Spark ignition natural gas engines—A review

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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 conditions with different combustion and emission characteristics. In this paper, the operating envelope, fuel economy, emissions, cycle-to-cycle variations in indicated mean effective pressure and strategies to achieve stable combustion of lean burn natural gas engines are highlighted. Stoichiometric natural gas engines are briefly reviewed. To keep the output power and torque of natural gas engines comparable to those of their gasoline or Diesel counterparts, high boost pressure should be used. High activity catalyst for methane oxidation and lean deNOx system or three way catalyst with precise air–fuel ratio control strategies should be developed to meet future stringent emission standards.

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CO2 is a greenhouse gas in the exhaust gases from SI engines. Improving fuel economy, using a fuel with higher hydrogen to carbon ratio (H/C) or using a renewable fuel can all reduce CO2 emissions from engines. The fuel economy of SI engines can be improved by operating the engine with diluted mixtures through extra air or exhaust gas recirculation (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 research octane number (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 ...

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% below 1990 levels in 38 industrialized countries. Therefore, how to reduce hazardous emissions and greenhouse gases from engines has now become a research focus. If driven according to the certifying cycle, modern spark ignition (SI) engines with three way catalyst emit very low amounts of hazardous emissions, along with large amounts of water and carbon dioxide (CO2) emissions.

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lean conditions in comparison to stoichiometric conditions [1] lead to a lower knock tendency of natural gas engines, 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 an improvement in thermal efficiency [4]. In the meantime, natural gas engines can achieve CO₂ levels below those of Diesel engines at the same air–fuel ratio, while keeping almost the same thermal efficiency under very lean conditions [5,6]. CO₂ emissions of natural gas engines can be reduced by more than 20% compared with gasoline engines at equal power [7].

In addition, particulate matter (PM) and nitrogen oxides (NOₓ) emissions have serious health and environmental implications when present in high enough concentrations, causing and exacerbating human respiratory illnesses such as asthma [8]. NG engines produce lower PM than Diesel engines do [9], since natural gas does not contain aromatic compounds such as benzene and contains less dissolved impurities (e.g., sulphur compounds) than petroleum fuels do [5]. A relatively low flame speed and low temperature combustion of NG engines [10] help to mitigate engine out NOₓ emissions when operating with high compression ratio or when the engine is supercharged [11]. Very low levels of NOₓ and carbon monoxide (CO) emissions can be achieved at lean equivalence ratios [12]. Moreover, engine out unburned hydrocarbons (HC) emissions can also be reduced below the corresponding levels for gasoline engines, since the gaseous state of compressed natural gas (CNG) avoids wall wetting effects on intake manifold and cylinder liner, especially at cold start conditions, which improves cold startability of CNG engines [13] and induces fuel consumption savings. The smaller percentage of HC emissions from oil film adsorption–desorption phenomena also contributes to the reduction of engine out HC emissions compared to those of gasoline base line engines [7,14]. Low density and high dispersal rates also are advantages when safety is considered, since an explosive mixture is unlikely to form in the event of a leak [11].

Even if natural gas cannot prove itself as an intrinsically better fuel than gasoline and Diesel fuels in terms of engine emissions, natural gas vehicles that operate on CNG fuel are expected to find widespread use because the sources of natural gas are far bigger than those of oil, and natural gas will be available at a competitive cost for a long time [15]. Consequently, various research projects have been 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. The operating envelope of lean burn engines

Low emission spark ignition CNG engines can be achieved by the lean burn engine and the stoichiometric engine with a three way catalyst. In a lean burn natural gas engine, air–fuel ratio is extremely critical. As the mixture is leaned out beyond a critical point to suppress NOₓ emissions, the burning rate in the lean conditions is reduced compared to that under stoichiometric conditions, which results in an increase in the overall combustion duration and, in turn, leads to increased heat transfer losses to the cylinder walls and decreased thermal efficiency [7]. Slow flame propagation velocity results in slow heat release, which causes combustion instability with high cyclic variations [16]. Hence, the efficiency gain of lean operation cannot be realized. When flame propagation stops, misfire cycles and incomplete combustion may occur in the end zone and result in high HC emissions [21]. On the other hand, a slight error in air–fuel ratio near the lean limits can drive the engine into misfire, which causes dramatic increases in exhaust emissions, engine roughness and poor throttle response [22].

