Technology Characterization: Reciprocating Engines

Prepared for:

Environmental Protection Agency
Combined Heat and Power Partnership Program
Washington, DC

Prepared by:
Energy and Environmental Analysis, Inc. an ICF Company
1655 N. Fort Myer Dr., Suite 600
Arlington, Virginia 22209

December 2008

Disclaimer:

The information included in these technology overviews is for information purposes only and is gathered from published industry sources. Information about costs, maintenance, operations, or any other performance criteria is by no means representative of agency policies, definitions, or determinations for regulatory or compliance purposes.

TABLE OF CONTENTS

Technology Characterization – Reciprocating Engines

Introduction and Summary

Reciprocating internal combustion engines are a widespread and well-known technology. North American production exceeds 35 million units per year for automobiles, trucks, construction and mining equipment, marine propulsion, lawn care, and a diverse set of power generation applications. A variety of stationary engine products are available for a range of power generation market applications and duty cycles including standby and emergency power, peaking service, intermediate and baseload power, and combined heat and power (CHP). Reciprocating engines are available for power generation applications in sizes ranging from a few kilowatts to over 5 MW.

There are two basic types of reciprocating engines – spark ignition (SI) and compression ignition (CI). Spark ignition engines for power generation use natural gas as the preferred fuel, although they can be set up to run on propane, gasoline, or landfill gas. Compression ignition engines (often called diesel engines) operate on diesel fuel or heavy oil, or they can be set up to run in a dual-fuel configuration that burns primarily natural gas with a small amount of diesel pilot fuel.

Diesel engines have historically been the most popular type of reciprocating engine for both small and large power generation applications. However, in the United States and other industrialized nations, diesel engines are increasingly restricted to emergency standby or limited duty-cycle service because of air emission concerns. Consequently, the natural gas-fueled SI engine is now the engine of choice for the higher-duty-cycle stationary power market (over 500 hr/yr) and is the primary focus of this report.

Current generation natural gas engines offer low first cost, fast start-up, proven reliability when properly maintained, excellent load-following characteristics, and significant heat recovery potential. Electric efficiencies of natural gas engines range from 30 percent LHV for small stoichiometric engines (<100 kW) to over 40 percent LHV for large lean burn engines (> 3 MW)[1]. Waste heat recovered from the hot engine exhaust and from the engine cooling systems produces either hot water or low pressure steam for CHP applications. Overall CHP system efficiencies (electricity and useful thermal energy) of 65 to 80 percent are routinely achieved with natural gas engine systems.

Reciprocating engine technology has improved dramatically over the past three decades, driven by economic and environmental pressures for power density improvements (more output per unit of engine displacement), increased fuel efficiency and reduced emissions. Computer systems have greatly advanced reciprocating engine design and control, accelerating advanced engine designs and making possible more precise control and diagnostic monitoring of the

[1] Lower Heating Value. Most of the efficiencies quoted in this report are based on higher heating value (HHV), which includes the heat of condensation of the water vapor in the combustion products. In engineering and scientific literature the lower heating value (LHV – which does not include the heat of condensation of the water vapor in the combustion products) is often used. The HHV is greater than the LHV by approximately 10% with natural gas as the fuel (i.e., 50% LHV is equivalent to 45% HHV). HHV efficiencies are about 8% greater for oil (liquid petroleum products) and 5% for coal.

engine process. Stationary engine manufacturers and worldwide engine R&D firms continue to drive advanced engine technology, including accelerating the diffusion of technology and concepts from the automotive market to the stationary market.

The emissions signature of natural gas SI engines in particular has improved significantly in the last decade through better design and control of the combustion process and through the use of exhaust catalysts. Advanced lean burn natural gas engines are available that produce NO_x levels as low as 50 ppmv @ 15 percent O_2 (dry basis).

Applications

Reciprocating engines are well suited to a variety of distributed generation applications. Industrial, commercial, and institutional facilities in the U.S. and Europe for power generation and CHP. Reciprocating engines start quickly, follow load well, have good part load efficiencies, and generally have high reliabilities. In many cases, multiple reciprocating engine units further increase overall plant capacity and availability. Reciprocating engines have higher electrical efficiencies than gas turbines of comparable size, and thus lower fuel-related operating costs. In addition, the first costs of reciprocating engine gensets are generally lower than gas turbine gensets up to 3-5 MW in size. Reciprocating engine maintenance costs are generally higher than comparable gas turbines, but the maintenance can often be handled by in-house staff or provided by local service organizations.

Potential distributed generation applications for reciprocating engines include standby, peak shaving, grid support, and CHP applications in which hot water, low pressure steam, or waste-heat-fired absorption chillers are required. Reciprocating engines are also used extensively as direct mechanical drives in applications such as water pumping, air and gas compression and chilling/refrigeration.

Combined Heat and Power

While the use of reciprocating engines is expected to grow in various distributed generation applications, the most prevalent on-site generation application for natural gas SI engines has traditionally been CHP, and this trend is likely to continue. The economics of natural gas engines in on-site generation applications is enhanced by effective use of the thermal energy contained in the exhaust gas and cooling systems, which generally represents 60 to 70 percent of the inlet fuel energy.

There are four sources of usable waste heat from a reciprocating engine: exhaust gas, engine jacket cooling water, lube oil cooling water, and turbocharger cooling. Recovered heat is in the form of hot water or low pressure steam (<30 psig). The high temperature exhaust can generate medium pressure steam (up to about 150 psig), but the hot exhaust gas contains only about one half of the available thermal energy from a reciprocating engine. Some industrial CHP applications use the engine exhaust gas directly for process drying. Generally, the hot water and low pressure steam produced by reciprocating engine CHP systems is appropriate for low temperature process needs, space heating, potable water heating, and to drive absorption chillers providing cold water, air conditioning or refrigeration.

There are many engine-based CHP systems operating in the United States in a variety of applications including universities, hospitals, water treatment facilities, industrial facilities, and commercial and residential buildings. Facility capacities range from 30 kW to 30 MW, with many larger facilities comprised of multiple units. Spark ignited engines fueled by natural gas or other gaseous fuels represent 84 percent of the installed reciprocating engine CHP capacity.

Thermal loads most amenable to engine-driven CHP systems in commercial/institutional buildings are space heating and hot water requirements. The simplest thermal load to supply is hot water. The primary applications for CHP in the commercial/institutional and residential sectors are those building types with relatively high and coincident electric and hot water demand such as colleges and universities, hospitals and nursing homes, multifamily residential buildings, and lodging. If space heating needs are incorporated, office buildings, certain warehousing and mercantile/service applications can be economic applications for CHP. Technology development efforts targeted at heat activated cooling/refrigeration and thermally regenerated desiccants expand the application of engine-driven CHP by increasing the thermal energy loads in certain building types. Use of CHP thermal output for absorption cooling and/or desiccant dehumidification could increase the size and improve the economics of CHP systems in existing CHP markets such as schools, multifamily residential buildings, lodging, nursing homes and hospitals. Use of these advanced technologies in applications such as restaurants, supermarkets and refrigerated warehouses provides a base thermal load that opens these applications to CHP.

A typical commercial application for reciprocating engine CHP is a hospital or health care facility with a 1 MW CHP system comprised of multiple 200 to 300 kW natural gas engine gensets. The system is designed to satisfy the baseload electric needs of the facility. Approximately 1.6 MW thermal (MW_{th}) of hot water is recovered from engine exhaust and engine cooling systems to provide space heating and domestic hot water to the facility, and to drive absorption chillers for space conditioning during summer months. Overall efficiency of this type of CHP system can exceed 70 percent.

Industry also uses engine-driven CHP in a variety of industrial applications where hot water or low pressure steam is required. A typical industrial application for engine CHP would be a food processing plant with a 2 MW natural gas engine-driven CHP system comprised of multiple 500 to 800 kW engine gensets. The system provides baseload power to the facility and approximately 2.2 MW_{th} low pressure steam for process heating and washdown. Overall efficiency for a CHP system of this type approaches 75 percent.

Technology Description

Basic Engine Processes

There are two primary reciprocating engine designs relevant to stationary power generation applications – the spark ignition Otto-cycle engine and the compression ignition Diesel-cycle engine. The essential mechanical components of the Otto-cycle and Diesel-cycle are the same. Both use a cylindrical combustion chamber in which a close fitting piston travels the length of the cylinder. The piston connects to a crankshaft that transforms the linear motion of the piston into the rotary motion of the crankshaft. Most engines have multiple cylinders that power a single crankshaft.

The primary difference between the Otto and Diesel cycles is the method of igniting the fuel. Spark ignition engines (Otto-cycle) use a spark plug to ignite a pre-mixed air fuel mixture introduced into the cylinder. Compression ignition engines (Diesel-cycle) compress the air introduced into the cylinder to a high pressure, raising its temperature to the auto-ignition temperature of the fuel that is injected at high pressure.

