Natural Gas Fired Reciprocating Engines for Power Generation: Concerns and Recent Advances

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Additional information is available at the end of the chapter

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1. Introduction

Concerns about the dwindling US energy deposits, uncertainties about the global energy supply chains, human health concerns, Green House Gas emissions and other environmental concerns have brought energy efficiency and clean technologies to the forefront. The energy supply matrix for the United States (see Figure 1) is a diverse mix - fossil fuels such as coal, natural gas, and petroleum are interspersed with other traditional and non-traditional resources. Out of these, due to its availability in abundance, easy transportation through pipelines, and clean burning nature, natural gas features prominently with total annual consumption of 22.5 Quads (1 Quad = 1 x 10^{15} BTU), i.e., 23.2% of the total US energy consumption. Approximately a quarter of this natural gas is used for electricity generation by peaking power plants and various distributed generation centers spread throughout the country. Additionally due to the fact that high efficiency (as high as ~ 80%) can be achieved via. Combined Heat and Power (CHP), the fraction of natural gas used in Distributed Generation continues to increase.

For centralized power generation, the prime movers of choice are large gas turbines as they offer very low maintenance. These turbines tend to be > 20 MW in size and are typically ~ 30% efficient. For most of the distributed power generation applications, the prime mover requirements are smaller than 20 MW and the choices remain reciprocating engines, fuel cells and microturbines (See figure 2). Fuel cells are low-polluting, and highly efficient (~ 60%), but often require very high capital costs. Microturbines, on the other hand, are low-polluting but have very low efficiencies (~30%). Reciprocating engines offer very low capital costs and further have very high efficiencies (~42%) but NO_x emissions are a concern. Also, the maintenance requirements are higher as compared to the other two prime movers.

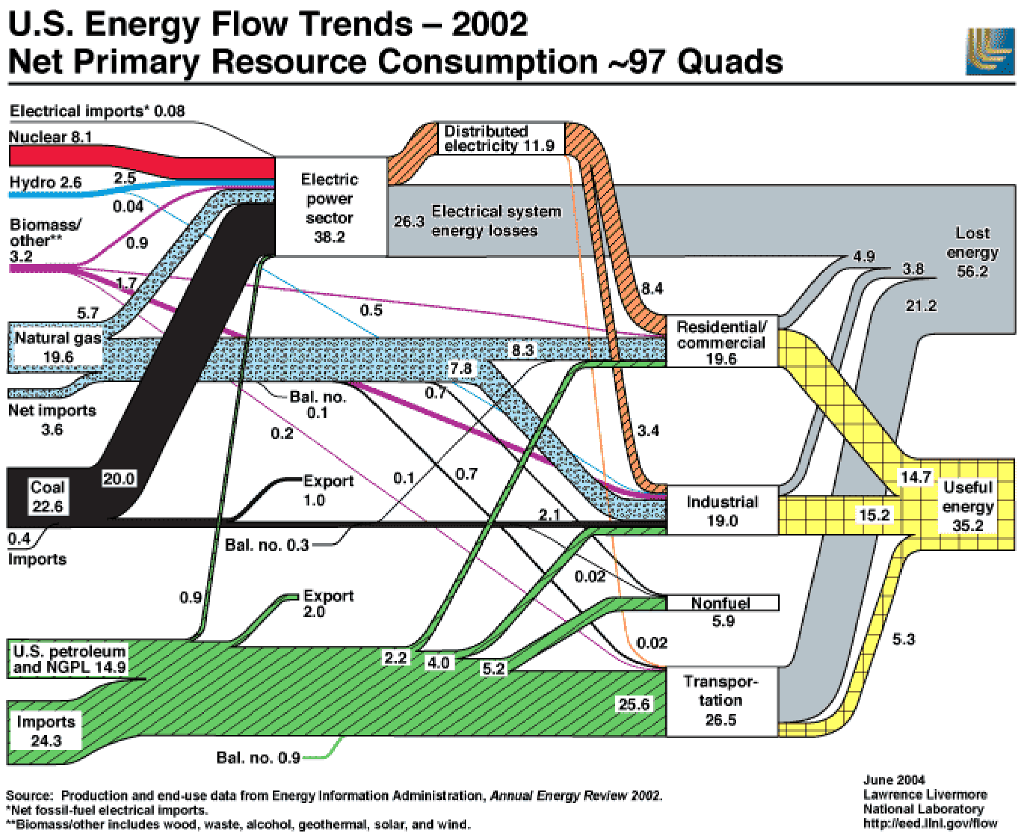


Figure 1. Sanke diagram showing the net US energy flow trends as of 2002. Courtesy: Lawrence Livermore National Lab.

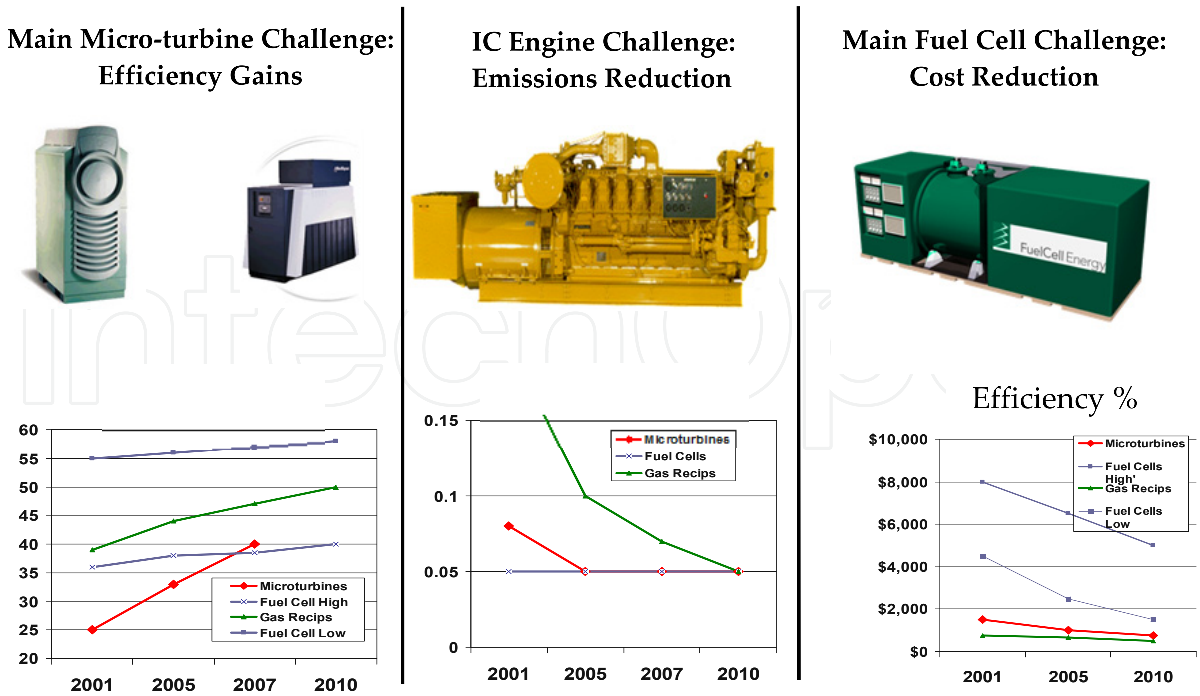


Figure 2. A performance comparison of prime movers used for distributed power generation. (Projections based on 2002 data)

Per recent DOE estimates, over 10,000 stationary reciprocating engines fueled by natural gas are already deployed in various parts of the US for distributed power generation. These are usually large bore engines, with bore sizes > 6.5 inches, and usually are within the power range of 0.5 to 20 MW. It is estimated that over 80% of such engines are of the power range 1.5 MW or lower. In order to accommodate the fact that these engines need to have high reliability, ensuring availability of >95 % throughout the year, most of these engine are of robust design with large lubrication oil reservoirs and very high life components such as bearings and spark plugs equipped with noble metal electrodes.

2. Combustion characteristics of natural gas

Before we proceed further, it behooves us to review the combustion characteristics of natural gas and the indicies commonly used in the gas engine industry to quantify them.

2.1. Natural gas combustion characteristics

The composition of natural gas as distributed in the United States varies from region to region and throughout the year. The variation of the main components of the gas as a result of 120 samples drawn at different locations is shown in Table 1 [1]. As noticed from this tabulation, methane is the primary component with ethane, propane and butane as the major components. Hydrocarbons larger than C_4H_{10} occur in trace quantities. Propane is added by some companies as a peak shaving measure and can reach concentrations as high as 23%. Nitrogen is added to aid in the pumping of natural gas through pipelines.