Several factors related to lean misfire limits include incylinder air motion, available ignition energy, natural gas composition, mixture temperature at ignition, residual fraction and water from humidity in the air–fuel 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 [19]. With too much excess air, the combustion temperature in the cylinder drops, which results in an increase in engine out CO and HC emissions due to slower flame initiation and propagation. For extremely lean mixtures, flame quenching, partial burn and combustion instabilities could be expected [7,23]. High amounts of residual gases from the previous combustion cycle also limit engine combustion stability at low and medium loads due to dilution. High loads and high amounts 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 effi-

![Fig. 1. Limitations in load and dilution for the engine mapping [15].](Image)
efficiency, an increase of some emissions and even leads to destruction of the engine under heavy knock operation. Knock limits the increase in load when the amount of dilution decreases [15]. The optimization of valve duration and timing is an effective solution for controlling the knock limits and for improving thermal efficiency. Early closure of inlet valves reduces the effective compression ratio and, thus, decreases knock sensitivity by reducing the in-cylinder mixture temperature, while the expansion ratio is kept high, giving high efficiency. 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 N₂ and CO₂ can improve the knock rating and then enable the advance of knock limited spark timing. However, CO₂ shows a twice higher effect than N₂ [24]. On the other hand, EGR allows a reduction in knocking tendency and can improve engine efficiency through increased compression ratio [18]. The combination of the Miller cycle and EGR improves brake mean effective pressure (BMEP) to 1.7 MPa for the stoichiometric condition [25].

Cycle efficiency (\(\eta_{\text{cycle}}\)) can be improved at lean burn conditions, but combustion efficiency (\(\eta_{\text{comb}}\)) suffers due to combustion instability and unburned mixture in the end zone. NOx and THC (total hydrocarbon) 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 [21]. The lean misfire limits of natural gas and air mixtures 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 the cylinder wall during the 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 80 °C by adapting only the air–fuel ratio shown in Fig. 3. Approximately, every 10 °C reduction in mixture temperature requires 1% reduction in air–fuel ratio. Therefore, an engine management system is required to meet low NOX emission levels under all circumstances. During a cold start, mixture enrichment is required to avoid misfire when the engine is normally run at an elevated intake manifold temperature [26].

3. Lean burn natural gas engines

Although, the fuel economy of a lean burn natural gas engine is not as high as a Diesel engine, it is higher than that of a stoichiometric engine due to the increase of specific heat ratio [16–20]. The thermal efficiency of natural gas engines is dependent largely on lambda, compression ratio, burning rate and NOX emission levels. The optimum value of lambda for maximizing the trade-off between specific fuel consumption and specific NOX emissions strongly depends on the type and quality of ignition and combustion. At part loads, the use of a throttle for lambda control causes a significant deterioration in thermal efficiency. However, to a certain extent, engine load under lean burn conditions can be controlled without throttle through varying the air–fuel ratio. In turn, volumetric efficiency at part loads can be increased due to reduced pumping losses. Furthermore, combustion temperatures in lean burn conditions decrease and, therefore, lead to reduced NOX emissions. To keep NOX emissions low and obtain the best fuel economy, NG engines are often operated close to lean misfire limits. Because of an excess of oxygen available in the exhaust, an oxidation catalyst can be applied to oxidize CO and THC emissions and meet stringent CO and THC
emissions limits [21]. On the other hand, the use of skip fire at part loads can reduce or eliminate the effect of throttling loss [19], which may improve fuel economy.