Engines are further categorized by crankshaft speed (rpm), operating cycle (2- or 4-stroke), and whether turbocharging is used. Reciprocating engines are also categorized by their original design purpose – automotive, truck, industrial, locomotive and marine. Hundreds of small-scale stationary power, CHP, irrigation, and chiller applications, use automotive engine models. These are generally low-priced engines due to large production volumes. However, unless conservatively rated, these engines have limited durability. Truck engines have the cost benefit of production volume and are designed for reasonably long life (e.g., one million miles). A number of truck engines are available as stationary engines. Engines intended for industrial use are designed for durability and for a wide range of mechanical drive and electric power applications. Their sizes range from 20 kW up to 6 MW, including industrialized truck engines in the 200 to 600 kW range and industrially applied marine and locomotive engines above 1 MW.

Both the spark ignition and the diesel 4-stroke engines most relevant to stationary power generation applications complete a power cycle in four strokes of the piston within the cylinder:

1. Intake stroke – introduction of air (diesel) or air-fuel mixture (spark ignition) into the cylinder

2. Compression stroke – compression of air or an air-fuel mixture within the cylinder. In diesel engines, the fuel is injected at or near the end of the compression stroke (top dead center or TDC), and ignited by the elevated temperature of the compressed air in the cylinder. In spark ignition engines, the compressed air-fuel mixture is ignited by an ignition source at or near TDC.

3. Power stroke – acceleration of the piston by the expansion of the hot, high pressure combustion gases, and

4. Exhaust stroke – expulsion of combustion products from the cylinder through the exhaust port.

Types of Reciprocating Engines

Natural Gas Spark Ignition Engines – Spark ignition engines use spark plugs, with a high-intensity spark of timed duration, to ignite a compressed fuel-air mixture within the cylinder. Natural gas is the predominant spark ignition engine fuel used in electric generation and CHP applications. Other gaseous and volatile liquid fuels, ranging from landfill gas to propane to gasoline, can be used with the proper fuel system, engine compression ratio and tuning. American manufacturers began to develop large natural gas engines for the burgeoning gas transmission industry after World War II. Smaller engines were developed (or converted from diesel blocks) for gas gathering and other stationary applications as the natural gas infrastructure developed. Natural gas engines for power generation applications are primarily 4-stroke engines available in sizes up to about 5 MW.

Depending on the engine size, one of two ignition techniques ignites the natural gas:

- Open chamber – the spark plug tip is exposed in the combustion chamber of the cylinder, directly igniting the compressed fuel-air mixture. Open chamber ignition is applicable to any engine operating near the stoichiometric air/fuel ratio up to moderately lean mixtures.[2]

[2] Stoichiometric ratio is the chemically correct ratio of fuel to air for complete combustion, i.e., there is no unused fuel or oxygen after combustion.

- Precombustion chamber – a staged combustion process where the spark plug is housed in a small chamber mounted on the cylinder head. This cylinder is charged with a rich mixture of fuel and air, which upon ignition shoots into the main combustion chamber in the cylinder as a high energy torch. This technique provides sufficient ignition energy to light off very lean fuel-air mixtures used in large bore engines.[3]

The simplest natural gas engines operate with natural aspiration of air and fuel into the cylinder (via a carburetor or other mixer) by the suction of the intake stroke. High performance natural gas engines are turbocharged to force more air into the cylinders. Natural gas spark ignition engines operate at modest compression ratios (compared with diesel engines) in the range of 9:1 to 12:1 depending on engine design and turbocharging. Modest compression is required to prevent auto-ignition of the fuel and engine knock, which can cause serious engine damage.[4]

Using high energy ignition technology, very lean fuel-air mixtures can be burned in natural gas engines, lowering peak temperatures within the cylinders and resulting in reduced NO_x emissions. The lean burn approach in reciprocating engines is analogous to dry low-NO_x combustors in gas turbines. All major natural gas engine manufacturers offer lean burn, low emission models and are engaged in R&D to further improve their performance.

Natural gas spark ignition engine efficiencies are typically lower than diesel engines because of their lower compression ratios. However, large, high performance lean burn engine efficiencies approach those of diesel engines of the same size. Natural gas engine efficiencies range from about 28 percent (LHV) for small engines (<50 kW) to 42 percent (LHV) for the largest high performance, lean burn engines. Lean burn engines tuned for maximum efficiency may produce twice the NO_x emissions as the same engine tuned for minimum NO_x. Tuning for low NO_x typically results in a sacrifice of 1 to 1.5 percentage points in electric generating efficiency from the highest level achievable.

Many natural gas spark ignition engines are derived from diesel engines, i.e., they use the same block, crankshaft, main bearings, camshaft, and connecting rods as the diesel engine. However, natural gas spark ignition engines operate at lower brake mean effective pressure (BMEP) and peak pressure levels to prevent knock.[5] Due to the derating effects from lower BMEP, the spark ignition versions of diesel engines often produce only 60 to 80 percent of the power output of the parent diesel. Manufacturers often enlarge cylinder bore about 5 to 10 percent to increase the power, but this is only partial compensation for the derated output. Consequently, the $/kW capital costs of natural gas spark ignition engines are generally higher than the diesel engines from which they were derived. However, by operating at lower cylinder pressure and bearing loads as well as in the cleaner combustion environment of natural gas, spark ignition engines generally offer the benefits of extended component life compared to their diesel parents.

Diesel Engines - Compression ignition diesel are among the most efficient simple-cycle power generation options on the market. Efficiency levels increase with engine size and range from

---

[3] Lean mixture is a mixture of fuel and air in which an excess of air is supplied in relation to the amount needed for complete combustion; similarly, a rich mixture is a mixture of fuel and air in which an excess of fuel is supplied in relation to the amount needed for complete combustion.
[4] Knock is produced by explosive auto-ignition of a portion of the fuel in the cylinder due to compression and heating of the gas mixture ahead of the flame front. The term knock and detonation are often used interchangeably.
[5] Brake mean effective pressure (BMEP) can be regarded as the "average" cylinder pressure on the piston during the power stroke and is a measure of the effectiveness of engine power output or mechanical efficiency.

about 30 percent (HHV) for small high-speed diesels up to 42 to 48 percent (HHV) for the large bore, slow speed engines. High speed diesel engines (1,200 rpm) are available up to about 4 MW in size. Low speed diesels (60 to 275 rpm) are available as large as 65 MW.

Diesel engines typically require compression ratios of 12:1 to 17:1 to heat the cylinder air to a temperature at which the injected fuel will ignite. The quality of fuel injection significantly affects diesel engine operating characteristics, fuel efficiency, and emissions. Fine atomization and good fuel dispersion by the injectors are essential for rapid ignition, ideal combustion and emissions control. Manufacturers are increasingly moving toward electronically controlled, high pressure injection systems that provide more precise calibration of fuel delivery and accurate injection timing.

Depending on the engine and fuel quality, diesel engines produce 5 to 20 times the NOx (on a ppmv basis) of a lean burn natural gas engine. Emergency generators on marine engines often emit over 20 lb n NO_x/MWh and present on road engines emit less than 13 lbs NO_x/MWh. New diesel engines using low sulfur diesel will achieve rates of approximately 0.65 lb NO_x/MWh. Diesel engines also produce assorted heavy hydrocarbons and particulate emissions. However, diesel engines produce significantly less CO than lean burn gas engines. The NO_x emissions from diesels burning heavy oil are typically 25 to 30 percent higher than diesels using distillate oil. Common NO_x control techniques include delayed fuel injection, exhaust gas recirculation, water injection, fuel-water emulsification, inlet air cooling, intake air humidification, and compression ratio and/or turbocharger modifications. In addition, an increasing number of larger diesel engines are equipped with selective catalytic reduction and oxidation catalyst systems for post-combustion emissions reduction.

High speed diesel engines generally require high quality fuel oil with good combustion properties. No. 1 and No. 2 distillate oil comprise the standard diesel fuels. Low sulfur distillate is preferred to minimize SO_2 emissions. High speed diesels are not suited to burning oil heavier than distillate. Heavy fuel oil requires more time for combustion and the combination of high speed and contaminants in lower quality heavy oils cause excessive wear in high speed diesel engines. Many medium and low speed diesels designs burn heavier oils including low grade residual oils or Bunker C oils.

