Catalog of CHP Technologies

Section 2. Technology Characterization – Reciprocating Internal Combustion Engines

U.S. Environmental Protection Agency
Combined Heat and Power Partnership

March 2015

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The September 2017 revision incorporated a new section on packaged CHP systems (Section 7).

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Section 2. Technology Characterization – Reciprocating Internal Combustion Engines

2.1 Introduction

Reciprocating internal combustion engines are a well-established and widely used technology. Worldwide production for reciprocating internal combustion engines is over 200 million units per year.[7] Reciprocating engines include both diesel and spark-ignition configurations. They are important for both transportation and for stationary uses. Their sizes range from fractional horsepower engines to 5-story tall marine propulsion systems weighing over 5 million pounds and producing over 80 megawatts (MW) of power. The long history of technical development and high production levels have contributed to making reciprocating engines a rugged, reliable, and economic choice as a prime mover for CHP applications.

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. Electronic Power Control Modules (PCMs) have made possible more precise control and diagnostic monitoring of the engine process. Stationary engine manufacturers and worldwide engine R&D firms continue to drive advanced engine technology, including accelerating the diffusion of innovative technology and concepts from the automotive market to the stationary market.

The features that have made reciprocating engines a leading prime mover for CHP and other distributed generation applications are summarized in **Table 2-1.**

Table 2-1. Reciprocating Engine Characteristics

Size range	Reciprocating engines are available in sizes from 10 kW to over 18 MW.
Thermal output	Reciprocating engines can produce hot water, low pressure steam, and chilled water (using an absorption chiller).
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 require minimal auxiliary power requirements. Generally only batteries or compressed air 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.

[7] Power Systems Research, EnginLinkTM 2013

2.2 Applications

Reciprocating engines are well suited to a variety of distributed generation applications, and are used throughout industrial, commercial, and institutional facilities 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, having multiple reciprocating engine units further increases 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 upfront costs of reciprocating engine gensets are generally lower than gas turbine gensets in sizes below 20 MW. 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.

2.2.1 Combined Heat and Power

There are over 2,000 active reciprocating engine combined heat and power (CHP) installations in the U.S. providing nearly 2.3 gigawatts (GW) of power capacity[8]. These systems are predominantly spark ignition engines fueled by natural gas and other gaseous fuels (biogas, landfill gas). Natural gas is lower in cost than petroleum based fuels and emissions control is generally more effective using gaseous fuels. Reciprocating engine CHP systems are commonly used in 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, and certain warehousing and mercantile/service applications can be economical 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 already strong CHP markets such as schools, multifamily residential buildings, lodging, nursing homes and hospitals. Use of these advanced technologies in other sectors such as restaurants, supermarkets and refrigerated warehouses provides a base thermal load that opens these sectors to CHP application.

Reciprocating engine CHP systems usually meet customer thermal and electric needs as in the two hypothetical examples below:

- 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

[8] ICF CHP Installation Database. Maintained for Oak Ridge National Laboratory by ICF International. 2013. *http://www.eea-inc.com/chpdata/index.html*

system is designed to satisfy the baseload electric needs of the facility. Approximately 1.6 MW of thermal energy (MW_{th}), in the form of hot water, is recovered from engine exhaust and engine cooling systems to provide space heating and domestic hot water to the facility as well as to drive absorption chillers for space conditioning during summer months. Overall efficiency of this type of CHP system can exceed 70 percent.

- 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.

2.2.2 Emergency/Standby Generators

Reciprocating engine emergency/standby generators are used in a wide variety of settings from residential homes to hospitals, scientific laboratories, data centers, telecommunication equipment, and modern naval ships. Residential systems include portable gasoline fueled spark-ignition engines or permanent installations fueled by natural gas or propane. Commercial and industrial systems more typically use diesel engines. The advantages of diesel engines in standby applications include low upfront cost, ability to store on-site fuel if required for emergency applications, and rapid start-up and ramping to full load. Because of their relatively high emissions of air pollutants, such diesel systems are generally limited in the number of hours they can operate. These systems may also be restricted by permit from providing any other services such as peak-shaving.

2.2.3 Peak Shaving

Engine generators can supply power during utility peak load periods thereby providing benefits to both the end user and the local utility company. The facility can save on peak power charges and the utility can optimize operations and minimize investments in generation, transmission, and distribution that are used only 0-200 hours/year. In a typical utility peak shaving program, a utility will ask a facility to run its on-site generator during the utility's peak load period, and in exchange, the utility will provide the facility with monthly payments.