	Methane	Ethane	Propane	Butane	Pentane	Hexane	Nitrogen	CO_2
Mean	93	3	1	0.5	0.1	0.5	1.5	0.5
Std. dev.	5.5	2.6	1.4	1	0.3	0.1	2.9	0.5
Min.	73	0	0	0	0	0	0	0
Max.	99	13	8	7	3	1	17	2

Table 1. Summary of natural gas variation within the US [1]

With such a varying composition, the following indices are used to characterize natural gas as a fuel for reciprocating engines.

Heating value

In the US, natural gas distribution companies try to hold the heating value, HHV at 1028 BTU/cu.ft., the volume being measured under standard conditions of 60°F and 14.73 psia. This corresponds to a LHV of 910 BTU/cu.ft.

Flammability limits

For natural gas the flammability limits are widely accepted to be between 5% and 15.6% by volume in air corresponding to equivalence ratios of 0.486 and 1.707 [2]. These limits marginally widen with increased concentrations of higher hydrocarbons.

Ignition limits

The auto ignition limits (which closely correlate with knock limits) and the spark ignition limits of methane-air mixtures at 490°C as determined using a Rapid Compression Machine are given in Figure 3. As noticed, though the lean flammability limit of methane-air mixtures is $\phi = 0.5$ ($\lambda = 2.0$), mixtures leaner than $\phi \sim 0.65$ ($\lambda \sim 1.54$) are very difficult to ignite using conventional ignition systems. A similar limit for stoichiometric mixtures diluted with exhaust gas is a EGR fraction of 22%. Significant efficiency and emission benefits can accrue with the extension of such limits.

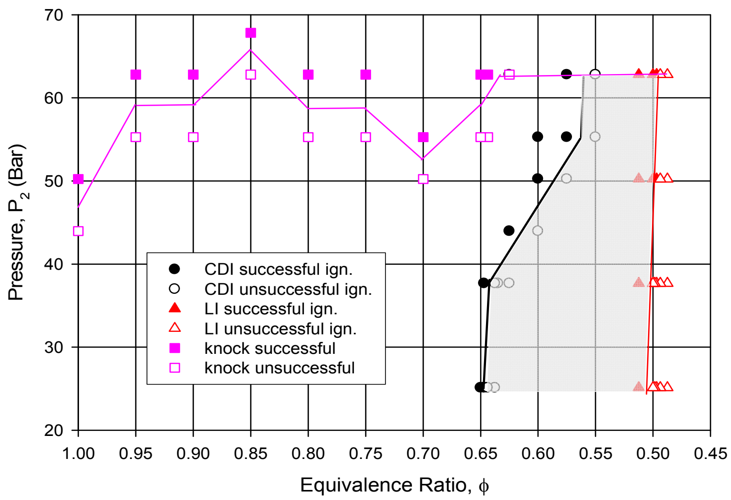


Figure 3. Ignition limits of methane-air mixtures at 490oC as determined using a RCM. [3]

Laminar Flame speed, SL

The laminar flame speed of natural gas as compared to other gaseous fuels is shown in Figure 4 below. As seen, due to the fact that the primary constituent of natural gas is methane, a very stable molecule, its flame speed closely mimics that of methane itself. Even for peak shaving conditions, wherein propane concentrations can be as high as 23%, the laminar flame speed is marginally larger. On the other hand, hydrogen laminar flame speed (not shown in the figure) is approximately 6 times that of methane. Significant engine efficiency improvements can be garnered by improving the flame speed of natural gas.

Methane Number (Knock resistance of fuels)

Knocking is a phenomenon closely associated with autoigntion of fuel air mixtures. In reciprocating engines, the end gas is compressed by the expanding flame front initiated by the spark kernel and auto ignites potentially at more than one location. This phenomenon could be damaging to the engine hardware and is manifest in pressure oscillations observed in the combustion pressure traces. As gasengines are operated close to the knock limit to extract maximum efficiency, knock resistance of the fuel is of prime concern. The most widely used index in the automitive industry to detrmine the knock potential of liquid fuels

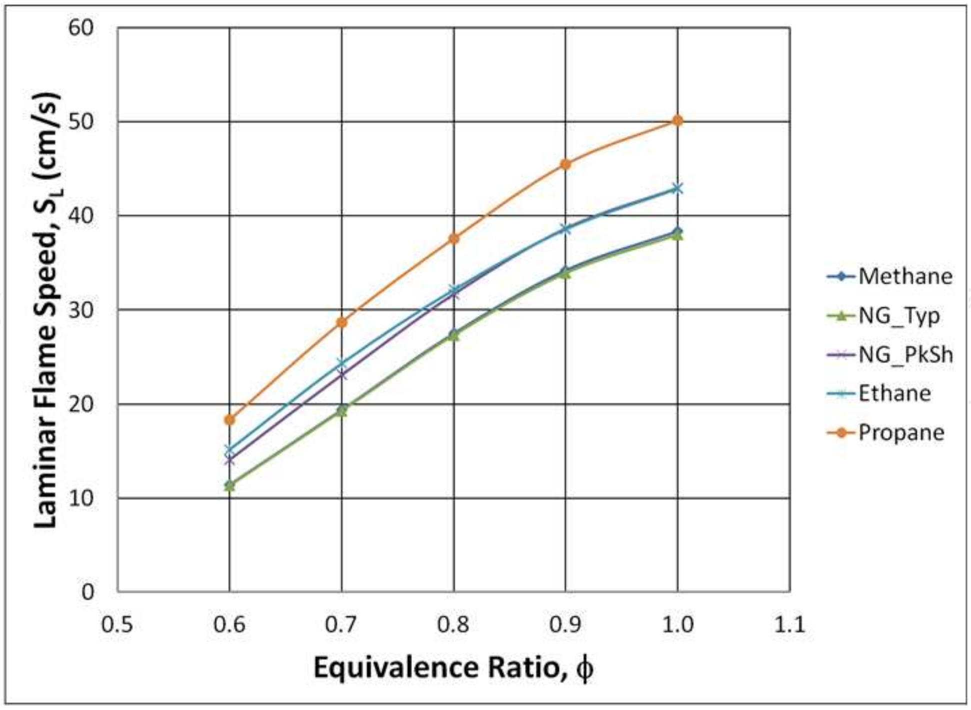


Figure 4. Laminar Flame Speed of typical natural gas and that representative of peak shaving composition as compared to other gaseous fuels. (Data points estimated using CHEMKIN software and GRIMECH 3.0 kinetic data.)

is Motor Octane Number (MON), which is determined as the knock tendency of a fuel between that of iso-Octane and n-Heptane. However, such an index is not representative of the range of values observed in gaseous fuels. As a result, the most widely used index in the industry is Methane Number (MN), which rates the knock resistance of the fuels between 100% methane and 100% Hydrogen. The other less widely used index is Butane number (BN), determined using methane and butane as primary reference fuels.

Gas	MON	MN	BN
Methane	122	100	0
Ethane	101	44	7.5
Propane	97	34	10
Butane	89	10	100

Table 2. Knock ratings of normal paraffins [4]

The knock ratings of various single component fuels are shown in Table 2. As seen, methane by itself being a stable molecule improves knock resistance, whereas long chain hydrocarbons like propane and butane inhibit it significantly. The MON equivalent of natural gas has been related to the reactive hydrogen to carbon ratio of the fuel as follows

$$MON = -406.14 + 508.04\,(H/C) - 173.55\,(H/C)^2 + 20.17\,(H/C)^3 \quad (1)$$

Further, empirical correlations relate the MON equivalent of natural gas to the methane number, as follows

$$MN = 1.624\ MON - 119.1 \tag{2}$$

Quadratic expressions are also available, relating the concentrations of individual component gases to the Methane number of the fuel [4]. Overall, the concentration of larger hydrocarbons in natural gas has a dominating effect and it significantly reduces the MN. As observed by Hack and McDonell, larger hydrocarbons also increase the heating value and the general emission propensity of natural gas [5].

As gas engines are operated close to their knock limit to maximize efficiency, the current compression ratios of these engines are limited to CR < 12. Only in the case of diesel pilot ignited engines, wherein diesel autoignition is encouraged, larger compression ratios are used.

Wobbe Index

This is defined as energy content per relative density. It is commonly used to determine compatibility of fuel systems with fuel type, especially in gas turbines and industrial burners. Wobbe index is less relevant to reciprocating engine combustion.

Among the combustion characteristics presented above for natural gas, poor ignitability and low flame speed are of chief concern. Significant improvements in engine performance can be garnered by addressing these issues.