Dual Fuel Engines – Dual fuel engines are diesel compression ignition engines predominantly fueled by natural gas with a small percentage of diesel oil as the pilot fuel. The pilot fuel auto-ignites and initiates combustion in the main air-fuel mixture. Pilot fuel percentages can range from 1 to 15 percent of total fuel input. Dual fuel operation is a combination of Diesel and Otto cycle operation, with reduction in the percentage of pilot fuel used it approaches the Diesel cycle more closely. Most dual fuel engines can be switched back and forth on the fly between dual fuel and 100 percent diesel operation. In general, because of lower diesel oil usage, NO_x, smoke and particulate emissions are lower for dual fuel engines than for straight diesel operation—particularly at full load. Particulate emissions reduce in line with the percentage reduction in diesel oil consumption while the level of NO_x reduction depends on combustion characteristics (see **Emissions** section). However, CO and unburned hydrocarbon emissions are often higher, partly because of incomplete combustion.

There are three basic types of dual fuel engines:

<u>Conventional, low pressure gas injection</u> engines typically require about 5 to 10 percent pilot fuel and may be derated to about 80 to 95 percent of the rated diesel capacity to avoid detonation. The diesel fuel injection system sets the minimum pilot fuel requirement.

Conventional diesel injectors cannot reliably turn down to less than 5 to 6 percent of the full load injection rate. Natural gas input is controlled at each cylinder by injecting gas before the air intake valves open. NO_x emissions of conventional dual fuel engines are generally in the 5 to 8 gm/kWh range (compared to lean burn natural gas engines with NO_x emissions in the 0.7 to 2.5 gm/kWh range).

High pressure gas injection engines attempt to reduce derating by injecting natural gas at very high pressures (3,600 to 5,100 psig) directly into the main combustion chamber as the pilot fuel is injected. However, the parasitic power for gas compression can be as high as 4 to 7 percent of the rated power output – partly offsetting the benefit of reduced derating. This technology has not proved particularly popular because of this issue and the additional equipment costs required for gas injection. Pilot fuel consumption is typically 3 to 8 percent and NO_x emissions are generally in the 5 to 8 gm/kWh range.

Micropilot prechamber engines are similar to spark ignition prechamber engines in that the pilot fuel injected into a prechamber provides a high energy torch that ignites the very lean, compressed fuel air mixture in the cylinder. Leaner mixtures than spark ignition engines are achievable since the energy provided by the diesel-fueled micropilot chamber is higher than that obtained with a spark ignition prechamber. Micropilot dual fuel engines with 1 percent pilot fuel can operate at or close to the diesel engine's compression ratio and BMEP, so little, if any, derating occurs. In this case the high power density and low $/kW cost advantage of the original diesel engine are retained and engine efficiency at 75 to 100 percent load is close to that of the 100 percent diesel engine. NO_x and other emissions are comparable to those of lean burn spark ignition prechamber engines (NO_x emissions in the 0.7 to 2.5 gm/kWh range). These engines must be equipped with conventional diesel fuel injectors in order to operate on 100 percent diesel.

Several independent developers and engine manufacturers are testing and commercializing dual fuel retrofit kits for converting existing diesel engines to dual fuel operation. The level of sophistication of these kits varies widely and some require major engine modifications. Derating, efficiencies, and emissions also vary widely and have yet to be fully tested or certified. However, dual fuel conversions are unlikely to be as low in emissions as dedicated natural gas engines. In addition, manufacturers may not honor warrantees on an engine that has been retrofitted by an independent third party.

Engine Speed Classifications – Reciprocating engines are classified as high-, medium-, or low-speed. **Table 1** presents the standard speed ranges in each class and the types and sizes of engines available. Engine driven electric generators typically must run at fixed (or synchronous) speeds to maintain a constant 50 or 60 Hertz (Hz) output, setting the engine speed needed within the classifications (i.e., a 60 Hz generator driven by a high speed engine would require engine speeds of 1200, 1800 or 3600 rpm versus a 50 Hz generator which requires engine speeds of 1000, 1500 or 3000 rpm)

Table 1. Reciprocating Engine Types by Speed (Available MW Ratings)

Speed Classification	Engine Speed, rpm	Stoic/ Rich Burn, Spark Ignition[6]	Lean Burn, Spark Ignition	Dual Fuel	Diesel
High Speed	1000-3600	0.01 – 1.5 MW	0.15 - 3.0 MW	1.0 - 3.5 MW[7]	0.01 – 3.5 MW
Medium Speed	275-1000	None	1.0 - 6.0 MW	1.0 – 25 MW	0.5 – 35 MW
Low Speed	58-275	None	None	2.0 – 65 MW	2 – 65 MW

Source: SFA Pacific, Inc.

Engine power output is proportional to engine speed, affording high speed engines the highest output per unit of displacement (cylinder size) and the highest power density. Consequently, high speed engines generally have the lowest $/kW production costs of the three types. The cost benefits of high speed engines must be weighed against other factors. Smaller high speed engines tend to have lower efficiencies than large bore, lower speed engines due in part to the higher surface area to volume ratio for small cylinders resulting in slightly higher heat losses. In addition, higher speed engines tend to have higher wear rates, resulting in shorter periods between minor and major overhauls. These factors are often less important than capital costs for limited duty cycle applications.

Medium speed stationary power engines are largely derived from marine and locomotive engines. Medium speed engines are higher in cost, but generally higher in efficiency than high speed engines. Because of their massive physical size and speed-related power reduction, low speed engines are increasingly being displaced by medium and high speed engines as the primary choice for stationary power applications.

Load Service Ratings – Reciprocating engine manufacturers typically assign three power ratings to engines depending on the intended load service:

- Standby - continuous full or cycling load for a relatively short duration (usually less than 100 hours) – *maximum power output rating*
- Prime – continuous operation for an unlimited time (except for normal maintenance shutdowns), but with regular variations in load – *80 to 85 percent of the standby rating*
- Baseload – continuous full-load operation for an unlimited time (except for normal maintenance shutdowns) – *70 to 75 percent of the standby rating.*

Design Characteristics

The features that have made reciprocating engines a leading prime mover for CHP and other distributed generation applications include:

Size range:	Reciprocating engines are available in sizes from 10 kW to over 5 MW.

[6] Stoichiometric or rich burn combustion is required for the use of 3-way catalytic converters for emissions control.
[7] Micropilot, prechamber dual fuel engines

Thermal output:	Reciprocating engines can produce hot water and low pressure steam.
Fast start-up:	The fast start-up capability of reciprocating engines allows timely resumption of the system following a maintenance procedure. In peaking or emergency power applications, reciprocating engines can quickly supply electricity on demand.
Black-start capability:	In the event of an electric utility outage, reciprocating engines requires minimal auxiliary power requirements. Generally only batteries are required.
Availability:	Reciprocating engines have typically demonstrated availability in excess of 95 *percent* in stationary power generation applications.
Part-load operation:	The high part-load efficiency of reciprocating engines ensures economical operation in electric load following applications.
Reliability and life:	Reciprocating engines have proven to be reliable power generators given proper maintenance.
Emissions:	Diesel engines have relatively high emissions levels of NO_x and particulates. However, natural gas spark ignition engines have improved emissions profiles.

Performance Characteristics

Electrical Efficiency

Table 2 summarizes performance characteristics for typical commercially available natural gas spark ignition engine CHP systems over a 100 kW to 5 MW size range. This size range covers the majority of the market applications for engine-driven CHP. Heat rates and efficiencies shown were taken from manufacturers' specifications and industry publications. Available thermal energy was taken directly from vendor specifications or, if not provided, calculated from published engine data on engine exhaust temperatures and engine jacket and lube system coolant flows. CHP thermal recovery estimates are based on producing hot water for process or space heating needs. As shown in the table, 50 to 60 percent of the waste heat from engine systems is recovered from jacket cooling water and lube oil cooling systems at a temperature too low to produce steam. This feature is generally less critical in commercial/institutional applications where it is more common to have hot water thermal loads. Steam can be produced from the exhaust heat if required (maximum pressure of 150 psig), but if no hot water is needed, the amount of heat recovered from the engine is reduced and total CHP system efficiency drops accordingly.

The data in the table show that electrical efficiency increases as engine size becomes larger. As electrical efficiency increases, the absolute quantity of thermal energy available to produce useful thermal energy decreases per unit of power output, and the ratio of power to heat for the CHP system generally increases. A changing ratio of power to heat impacts project economics and may affect the decisions that customers make in terms of CHP acceptance, sizing, and the desirability of selling power.