2.3 Technology Description

2.3.1 Basic 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.

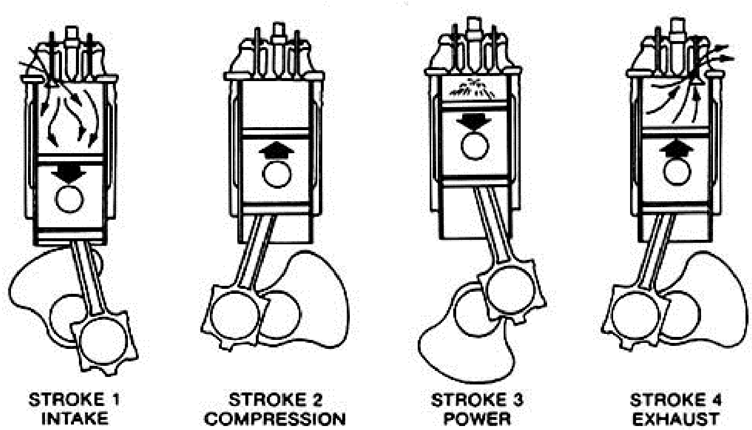
The main difference between the Otto and Diesel cycles is the method of igniting the fuel. Spark ignition engines (Otto-cycle) use a spark plug to ignite a pre-mixed air fuel mixture introduced into the cylinder. Compression ignition engines (Diesel-cycle) compress the air introduced into the cylinder 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 in revolutions per minute (rpm), operating cycle (2- or 4-stroke), and whether turbocharging is used. Reciprocating engines are also categorized by their original design purpose, such as 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 the economies of scale of large production volumes. Truck engines have the cost benefit of production volume and are designed for a 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.

There are 2-cycle engines in stationary power applications, particularly in standby service. However, most spark ignition and the diesel engines relevant to stationary power generation applications complete a power cycle in four strokes of the piston within the cylinder as shown in **Figure 2-1**.

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

Figure 2-1. 4-Stroke Reciprocating Engine Cycle



Source: http://www.globalspec.com/learnmore/motion_controls/engines_components/industrial_engines

2.3.2 Components

2.3.2.1 Engine System

Natural Gas Spark Ignition Engines

Current natural gas engines for power generation 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 (27 percent HHV) for small stoichiometric engines (<100 kW) to over 46 percent LHV (42 percent HHV) for large lean burn engines (> 3 MW).[9,10] 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 up to 80 percent (HHV) can be achieved.

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 18 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 for up to moderately lean mixtures.[11]

- **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.[12]

The simplest natural gas engines operate with a 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

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[9] The exact ratio of air to fuel that is required for complete combustion is called the stoichiometric ratio. If there is less or more air than needed for complete combustion the engine is called rich burn or lean burn respectively.

[10] Most 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). Higher Heating Values are about 6% greater for oil (liquid petroleum products) and 5% for coal.

[11] 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.

[12] 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.

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.

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 can exceed those of diesel engines of the same size. Natural gas engine efficiencies range from about 28 percent (LHV) for small engines (<50 kW) to 46 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.[13] Due to the derating effects from lower BMEP, the spark ignition versions of smaller diesel engines may produce only 60 to 80 percent of the power output of the parent diesel. Manufacturers often enlarge cylinder bore by about 5 to 10 percent to increase the power to levels which meet or exceed their diesel counterparts. The $/kW capital costs of large, high performance natural gas spark ignition engines are typically on a similar level to 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.

2.3.2.2 Diesel Engines

Diesel engines have historically been the most common 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 and also because of the high cost of fuel. 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.

Compression ignition diesel engines are among the most efficient simple-cycle power generation options on the market. Efficiency levels increase with engine size and range from 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 (>=1,000 rpm) are available for up to about 4 MW in size. Low speed diesel

[13] 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.

engines (60 to 275 rpm) are available as large as 80 MW. Medium speed diesel engines (400 – 1000 rpm) are available for up to approximately 17 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 NO_x (on a ppmv basis) of a lean burn natural gas engine. Diesel engines on marine engines often emit over 20 lbs NO_x/MWh and present on road engines emit less than 13 lbs NO_x/MWh. New diesel engines 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. Ultra-low sulfur diesel with sulfur contents of less than 0.15 ppm is now required for the new Tier 4 diesel engines to reduce sulfur emissions. High speed diesel engines 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 diesel designs burn heavier oils including low grade residual oils or Bunker C oils.