3.1. 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 the combustion process. It is well known that as the air–fuel mixture becomes leaner, the time required for the initial flame development and the period for rapid flame development increase. Charge dilution also affects flame propagation speed. Slow flame propagation velocity caused by a very lean mixture results in slow heat release, low combustion quality [21], higher cycle-to-cycle variations and poor combustion phasing [4]. Two important measures of cyclic variations are the coefficient of variation (COV) in indicated mean effective pressure (IMEP) and in peak cylinder pressure. Vehicle drivability problems are usually noticeable when the COV in IMEP exceeds about 10% [27].

Cycle-to-cycle variations in the combustion process tend to increase with increasingly lean mixture. The effect of equivalence ratio on the COV in peak pressure is shown in Fig. 4. Generally, the effect of equivalence ratio is much more pronounced on the COV in peak pressure. When spark ignition timing is 30° CA BTDC, COV in peak pressure reaches the minimum value, 4%, at 0.9 equivalence ratio and is increased to about 9% at 0.6 equivalence ratio. At the maximum brake torque (MBT) timings, combustion stability is partially regained due to lower COV in peak pressure at lean burn conditions. However, at the same operating condition, the COV in IMEP shown in Fig. 5 is less than 5% at 0.6 equivalence ratio at 30° CA BTDC spark ignition timing and is reduced to 2.3% at the MBT timing, which is an indication of good stability of combustion process at such lean conditions [28]. The comparison of the COV in IMEP for a gasoline engine, mixer type NG engine and port fuel injection NG engine shown in Fig. 5 indicates that the port fuel injection NG engine is much more stable due to better air–fuel ratio control [12,28]. Although, the COV decreases with increasing loads, it is still much higher than those under relatively rich conditions [12]. Consequently, consideration should be taken to ensure minimal cylinder to cylinder deviation in fueling when natural gas engines are developed [29].

Cyclic variations in combustion also result in variations in CO, HC and NOx emissions. The relative variations in HC and NOx levels are higher at very lean conditions.

Besides the fluctuations in local and cycle-to-cycle mixture, variations in flow motion appear to be the major cause of cyclic variations in combustion. Flow variation includes local turbulence and bulk flow according to its effect on the combustion process. The change in local turbulence intensity and scales from one cycle to another affects both flame growth rate and the rate of early turbulent flame development, which alters the combustion process significantly. 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]. The wall contact area of the small flame is of major importance for the flame speed in the flame development period due to the change of heat loss to the spark plug electrodes related to mean flow direction and magnitude [23]. Higher cycle-to-cycle variability and poor combustion phasing also set a limit on the use of very lean or high EGR mixtures [4]. Accordingly, the control of mixture formation is required in lean burn natural gas engines [6].
3.2. Achievement of stable combustion at lean burn conditions

It is well known that the mixture of methane and air has low burning velocity compared with that of gasoline and air [30,31]. Homogeneous lean burn mixtures result in lower flame propagation, occurrence of misfire, low mixture distribution quality in multi-cylinder engines and high unburned HC emissions in the exhaust. To implement successfully a lean burn strategy to natural gas engines with minimum exhaust emissions and maximum thermal efficiency, high compression ratio, high energy ignition system, increased swirl and turbulence at the end of the compression stroke, which leads to fast burning rates and, hence, increase in-cylinder turbulence at the end of the compression and catalytic coating on the combustion chamber can be used [20]. Controlling the fluid flow inside the combustion chamber is necessary to promote combustion under all operating conditions [32,33].

Increasing the organized charge motion in the cylinder leads to increased turbulence levels [34]. High levels of charge turbulence in the combustion chamber profoundly speeds up the combustion process in SI engines [35,36], reduces cyclic variability [34] and compensates for the decreased combustion quality under lean burn conditions [16,37], which can improve thermal efficiency.

The level of turbulence in the chamber just prior to ignition and during the combustion process has an important impact on the burning rate of air-fuel mixture. The turbulence intensity in the chamber is influenced by the design of combustion chamber, intake ducts and valves lift laws through the degree of swirl imparted to the mixture during the intake process and by the squish motion generated as the piston nears top dead center.

