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A Study on Spark Ignition Natural Gas Engines

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Abstract : Natural gas is a promising alternative fuel to meet strict engine emission regulations in many countries. Natural gas engines can operate at lean burn and stoichiometric burn conditions with different combustion and emission characteristics. In this paper, the fuel economy, emissions, misfire, knock and cycle-to-cycle variations in indicated mean effective pressure of lean burn natural gas engines are highlighted. Stoichiometric burn natural gas engines are briefly reviewed. To keep the output power and torque of natural gas engines comparable to that of gasoline engines, high boosting pressure should be used. High activity catalyst for methane oxidation and lean deNOx system or three way catalyst with precisely control strategies should be developed to meet stringent emission standards.

Key words : Natural gas; Emission; Lean burn; Stratification; Turbulence; Catalyst

1. Introduction

In recent years, air quality has become a particularly severe problem in many countries. Growing concern with exhaust emissions from internal combustion engines has resulted in the implementation of strict emission regulations in many industrial areas such as the United States and Europe. In the meantime, the Kyoto protocol calls for a reduction in greenhouse gas emissions between 2008 and 2012 to the levels that are 5.2 percent below 1990 levels in 38 industrialized countries. Therefore, how to reduce hazardous emissions and greenhouse gases from engines has now become a research focus. Modern spark ignition (SI) engines with three way catalyst emit very low amounts of hazardous emissions if driven according to the certifying cycle, along with large amounts of water and carbon dioxide (CO_2) emissions.

$CO_2$is a greenhouse gas in the exhaust gases of SI engines. Improving fuel economy, using a fuel with higher hydrogen to carbon ratio (H/C) or using a renewable fuel can all reduce CO_2 emissions from engines. The fuel economy of SI engines can be improved by operating the engine with diluted mixtures through extra air or exhaust gasrecircu-

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lation (EGR) due to low temperature combustion, low heat transfer losses and low pumping losses at part loads.

Direct injection SI engines have reduced pumping losses and heat transfer losses and hence, have low fuel consumption. Homogenous charge compression ignition (HCCI) gasoline engines using diluted mixtures can also improve their fuel economy.

Natural gas (NG), which is primarily composed of methane, is regarded as one of the most promising alternative fuels, due to its interesting chemical properties with high H/C ratio and high octane research number rating (about 130). When changing the fuel from diesel to natural gas, its H/C ratio is approximately changed from 1.8 to 3.7 to 4.0. Also, natural gas has relatively wide flammability limits. The lower peak combustion temperatures under ultra lean conditions in comparison to stoichiometric conditions[1] leads to lower knock tendency of natural gas engine, allowing a higher power for the same engine displacement by increasing the boost pressure level[2]. Accordingly, NG engines using high compression ratio, lean-burn mixture or high exhaust gas recirculation would be expected to outperform gasoline engines in torque, power[3],[4], and can allow a remarkable reduction in pollutant emissions and improvement in thermal efficiency[4]. In the meantime, natural gas engines can achieve CO_2levels below those of diesel engines at the same air fuel ratio, while keeping almost the same thermal efficiency under very lean conditions[5],[6]. CO_2 emissions of natural gas engine can be reduced by more than 20% compared with gasoline engine at equal power[7]. Very low levels of NOx and carbon monoxide (CO) emissions can be achieved at lean equivalence ratios[12]. Consequently, various research projects were undertaken all over the world to convert light-duty vehicles, passenger cars, heavy-duty trucks and buses, as well as locomotive engines to use natural gas.

2. Lean burn natural gasengines

2.1 The operating envelope of lean burn engines

When flame propagation stops, misfire cycles and incomplete combustion may occur in the end zone and result in high HC emissions[8]. On the other hand, when the mixture is near the lean limit, a slight error in air-fuel ratio can drive the engine into misfire, which causes dramatic increases in exhaust emissions, engine roughness, and poor throttle response[9].