3. Main operating concerns of reciprocating engines

The main concerns for natural gas fueled engines are

3.1. Emissions

Natural gas fueled engines are very low polluting as compared to other hydrocarbon fueled engines. However, as they are operated round the clock and throughout the year, the emissions emitted per year total to a very large quantity. Of chief concern are NO_x , CO and UHC emissions. NO_x emissions tend to increase with engine efficiency and have pathways of thermal, prompt, NNH and other such mechanisms. Under low-temperature conditions, NO_x formation is inhibited, whereas CO and VOC emissions increase. The final engine operation is optimized for maximum efficiency, while being compliant with local emission regulations.

In the US, the stationary engines were exempt from emission regulations till June 2006. The regulations instituted thereafter are summarized in Table 3. As noticed, the applicable regulations are based on the power of the engine, as well as, the intended application.

3.2 Efficiency

The prime operating cost of these stationary engines is the fuel cost, which can total up to six times the initial cost of the engine itself within a year of operation. As a result, the efficiency of the engine is of prime concern to the engine operator.

Stationary Spark-Ignited Generator Set Emission Regulations — **(NOx+HC) / CO (g/kWm-hr)** — **NOx / CO / VOC (g/kWm-hr)**

	kWe, 60Hz	Gross engine power kW	(HP)	2008	2009	2010	2011	2012
Standby	15–75 NG	19–97	(25–130)	—	(13.4) / 519 Part 90 V			
Standby	80–150 NG	97–179	(130–240)	—	(2.7) / 4.4 Part 1048 V[1] or		2.0 / 4.0 / 1.0 Part 60 V	
Standby	15–75 LP	19–97	(25–130)	—	(13.4) / 519 Part 90 M			
Standby	80–150 LP	97–179	(130–240)	—	(2.7) / 4.4 Part 1048 M[1]			
Prime	15–50 NG	19–75	(25–100)	—	2.0 / 4.0 / 1.0 Part 60 V or		(2.7) / 4.4 Part 1048 V[1]	
Prime	60–150 NG	75–179	(100–240)	—	(2.7) / 4.4 Part 1048 V[1]		1.0 / 2.0 / 0.7 Part 60 V	
Prime	15–150 LP	19–179	(25–240)	—	(2.7) / 4.4 Part 1048 M[1]			

M: Manufacture certification required
V: Manufacture voluntary certification or end user mandatory certification
1: Alternately can use emissions limit formula: (NOx+NMHC) x CO < 8.57 rounded to the nearest 0.1 g/kW-hr.
You may not select an HC+NOX emission standard hight than 2.7 g/kW-hr or a CO emission standard higher than 20.6 g/kW-hr.

Non-regulated
Interim
Final

Table 3. US emission regulations for stationary SI engines.

In order to meet the competing demands of emission regulation, fuel efficiency and system reliability, two prominent engine operational modes have evolved.

The first referred to as "Rich-burn" engines are primarily operated at equivalence ratios ≈ 1 in conjunction with a three-way catalyst to reduce CO, HC and NOx emissions. Rich-burn engines often use Exhaust Gas Recirculation to reduce in-cylinder combustion temperatures and as a result the engine-out NO_x emissions. To offset the loss in power density, intake air is boosted using a turbocharger. Efficiencies of Rich-burn engines are ~ 36% as of 2011. The primary advantage of Rich-burn engines is that they easily achieve compliance with emission regulations.

The second operational mode that has prominently evolved over the last two decades is "lean-burn." Such engines are often operated at very low equivalence ratios, $\phi < 0.65$ ($\lambda > 1.54$). The inert nitrogen gas in the excess air that is used acts as a diluent and reduces the in-cylinder combustion temperature for subsequent NO_x mitigation. Lean-burn engines are often boosted using a turbocharger to offset the loss in power density. The primary advantage of lean-burn engines is that they can achieve thermal efficiencies as high as 42% leading to significant fuel savings to a typical operator. Also, the engine-out emissions are so low that there is no need for an aftertreatment system for easy compliance with emission regulations.

3.3. Maintenance

Ignition: Of prime concern to the operation of stationary natural gas engines is the maintenance associated with spark plugs. Modern engines are optimized to meet the demands of increased efficiency, low emissions and high specific power, and are operated under conditions that are difficult to ignite. Problems with ignition can result in significant misfires leading to increased hydrocarbon emissions, and most importantly increased fuel consumption.

Friction: On account of the fact that stationary engines are not produced in large volume, technologically their designs have lagged the advancements in the automotive field. Besides leading to energy loss that can total up to ~ 7%, friction leads to reduced hardware life.

4. Recent technological advancements

Addressing the above concerns, US-DOE has embarked on a research program called Advanced Reciprocating Engine Systems (ARES) since 2002, with the research activities being conducted at various universities, National Labs and industry. The main goals of the ARES program are to achieve >50% efficiency, < 0.1 g/bhp-hr NO_x, increased fuel flexibility, and lowered capital costs by 2013. In the remainder of this chapter, some of the technologies pursued under the ARES program at Argonne will be presented, along with some important technologies pursued elsewhere.

4.1. Pretreatment technologies

4.1.1. Nitrogen Enriched Air (NEA)

In this technology, the preferential diffusion of oxygen over nitrogen through certain polymers is advantageously used [6]. Nonporous Polymeric membranes are instituted as micron thick coatings on the inner walls of capillary tubes. The coated capillaries, in turn, are packaged into bundles (see Fig. 5) with optimized geometries for low pressure drop and high yield. When air is passed through such membrane bundles, it is split it into a nitrogen-rich stream and an oxygen-rich stream. When the nitrogen-rich stream is used for engine combustion purposes, it leads to lower in-cylinder temperatures by inert gas dilution thereby reducing NO_x formation. NEA avoids harsh and corrosive exhaust emissions, such as particulates and acidic compounds, from being introduced into the engine intake manifold and thereby offers a clean alternative to the traditional Exhaust Gas Recirculation. Advantages that accrue from NEA include improved engine hardware and significant extensions to lubrication oil life.

The basic principle of membrane operation and various designs and operating characteristics are described in Poola et al. [7]. Research work done on a locomotive two stroke diesel engine by Poola and Sekar [8] corroborate variable air composition as a technique to reduce emissions.

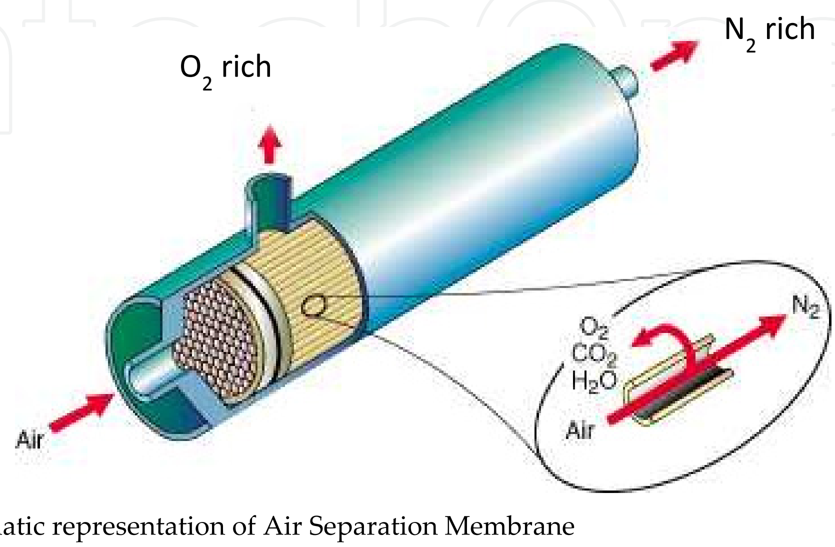


Figure 5. Schematic representation of Air Separation Membrane

Nitrogen enrichment tests with gas blends

Initial tests were conducted in a 1.87 Liter, 33 kW, 1800 rpm SI natural gas single-cylinder research engine. Nitrogen enrichment was achieved by inducting bottled nitrogen into the air stream [9]. Before proceeding further, let us define a few terms:

The amount of nitrogen enrichment (NEA) is expressed as the volume percent of nitrogen above the usual 79.05% in the intake air.

To account for the fact that the composition of air varies with nitrogen enrichment, it helps to define an oxygen based equivalence ratio,

$$\psi = \frac{\left(\dot{m}_f / \dot{m}_{O_2}\right)}{\left(\dot{m}_f / \dot{m}_{O_2}\right)}. \tag{3}$$

With nitrogen enrichment, the local combustion environment is starved of oxygen and leads to increased levels of ψ. This in turn, leads to reduced bulk gas temperatures and results in lower NOx formation. The test results for varying amounts of nitrogen enrichment for a fixed ignition timing of 20°BTDC, 1800 rpm and 12 bar BMEP are shown in Figure 6. As noticed with increasing nitrogen enrichment, NOx emissions decrease monotonically. Also, as shown in Figure 7, for a fixed ignition timing, the fuel conversion efficiency (FCE) decreases concomitantly with NOx emission. Analysis of the cylinder pressure traces shown in Figure 8 provides further insights. With increasing nitrogen enrichment the combustion event is both delayed as well as decelerated, which points to the possibility of further gains by optimizing the ignition timing.