Table 2. Gas Engine CHP - Typical Performance Parameters*

Cost & Performance Characteristics[8]	System 1	System 2	System 3	System 4	System 5
Baseload Electric Capacity (kW)	100	300	800	3,000	5,000
Total Installed Cost (2007 $/kW)[9]	$2,210	$1,940	$1,640	$1,130	$1,130
Electric Heat Rate (Btu/kWh), HHV[10]	12,000	9,866	9,760	9,492	8,758
Electrical Efficiency (percent), HHV	28.4%	34.6%	35.0%	36.0%	39.0%
Engine Speed (rpm)	1800	1800	1800	900	720
Fuel Input (MMBtu/hr)	1.20	4.93	9.76	28.48	43.79
Required Fuel Gas Pressure (psig)	<3	<3	<3	43	65
CHP Characteristics					
Exhaust Flow (1000 lb/hr)	1.4	6.3	12.1	48.4	67.1
Exhaust Temperature (Fahrenheit)	1,060	939	909	688	698
Heat Recovered from Exhaust (MMBtu/hr)	0.28	1.03	1.85	4.94	7.01
Heat Recovered from Cooling Jacket (MMBtu/hr)	0.33	1.13	2.45	4.37	6.28
Heat Recovered from Lube System (MMBtu/hr)	0.00	0.00	0.00	1.22	1.94
Total Heat Recovered (MMBtu/hr)	0.61	2.16	4.30	10.53	15.23
Total Heat Recovered (kW)	179	632	1,260	3,084	4,463
Form of Recovered Heat	Hot H_20	Hot H_20	Hot H_20	Hot H_20	Hot H_20
Total Efficiency (percent)[11]	79%	78%	79%	73%	74%
Thermal Output/Fuel Input (percent)	51%	44%	44%	37%	35%
Power/Heat Ratio[12]	0.56	0.79	0.79	0.97	1.12
Net Heat Rate (Btus/kWh)[13]	4,383	4,470	4,385	5,107	4,950
Effective Electrical Efficiency[14]	0.78	0.76	0.78	0.67	0.69

* *For typical systems commercially available in 2007*

Source: EEA/ICF

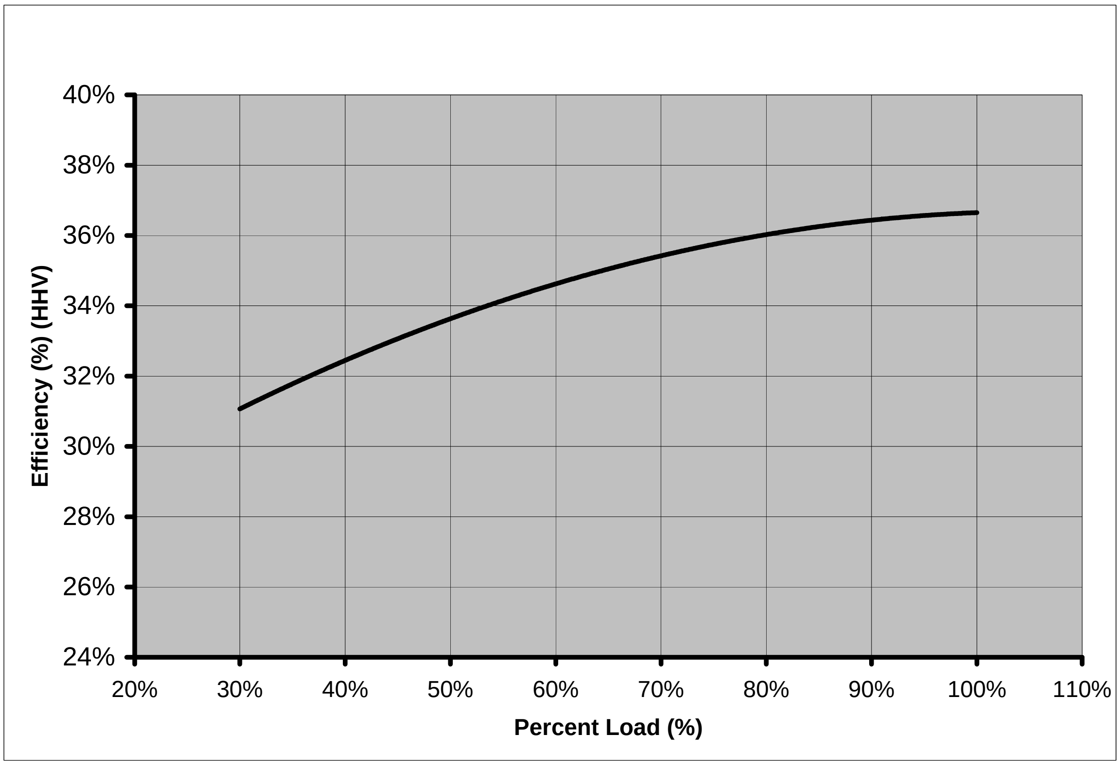
Part Load Performance

In power generation and CHP applications, reciprocating engines generally drive synchronous generators at constant speed to produce steady alternating current (AC) power. As load is reduced, the heat rate of spark ignition engines increases and efficiency decreases. **Figure 1**

[8] Characteristics for "typical" commercially available natural gas engine gensets. Data based on: IPower ENI85 – 85 kW; GE Jenbacher JMS 312 GS-N.L – 625 kW; GE Jenbacher JMS 320 GS-N.L – 1050 kW; Caterpiller G3616 LE – 3 MW; Wartsila 5238 LN - 5 MW; Energy use and exhaust flows normalized to nominal system sizes.
[9] Installed costs based on vendor quote or on CHP system producing hot water from exhaust heat recovery (280 F exhaust from heat recovery heat exchanger), and jacket and lube system cooling
[10] All engine manufacturers quote heat rates in terms of the lower heating value (LHV) of the fuel. However the purchase price of fuels on an energy basis is typically measured on a higher heating value basis (HHV). For natural gas, the average heat content of natural gas is 1030 Btu/kWh on an HHV basis and 930 Btu/kWh on an LHV basis – or about a 10% difference.
[11] Total CHP Efficiency = (net electric generated + net thermal energy recovered)/total engine fuel input
[12] Power/Heat Ratio = (CHP electric power output (Btus))/useful thermal output (Btus)
[13] Net Heat Rate = (Total fuel input to the CHP system - the fuel that would be normally used to generate the same amount of thermal output as the CHP system thermal output assuming an efficiency of 80%)/CHP electric output (kW).
[14] Effective Electrical Efficiency = (CHP electric power output)/(Total fuel into CHP system – total heat recovered/0.8); Equivalent to 3412 Btu/kWh/Net Heat Rate

shows the part load efficiency curve for a typical lean burn natural gas engine. The efficiency at 50 percent load is approximately 8 to 10 percent less than full load efficiency. As the load decreases further, the curve becomes somewhat steeper. While gas engines compare favorably to gas turbines, which typically experience efficiency decreases of 15 to 25 percent at half load conditions, multiple engines may be preferable to a single large unit to avoid efficiency penalties where significant load reductions are expected on a regular basis. Diesel engines exhibit even more favorable part load characteristics than spark ignition engines. The efficiency curve for diesel engines is comparatively flat between 50 and 100 percent load.

Figure 1. Part Load Efficiency Performance



Source: Caterpillar, EEA/ICF

Effects of Ambient Conditions on Performance

Reciprocating engines are generally rated at ISO conditions of 77°F and 0.987 atmospheres (1 bar) pressure. Like gas turbines, reciprocating engine performance – both output and efficiency – degrades as ambient temperature or site elevation increases. While the effect on gas turbines can be significant, it is less so on engines. Reciprocating engine efficiency and power are reduced by approximately 4 *percent* per 1,000 feet of altitude above 1,000 feet, and about 1 *percent* for every 10°F above 77°F.

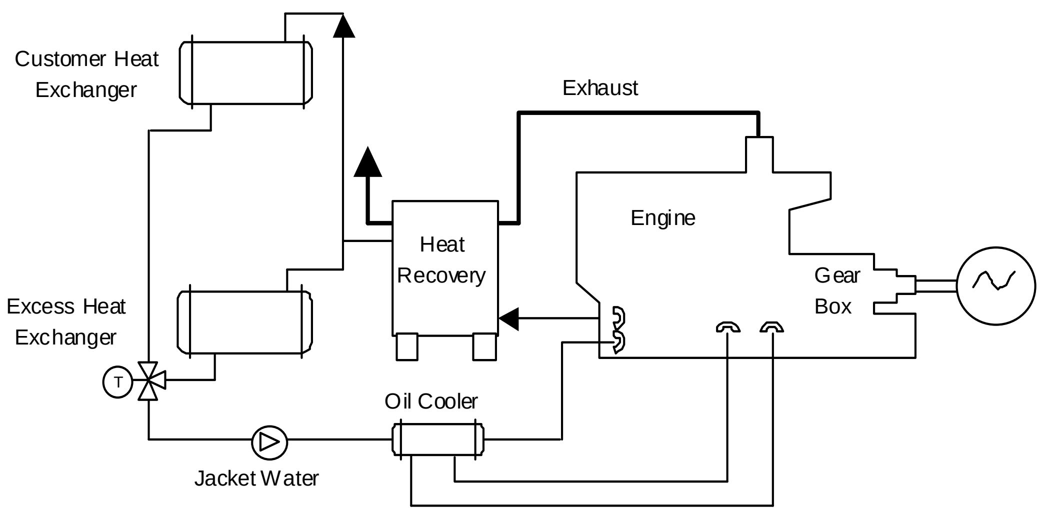
Heat Recovery

The economics of engines in on-site power generation applications often depend on effective use of the thermal energy contained in the exhaust gas and cooling systems, which generally represents 60 to 70 percent of the inlet fuel energy. Most of the waste heat is available in the engine exhaust and jacket coolant, while smaller amounts can be recovered from the lube oil cooler and the turbocharger's intercooler and aftercooler (if so equipped). The most common use of this heat is to generate hot water or low pressure steam for process use or for space heating, process needs, domestic hot water or absorption cooling. However, the engine exhaust gases can also be used as a source of direct energy for drying or other direct heat processes.