2.3.2.3 Dual Fuel Engines

Dual fuel engines are predominantly fueled by natural gas with a small percentage of diesel oil added. There are two main configurations for introducing the gaseous fuel in a dual fuel engine. These engines can be purpose built or conversions of diesel engines. Such engines can be switched to 100 percent diesel operation. Dual fuel engines provide a multi-use functionality. Operation on predominantly cheaper and cleaner burning natural gas allows the engine to be used in CHP and peak shaving applications, while operation on 100 percent diesel allows the engine to also meet the onsite fuel requirements of emergency generators. The dual function adds benefit in applications that have specific emergency generator requirements such as in hospitals or in public buildings.

There are three main configurations for introducing the gaseous and pilot diesel fuel: 1) low pressure injection with the intake air, 2) high pressure injection after the intake air has been compressed by the piston, and 3) micropilot prechamber introduction of the diesel fuel. New dual-fuel engines are offered in oil and gas production markets to reduce operating costs. Dual-fuel retrofits of existing diesel engines are also offered as a means to reduce both operating costs and emissions for extending the hours of use for limited duty engines such as emergency and peaking applications. Dual fuel is not widely used for CHP applications.

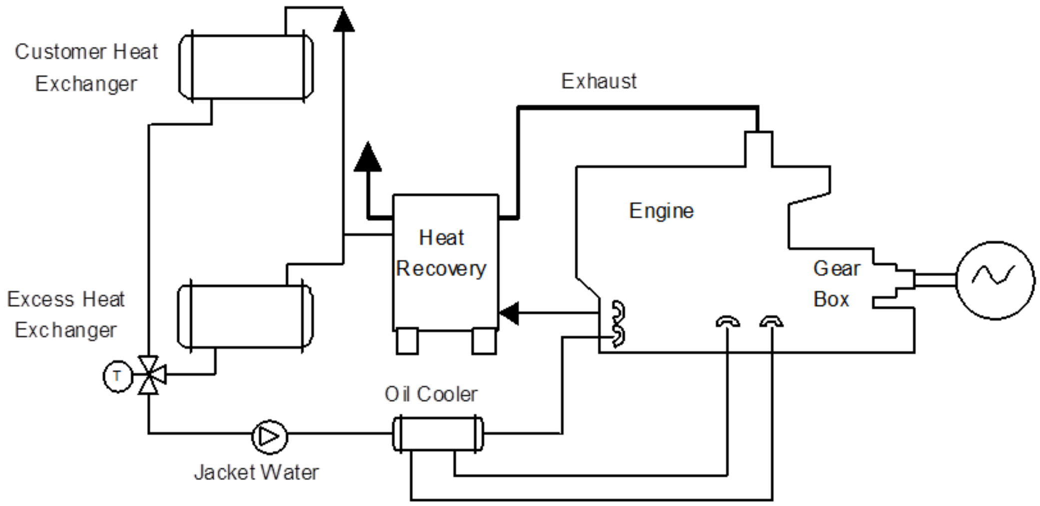
2.3.2.4 Heat Recovery

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). As shown in the previous table, 45 to 55 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 400 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.

Heat in the engine jacket coolant accounts for up to 30 percent of the energy input and is capable of producing 190 to 230 °F hot water. Some engines, such as those with high pressure or ebullient cooling systems, can operate with water jacket temperatures of up to 265°F. Engine exhaust heat represents 30 to 50 percent of the available waste heat. Exhaust temperatures for the example systems range from 720 to 1000°F. By recovering heat in the cooling systems and exhaust, around 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 2-2**. 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 a 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 after-cooling may be either separate or part of the jacket cooling system.

Figure 2-2. 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 of up to about 230°F or steam up to 400 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.

2.4 Performance Characteristics

Table 2-2 summarizes performance characteristics for typical commercially available natural gas spark ignition engine CHP systems over a 100 kW to 9 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.

Most 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.*

The ratings shown are for baseload operation.