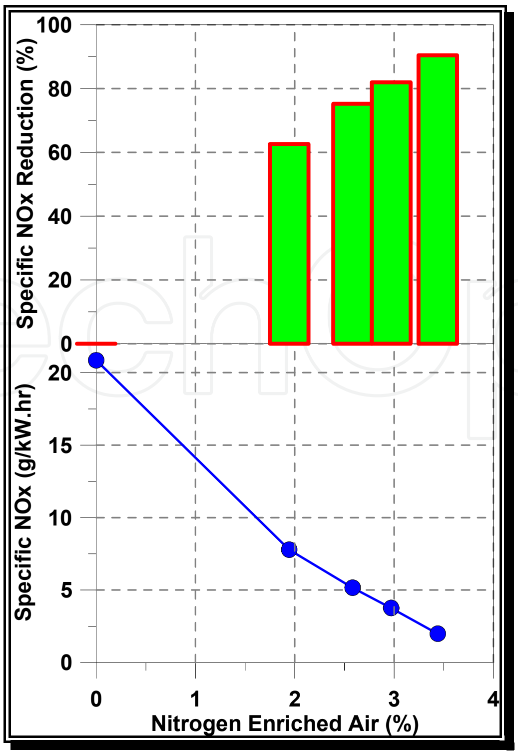


Figure 6. Specific NOx as function of Nitrogen Enrichment.

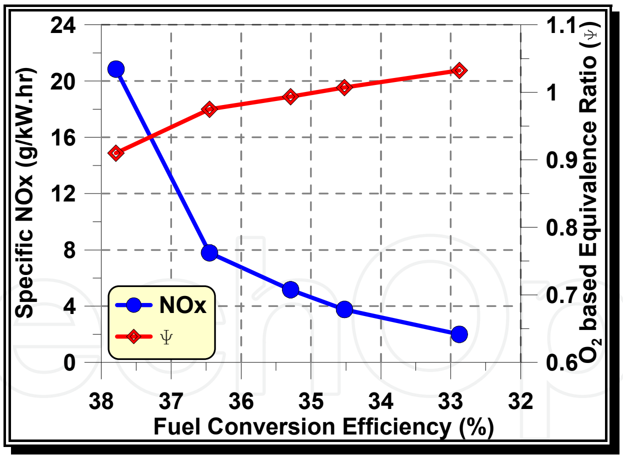


Figure 7. Spec. NOx and ψ as a function of FCE

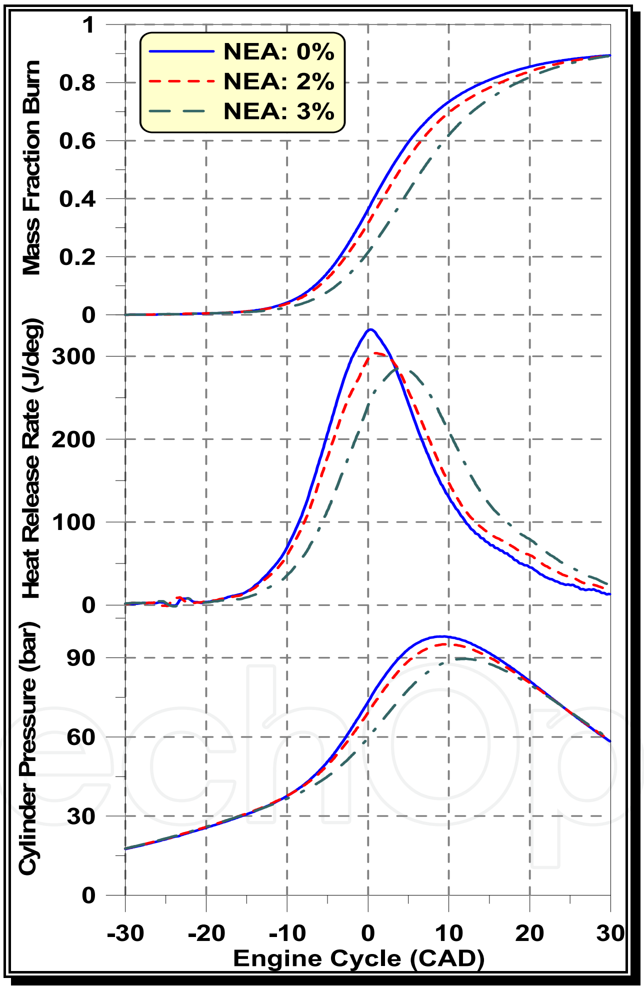


Figure 8. Combustion related data as a function of engine cycle. Ignition timing = 20°BTDC

The optimal performance results were obtained through ignition timing sweeps performed between 25º and 45º BTDC. Within these limits, based on the equivalence ratio, ignition timing could be varied between knock limit and that value limiting the COV of ignition below the industry accepted 5%. In most of the cases, engine knock was avoided with nitrogen enrichment. The optimal performance results identified through such timing sweeps are shown in Figure 9. It is interesting to note that for similar air/fuel ratio, identified by the circles on the graph, NOx decreases by approximately 50% with less than 0.5% FCE penalty. Similar NOx reduction, around $\psi = 1.08$ is an appreciable 84%, but is accompanied by ≈ 2% FCE. Operation for intermediate values of ψ is not foreseen due to unacceptably high NOx emissions.

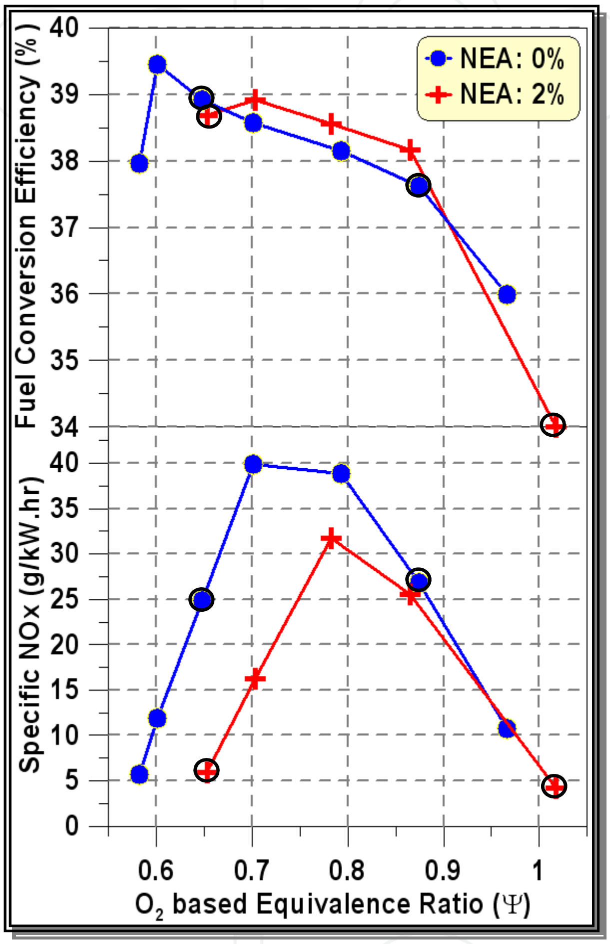


Figure 9. Specific NOx and FCE as a function of ψ

Nitrogen enrichment tests with air separation membrane

Encouraged by the above results, tests were performed by using nitrogen enriched air provided by an air separation membrane. As a compromise between the parasitic power requirements to achieve high levels of nitrogen enrichment and the amount of reduction in NOx emissions, it was decided to fix the nitrogen enrichment at 2%. For further details of the experiment, the reader is referred to reference 27. In these set of tests, both laser ignition as well as conventional CDI were used. The test results are as shown in Figure 10. For either method of ignition, low BSFC and specific NOx were obtained for lean operation. A combination of LI and NEA led to NOx reductions up to 89%, as compared to 70% reductions achievable just by using NEA [10].