Heat in the engine jacket coolant accounts for up to 30 percent of the energy input and is capable of producing 200 to 210°F hot water. Some engines, such as those with high pressure or ebullient cooling systems, can operate with water jacket temperatures up to 265°F. Engine exhaust heat represents from 30 to 50 percent of the available waste heat. Exhaust temperatures of 850 to 1200°F are typical. By recovering heat in the cooling systems and exhaust, approximately 70 to 80 percent of the fuel's energy can be effectively utilized to produce both power and useful thermal energy..

Closed-loop cooling systems - The most common method of recovering engine heat is the closed-loop cooling system as shown in **Figure 2**. These systems are designed to cool the engine by forced circulation of a coolant through engine passages and an external heat exchanger. An excess heat exchanger transfers engine heat to a cooling tower or radiator when there is excess heat generated. Closed-loop water cooling systems can operate at coolant temperatures from 190 to 250°F. Depending on the engine and CHP system's requirements, the lube oil cooling and turbocharger aftercooling may be either separate or part of the jacket cooling system.

Figure.2. Closed-Loop Heat Recovery System



Ebullient Cooling Systems - Ebullient cooling systems cool the engine by natural circulation of a boiling coolant through the engine. This type of cooling system is typically used in conjunction with exhaust heat recovery for production of low-pressure steam. Cooling water is introduced at the bottom of the engine where the transferred heat begins to boil the coolant generating two-phase flow. The formation of bubbles lowers the density of the coolant, causing a natural circulation to the top of the engine.

The coolant at the engine outlet is maintained at saturated steam conditions and is usually limited to 250°F and a maximum of 15 psig. Inlet cooling water is also near saturation conditions and is generally 2 to 3°F below the outlet temperature. The uniform temperature throughout the coolant circuit extends engine life and contributes to improved combustion efficiencies.

Exhaust Heat Recovery – Exhaust heat is typically used to generate hot water to about 230°F or low-pressure steam (up to 150 psig). Only a portion of the exhaust heat can be recovered since exhaust gas temperatures are generally kept above temperature thresholds to prevent the corrosive effects of condensation in the exhaust piping. For this reason, most heat recovery units are designed for a 250 to 350°F exhaust outlet temperature.

Exhaust heat recovery can be independent of the engine cooling system or coupled with it. For example, hot water from the engine cooling can be used as feedwater or feedwater preheat to the exhaust recovery unit. In a typical district heating system, jacket cooling, lube oil cooling, single stage aftercooling and exhaust gas heat recovery are all integrated for steam production.

Performance and Efficiency Enhancements

BMEP and Engine Speed

Engine power is related to engine speed and the BMEP during the power stroke. BMEP can be regarded as an "average" cylinder pressure on the piston during the power stroke, and is a measure of the effectiveness of engine power output or mechanical efficiency. Engine manufacturers often include BMEP values in their product specifications. Typical BMEP values are as high as 230 psig for large natural gas engines and 350 psig for diesel engines. Corresponding peak combustion pressures are about 1,750 psig and 2,600 psig respectively. High BMEP levels increase power output, improve efficiency, and result in lower specific costs ($/kW).

BMEP can be increased by raising combustion cylinder air pressure through increased turbocharging, improved after-cooling, and reduced pressure losses through improved air passage design. These factors all increase air charge density and raise peak combustion pressures, translating into higher BMEP levels. However, higher BMEP increases thermal and pneumatic stresses within the engine, and proper design and testing is required to ensure continued engine durability and reliability.

Turbocharging

Essentially all modern engines above 300 kW are turbocharged to achieve higher power densities. A turbocharger is basically a turbine-driven intake air compressor. The hot, high velocity exhaust gases leaving the engine cylinders power the turbine. Very large engines typically are equipped with two turbochargers. On a carbureted engine, turbocharging forces more air and fuel into the cylinders, increasing engine output. On a fuel injected engine, the mass of fuel injected must be increased in proportion to the increased air input. Cylinder pressure and temperature normally increase as a result of turbocharging, increasing the tendency for detonation for both spark ignition and dual fuel engines and requiring a careful balance between compression ratio and turbocharger boost level. Turbochargers normally boost inlet air pressure on a 3:1 to 4:1 ratio. A wide range of turbocharger designs and models are used. Heat exchangers (called aftercoolers or intercoolers) are often used on the discharge air from the turbocharger to keep the temperature of the air to the engine under a specified limit. Intercooling on forced induction engines improves volumetric efficiency by increasing the density of intake air to the engine (i.e. cold air charge from intercooling provides denser air for combustion thus allowing more fuel and air to be combusted per engine stroke increasing the output of the engine).

Capital Cost

This section provides typical study estimates for the installed cost of natural gas spark-ignited, reciprocating engine-driven generators in CHP applications. Capital costs (equipment and installation) are estimated for the five typical engine genset systems ranging from 100 kW to 5 MW for each configuration. These are “typical” budgetary price levels; it should also be noted that installed costs can vary significantly depending on the scope of the plant equipment, geographical area, competitive market conditions, special site requirements, emissions control requirements, prevailing labor rates, and whether the system is a new or retrofit application.

In general, engine gensets do not show the economies of scale that are typical when costing industrial equipment of different sizes. Smaller genset packages are typically less costly on a unit cost basis ($/kW) than larger gensets. Smaller engines typically run at a higher RPM than larger engines and often are adapted from higher volume production runs from other markets such as automotive or truck engines. These two factors combine to make the engine package costs lower than the larger, slow-speed engines.

The basic genset package consists of the engine connected directly to a generator without a gearbox. In countries where 60 Hz power is required, the genset operates at multiples of 60 – typically 1800 rpm for smaller engines and 900 or 720 rpm for the large engines. In areas where 50 Hz power is used such as Europe and Japan, the engines run at speeds that are multiples of 50 – typically 1500 rpm for the small engines. The smaller engines are skid mounted with a basic control system, fuel system, radiator, fan, and starting system. Some smaller packages come with an enclosure, integrated heat recovery system, and basic electric paralleling equipment. The cost of the basic engine genset package plus the costs for added systems needed for the particular application comprise the total equipment cost. The total plant cost consists of total equipment cost plus installation labor and materials (including site work), engineering, project management (including licensing, insurance, commissioning and startup), and financial carrying costs during the 6 to 18 month construction period.

Table 3 provides cost estimates for combined heat and power applications. The CHP system is assumed to produce hot water, although the multi-megawatt size engines are capable of producing low-pressure steam. The heat recovery equipment consists of the exhaust silencer that extracts heat from the exhaust system, process heat exchanger for extracting heat from the engine jacket coolant, circulation pump, control system, and piping. These cost estimates include interconnection and paralleling. The package costs are intended to reflect a generic representation of popular engines in each size category. The engines all have low emission, lean-burn technology with the exception of the 100 kW system, which is a rich burn engine that would require a three way catalyst in most urban installations. The interconnect/electrical costs reflect the costs of paralleling a synchronous generator, though many 100 kW packages available today use induction generators that are simpler and less costly to parallel.[15] Labor/materials represent the labor cost for the civil, mechanical, and electrical work and materials such as ductwork, piping, and wiring. Project and construction management also includes general contractor markup and bonding and performance guarantees. Contingency is assumed to be 5 percent of the total equipment cost in all cases.

[15] *Reciprocating Engines for Stationary Power Generation: Technology, Products, Players, and Business Issues,* GRI, Chicago, IL and EPRIGEN, Palo Alto, CA: 1999. GRI-99/0271, EPRI TR-113894.

