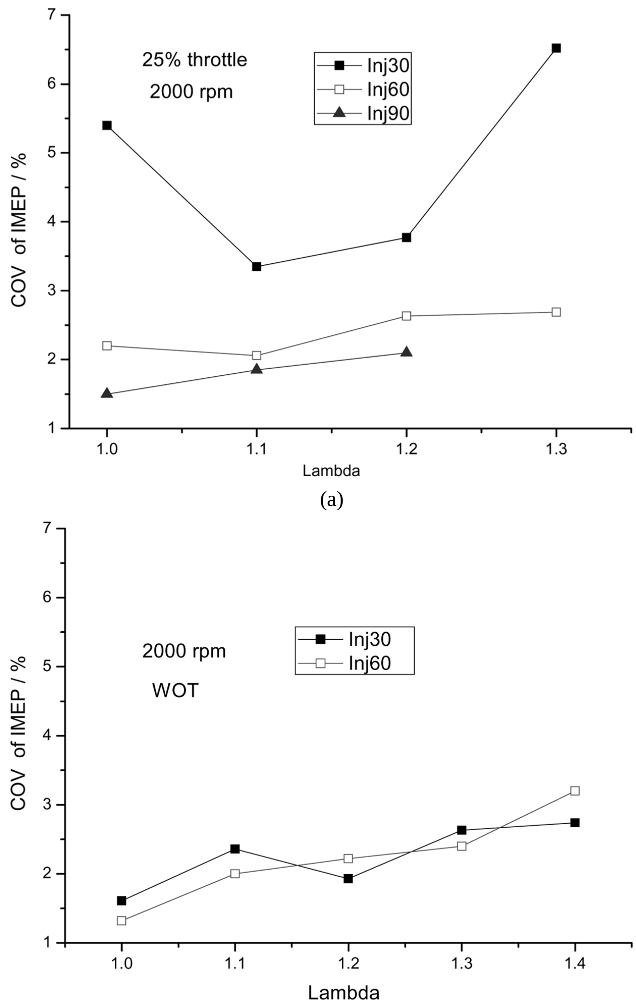
At the two injection timings, IMEP is very close especially at given lean burn conditions, which indicates that injection timings have little effect on the availability of heat energy released from the mixture.

Cyclic variability is characterized by the non-repeatability of consecutive indicator diagram of spark ignition engines. The COV of IMEP was calculated based on the following equation.

$$COV = \frac{\sigma_{IMEP}}{\overline{IMEP}} \times 100\% \qquad (1)$$

Where $\overline{IMEP}$ is the mean IMEP; σ_{IMEP} is the standard deviation in IMEP.

Fig. 6 illustrates the relationship between the COV of IMEP and lambda at MBT spark timings under different operating conditions. Residual gas fraction is high at 25% throttle, the COV of IMEP usually increases. However, at 25% throttle, the COV of IMEP decreases with delayed fuel injection timing at given



(a)

(b)

Fig. 6. COV of IMEP versus lambda at MBT spark timing under different operating conditions.

lambdas. At delayed fuel injection timing, the time available to mix air and natural gas in the cylinder is short. In particular, the mixture forming in the cylinder at the late stage of intake stroke is more heterogeneous, ignitable mixture is probably formed around the spark plug at the time of spark discharge. The effect of fuel injection timing on the COV of IMEP offsets that from residual gas. Consequently, late fuel injection timing leads to low COV of IMEP at 25% throttle. Furthermore, independent of fuel injection timing, the leaner the mixture, the less the amount of fuel in the cylinder becomes. Therefore, the COV of the IMEP increases in most cases although MBT spark timings are earlier.

At WOT, however, the COV of IMEP increases with lambda in most cases. Lower combustion temperature and slow burning rate result in slow heat release and long combustion duration under lean conditions. Therefore, non-repeatable combustion process increases with increased lambda.[15] At relatively richer conditions, there is more fuel in the cylinder, higher cylinder temperatures at the timing of spark discharge help the flame development at the early stage of combustion, which facilitates the improvement of combustion stability.[14] Furthermore, CA10 shown in Fig. 3 occur late or even after top dead center at lean conditions, which worsens the initial flame development and decreases combustion stability. However, the effect of spark ignition timings on COV of IMEP decreases when CA10 occurs before top dead center at lean burn condition.

In the case of the two fuel injection timings at WOT, the relationship between the COV of IMEP and fuel injection timing is much complicated. When injection duration increases in those conditions, the flow velocity of the intake air keeps changing, it alters the mixing process during the intake stroke. As a result, the turbulence intensity of the mixture forming at different timing is changed with fuel injection timing, cyclic variability is affected. When ignitable air fuel mixture forms in the vicinity of the spark plug at spark discharge, stable combustion occurs. It indicates that changing fuel injection timings may improve combustion stability of a port fuel natural gas engine.