Table 2-2. Gas Spark Ignition Engine CHP - Typical Performance Parameters

Cost & Performance Characteristics[14]	System				
	1	2	3	4	5
Baseload Electric Capacity (kW)	100	633	1,121	3,326	9,341
Total Installed Cost in 2013 ($/kW) [15]	$2,900	$2,837	$2,366	$1,801	$1,433
Electrical Heat Rate (Btu/kWh), HHV[16]	12,637	9,896	9,264	8,454	8,207
Electrical Efficiency (%), HHV	27.0%	34.5%	36.8%	40.4%	41.6%
Engine Speed (rpm)	2,500[17]	1,800	1,800	1,500[18]	720
Fuel Input (MMBtu/hr), HHV	1.26	6.26	10.38	28.12	76.66
Required Fuel Gas Pressure (psig)	0.4-1.0	> 1.16	> 1.74	> 1.74	75
CHP Characteristics					
Exhaust Flow (1000 lb/hr)	1.2	7.89	13.68	40.17	120
Exhaust Temperature (Fahrenheit)	1,200	941	797	721	663
Heat Recovered from Exhaust (MMBtu/hr)	0.21	1.48	2	5.03	10
Heat Recovered from Cooling Jacket (MMBtu/hr)	0.46	0.72	1.29	1.63	4.27
Heat Recovered from Lube System (MMBtu/hr)	Incl.	0.27	0.44	1.12	5.0
Heat Recovered from Intercooler (MMBtu/hr)	n/a	0.31	0.59	2.89	7.54
Total Heat Recovered (MMBtu/hr)	0.67	2.78	4.32	10.67	26.81
Total Heat Recovered (kW)	196	815	1,266	3,126	7857

[14] Characteristics are for representative natural gas engine gensets commercially available in 2013. Data based on (1) Tecogen Inverde Ultra 100, (2) GE Jenbacher (GEJ) JMS-312C65; (3) GEJ JMS-416B85, (4) GEJ JMS-620F01, and (5) Wartsila 20V34SG

[15] Details on installed costs are provided later in **Table 2-4.**

[16] 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 measured on a higher heating value basis (HHV). For natural gas, the average heat content is 1030 Btu/scf on an HHV basis and 930 Btu/scf on an LHV basis – a ratio of approximately 0.9 (LHV / HHV).

[17] At rated load. The unit operates at variable speeds from 1,000 to 3,000 rpm, with a peak output of 125 kW while producing 60 Hz power through the inverter.

[18] The unit operates through a gearbox to produce 60 Hz power.

Table 2-2. Gas Spark Ignition Engine CHP - Typical Performance Parameters

Cost & Performance Characteristics[14]	System				
	1	2	3	4	5
Form of Recovered Heat	H_2O	H_2O	H_2O	H_2O	H20, steam
Total Efficiency [%][19]	80.0%	78.9%	78.4%	78.3%	76.5%
Thermal Output / Fuel Input [%]	53.0%	44.4%	41.6%	37.9%	35.0%
Power / Heat Ratio[20]	0.51	0.78	0.89	1.06	1.19

Source: Compiled by ICF from vendor supplied data.

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.