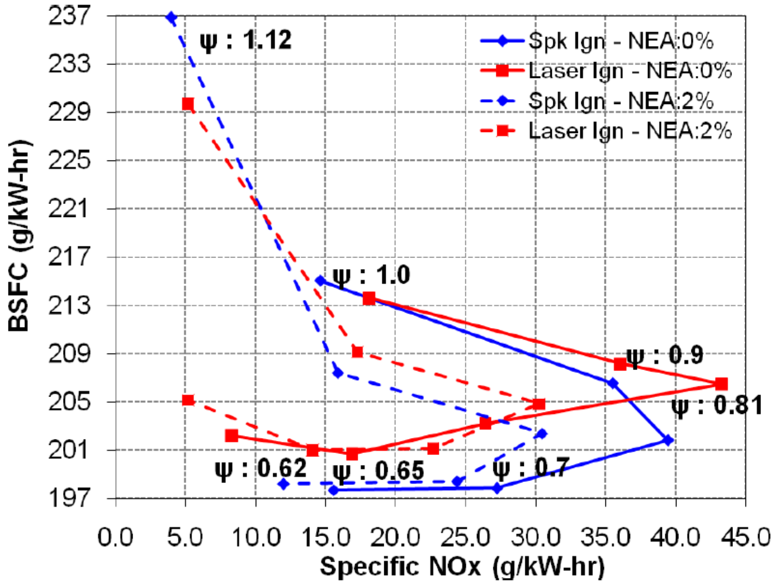


Figure 10. BSFC versus Specific NOx at different ψ.

Nitrogen enrichment vs. EGR

Tests were also conducted to compare the efficacy of NEA and EGR to reduce NOx while maintaining the combustion efficiency. To facilitate simulated EGR under turbocharged conditions in a single-cylinder engine, an EGR system using a reciprocating compressor was developed. Such a system facilitated exhaust gas recirculation up to 29%. Results from such tests are summarized in Figure 11. It is seen that though both EGR and NEA offer comparable efficacy towards NOx reduction, their ability to retain concomitant FCE could not be estimated accurately. Better models for auxiliary power requirements in either case are required to facilitate such a comparison.

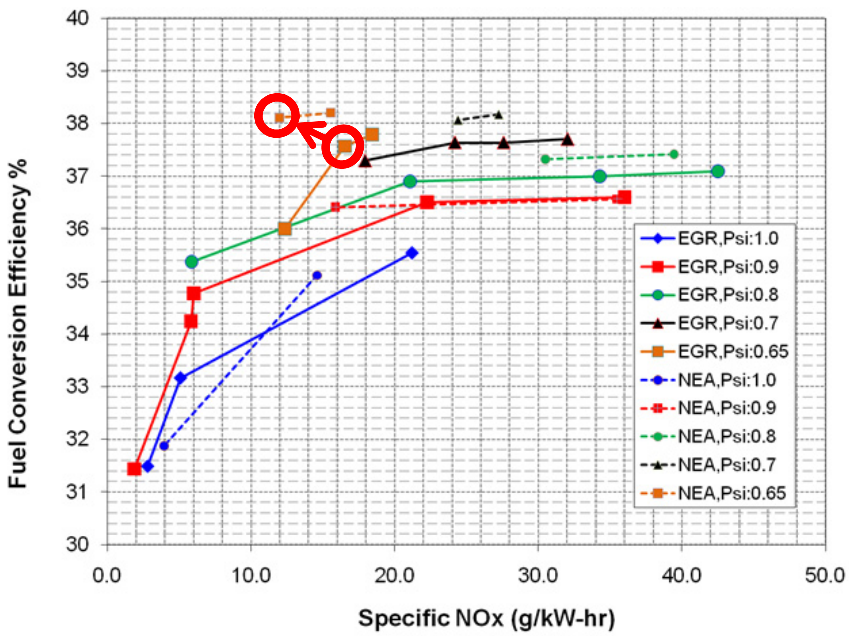


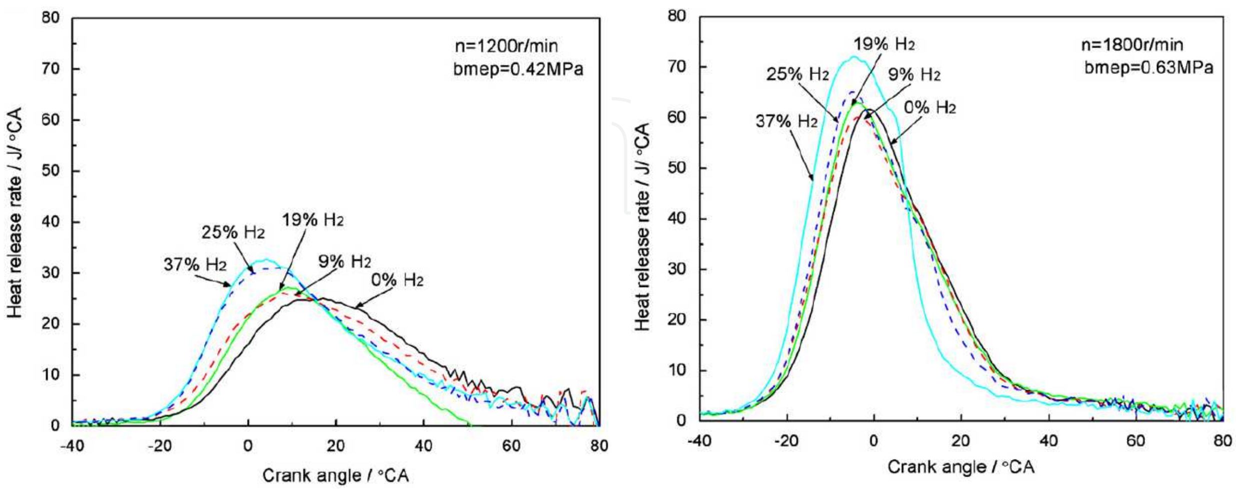
Figure 11. Performance tradeoff with EGR or NEA.

4.1.2. Hydrogen addition

Though natural gas is characterized by high octane number of 122 and clean burning nature, its combustion in IC engines is limited by low flame speed and poor ignitability. The low flame speed limits the rate at which the flame spreads from the ignition kernel to consume the complete in-cylinder charge. This is manifest in the form of low rates of heat release, and results in lower engine efficiencies. Ignitability, on the other hand, is limited by the capabilities of the ignition system used and at the outer extremes is limited by the flammability limits of the fuel. On the other hand hydrogen gas has flame speeds that are 6 times larger than those of natural gas. Also, hydrogen has a lean flammability limit that extends up to◎◎ = 0.1. Significant operational gains can be achieved, with the addition of hydrogen to the natural gas, as it improves both ignition, as well as, flame speed.

In a majority of gas engines, natural gas is inducted in to the intake plenum, and as a result occupies $\phi < 9.5\%$ volumetric portion of the intake charge. Such a large fraction contributes to the breathing losses, resulting in low efficiencies, especially at low-loads. To offset such an effect, gaseous fuel is injected directly in to the cylinder, usually optimizing the parasitic loss associated with compression of the fuel. As a result, some of the hydrogen addition tests have used in-cylinder injection.

Wang and coworkers [11] have conducted tests in a engine fueled with various fractions of hydrogen addition. They observed that with increasing amount of hydrogen addition, the phase of the heat release curve advanced, and combustion duration decreased (see Figure 12). This phenomenon was more pronounced at low engine speed suggesting efficacy of flame speed enhancement at relatively low cylinder air motion. As the flame speed increases exponentially with hydrogen content, maximum mean gas temperature and rate of pressure rise increased rapidly. As a result, HC, CO and CO_2 emissions decreased rapidly, whereas NO_x emissions increased. Optimal hydrogen concentrations were found to be around 20% as a best compromise between emissions and efficiency.



(a) (b)

Figure 12. Combustion pressure curves with the addition of Hydrogen at (a) low speed and low loads, and (b) high speed and high load [11]

To offset the increase in NO_x emissions, low-temperature combustion may be promoted using Exhaust Gas Recirculation. Recently Hu et al. [ref. Hu 2009] conducted studies in a stoichiometric engine equipped with a three-way catalyst using EGR rates up to 40% and hydrogen fractions up to 40%. They have observed that for a given hydrogen fraction, NOx concentrations decrease with increasing EGR fraction. CO and CO_2 while remaining less influenced by EGR, decreased significantly with hydrogen addition. As shown in Fig. 13, an optimization of EGR fraction, hydrogen addition, and ignition timing results in high efficiencies and low emissions. Optimal engine operation at 2000 rpm was achieved, for EGR fractions in the range 10-20% and hydrogen fractions in the range 30-40%.