Various turbulence enhancing combustion chamber geometries, in particular, the pistons with unusual bowl shapes to generate squish flows or enhance the effects of swirl, have been widely investigated [38–41]. Johansson et al. found that combustion chambers with a square cross section in the piston bowl have the fastest combustion and, hence, late ignition timing for MBT. The flat chamber without squish is clearly the slowest with twice as long duration for the combustion [23]. Using a pre-chamber can ignite very lean natural gas mixture and eliminate the negative pumping loop of throttle controlled carbureted engine [20]. In the meantime, the increased turbulence produced by the prechamber has strong effect on power output at the same air-fuel ratio mixture. Compared to the conventional gas engine, stable ignition and quick flame propagation are greatly improved due to the prechamber, which leads to high burning rates and short burning period in the main combustion chamber [42].

The piston squish area has a direct effect on in-cylinder air motion. Increases in squish area can give a high level of shear and, hence, increase in-cylinder turbulence at the end of compression stroke, which leads to fast burning rates and an average 1.5% reduction in brake specific fuel consumption (BSFC), or 1.5% increase in power output under wide open throttle conditions in comparison with the slowest burning cases [33].

However, specific emission levels are generally higher for a fast burning combustion chamber. In some cases, a combustion chamber with the highest turbulence level causes the highest emission levels [33]. On the other hand, the exhaust gases are colder due to the increase in effective expansion ratio. Even if a fast combustion chamber is more tolerant to diluted mixtures, the cold exhaust gases result in too low boost pressure at highly diluted conditions [15].

The different forms of turbulence affect the combustion process quite differently [43]. Very high turbulence levels may extinguish the flame completely, particularly with very lean or high EGR mixture [44] while the same high turbulence levels located away from the spark might enhance the flame propagation [33]. It is important, therefore, to control the fluid velocity at the spark plug location because too high turbulence levels can attenuate the rapid early flame kernel development. It is desirable to control engine turbulence to enhance its positive effects and minimize its negative effects, thus optimizing combustion of very lean or high EGR mixtures.

Increased charge motion can be obtained through the use of intake ports to produce swirl or tumble, or through the use of shrouded intake valves to produce swirl. By using tumble enhancing piston crown geometry and optimizing the shape of the swirl control valve as shown in Fig. 6, stable combustion can be achieved at an excess air ratio of approximately 1.6 shown in Fig. 7, and fuel economy is improved approximately 10% compared with a stoichiometric condition at MBT [7].

The results obtained from an engine with a straight intake port and a swirl intake port show that in-cylinder fuel distributions in the swirl plane can be controlled by where the fuel is injected into the straight intake port, since they affect the ignitability of the in-cylinder charge. Stable ignition at extended lean limits can be realized if the fuel gas were injected into regions A and B in Fig. 8 because a rich region beneath an intake valve can be transported.

![Fig. 6. Intake gas flow [7].](image-url)
to the vicinity of the spark plug at spark ignition timing [45].

To further extend the lean limit of natural gas engines, it is possible partially to stratify the air–fuel charge such that the mixture in the region of the spark plug is richer than the surrounding ultra lean homogeneous mixture [46,47]. Additional fuel injection to produce a rich mixture near the spark plug has been shown successfully [47–50].

Reynolds et al. developed a partially stratified charge (PSC) system shown in Fig. 9 to improve the lean burn combustion process on a natural gas engine. High pressure natural gas (20 bar) was supplied to the spark plug injector. A fast response solenoid valve was used to control the injection timing and mass flow. A capillary tube with 0.53 mm internal diameter directed the injected fuel to the custom made spark plug injector and a check valve installed close to the plug to prevent back flow of combustion gases. Lean burn natural gas engines can utilize either pre-mixed or stratified charge combustion strategies. Without PSC, the COV in IMEP is higher than 10% at lambda = 1.66, the engine stability is unacceptable. However, the partial stratification through pure fuel injection increases diffusion combustion and reduces the COV in IMEP to below 5%, which is similar to a Diesel engine. However, at a given air–fuel ratio, partial stratification increases NO\(_x\) emissions due to higher in-cylinder temperature and pressure. The magnitude of NO\(_x\) increase is highly dependent on the quantity of partial stratification. Higher PSC flow rates correspond to increased NO\(_x\) emissions as shown in Fig. 10. Although