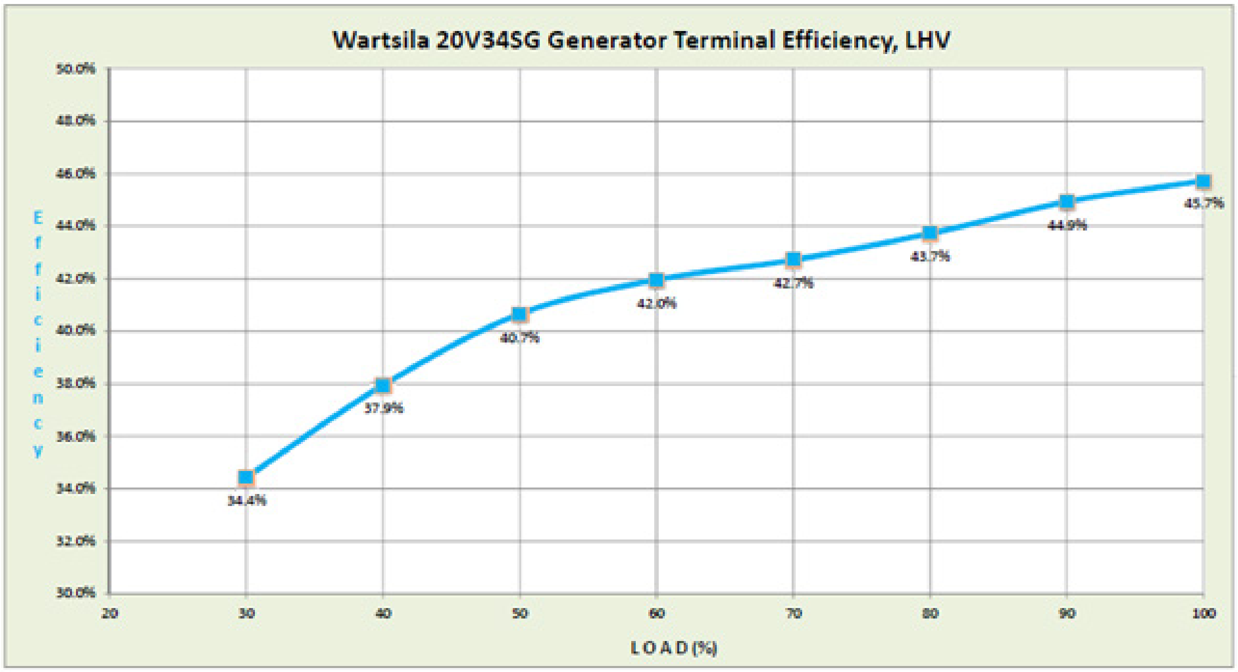
2.4.1 Part Load Performance

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 2-3** 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.

[19] Total CHP Efficiency = (net electric generated + net thermal energy recovered)/total engine fuel input.

[20] Power/Heat Ratio = (CHP electric power output (Btus))/useful thermal output (Btus)

Figure 2-3. Part Load Generator Terminal Efficiency, System 5



Source: Wartsila[21]

2.4.2 Effects of Ambient Conditions on Performance

Reciprocating engines are generally rated at ISO conditions of 77 °F and 0.987 atmospheres (1 bar) pressure. (Gas turbines are rated at 59 °F.) Like gas turbines, reciprocating engine performance – measured for 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.

2.4.3 Engine Speed Classifications

Reciprocating engines are classified as high-, medium-, or low-speed. **Table 2-3** 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).

[21] Wartsila gas-fired engines. *http://www.wartsila.com/en/power-plants/technology/combustion-engines/gas-engines*.

Table 2-3. Reciprocating Engine Types by Speed (Available MW Ratings)

Speed Classification	Engine Speed	Stoic/Rich Burn, Spark Ignition[22]	Lean Burn, Spark Ignition	Dual Fuel	Diesel
High Speed	1000-3600 rpm	0.01 – 1.5 MW	0.15 - 3.0 MW	1.0 - 3.5 MW[23]	0.01 – 3.5 MW
Medium Speed	275-1000 rpm	None	1.0 – 18 MW	1.0 – 17 MW	0.5 – 18 MW
Low Speed	58-275 rpm	None	None	2.0 – 65 MW	2 – 84 MW

Source: SFA Pacific, Inc., Wartsila

Engine power output is partly proportional to engine speed, which affords 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 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 high cost of installation, low speed engines are increasingly being displaced by medium and high speed engines as the primary choice for stationary power applications; with low speed engines being left to their primary market as marine propulsion engines.

Most 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.*

2.4.4 Performance and Efficiency Enhancements

2.4.4.1 Brake Mean Effective Pressure (BMEP) and Engine Speed

Engine power is related to engine speed and the BMEP during the power stroke. BMEP, as described above, 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 320 psig for large natural gas engines and 350 psig for diesel engines. Corresponding peak combustion pressures

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

are about 2,400 psig and 2,600 psig respectively. High BMEP levels increase power output, improve efficiency, and result in lower capital costs ($/kW).

BMEP can be increased by raising combustion cylinder air pressure through increased turbocharging, improved aftercooling, 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.

2.4.4.2 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, which increases the 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 normally 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).

2.4.5 Capital Costs

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 9 MW. 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.

The basic generator 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 or 514 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. In **Table 2-4**, System 4 is based on a German design, and operates at 1,500 rpm and produces 60 Hz power through a gearbox. 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 4 to 18 month construction period. All engines are in low NO_x configuration. System 1, a stoichiometric (rich burn) engine, uses a three-way catalyst to reduce emissions to their final level. The other systems are all lean burn engines and are shown with a SCR, CO catalyst, and continuous emissions monitoring system (CEMS) that are required in environmentally sensitive areas such as Southern California and the Northeastern U.S.

Table 2-4 provides cost estimates for combined heat and power applications based on a single unit engine. 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 economizer 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 interconnection/electrical costs reflect the costs of paralleling a synchronous generator for the larger systems. The 100 kW system uses an inverter based generator that has been pre-certified for interconnection in most areas. Labor/materials represent the labor cost for the civil, mechanical, and electrical work as well as 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. Cost estimates for multiple unit installations have lower unit costs than single unit installations.