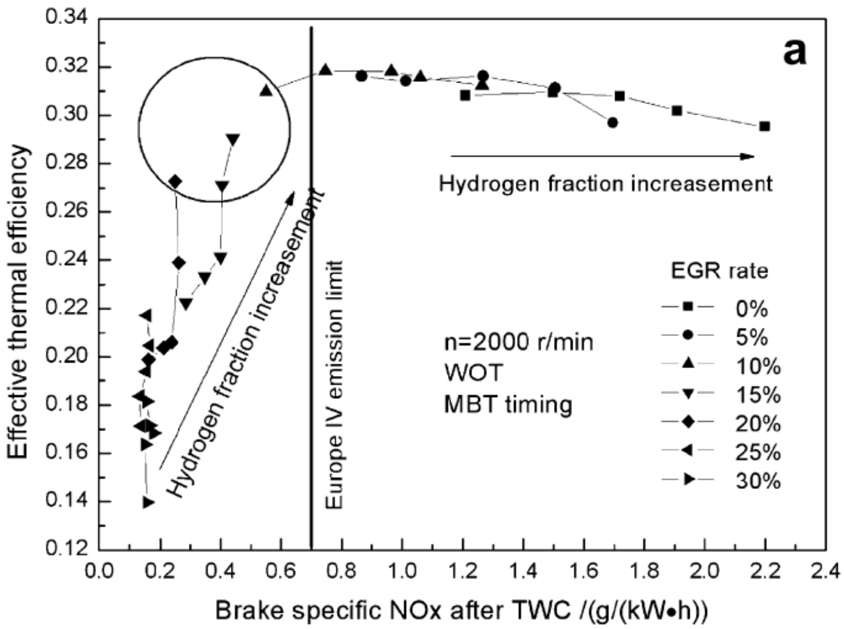


Figure 13. NOx vs. efficiency tradeoff for an engine with EGR and hydrogen addition.[12]

Similar results were evident through tests conducted though tests conducted on lean-burn engine [13] . In spite of the potential to improve the performance significantly, use of hydrogen addition in practical engines has evaded reduction to practice. A review of the practical aspects associated with hydrogen storage and transportation, reveals that on-site hydrogen generation is more suitable for use with these engines. Among the possible methods of hydrogen generation (a) Steam methane reforming, (b) partial oxidation (POX), (c) autothermal reforming, and (d) exhaust gas fuel reforming [14]. All of these methods use a catalyst and are somewhat sensitive to the temperature, carbon poisoning, and sulfur poisoning and it appears the former two being less complex are most reliable for use with stationary engines. Gas Technology Institute has evaluated the prospect of using engine exhaust heat, which for a natural gas engine can be in excess of 550°C, to reform a slipstream of natural gas fuel [15]. As shown in Figure 13, the slipstream is reformed to H_2 and CO mixture which is further enriched by a water shift reaction further downstream. Though excellent gains in emission reductions and efficiency improvements up to 3% points were possible, long term performance of the catalyst degraded due to carbon poisoning. An alternative path, though less pursued, is to use two engines in tandem: A PO_x engine that operates under rich conditions, partially oxidizes the fuel, the exhaust of which is fed to a larger engine that consumes additional fuel [16]. Research continues in identifying a potential pathway for on-site hydrogen generation.

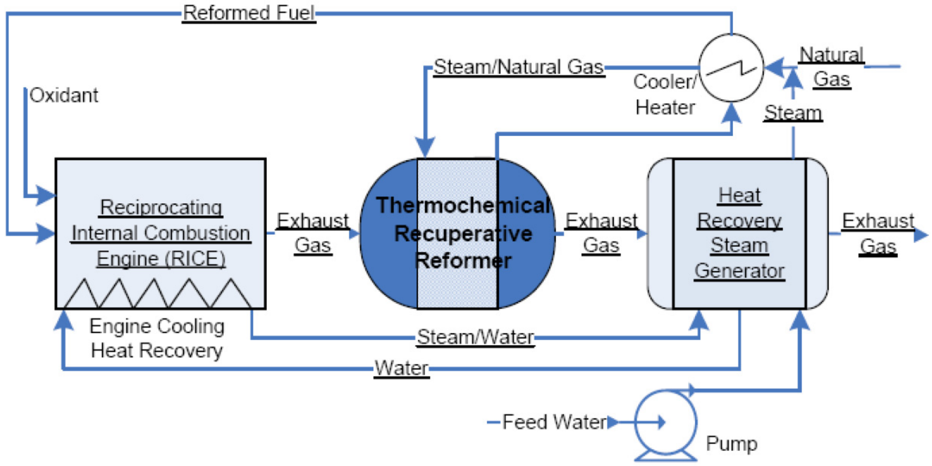


Figure 14. Schematic representation of TCR use with engine.[15]

4.2. Aftertreatment

Catalyst aftertreatment is a logical approach to achieve emission requirements (0.1g NOx/hp-hr: <7 ppm NOx) with minimal impact on engine design. In addition, catalyst technology may be able to abate other pollutants such as formaldehyde, residual VOC and unburned CH_4 from natural gas engines at the same time as it controls NO_x. Though the performance of a three-way catalyst under stoichiometric operation is proven, the associated engine efficiencies are somewhat low. Unfortunately, the presence of excess oxygen in the exhaust precludes using three-way catalyst technology for lean-burn operation.

Today, the only proven aftertreatment technology that can provide substantial NO_x reduction (90+%) in lean burn exhaust is the SCR catalyst system that requires supplemental addition of ammonia/urea as a reductant to the exhaust stream. However, there are a significant number of implementation issues associated with urea SCR that make it a less than ideal technology to control NO_x emissions. These issues include storage, corrosion, transportation, infrastructure, and concerns over release (slip) of ammonia into the air or oxidation of ammonia at high temperatures which yields further NO_x emissions.

A more attractive aftertreatment technology is NSR (NOx Storage and Reduction) catalysis (also referred to as NO_x Adsorbers), wherein NO_x is stored by reaction with an alkaline earth to form the solid nitrate. In a reduction step under rich condition, an injection of reducing agents releases and reduces the stored NO_x, converting it to N_2. Therefore, the catalyst needs to be regenerated occasionally to restore the NO_x storage capacity. The preferred reducing reagents; CO or H_2 are not present in sufficiently high amounts in natural gas engine exhaust. The challenge is to regenerate the catalyst with an available reductant (CH_4). Because NSR catalysts consist of alkali metal or alkaline earth metal oxides, sulfur oxides in the exhaust stream readily form stable sulfates with the catalyst material, causing a loss in deNOx performance. In spite of the best efforts under the ARES program, efforts to improve the NO_x reduction efficiency while improving the resistance to sulfur poisoning and reducing the fuel penalty at reasonable levels have met with partial success [17, 18].

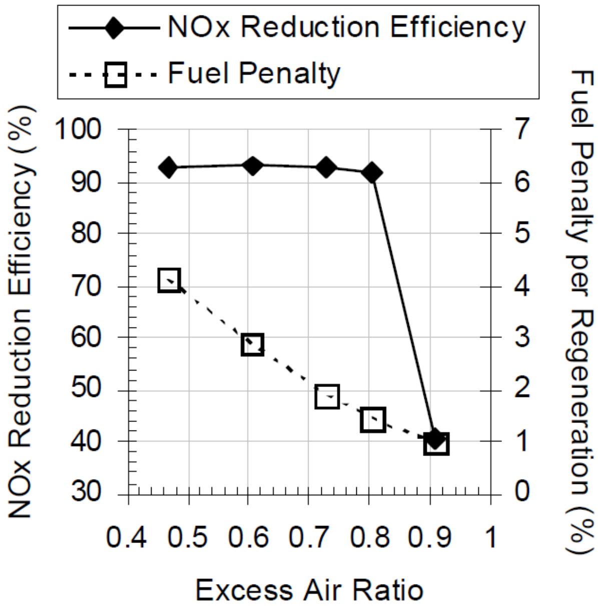


Figure 15. NOx reduction efficiency during sorption and fuel penalty per regeneration as a function of excess air ratio (⊚) [17].

4.3. Friction reduction

In a typical reciprocating engine, approximately 7% of fuel energy is wasted due to frictional losses. Approximately half of these losses can be attributed to the piston Ring-Cylinder Liner (PRCL) interface where mixed mode lubrication occurs at top-dead center and bottom-dead center. It is estimated that approximately 50% of the friction at the PRCL interface can be reduced with (i) the use of improved lubricants, (ii) improved ring pack designs, and (iii) Laser Surface Texturing (LST). In the case of stationary engines that operate throughout the year, this can lead to significant fuel savings and significant GHG emissions reduction. Research in improved lubricants continues as and will not be discussed here. Ring packs were optimized through modeling and improvements up to 0.5% points in efficiency were observed through tests on a 6-cylinder engine [19].

LST on the other hand, stands out as one of the promising tribological advancements in the last 30 years. In this technology, sections of the cylinder are covered with a 2-D array of micron size dimples, approximately 100 μm dia and 5 μm deep, which in turn act as micro-hydrodynamic bearings in the case of full/mixed lubrication, micro-reservoirs for lubricant in the case of starved lubrication, and as micro-traps for absorbing wear particles. Through numerous modeling and experimental efforts the depth, size and spacing of the dimples have been optimized to result in friction reduction at the PRCL interface up to 50% [20, 21].