Fig. 7 illustrates the engine-out THC emissions at different operating conditions. THC emissions at 60ºCA ATDC fuel injection timing are lower than those at 30ºCA ATDC fuel injection timing at a fixed lambda at MBT spark timings. Most engine-out hydrocarbon emissions are related to the storage and subsequent release of unburned mixture from the combustion chamber crevices in a spark ignition natural gas engine with good combustion quality.[21] Since the COV of IMEP is less than 5 percent at WOT, less bulk quenching and boundary layer quenching occur in the cylinder. It indicates that late fuel injection can reduce THC emissions from the crevice since the fuel has less time to contact with cylinder walls and to fill the combustion chamber crevices. On the other hand, the absolute concentration of engine- out hydrocarbons is dependent on the extent which the returned crevice gases are oxidized, which is related to the in-cylinder temperature history starting from the time of crevice gas release.[22] Late fuel injection timing causes high exhaust temperature shown in Fig. 8, which indicates that the temperatures in the cylinder are high in the late expansion stroke and exhaust stroke. It leads to increased crevice hydrocarbon oxidation rates. As a result, engine-out THC emissions are reduced.

At lean burn conditions, excess oxygen available in the cylin-

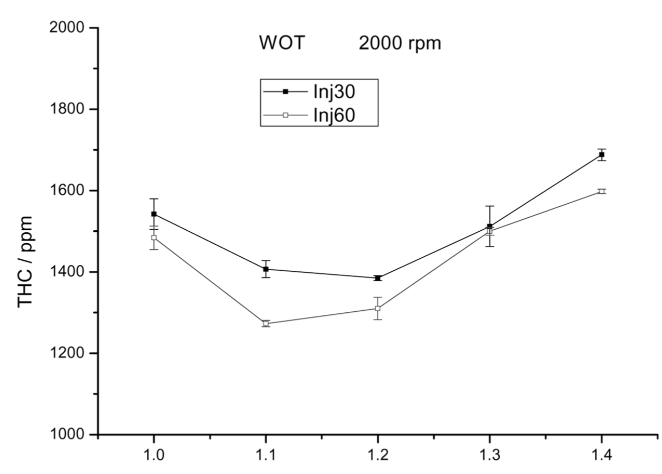


Fig. 7. Engine-out THC emissions versus lambda at MBT spark timing under different operating conditions.

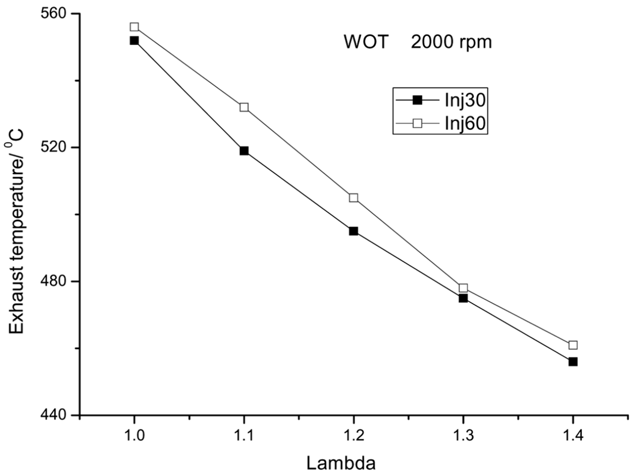


Fig. 8. Exhaust temperature versus lambda at MBT spark timing under different operating conditions.

der can improve combustion process and is beneficial to the oxidation of unburned hydrocarbon compounds.[24] Moreover, less unburned fuel is trapped in the crevice in the cylinder and is released in the late exhaust stroke, which is helpful to the reduction in THC emissions. However, slow combustion velocity, relatively high COV of IMEP and long combustion duration offset THC emissions reduction in fuel-lean conditions. It indicates that bulk quenching and boundary layer quenching are the main reasons for the increase of THC emissions at much lean conditions.

Fig. 9 shows the relationship between engine-out CO emissions and lambda under different operating conditions. At stoichiometric conditions, CO emissions are the highest due to relatively rich mixture combustion. With the increase of lambda, engine-out CO emissions significantly decrease and gradually increase again. However, engine-out CO emissions are much lower at lean conditions than those at stoichiometric conditions since there is excess oxygen available to oxidize CO in the former cases. On the other hand, the rate of CO formation is a function of the unburned gaseous fuel availability and mixture temperature, which control the rate of fuel decomposition and oxidation. The dominant oxidation reaction for CO is[15]:

$$CO + OH \leftrightarrow CO_2 + H \tag{2}$$

CO oxidation begins only relatively late in the reaction process after all of the original fuel and intermediate HC compounds have been burned.[22] Since THC emissions shown in Fig.7 increase with lambda when relative air fuel ratio is greater than 1.1, the presence of unburned hydrocarbons in the reaction zone inhibits CO oxidation according to reaction (2). This is a reason why CO emissions increase under lean conditions.