Table 2-4. Estimated Capital Cost for Typical Gas Engine Generators in Grid Interconnected CHP Applications

Capital Cost, $/kW	System				
	1	2	3	4	5
Nominal Capacity (kW)	100	633	1121	3326	9341
Equipment (Costs in 2013 ($/kW))					
Gen Set Package	$1,400	$400	$375	$350	$575
Heat Recovery	$250	$500	$500	$500	$175
Interconnect/Electrical	$250	$140	$100	$60	$25
Exhaust Gas Treatment		$750	$500	$230	$150
Total Equipment	$1,900	$1,790	$1,475	$1,140	$925
Labor/Materials	$500	$448	$369	$285	$231
Total Process Capital	$2,400	$2,238	$1,844	$1,425	$1,156
Project and Construction Management	$125	$269	$221	$171	$139
Engineering and Fees	$250	$200	$175	$70	$30
Project Contingency	$95	$90	$74	$57	$46
Project Financing	$30	$42	$52	$78	$62
Total Plant Cost ($/kW)	**$2,900**	**$2,837**	**$2,366**	**$1,801**	**$1,433**

Source: Compiled by ICF from vendor-supplied data

2.4.6 Maintenance

Maintenance costs vary with type, speed, size and number of cylinders of an engine. These costs 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 either be 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 1 to 2.5 cents/kWh depending on engine size, speed and service. Many service contracts now include remote monitoring of engine performance and conditions in addition to allowing 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 filters, 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 2-5**) 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. Maintenance intervals are shown in **Table 2-5**.

Table 2-5. Representative Overhaul Intervals for Natural Gas Engines in Baseload Service

	Time Between Overhaul (thousand operating hours) as a Function of Engine Speed (rpm)				
	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 2-6** 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. Engine maintenance can be broken into fixed components that need to be performed on a recurring basis regardless of the engine run time and variable components that depend on the hours of operation. The vendors quoted all O&M costs on a variable basis for a system in baseload operation.

Table 2-6. Typical Natural Gas Engine Maintenance Costs ($2013/kWh)

	System				
	1	**2**	**3**	**4**	**5**
Nominal Capacity (kW]	100	633	1121	3326	9341
Service Contract	$0.023 - $0.025	$0.020	$0.018	$0.015	$0.0075
Consumables	included	$0.001	$0.001	$0.001	.001
Total O&M Costs, 2013 $/kWh	$0.023 - $0.025	$0.021	$0.019	$0.016	.0085

Source: Compiled by ICF from vendor supplied data

2.4.7 Fuels

In addition to operation on natural gas, 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
- **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 lower 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 for fuels with lower octane rating such as propane** – This is often characterized by a calculated value known as the Methane Number (MN). Different manufacturers may calculate Methane Number differently. Gases with heavier hydrocarbon components (Propane, Ethane, Butane, etc.) have a lower Methane Number as they will tend to autoignite more easily.

- 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 2-7 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 2-7. Major Constituents and LHV of Gaseous Fuels

Fuel Component / LHV	Natural Gas	LPG	Digester Gas	Landfill Gas
Methane, CH_4, % vol.	80 – 97	0	35 – 65	40 – 60
Ethane, C_2H_6, % vol.	3 – 15	0 – 2	0	0
Propane, C_3H_8, % vol.	0 – 3	75 - 97	0	0
Butane, C_4H_{10}, % vol.	0 – 0.9	0 - 2	0	0
Higher C_xH_{2x+2}, % vol.	0 – 0.2	0 - 20[24]	0	0
CO_2, % vol.	0 – 1.8	0	30 – 40	40 - 60
N_2, % vol.	0 – 14	0	1 - 2	0 - 13
H_2, % vol.	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 such as siloxanes); 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 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.

2.4.7.1 Liquefied Petroleum Gas

Liquefied petroleum gas (LPG) is composed primarily of propane and/or butane. While propane and butane ratings are higher than gasoline, most stationary spark ignition engines are designed with higher compression ratios that optimize operation with natural gas and its associated high methane number. Use of fuels with lower methane numbers like LPG in natural gas engines requires retarding of ignition timing and other appropriate adjustments to avoid detonation (*knocking*). LPG often serves as a back-up fuel where there is a possibility of interruption in the natural gas supply. Off-spec LPG may require cooling to condense out larger volumes of butane or heavier hydrocarbons that would aggravate engine knock. High butane content LPG is recommended only for low compression, naturally aspirated engines.