![Fig. 7. Lean burn characteristics under partial load condition [7].](image)

![Fig. 8. Four regions of intake passage [45].](image)

![Fig. 9. The PSC spark plug injector: (1) metal body of plug; (2) insulator; (3) capillary tube and (4) slot in threads connects to delivery hole (5) [11].](image)

![Fig. 10. NO\(_x\) emissions versus injection timing (1500 rpm, wide open throttle, lambda = 1.65, ignition at 50° CA BTDC) [51].](image)
PSC does tend to increase NOx levels, engine out NOx emissions are low, due to earlier heat release and improved combustion. Retarding spark can reduce NOx without performance penalties [11]. When the engine operated at lambda of 1.66, pure fuel PSC increased gross indicated mean effective pressure by approximately 15%, and improved brake specific fuel consumption by a corresponding amount [11]. The highest heat release rate and, hence, the fastest combustion event were observed when pure fuel was micro-injected. A progressive improvement in engine performance was observed as the micro-injected charge was made richer, but premixed air–fuel charges were less effective than pure fuel injection [51]. In the meantime, THC emissions decrease with increasing air–fuel ratio due to decreased unburned fuel trapped in crevices. Also, the excess air in the combustion chamber makes the post oxidation of the crevice gas more effective during the expansion and exhaust strokes. When the mixture is very lean, however, THC emissions tend to increase due to very low flame propagation, incomplete combustion and misfire.

Different from conventional high energy ignition systems, laser induced sparks have high surface area with peak temperature and pressure on the order of 10⁵ K and 10⁷ atm, respectively. When a laser ignition system is used, the misfire limit can be significantly extended and the total operating envelope is increased by 46%, and half the minimum level of NOx emissions using the conventional spark ignition system can be achieved with no appreciable degradation in thermal efficiency while hydrocarbon emissions are comparable. However, early ignition using a laser ignition system results in a slight decrease in the knock limit [52]. In addition, at low load and low speed, two spark plugs per cylinder help to stabilize combustion at delayed spark advance to light off catalyst during cold condition [14].

3.3. Spark ignition timings

The MBT timing should take into account the reduction of flame speed and the increase of combustion duration under lean conditions. Usually, MBT spark advance varies with the composition of natural gas and air–fuel ratio for low fuel consumption and emissions. If ignition timing were overly retarded in a lean burn engine, the mixture temperature in the end zone could drop below the misfire temperature due to expansion and quenching. When the misfire margin approaches zero, occasional misfire and poor combustion quality with increased THC emissions occur at lean burn, late ignition timing, and low mixture temperature conditions. Therefore, to provide adequate combustion quality and stable combustion, using lower lambda and advance timing at part load is necessary as compared to the lean lambda at full load [21]. The MBT spark timing for natural gas is advanced between 2° and 10° crank angle more than that for gasoline [53]. The spark timing under lean conditions must be advanced compared to stoichiometric operation, which may then lead to knock. In the meantime, advancing the spark timing can effectively extend lean limits while keeping thermal 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 inevitably 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 that simultaneously controls the air–fuel ratio and the ignition timing would be well suited for such an application [22].

Fig. 11 [28] shows the relationship between MBT timings and equivalence ratios. 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, hence, influence the MBT ignition timings. Fig. 12 shows the MBT ignition timings obtained from a natural gas engine with two intake ports shown in Fig. 8. The differences of the MBT ignition timings between region A and region B and between region C and region D are quite small, which suggests that the combustion histories after the flame development periods may be different. 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 also retarded 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

![Fig. 11. MBT spark timings at different equivalence ratios [28.]](image-url)
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 [45].