Table 3. Estimated Capital Cost for Typical Gas Engine Generators in Grid Interconnected, Combined Heat and Power Application (2007 $/kW)

Cost Component	System 1	System 2	System 3	System 4	System 5
Nominal Capacity (kW)	100	500	1000	3000	5000
Costs ($/kW)					
Equipment					
Gen Set Package	$1,000	$880	$760	$520	$590
Heat Recovery	$110	$240	$190	$80	$50
Interconnect/Electrical	$260	$60	$40	$30	$20
Total Equipment	$1,370	$1,180	$990	$630	$660
Labor/Materials	$340	$300	$250	$240	$250
Total Process Capital	$1,710	$1,480	$1,240	$870	$910
Project and Construction Management	$200	$180	$150	$90	$70
Engineering and Fees	$200	$180	$150	$90	$70
Project Contingency	$70	$60	$50	$30	$30
Project Financing (interest during construction)	$30	$40	$50	$50	$50
Total Plant Cost ($/kW)	$2,210	$1,940	$1,640	$1,130	$1,130

Source: EEA/ICF

Maintenance

Maintenance costs vary with type, speed, size and numbers of cylinders of an engine and typically include:

- Maintenance labor
- Engine parts and materials such as oil filters, air filters, spark plugs, gaskets, valves, piston rings, electronic components, etc. and consumables such as oil.
- Minor and major overhauls.

Maintenance can be either done by in-house personnel or contracted out to manufacturers, distributors or dealers under service contracts. Full maintenance contracts (covering all recommended service) generally cost between 0.7 to 2.0 cents/kWh depending on engine size, speed and service. Many service contracts now include remote monitoring of engine performance and condition and allow for predictive maintenance. Service contract rates typically are all-inclusive, including the travel time of technicians on service calls.

Recommended service is comprised of routine short interval inspections/adjustments and periodic replacement of engine oil and filter, coolant and spark plugs (typically 500 to 2,000 hours). An oil analysis is part of most preventative maintenance programs to monitor engine wear. A top-end overhaul is generally recommended between 8,000 and 30,000 hours of operation (see **Table 4**) that entails a cylinder head and turbocharger rebuild. A major overhaul is performed after 30,000 to 72,000 hours of operation and involves piston/liner replacement, crankshaft inspection, bearings and seals (**Table 4**).

Table 4. Representative Overhaul Intervals for Natural Gas Engines in Baseload Service

	Time Between Overhauls – (Thousand Operating Hours)				
Engine Speed	720 rpm	900 rpm	1200 rpm	1500 rpm	1800 rpm
Minor Overhaul	> 30	15 - 36	24 – 36	10 - 20	8 - 15
Major Overhaul	> 60	40 - 72	48 - 60	30 – 50	30 - 36

Source: SFA Pacific, Inc.

Maintenance costs presented in **Table 5** are based on engine manufacturer estimates for service contracts consisting of routine inspections and scheduled overhauls of the engine generator set. Costs are based on 8,000 annual operating hours expressed in terms of annual electricity generation.

Table 5. Typical Natural Gas Engine Maintenance Costs*

Maintenance Costs[16]	System 1	System 2	System 3	System 4	System 5
Electricity Capacity, kW	100	300	800	3000	5000
Variable (service contract), 2007 $/kWh	0.02	0.015	0.012	0.01	0.009
Variable (consumables), 2007 $/kWh	0.00015	0.00015	0.00015	0.00015	0.00015
Fixed, 2007 $/kW-yr	15	7	5	2	1.5
Fixed, 2007 $/kWh @ 8000 hrs/yr	0.0019	0.0009	0.0006	0.0003	0.0002
Total O&M Costs, 2007 $/kWh	0.022	0.016	0.013	0.010	0.009

* Typical maintenance costs for gas engine gensets 2007
Source: EEA/ICF

Fuels

Spark ignition engines operate on a variety of alternative gaseous fuels including:

- Liquefied petroleum gas (LPG) – propane and butane mixtures
- Sour gas - unprocessed natural gas as it comes directly from the gas well
- Biogas – any of the combustible gases produced from biological degradation of organic wastes, such as landfill gas, sewage digester gas, and animal waste digester gas

[16] Maintenance costs presented in **Table 5** are based on 8,000 operating hours expressed in terms of annual electricity generation. Fixed costs are based on an interpolation of manufacturers' estimates. The variable component of the O&M cost represents the inspections and overhaul procedures that are normally conducted by the prime mover original equipment manufacturer through a service agreement usually based on run hours.

- Industrial waste gases – flare gases and process off-gases from refineries, chemical plants and steel mill
- Manufactured gases – typically low- and medium-Btu gas produced as products of gasification or pyrolysis processes.

Factors that impact the operation of a spark ignition engine with alternative gaseous fuels include:

- Volumetric heating value – Since engine fuel is delivered on a volume basis, fuel volume into the engine increases as heating value decreases, requiring engine derating on fuels with very low Btu content. Derating is more pronounced with naturally aspirated engines, and depending on air requirements, turbocharging partially or totally compensates.
- Autoignition characteristics and detonation tendency
- Contaminants that may impact engine component life or engine maintenance, or result in air pollutant emissions that require additional control measures.
- Hydrogen-containing fuels may require special measures (generally if hydrogen content by volume is greater than 5 percent) because of hydrogen's unique flammability and explosion characteristics.

Table 6 presents representative constituents of some of the alternative gaseous fuels compared to natural gas. Industrial waste and manufactured gases are not included in the table because their compositions vary widely depending on their source. They typically contain significant levels of H_2 and/or CO. Other common constituents are CO_2, water vapor, one or more light hydrocarbons, and H_2S or SO_2.

Table 6. Major Constituents of Gaseous Fuels

	Natural Gas	LPG	Digester Gas	Landfill Gas
Methane, CH_4, (percent)	80 – 97	0	35 – 65	40 – 60
Ethane, C_2H_6, (percent)	3 – 15	0 – 2	0	0
Propane, C_3H_8, (percent)	0 – 3	75 - 97	0	0
Butane, C_4H_{10}, (percent)	0 – 0.9	0 - 2	0	0
Higher C_xH_x, (percent)	0 – 0.2	0 - 20[17]	0	0
CO_2, (percent)	0 – 1.8	0	30 – 40	40 - 60
N_2, (percent)	0 – 14	0	1 - 2	0 - 13
H_2, (percent)	0 – 0.1	0	0	0
LHV, (Btu/scf)	830 - 1075	2500	300 - 600	350 - 550

Source: SFA Pacific, Inc.; North American Combustion Handbook

Contaminants are a concern with many waste fuels, specifically acid gas components (H_2S, halogen acids, HCN; ammonia; salts and metal-containing compounds; organic halogen-, sulfur-, nitrogen-, and silicon-containing compounds); and oils. In combustion, halogen and sulfur compounds form halogen acids, SO_2, some SO_3 and possibly H_2SO_4 emissions. The acids can

[17] High levels of heavier hydrocarbons are found in LPG derived from refinery processing

also corrode downstream equipment. A substantial fraction of any fuel nitrogen oxidizes into NO_x in combustion. To prevent corrosion and erosion of components, solid particulates must be kept to very low concentrations. Various fuel scrubbing, droplet separation and filtration steps will be required if any fuel contaminant levels exceed manufacturers specifications. Landfill gas in particular often contains chlorine compounds, sulfur compounds, organic acids and silicon compounds, which dictate pretreatment.

Once treated and acceptable for use in the engine, emissions performance profiles on alternative fuels are similar to natural gas engine performance. Specifically, the low emissions ratings of lean burn engines can usually be maintained on alternative fuels.

LPG

LPG is composed primarily of propane and/or butane. Propane used in natural gas engines, requires retarding of ignition timing and other appropriate adjustments. LPG often serves as a back-up fuel where there is a possibility of interruption in the natural gas supply. LPG is delivered as a vapor to the engine. LPG's use is limited in high-compression engines because of its relatively low octane number. In general, LPG for engines contains 95 percent propane by volume with an HHV of 2,500 Btu/scf, and with the remaining 5 percent lighter than butane. Off-spec LPG may require cooling to condense out larger volumes of butane or heavier hydrocarbons.

High butane content LPG is recommended only for low compression, naturally aspirated engines. Significantly retarded timing avoids detonation.

Field Gas

Field gas often contains more than 5 percent by volume of heavy ends (butane and heavier), as well as water, salts and H_2S and usually requires some scrubbing before use in natural gas engines. Cooling may be required to reduce the concentrations of butane and heavier components. Field gas usually contains some propane and normally is used in low compression engines (both naturally aspirated and turbocharged). Retarted ignition timing eliminates detonation.

Biogas

Biogases (landfill gas and digester gas) are predominantly mixtures of methane and CO_2 with HHV in the range of 300 to 700 Btu/scf. Landfill gas also contains a variety of contaminants as discussed earlier. Biogases are produced essentially at atmospheric pressure so must be compressed for delivery to the engine. After compression, cooling and scrubbing or filtration are required to remove compressor oil, condensate, and any particulates that may have been entrained in the original gas. Scrubbing with a caustic solution may be required if acid gases are present. Because of the additional requirements for raw gas treatment, biogas powered engine facilities are more costly to build and operate than natural gas-based systems.