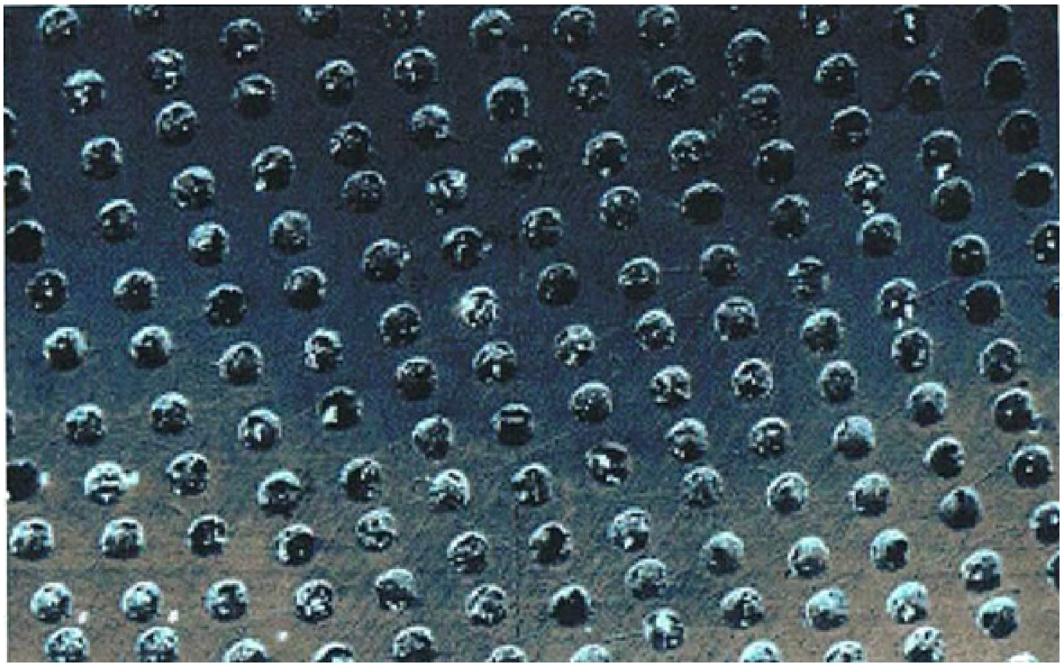


Figure 16. Photograph of laser textured surface.

4.4. Ignition

As mentioned earlier, to reduce in-cylinder NO_x formation, Low-Temperature Combustion conditions are preferred. This can be achieved by using excess combustion air as used under lean-burn conditions, or with the use of EGR as used under rich-burn conditions. In either case, to offset the loss in specific power of the engine, engine intake pressures are boosted using a turbocharger. Under such conditions, the in-cylinder pressures at the time of ignition tend to be high. The voltage required to strike a spark across the spark gap, which is governed by Paschen's law ($V_{discharge} \propto$ pressure x distance between electrodes), increases to a point that it surpasses the 40 kV design limit of commercial Capacitance Discharge Ignition (CDI) Systems. Also, with the use of high voltage ignition systems the erosion of spark electrodes is accelerated (see Figure 17). Depending upon the manufacturer, typical Iridium and Platinum tipped spark plugs are currently replaced every 1000-3000 hrs of operation. To reduce the maintenance burden, engine manufacturers would like to extend this interval to 8000 hrs of engine operation. Additionally, confirming to market demands for higher specific power from a given engine frame, manufacturers would like to operate these engines under high in-cylinder pressures (see Fig. 18). Such conditions, along with the need for reliable ignition, and the concomitant need for reduced maintenance warrant the use of advanced ignition strategies.

Numerous entities around the world have proposed and evaluated numerous advanced ignition strategies to extend ignition limits, enhance the ignition reliability, improve ignition stability and reduce maintenance requirements. Most prominent ones will be discussed below in the context of their power draw, initial cost, long-term durability and ability to ignite lean/ highly diluted mixtures.

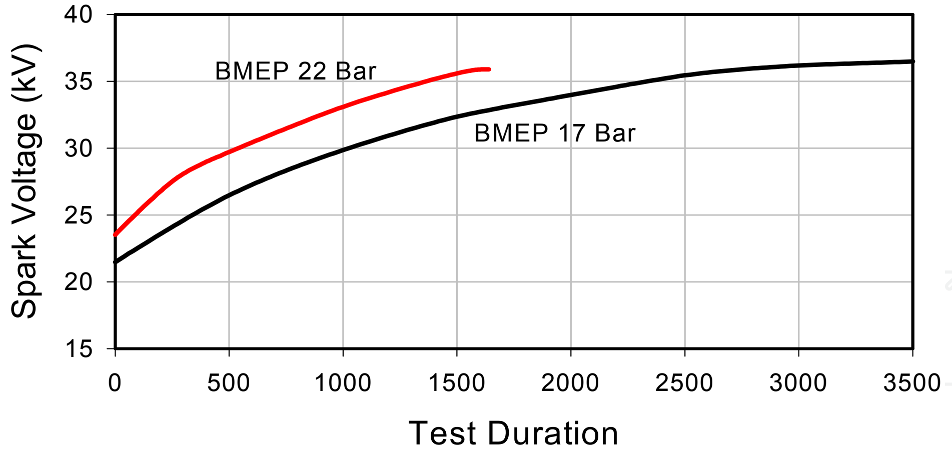


Figure 17. Ignition voltage vs. engine operational duration. [22]

ASME ICE-Vol. 35-2, 2000]

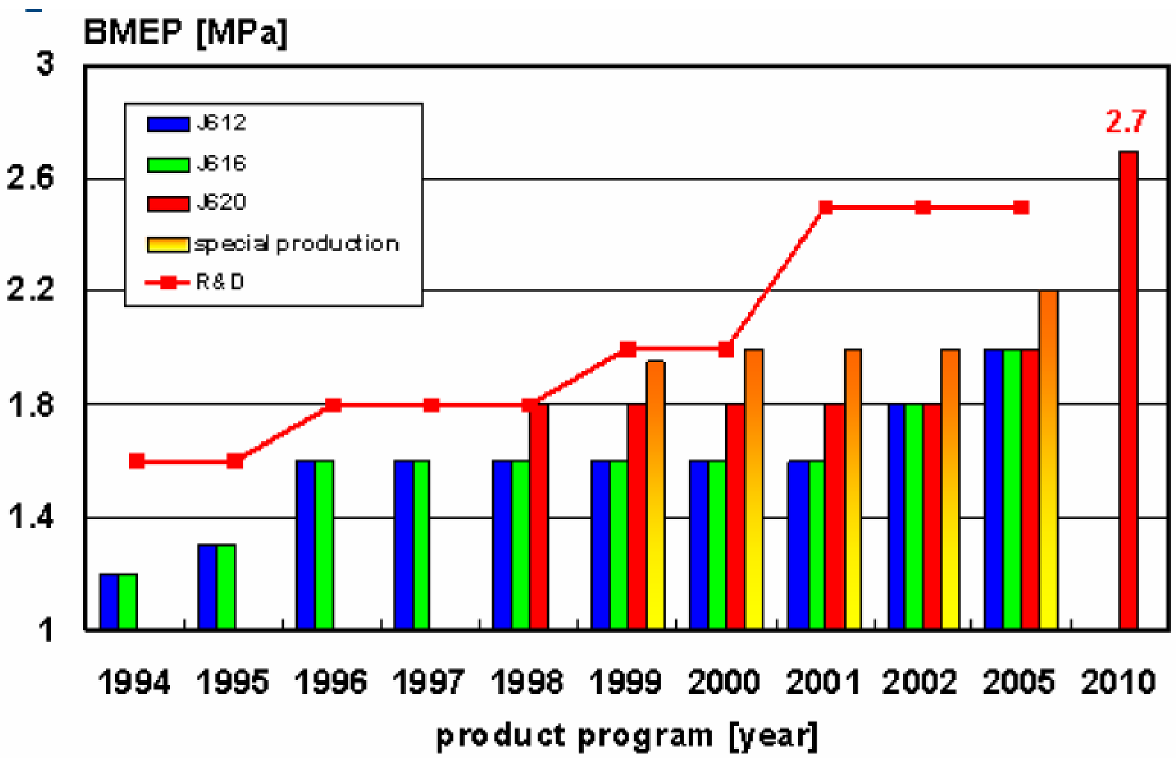


Figure 18. Engine BMEP trend at GE-Jenbacher [23]

4.4.1. Plasma jet / Rail plug / Corona discharge ignition

A number of studies have been performed evaluating the high energy plasma ignition systems [24, 25]. Most of such systems prove promising in extending the lean ignitability limits and thereby improving the performance of the engine. However, their power draw tends to be very large and unattractive. Also, high-energy plasma generation results in accelerated wear of the spark plug electrodes.