Using a lean burn strategy with a carefully optimized combustion chamber geometry, ignition system, spark ignition timings and oxidation catalytic converter, natural gas engines can meet the increasingly stringent exhaust emissions targets, for example, the proposed US EPA 2007 regulation to reduce NO\textsubscript{x} and PM for heavy duty truck and bus engines [54].

4. Stoichiometric natural gas engines

Compared to SI gasoline engines, engine out NO\textsubscript{x} emissions for stoichiometric natural gas engines with early spark timing are lower due to lower combustion velocity while engine out THC emissions stay low at full loads [14]. However, the thermal efficiency of stoichiometric natural gas engines is lower than that at lean conditions. One way to get better fuel economy than pure stoichiometric operation is to use EGR.

Usually, the flame conditions during the early flame development are nearly laminar due to small size of the flame compared to the integral length of turbulence. When EGR is applied to a natural gas engine, the laminar flame speed is more reduced compared to lean operation, due to lower oxygen concentration in the mixture. Since EGR has a strong influence on the early combustion, combustion duration increases with increased EGR. Therefore, a fast combustion chamber is required under the operating conditions with high EGR rates.

Engine out HC emissions increase with the increase of EGR since the modes with the highest amounts of EGR have slower and colder combustion, but HC emissions decrease with increased BMEP as shown in Fig. 13 in stoichiometric conditions.

Stoichiometric natural gas engine with EGR and three way catalyst is a preferred strategy to meet future stringent emission regulations [55]. Einewall et al. found that NO\textsubscript{x} emissions can be reduced by 99.9% and HC emissions by 90–97% by operating the stoichiometric engine with EGR and a three way catalyst compared to the lean high efficiency strategy [15].

When three way catalysts with higher catalytic conversion are used, air–fuel ratio should be precisely controlled. A sequential multi-port injection system (MPI) can help to precisely control air–fuel ratio as closely as possible to the stoichiometric air–fuel ratio under any engine operating conditions by utilizing air–fuel sensor, heated oxygen sensor and fuel injection control. For the exhaust system fitted with a close coupled converter and an under floor three way catalytic converter, in order to achieve a higher catalytic conversion efficiency, an air–fuel ratio sensor on the upstream of a close coupled converter can be used to control the mixture at stoichiometric air–fuel ratio, and a heated oxygen sensor on the downstream of an under floor converter can be employed to compensate precisely the air–fuel ratio. If the air–fuel ratio is out of the lean side of the window at all, NO\textsubscript{x} conversion efficiency sharply decreases. Therefore, the air–fuel ratio is controlled to shift toward the slightly rich side. Ignition timing retard is applied after engine start in open loop mode in order to increase exhaust temperature and to shorten catalyst light off times [13].

The basic demands of natural gas engines in power generation, cogeneration and vehicle markets are low exhaust emissions (especially NO\textsubscript{x}), high brake thermal efficiency, low fuel consumption, high power density, low investment and low maintenance cost [56]. The lean burn natural gas engines can achieve high thermal efficiency and moderately low NO\textsubscript{x} emission without deNO\textsubscript{x} system. To meet stringent emission regulations, lean burn engines need a rather complex deNO\textsubscript{x} system like selective catalytic reduction (SCR) method, which requires additional reductants. This potentially favors the stoichiometric engine equipped with three way catalyst, but the stoichiometric engine with three way catalyst shows rather inferior performance to the lean...
burn engine in the form of lower BMEP and lower brake thermal efficiency. It is, therefore, important to develop innovative methods to achieve a high BMEP, low specific fuel consumption stoichiometric engine, with performance competitive to that of Diesel engines [57].

To meet the demands of increased efficiency and reduced exhaust emissions, future clean internal combustion engine technology may combine some of the best features found in different types of conventional technologies. The operating mode transition from lean burn to stoichiometric burn spark ignition engines or from SI engines to HCCI engines may be selected, depending on speed, load and emissions requirements.