Industrial Waste Gases

Industrial waste gases that are common reciprocating engine fuels include refinery gases and process off-gases. Refinery gases typically contain components such as H_2, CO, light hydrocarbons, H_2S, and ammonia, as well as CO_2 and N_2. Process off-gases include a wide variety of compositions. Generally, waste gases are medium- to low-Btu content. Medium-Btu gases generally do not require significant engine derating; low-Btu gases usually require derating.

Depending on their origin and contaminants, industrial gases sometimes require pretreatment comparable to that applied to raw landfill gas. Particulates (e.g., catalyst dust), oils, condensable gases, water, C_4+ hydrocarbons and acid gases may all need to be removed. Process offgases are usually available at pressures of several atmospheres or higher, which are generally satisfactory for delivery to an on-site or nearby reciprocating engine facility.

Availability

Reciprocating engines are maintenance intensive but, they can provide high levels of availability, even in high load factor applications. While natural gas engine availabilities vary with engine type, speed and fuel quality, **Table 7** illustrates typical availability numbers based on a survey of natural gas engine gensets in CHP applications.

Table 7. Availabilities and Outage Rates for Natural Gas Engines

	Gas Engines 80 – 800 kW	Gas Engines >800 kW
Availability Factor (percent)	94.5	91.2
Forced Outage Rate (percent)	4.7	6.1
Scheduled Outage Rate (percent)	2.0	3.5

Source: GRI (Liss, 1999)

The use of multiple units or back-up units at a site can further increase the availability of the overall facility. Some engine manufacturers offer engine exchange programs or other maintenance options that increase the ability to promptly deliver and install replacement units on short notice, typically increasing facility availabilities to greater than 95 percent.

Emissions

Exhaust emissions are the primary environmental concern with reciprocating engines. The primary pollutants are oxides of nitrogen (NO_x), carbon monoxide (CO), and volatile organic compounds (VOCs – unburned, non-methane hydrocarbons). Other pollutants such as oxides of sulfur (SO_x) and particulate matter (PM) are primarily dependent on the fuel used. The sulfur content of the fuel determines emissions of sulfur compunds, primarily SO_2. Engines operating on natural gas or desulfurized distillate oil emit insignificant levels of SO_x. In general, SO_x emissions are an issue only in large, slow speed diesels firing heavy oils. Particulate matter (PM) can be an important pollutant for engines using liquid fuels. Ash and metallic additives in the fuel contribute to PM in the exhaust.

Nitrogen Oxides (NO_x)

NO_x emissions are usually the primary concern with natural gas engines and are a mixture of (mostly) NO and NO_2 in variable composition. In measurement, NO_x is reported as parts per million by volume in which both species count equally (e.g., ppmv at 15 percent O_2, dry). Other common units for reporting NO_x in reciprocating engines are gm/hp-hr and gm/kWhr, or as an output rate such as lbs/hr. Among natural gas engine options, lean burn natural gas engines produce the lowest NO_x emissions directly from the engine. However, rich burn engines can more effectively make use of three way catalysts to produce very low emissions. If lean burn engines must meet extremely low emissions levels, as in California CARB 2007 standards of .07 lb/MWh then selective catalytic reduction must be added. Rich burn engines would qualify for this standard by taking a CHP credit for avoided boiler emissions. In addition, a commercial rich burn engine with cold exhausts gas recirculation and three way catalyst has been tested below the CARB 2007 standard without the CHP credit. Operation at this ultra-low emissions level still in a commercial installation needs further development and refinement of control systems. **Table 8** presents representative NO_x emissions from reciprocating engines without add on controls.

Table 8. Representative NO_x Emissions from Reciprocating Engines (w/o add on controls)

Engines	Fuel	NO_x (ppmv)	NO_x (lb/MWh)
Diesel Engines (high speed & medium speed)[18]	Distillate	450 - 1350	3 - 8
Diesel Engines (high speed & medium speed)[19]	Heavy Oil	900 – 1800	5 - 9
Rich Burn, Spark Ignition, natural gas[20]			0.096
Lean Burn, Spark Ignition, natural gas Engine[21]	Natural Gas	45 - 150	1.25

Source: SFA Pacific, Inc., EEA/ICF

Three mechanisms form NO_x: thermal NO_x, prompt NO_x, and fuel-bound NO_x. The predominant NO_x formation mechanism associated with reciprocating engines is thermal NO_x. Thermal NO_x is the fixation of atmospheric oxygen and nitrogen, which occurs at high combustion temperatures. Flame temperature and residence time are the primary variables that affect thermal NO_x levels. The rate of thermal NO_x formation increases rapidly with flame temperature. Early reactions of nitrogen modules in the combustion air and hydrocarbon radicals from the fuel form prompt NO_x. It forms within the flame and typically is approximately 1 ppm at 15 percent O_2, and is usually much smaller than the thermal NO_x formation. Fuel-bound NO_x forms when the fuel contains nitrogen as part of the hydrocarbon structure. Natural gas has negligible chemically bound fuel nitrogen. Fuel-bound NO_x can be at significant levels with liquid fuels.

The control of peak flame temperature through lean burn conditions has been the primary combustion approach to limiting NO_x formation in gas engines. Diesel engines produce higher combustion temperatures and more NO_x than lean burn gas engines, even though the overall

[18] Efficiency range: 37 to 44% LHV
[19] Efficiency range: 42 to 48% LHV
[20] Efficiency, 31% LHV
[21] Efficiency 40% LHV

diesel engine air/fuel ratio may be very lean. There are three reasons for this: (1) heterogeneous near-stoichiometric combustion; (2) the higher adiabatic flame temperature of distillate fuel; and (3) fuel-bound nitrogen. The diesel fuel is atomized as it is injected and dispersed in the combustion chamber. Combustion largely occurs at near-stoichiometric conditions at the air-droplet and air-fuel vapor interfaces, resulting in maximum temperatures and higher NO_x. In contrast, lean-premixed homogeneous combustion used in lean burn gas engines results in lower combustion temperatures and lower NO_x production.

For any engine there are generally trade-offs between low NO_x emissions and high efficiency. There are also trade-offs between low NO_x emissions and emissions of the products of incomplete combustion (CO and unburned hydrocarbons). There are three main approaches to these trade-offs that come into play depending on regulations and economics. One approach is to control for lowest NO_x accepting a fuel efficiency penalty and possibly higher CO and hydrocarbon emissions. A second option is finding an optimal balance between emissions and efficiency. A third option is to design for highest efficiency and use post-combustion exhaust treatment.

Carbon Monoxide (CO)

CO and VOCs both result from incomplete combustion. CO emissions result when there is inadequate oxygen or insufficient residence time at high temperature. Cooling at the combustion chamber walls and reaction quenching in the exhaust process also contribute to incomplete combustion and increased CO emissions. Excessively lean conditions can lead to incomplete and unstable combustion and high CO levels.

Unburned Hydrocarbons

Volatile hydrocarbons also called volatile organic compounds (VOCs) can encompass a wide range of compounds, some of which are hazardous air pollutants. These compounds are discharged into the atmosphere when some portion of the fuel remains unburned or just partially burned. Some organics are carried over as unreacted trace constituents of the fuel, while others may be pyrolysis products of the heavier hydrocarbons in the gas. Volatile hydrocarbon emissions from reciprocating engines are normally reported as non-methane hydrocarbons (NMHCs). Methane is not a significant precursor to ozone creation and smog formation and is not currently regulated.

Carbon Dioxide (CO_2)

While not considered a pollutant in the ordinary sense of directly affecting health, emissions of carbon dioxide (CO_2) are of concern due to its contribution to global warming. Atmospheric warming occurs since solar radiation readily penetrates to the surface of the planet but infrared (thermal) radiation from the surface is absorbed by the CO_2 (and other polyatomic gases such as methane, unburned hydrocarbons, refrigerants and volatile chemicals) in the atmosphere, with resultant increase in temperature of the atmosphere. The amount of CO_2 emitted is a function of both fuel carbon content and system efficiency. The fuel carbon content of natural gas is 34 lbs carbon/MMBtu; oil is 48 lbs carbon/MMBtu; and (ash-free) coal is 66 lbs carbon/MMBtu.

Emissions Control Options

NO_x control has been the primary focus of emission control research and development in natural gas engines. The following provides a description of the most prominent emission control approaches.

Combustion Process Emissions Control

Control of combustion temperature has been the principal focus of combustion process control in gas engines. Combustion control requires tradeoffs – high temperatures favor complete burn up of the fuel and low residual hydrocarbons and CO, but promote NO_x formation. Lean combustion dilutes the combustion process and reduces combustion temperatures and NO_x formation, and allows a higher compression ratio or peak firing pressures resulting in higher efficiency. However, if the mixture is too lean, misfiring and incomplete combustion occur, increasing CO and VOC emissions.