Several factors related to the lean misfire limit include in-cylinder air motion, available ignition energy, natural gas composition, the mixture temperature at ignition, residual fraction and water from humidity present in the fuel-air mixture. In addition, the effect of mixture preparation on the homogeneity of the cylinder charge is important. Fig. 1 shows the operating range under different conditions. The lambda limits for natural gas engines are based primarily on lean misfire and increased fuel consumption under lean burn conditions and on the levels of NOx and knock and increased fuel consumption under fuel-rich conditions[10]. High amount of residual gases from the previous combustion cycle limits engine combustion stability at low and medium loads due to dilution. High loads and high amount of additional air or EGR results in too low exhaust energy for sufficient boost pressure[15]. Knock is the self-ignition of the end-gas ahead of the propagating flame front. It results in lower engine efficiency, an increase of some emissions and even leads to the At part loads, the reduction in the amount of intake also reduces pumping losses due to the higher intake pressure needed. Without modifying intake ducts, reducing the intake stroke also results in lower swirl number and increased efficiency as well[5]. Adding inert gases such as N_2 and CO_2 can improve the knock rating and then enable the increase of knock limited spark timing. But CO_2 shows a twice higher effect than N_2 .

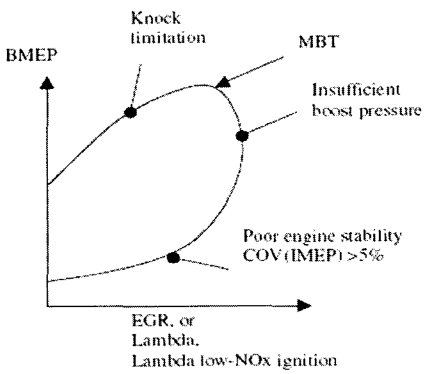

Fig. 1 Limitations in load and dilution for the engine mapping[15]

Cycle efficiency can be improved at lean burn conditions, but combustion efficiency suffers due to combustion instability and unburned mixture in the end zone. NOx andTHC emissions levels and cycle efficiency are plotted versus lambda in Fig. 2. THC emissions levels increase for lean mixtures, in particular after lambda is higher than 1.6.

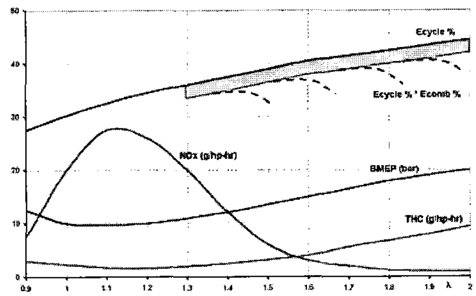

Fig. 2 Effect of lambda on natural gas engine performance[10]

The lean misfire limits of natural gas and air mixture increase with temperature, so that the engine can be run at an elevated intake manifold temperature without producing more NOx emissions. The reason destruction of engine under heavy knock operation. Knock limits the increase in load when the amount of dilution decreases[15]. Adding inert gases such as N_2 and CO_2 can improve the knock rating and then enable the increase of knock limited spark timing. But CO_2 shows a twice higher effect than N_2[11].

The lean misfire limits of natural gas and air mixture increase with temperature, so that the engine can be run at an elevated intake manifold temperature without producing more NOx emissions.

The reason is that a hot mixture is less heated by cylinder wall during intake stroke than a cool mixture, the volumetric efficiency of an engine increases with manifold temperature thus reducing the increase in manifold pressure required for the leaner and hotter mixture. Although CO and HC emissions show a slightly increasing tendency with leaner mixture at higher intake manifold temperatures, they hardly affect the absolute emission levels. It is possible to maintain specific NOx emissions at a level of 140 g/GJ in the temperature range from 25 to 80ºC by adapting only the air-fuel ratio as shown in Fig. 3. Approximately, every 10K reduction in mixture temperature requires 1% reduction in air- fuel ratio. Therefore, an engine management system is required to meet low NOx emission level under all circumstances. During a cold start, mixture enrichment is required to avoid misfire when engine is normally run at an elevated intake manifold temperature.

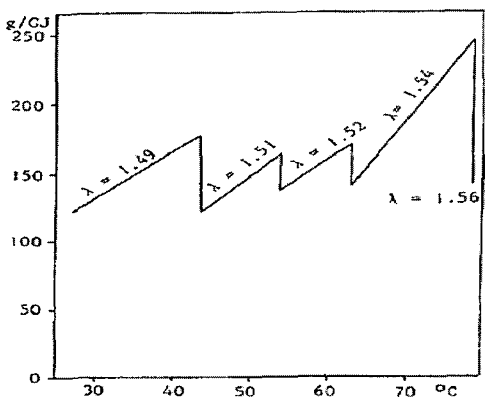

Fig. 3 NOx production for different intake-manifold temperatures with adapted air-to-fuel ratios