2.4.7.2 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

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

some propane and normally is used in low compression engines (both naturally aspirated and turbocharged). Retarded ignition timing eliminates detonation.

2.4.7.3 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 or somewhat below 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.

A key contaminant in biogas is a class of compounds called siloxanes, a subgroup of silicones containing Si-O bonds with organic radicals. These compounds are widely used for a variety of industrial processes and are also commonly added to consumer products, including detergents, shampoos, cosmetics, paper coatings, and textiles. Siloxanes in wastewater do not break down in wastewater treatment facilities or in landfills. As sludge undergoes anaerobic digestion, it may be subjected to temperatures of up to 150 ^{o}F. At these temperatures, siloxanes volatilize and enter the gas stream. Subjected to the heat of combustion in a reciprocating engine (turbine or microturbine), siloxanes leave behind hard deposits of silica on pistons and valve assemblies causing abrasion and impact damage that reduce the life and efficiency of the engine. Siloxanes need to be removed using refrigeration or sorbents such as activated carbon, alumina, synthetic resins, or liquid sorbents.[25]

For engines operating on biogas, additional capital investment is required for this fuel clean-up, compression, and sometimes derating of the engine capacity due to the lower thermal energy content of the fuel. For a 1,000 kW reciprocating engine, the added equipment and installation cost is about $600/kW.[26] Smaller systems can require nearly the same amount of equipment, so unit costs go up rapidly on smaller installations.

Improved engine design and hardened valve seats reduce siloxane damage on engines, thereby reducing the need for complete removal.

2.4.7.4 Industrial Waste Gases

Industrial waste gases that are common used as 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.

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[25] Dick McCarrick, "Siloxanes and Biogas," Environmental Leader (online edition), July 10, 2012. *http://www.environmentalleader.com/2012/07/10/siloxanes-and-biogas/*

[26] *Opportunities for and Benefits of Combined Heat and Power at Wastewater Treatment Facilities*, Eastern Research Group and EEA, Inc. (now ICF) for the U.S. EPA, 2007.

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.

2.4.8 System Availability

The percentage of time that a system is either up and running or available for use is referred to as its availability. Systems are unavailable during periods of scheduled maintenance or forced outages. 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 2-8** illustrates typical availability numbers based on a survey of natural gas engine gensets in CHP applications.

Table 2-8. Availabilities and Outage Rates for Natural Gas Engines

Reciprocating Engines	< 100 kW	100-800 kW	800-9000 kW
Systems Surveyed	14	8	18
Availability, %	97.93%	95.99%	98.22%
Forced Outage Rate, %	1.76%	1.98%	0.85%
Scheduled Outage Rate, %	0.73%	2.47%	1.12%

Source: ICF[27]

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. The use of multiple units or back-up units at a site can further increase the availability of the overall facility over 99 percent.

2.5 Emissions and Emissions Control Options

Emissions of criteria pollutants – oxides of nitrogen (NO_x), carbon monoxide (CO), and volatile organic compounds (VOCs – unburned, non-methane hydrocarbons) – are the primary environmental concern with reciprocating engines operating on natural gas. Emissions of sulfur compounds (SO_x) depend only on the sulfur content of the fuel. SO_x emissions are an issue only in large, slow speed diesels firing heavy oils. SO_x emissions from natural gas engines are assumed to be less than 0.0006 lb/MMBtu.[28] 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. Particulate emissions from 4-stroke lean burn natural gas engines are 4,000 times lower than for an uncontrolled diesel engine.

[27] Distributed Generation Operational Reliability and Availability Database, EEA, Inc. (now part of ICF), January 2004
[28] EPA AP-42, Natural Gas Fired Reciprocating Engines.

2.5.1 Emissions Characteristics

2.5.1.1 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 g/hp-hr and g/kWh, or as an output rate such as lb/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 (TWC) 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. Lean burn engines can meet the standard using selective catalytic reduction (SCR). Both rich burn and lean burn engines have been certified for operation in Southern California meeting the stringent California Air Resources Board (CARB) 2007 standards.

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 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.[29] 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.

2.5.1.2 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. Therefore, control of NO_x through lean combustion can increase CO and VOC emissions out of the engine.

[29] Kirby Chapman, *Cost Effective Reciprocating Engine Emissions Control and Monitoring for E&P Field and Gathering Engines*, Kansas State University, 2003.

2.5.1.3 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).

2.5.1.4 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 climate change. 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. As converted to CO_2 in the exhaust, these values are 117 lb/MMBtu for natural gas, 160 lb/MMBtu for diesel oil, and 205-226 lb/MMBtu for coal.