It is also shown that engine-out CO emissions at 60ºCA ATDC fuel injection timing are lower than those at 30ºCA ATDC fuel injection timing at the same lambdas. Fuel injection timing can change the mixture distribution in the cylinder and influence combustion process. It has less time to disperse fuel in the cylinder in the case of late fuel injection timing and the likelihood that fuel contacts cylinder walls is reduced. It decreases the amount of fuel burned in the late power stroke and exhaust stroke and facilitate complete combustion. Consequently, engine-out CO emissions can be decreased.

The reactions governing NOx formation are favored by long residence time, high oxygen concentration in the mixture and high charge temperature.[15] NOx emissions at MBT spark timing are plotted against lambda in Fig. 10. As expected, NOx emissions decrease with increased lambda except at lambda=1.1 and 30 ºCA ATDC fuel injection timing. Low combustion temperatures at lean burn conditions contribute to the reduction of engine-out NOx emissions. However, at lambda=1.1 and 30 ºCA ATDC fuel injection timing, the excess oxygen available in the mixture may facilitate combustion process and NOx emissions increase. Besides, the combustion duration is shorter and combustion process occurs in less volume compared to that at lambda=1. As a result, higher combustion temperatures present and offset the decrease in combustion duration to some extent, NOx emissions increase.

At the two fuel injection timings, engine-out NOx emissions are dependent on ignition timings, combustion duration and air fuel ratio. At lean air fuel ratio conditions, engine-out NOx emissions at 30ºCA ATDC fuel injection timing are higher than

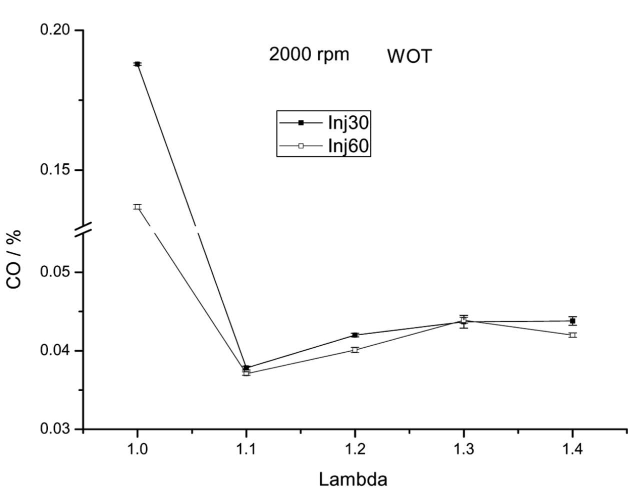


Fig. 9. Engine-out CO emissions versus lambda at MBT spark timing under different operating conditions.

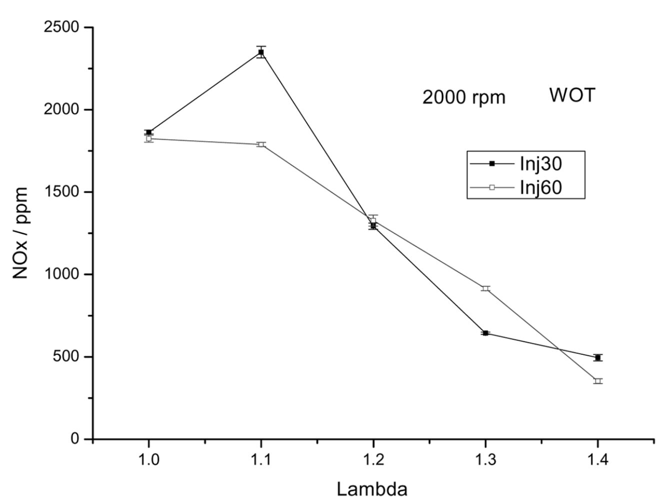


Fig. 10. Engine-out NOx emissions versus lambda at MBT spark timing under different operating conditions.

those at 60ºCA ATDC fuel injection timing when combustion duration is shorter. It indicates that the change in fuel injection timing can be a way to reduce NOx emissions in some cases.

4. Conclusions

The combustion and emission characteristics of a natural gas engine at MBT spark timings and two fuel injection timings were investigated and the main conclusions are summarized as follows:

1. Fuel injection timing influences combustion processes. MBT spark timings are affected by fuel injection timing and lambda. Lean burn conditions correspond to early MBT spark timings regardless of fuel injection timings.
2. The COV of IMEP is dependent on air fuel ratio, throttle position and fuel injection timings. In most cases, the COV of IMEP increases with lambda. At part loads, the COV of IMEP decrease with delayed fuel injection timing. However, the effect of fuel injection timing on the COV of IMEP decreases at WOT.
3. Fuel injection timing affects engine-out emissions. Late fuel injection timings can reduce CO and hydrocarbon emissions. But engine-out NOx emissions can be reduced only at late fuel injection timing conditions in some cases.

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