2.2 Cyclic variations under lean burn conditions

A particular important problem of lean-burn engines is the extent of cyclic variations in cylinder pressure due to cyclic variations in combustion process. It is generally known that as air-fuel mixture becomes leaner, the time required for the initial flame development and the period for rapid flame development increase. Two important measures of cyclic variations are the coefficient of variation (COV) in indicated mean effective pressure (IMEP) and in peak cylinder pressure. Vehicle derivability problems are usually noticeable when COV in IMEP exceeds about 10 percent. Cycle-to-cycle variations in combustion process tend to increase with increased lean mixture. The effect of equivalence ratio on the COV in peak pressure is shown in Fig. 4.

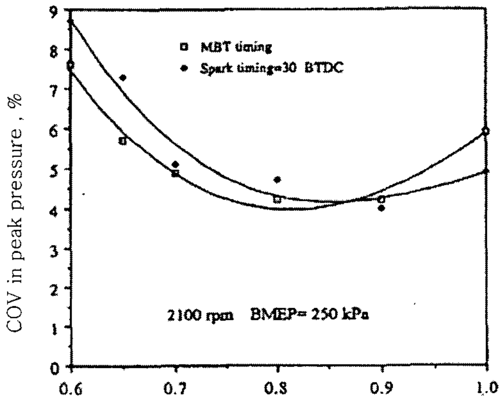

Fig. 4 COV in peak pressure equivalence ratio[15]

Generally, the effect of equivalence ratio is much more pronounced on the COV in peak pressure. When spark ignition timing is 30º CA BTDC, COV in peak pressure reaches its minimum value, 4%, at 0.9 equivalence ratio and is increased to 9% at 0.6 equivalence ratio. At the maximum brake torque (MBT) timings, engine stability is partially regained due to lower COV in peak pressure at lean burn conditions. However, the coefficient of variation in indicated mean effective pressure (IMEP) shown in Fig. 5 is less than 5% at 0.6 equivalence ratio at 30º CA BTDC spark ignition timing and is reduced to 2.3% at MBT timing, which is indication of good stability of combustion process at such lean conditions. The comparison of the COV in IMEP for a gasoline engine, mixer type NG engine and the fuel injected NG engine shown in Fig. 4 indicates that the fuel injected NG engine is much more stable due to better air-fuel ratio control[12]. Although the COV decreases with increasing loads, it is still much higher than those under the relatively rich conditions[12].

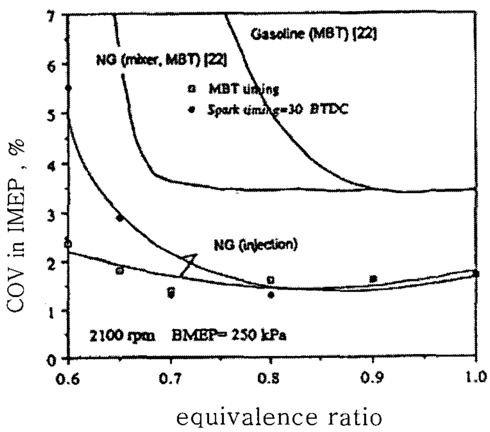

Fig. 5 COV in IMEP vs. equivalence ratio

Extreme lean mixture operation even results in the highest COV in HC emissions. Variations in the bulk flow intensity or direction lead to notable heat loss or flame front quenching due to contact with electrodes and the chamber wall, and hence result in significant cyclic variability[4]. Therefore, the flow at spark plugs to stabilize ignition and support flame initiation, and the flow in the combustion chamber to favor flame propagation should be taken into account [6]. Higher cycle-to-cycle variability and poor combustion phasing set a limit on the use of very lean or high EGR mixtures[4].

2.3 Spark ignition timings

The MBT timings should take into account the reduction of flame speed and the increase ofcombustion duration under lean conditions. The MBT spark timing for natural gas is advanced between 2 and 10º crank angle more than that for gasoline. The spark timings under lean conditions require to be advanced compared to stoichiometric operation, which may then lead to knock. Advancing the spark timing is also an effective way to extend lean limits while keeping the efficiency high. Since MBT spark timings are based on average conditions, some cycles may be slower or faster than the average cycle.At extreme lean operating conditions, where the flame development time and the rapid burn periods are long, cyclic variations effectively limit engine operation. More spark advance and ignition energy are required. As a result, the optimum air-fuel ratio is related to the ignition energy and spark timing, an adaptive control system which simultaneously controls the air-fuel ratio and the ignition timing would be well suited for such an applica- tion.