5. Problems needed to be solved

There are several major problems when using lean burn natural gas engines. First, the set point for the best compromise between emissions and fuel economy is not clear, although wide range exhaust gas oxygen sensors have recently become available. Second, even if this set point is known for a given fuel and operating condition, the optimum air–fuel ratio changes with both operating conditions and fuel properties [22]. Third, the exhaust temperatures of natural gas engines operating in lean burn conditions are below 750 K at most operating conditions, comparable to the base Diesel engines [12]. The lower exhaust temperatures increase the difficulties in methane oxidation and result in low THC conversion efficiency. If catalyst inlet temperatures were always very high (>550 °C), low THC conversion could be attributed to the increase of space velocity (ratio of exhaust gas flow rate to catalyst volume), since the increase of space velocity decreases the contact time between the reactants and the catalyst [58]. 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 non-methane hydrocarbons [14]. 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 optimization of the mixture formation and the internal flow characteristics is necessary, which may lead to fuel consumption increasing. In addition, the lack of NOx emission catalytic control in lean conditions would make problematic the compliance with future stricter emission limits [58].

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]. Unless a direct injection fuelling approach is adopted, the volumetric efficiency of a port fuel injection (PFI) engine is reduced by between 10% and 15%, depending on engine optimizations [11,14]. Hence, a PFI CNG engine will suffer power output losses over 10% compared to the same size gasoline engine due to the displacement of the air by the gas fuel [59]. 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 [59]. Intake and exhaust valves with high lift can also be used to increase torque and power [13]. In order to offset power density losses due to lean operation and keep output power and torque comparable to those of a gasoline or Diesel engine, high turbocharger boost is often used to increase the amount of air in the cylinder [60]. However, lean burn at high pressure conditions requires significant increased ignition energy to ignite mixtures when conventional spark plug technology is used, which significantly reduces the service life of conventional spark plugs. The achievable boosting pressure and lean burn limit is, therefore, primarily limited by the spark plug. Shortened and unpredictable spark plug life causes increasing maintenance cost [56]. Hence, durable spark plugs are needed to be developed for high efficiency lean burn natural gas engines.

CNG lacks the latent heat of evaporation to cool intake charge. Accordingly, the temperatures of piston, cylinder wall, valve and valve seat increase. Figs. 14 and 15 show the results obtained from a four cylinder 2.2 L, 4 valve CNG engine modified from a Camry gasoline engine at wide open throttle (WOT). Because of the absence of the liquid fuel spray acting as a cooling agent or lubrication property for CNG engines like that of gasoline, intake valves run hotter, which results in increased intake valve and seat wear rates. When run on gaseous fuels, standard gasoline exhaust valves are also subject to increased recession. Exhaust valve and seat materials should be developed for natural gas engines [58]. Intake valves are also subject to increasing deposit build up due to the lack of liquid fuel spray with detergent additive as a cleaner. This means that assurance of reliability against thermal loads and wear of sliding parts is a serious issue. The durability and reliability of a new CNG engine can be achieved and even improved,
mainly by selecting suitable materials and surface treatment methods for the valves, valve seats, spark plugs and other parts [7]. The self lubrication of exhaust valves and valve seats can improve wear [13]. Additionally, high strength pistons made of heat resistant material and modified water jacket design in the cylinder head can be employed to reduce deformation. The piston pin boss in a taper shape and the application of surface treatments to piston top surface and skirt can also be utilized to improve seizure resistance [13].

From the fuel storage standpoint, at the same energy content, CNG at 20.7 MPa occupies about five times the volume of Diesel fuel and four times the volume of petrol, which poses a significant problem in limiting the driving range in vehicular applications [16].

6. Conclusions

1. Lean burn is an effective way to improve fuel efficiency and reduce NOx emissions. Lean burn limits are dependent on combustion chamber geometry, ignition timings, ignition energy and turbulence. Cycle-to-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 precise 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 option.

3. Brake mean effective pressures of natural gas engines are limited by knocking and thermal loading. EGR can improve knock limit by reducing combustion temperature. The combination of Miller cycle and EGR is effective in increasing BMEP of natural gas engines.

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References