Lean burn engine technology was developed during the 1980s as a direct response to the need for cleaner burning gas engines. As discussed earlier, thermal NO_x formation is a function of both flame temperature and residence time. The focus of lean burn developments was to lower combustion temperature in the cylinder using lean fuel/air mixtures. Lean combustion decreases the fuel/air ratio in the zones where NO_x is produced so that peak flame temperature is less than the stoichiometric adiabatic flame temperature, therefore suppressing thermal NO_x formation. Most lean burn engines use turbocharging to supply excess air to the engine and produce the homogeneous lean fuel-air mixtures. Lean burn engines generally use 50 to 100 percent excess air (above stoichiometric). The typical emissions rate for lean burn natural gas engines is between 0.5 to 2.0 gm/bhph.

As discussed above, an added performance advantage of lean burn operation is higher output and higher efficiency. Optimized lean burn operation requires sophisticated engine controls to ensure that combustion remains stable and NO_x reduction is maximized while minimizing emissions of CO and VOCs. **Table 9** shows data for a large lean burn natural gas engine that illustrates the tradeoffs between NO_x emissions control and efficiency. At the lowest achievable NO_x levels (45 to 50 ppmv), almost 1.5 percentage points are lost on full rated efficiency.

Table 9. NO_x Emissions versus Efficiency Tradeoffs[22]

Engine Characteristics	Low NO_x	High Efficiency
Capacity (MW)	5.2	5.2
Speed (rpm)	720	720
Efficiency, LHV (percent)	40.7	42.0
Emissions:		
NO_x (gm/kWh) (ppmv @ 15 percent O_2)	0.7 46	1.4 92
CO (gm/kWh) (ppmv @ 15 percent O_2)	3.2 361	2.0 227
NMHC (gm/kWh) (ppmv @ 15 percent O_2)	0.9 61	0.6 39

Combustion temperature can also be controlled to some extent in reciprocating engines by one or more of the following techniques:

- Delaying combustion by retarding ignition or fuel injection
- Diluting the fuel-air mixture with exhaust gas recirculation (EGR), which replaces some of the air and contains water vapor that has a relatively high heat capacity and absorbs some of the heat of combustion.
- Introducing liquid water by direct injection or via fuel oil emulsification – evaporation of the water cools the fuel-air mixture charge.
- Reducing the inlet air temperature with a heat exchanger after the turbocharger or via inlet air humidification.
- Modifying valve timing, compression ratio, turbocharging, and the combustion chamber configuration

Water injection and EGR reduce diesel NO_x emissions 30 to 60 percent from uncontrolled levels. The incorporation of water injection and other techniques to lean burn gas engines is the focus of ongoing R&D efforts with several engine manufacturers and is being pursued as part of the Department of Energy's Advanced Reciprocating Engine Systems (ARES) program. One of the goals of the program is to develop a 45 percent efficient (HHV) medium sized natural gas engine operating at 0.3 lb NOx/MWh (0.1 gm NOx/bhph).

Post-Combustion Emissions Control

There are several types of catalytic exhaust gas treatment processes that are applicable to various types of reciprocating engines.

[22] Based on engine manufacturer's data – Wartsila 18V34SG Prechamber Lean Burn Gas Engine.

Three - Way Catalyst

The catalytic three-way conversion process (TWC) is the basic automotive catalytic converter process that reduces concentrations of all three major criteria pollutants – NO_x, CO and VOCs. The TWC is also called non-selective catalytic reduction (NSCR). NO_x and CO reductions are generally greater than 90 percent, and VOCs are reduced approximately 80 percent in a properly controlled TWC system. Because the conversions of NO_x to N_2 and CO and hydrocarbons to CO_2 and H_2O will not take place in an atmosphere with excess oxygen (exhaust gas must contain less than 0.5 percent O_2), TWCs are only effective with stoichiometric or rich-burning engines. Typical “engine out” NO_x emission rates for a rich burn engine are 10 to 15 gm/bhp-hr. NO_x emissions with TWC control are as low as 0.15 gm/bhp-hr.

Stoichiometric and rich burn engines have significantly lower efficiency than lean burn engines (higher carbon emissions) and only certain sizes (<1.5 MW) and high speeds are available. The TWC system also increases maintenance costs by as much as 25 percent. TWCs are based on noble metal catalysts that are vulnerable to poisoning and masking, limiting their use to engines operated with clean fuels – e.g., natural gas and unleaded gasoline. In addition, the engines must use lubricants that do not generate catalyst poisoning compounds and have low concentrations of heavy and base metal additives. Unburned fuel, unburned lube oil, and particulate matter can also foul the catalyst. TWC technology is not applicable to lean burn gas engines or diesels.

Selective Catalytic Reduction (SCR)

This technology selectively reduces NO_x to N_2 in the presence of a reducing agent. NO_x reductions of 80 to 90 percent are achievable with SCR. Higher reductions are possible with the use of more catalyst or more reducing agent, or both. The two agents used commercially are ammonia (NH_3 in anhydrous liquid form or aqueous solution) and aqueous urea. Urea decomposes in the hot exhaust gas and SCR reactor, releasing ammonia. Approximately 0.9 to 1.0 moles of ammonia is required per mole of NO_x at the SCR reactor inlet in order to achieve an 80 to 90 percent NO_x reduction.

SCR systems add a significant cost burden to the installation cost and maintenance cost of an engine system, and can severely impact the economic feasibility of smaller engine projects. SCR requires on-site storage of ammonia, a hazardous chemical. In addition ammonia can “slip” through the process unreacted, contributing to environmental health concerns.

Oxidation Catalysts

Oxidation catalysts generally are precious metal compounds that promote oxidation of CO and hydrocarbons to CO_2 and H_2O in the presence of excess O_2. CO and non-methane hydrocarbon analyzer (NMHC) conversion levels of 98 to 99 percent are achievable. Methane conversion may approach 60 to 70 percent. Oxidation catalysts are now widely used with all types of engines, including diesel engines. They are being used increasingly with lean burn gas engines to reduce their relatively high CO and hydrocarbon emissions.

Lean –NO_x Catalysts

Lean-NO_x catalysts utilize a hydrocarbon reductant (usually the engine fuel) injected upstream of the catalyst to reduce NO_x. While still under development, it appears that NO_x reduction of 80 percent and both CO and NMHC emissions reductions of 60 percent may be possible. Long-term testing, however, has raised issues about sustained performance of the catalysts. Current lean-NO_x catalysts are prone to poisoning by both lube oil and fuel sulfur. Both precious metal and base metal catalysts are highly intolerant of sulfur. Fuel use can be significant with this technology – the high NO_x output of diesel engines would require approximately 3 percent of the engine fuel consumption for the catalyst system.

Gas Engine Emissions Characteristics

Table 10 shows typical emissions for each of the five gas engine systems. The emissions presented assume available exhaust treatment. System 1, 100 kW engine, is a high speed, rich burn engine. Use of a TWC system with EGR provides NOx emissions of just under 0.1 lb NO_x per MWh. Lean burn systems use an SCR system providing 30 percent emissions reduction. Higher levels of emissions reduction are available up to 90 percent reduction.

With current commercial technology, highest efficiency and lowest NO_x are not achieved simultaneously. Therefore many manufacturers of lean burn gas engines offer different versions of an engine – a low NO_x version and a high efficiency version – based on different tuning of the engine controls and ignition timing. Achieving highest efficiency operation results in conditions that generally produce twice the NO_x as low NO_x versions (e.g., 1.0 gm/bhp-hr versus 0.5 gm/bhp-hr). Achieving the lowest NO_x typically entails sacrifice of 1 to 2 points in efficiency (e.g., 38 percent versus 36 percent). In addition, CO and VOC emissions are higher in engines optimized for minimum NO_x.

Table 10. Gas Engine Emissions Characteristics with Available Exhaust Control Options*

Emissions Characteristics	System 1	System 2	System 3	System 4	System 5
Electricity Capacity (kW)	100	300	1000	3000	5000
Electrical Efficiency (HHV)	28.4%	31.1%	35.0%	36.0%	39.0%
Engine Combustion	Rich	Rich	Lean	Lean	Lean
NO_x, (lb/MWh)	0.10	0.50	1.49	1.52	1.24
CO, (lb/MWh)	0.32	1.87	0.87	0.78	0.75
VOC, (lb/MWh)	0.10	0.47	0.38	0.34	0.22
CO_2, (lb/MWh)	1,404	1,284	1,142	1,110	1,024

* *For typical systems commercially available in 2007.*
Source: EEA/ICF