Fig. 6 shows the relationship between MBT timings and equivalence ratio. As the mixture becomes leaner, MBT timings advance. When the engine load is increased, the increased cylinder pressure and temperature help to promote flame propagation. Thereby, MBT timings need somewhat lesser advance than the others. Injection positions in the intake port also affect the mixture formation in the cylinder and therefore influence the MBT ignition timings. MBT ignition timings obtained from a natural gas engine with two intake ports. The differences of the MBT ignition timings between region A and region B, and between region C and region D arequite small, which suggests that the combustion histories after the flame-development periods may change. The steady state simulation results showed that in the case of region B, richer mixture was mainly distributed around the outer portion of the cylinder. And MBT was alsoretarded the most. As a result, the mixture could not be burned completely due to insufficient time for combustion and unfavorable spatial mixture distribution, and led to the worst efficiency at all conditions. The highest thermal efficiency was obtained in the case of region A because the combustion events occurred around the cylinder center due to the lower cooling loss caused by effective charge stratification and stable ignition.

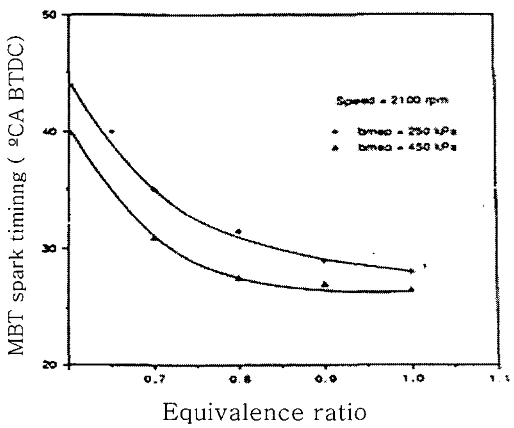

Fig. 6 MBT spark timings at different equivalence ratios[16]

3. Problems needed to be solved

There are several major problems when using lean burn natural gas engines. First, the setpoint for the best compromise between emissions and economy is not clear although wide-range exhaust gas oxygen sensors have recently become available. Second, even if this setpoint was known for a given fuel and operating condition, the optimum air-fuel ratio changes with both operating conditions and fuel properties[14].

Furthermore, HC emissions from natural gas engines are mainly composed of methane. This would not be necessary if standard emissions regulations would impose different limits for methane and nonmethane hydrocarbons[17]. However, methane is the most difficult hydrocarbon to be oxidized, the large methane fraction of the HC emissions from NG engines could be a problem for three-way catalysts. Therefore, in order to minimize HC emissions, some management strategies such as the optimization of the mixture formation and the internal flow characteristics is necessary, which may lead to fuel consumption increasing.

Natural gas as a vehicle fuel has low energy density and exists in a gaseous state in the intake manifold. The lack of latent heat of evaporation of natural gas decreases volumetric efficiency by about 3% compared to sequential injection gasoline engines[13]. In fact, these performance losses can be recovered in part by increasing the compression ratio since natural gas engines can be safely operated at compression ratios approaching 15:1 provided gas quality is maintained. Intake and exhaust valves with high lift can also be used to increase torque and power[18].

5. Conclusions

1. Lean burn is an effective way to improve fuel efficiency and reduce NOx emissions. Lean burn limits are dependent on the configuration of combustion chamber, ignition timings, ignition energy and turbulence. Cycle-by-cycle variation in indicated mean effective pressure should be controlled to operate natural gas engines under lean burn conditions. To enhance the power density of natural gas engines, turbocharging technology should be used. To meet stringent emission regulations, lean burn engines need a rather complex deNOx system like selective catalytic reduction method.
2. Stoichiometric natural gas engines equipped with three-way catalyst can meet future stringent emission regulations. But it needs precisely air-fuel ratio control strategies and highly efficient catalyst to oxidize methane. To get better fuel economy at pure stoichiometric SI operation conditions, the addition of EGR to a stoichiometric mixture is one way.
3. Brake mean effective pressure of natural gas engines are limited by knocking and thermal loading. EGR can improve knock limit by reducing exhaust temperature. The combination of Miller cycle and EGR is effective in increasing BMEP of natural gas engines.

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