To offset the above shortcomings, several groups have tried using the corona discharge phase of gas breakdown which ionizes the gas prior to a full arc discharge [26 - 28]. Though Corona discharge offers promise, the system needs to be carefully designed specific to an engine to avoid inadvertent arcing between the electrode and the engine cylinder head.

4.4.2. Diesel pilot ignition

In this method of ignition, a very small quantity (< 4% of total fuel) of diesel is injected directly into the cylinder, which under goes auto-ignition and serves as a high-energy ignition kernel (up to 5000 times that achieved in SI) to ignite the lean natural gas-air charge introduced in to the cylinder. The load is varied by regulating the natural gas inducted into the intake air instead of throttling the intake charge. As a result, higher compression ratios can be used resulting in higher efficiencies. The amount of NO_x resulting from such a method of ignition though very low is limited by the NO_x formation resulting from the diesel pilot itself. Krishnan et al. [29] have tried using very advanced timing for the diesel pilot in order to reduce the NO_x formation. Others have tried to reduce the amount of the diesel pilot as a fraction of the total fuel introduced in to the cylinder with promising results [30].

4.4.3. Laser ignition

Among all of the ignition technologies discussed so far, laser ignition remains unparalleled in its ability to extend the ignition limits. As seen in Figure 3, laser ignition can extend the lean ignition limits of methane-air mixtures all the way to the lean flammability limit of $\phi = 0.5$.

Another advantage of laser ignition is the fact that it improves ignitability of fuel-air mixtures as pressure increases. As shown in Figure 19, for conventional ignition systems, the spark voltage which determines the success of ignition events, increases with pressure. In the case of laser ignition, laser pulse energy which becomes the governing factor decreases with increase in pressure. Such a trend proves beneficial in increasing the BMEP of the engines and thereby allows enhancing thermal efficiency.

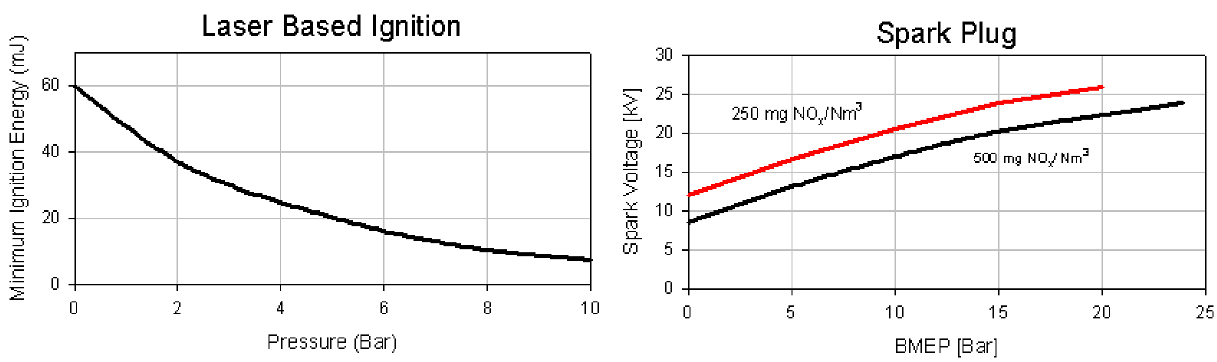


Figure 19. Pressure dependence of laser ignition (LI) and conventional spark ignition (SI). Please note that BMEP is given to be representative of in-cylinder pressure. [22]

Laser ignition tests performed in natural gas fueled single-cylinder engines have shown (i) shorter ignition delays, (ii) accelerated rates of combustion, and (iii) lean ignition limit extensions. However, as shown in Figure 20, the most significant advantage is the fact that laser ignition can result in NOx reductions up to 70% for a given engine efficiency, or alternately up to 3% point improvement in efficiency for a given NOx level [31]. The associated cost savings due to reduced fuel consumption, and avoidance of expensive after treatment systems, have provided an impetus for the development of practical laser ignition systems.

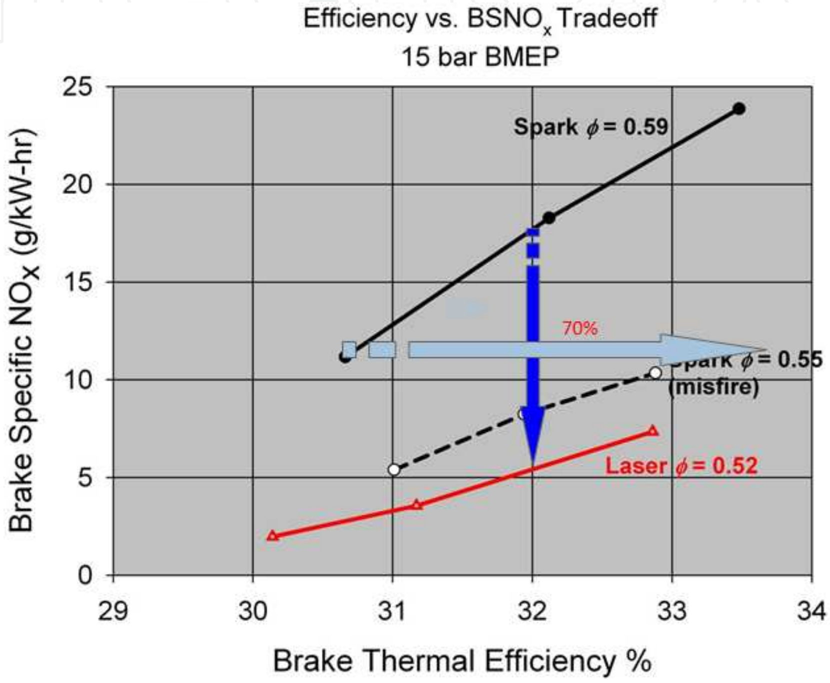


Figure 20. BSNOx vs. thermal efficiency improvement with laser ignition. [31]

The earliest known attempt at laser ignition was that performed by Dale et al. in 1978 [32], when they used a 14 ft. long CO_2 laser. At that time, laser ignition was considered impractical due to the high-cost and large size of the lasers, and interest in laser ignition remained luke warm. Since then, numerous developments in optics and lasers have resulted that make lasers efficient, of small size, low-cost and relatively insensitive to heat and vibration. As a result, interest in this field reemerged around 2000 and various organizations have made attempts to develop a viable laser ignition system. Kroupa et al. [32] and simultaneously Woodruff & Dustin McIntyre et al. [33] have tried to develop an end-pumped laser-per-cylinder Nd:YAG system. Weinrotter, Herdin et al. [34] have tried to develop a split system wherein a long laser pulse from a pump laser is transmitted through a optical fiber to end pump a laser gain medium equipped with a saturable absorber installed in the spark plug well of the engine. Bihari et al. through their attempts found it difficult to transmit high-power laser pulses through optical fibers and have embarked in developing a system that uses free-space transmission [31]. Recently, Taira et al [35, 36] have successfully demonstrated microlaser ignition systems, by using new age lasing materials called optical ceramics, and by using pulse trains of low-energy pulses to ignite instead of a single high-energy laser pulse. As of 2011, attempts to reduce laser ignition to practice continue throughout the world.

5. Conclusions

Technologies for improving the efficiency and reducing emissions from stationary natural gas engines have been presented. Notable omissions from this review are (i) use of advanced combustion cycles, such as, Miller cycle [37], (ii) use of high velocity compression, (iii) use of stratified charge [38], and (iv) use of enhanced turbulence levels [39]. As natural gas becomes widely available in abundance, development of technologies for further improvement of these engines is likely to continue well into the future. Advancements in materials and manufacturing processes for engine components, newer technologies for preprocessing natural gas as a fuel, and robust aftertreatment technologies are likely to yield efficiencies > 50%, while achieving easy compliance with emission regulations. Though not discussed in this chapter, technologies that harness energy from waste heat will further improve thermal efficiencies of these engines.

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The authors research and develop novel ways of improving the efficiency and reducing emissions from stationary natural gas fueled engines that are primarily used for distributed power generation. Of particular importance are their efforts to develop a novel laser based ignition system, and nitrogen enrichment of engine intake air using polymeric membranes. These researchers are currently involved in developing engine strategies for efficient energy conversion from various alternative gaseous fuels – various digester gases, as well as, synthesis gases.

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