2.5.2 Emissions Control Options

Emissions from natural gas SI engines have 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 1.8 lb/MWh and CO emissions of 8.1lb/MWh before any exhaust gas treatment. Adding selective catalytic reduction (SCR) and a CO oxidation catalyst can allow lean burn reciprocating engines to meet the very stringent California South Coast emissions standards of 0.07 lb/MWh for NO_x and 1.0 lb/MWh for CO.

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.

2.5.2.1 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 uncontrolled emissions rate for lean burn natural gas engines is between 1.5-6.0 lb/MWh.

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 2-9** 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 2-9. Uncontrolled NO_x Emissions versus Efficiency Tradeoffs

Engine Characteristics	Low NO_x	High Efficiency
Capacity (MW)	9.3	9.3
Speed (rpm)	720	720
Efficiency, LHV (percent)	44.1	45.7
Emissions:		
NO_x (g/kWh) (ppmv @ 15% O_2)	0.62 45	1.2 90
CO (g/kWh) (ppmv @ 15 % O_2)	1.9 226	1.3 158
NMHC (g/kWh) (ppmv @ 15% O_2)	1.0 209	0.71 153

Data Source: Based on engine manufacturer's data – Wartsila 20V34SG Prechamber Lean Burn Gas Engine[30]

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 for several engine manufacturers and is being pursued as part of the Department of

[30] Wartsila Gas-fired Engines. http://www.wartsila.com/en/power-plants/technology/combustion-engines/gas-engines#expandable_id

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 NO_x/MWh (0.1 g NO_x/bhph).

2.5.2.2 Post-Combustion Emissions Control

There are several types of catalytic exhaust gas treatment processes that are applicable to various types of reciprocating engines. **Table 2-10** shows the methods in use today, the applicable engine types, and the pollutant reduction achievable.

Table 2-10. Post-Combustion Exhaust Gas Cleanup Options

Emission Control Technology	Applicable Engine Type	Typical Performance Reductions, %			
		CO	NMHC	NO_x	PM
Diesel Oxidation Catalyst (DOC)	Diesel	90	80	0	20
Catalyzed Diesel Particulate Filter (DPF)	Diesel	90	90	0	90+
Non-selective Catalytic Reduction (NSCR)	Rich Burn Natural Gas	90	80	95	0
NG Oxidation Catalyst	Lean Burn Natural Gas	95	95	0	0
Selective Catalytic Reduction (SCR)	Lean Burn Diesel or Natural Gas	0	0	95	0

Reference: Jay Warner and Gary Bremigan, *System Solutions for Optimizing Exhaust Emission Control Systems*, Universal Acoustic & Emissions Control Technologies USA, 2010

2.5.2.3 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 95 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.

2.5.2.4 Diesel Particulate Filter

While not an issue for spark ignition engines firing gaseous fuels, compression ignition engines fueled by diesel or heavy oil produce particulates that must be controlled. Diesel particulate filters can reduce over 90 percent of particulate (soot) emissions from diesel engines. There are a variety of filter materials and regeneration strategies used. Currently, there are no commercially available particulate control devices available for large, medium speed diesel engines.[31]

2.5.2.5 Three–Way Catalyst (Non Specific Catalytic Reduction)

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

[31] Private Communication, Wartsila, January 2014.

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, the conversion of 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 g/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.

2.5.2.6 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 mole 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 are considered commercial today and represent the only technology that will reduce NO_x emissions to the levels required in Southern California and the Northeast U.S. Still, SCR adds significantly to the capital and operating cost of a reciprocating engine CHP system. As shown previously in **Table 2-4**, SCR with oxidation catalyst and associated continuous energy monitoring system adds between $150-$700/kW to the capital cost for a lean burn reciprocating engine CHP installation. The cost burden is higher for smaller engines.

2.5.3 Gas Engine Emissions Treatment Comparison

Table 2-11 shows achievable emissions for each of the five representative gas engine systems. The emissions presented assume available exhaust treatment. System 1, the 100 kW engine, is a high speed, rich burn engine. Use of a TWC system with EGR provides NO_x emissions of just under 0.07 lb NO_x per MWh after credit is taken for the thermal energy provided.[32] The Lean burn engine systems use an SCR/CO system providing emissions reduction that meets the CARB 2007 emissions limits without consideration of the thermal energy credit.

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

[32] CARB 2007 emissions regulations allow CHP systems to include both the electric and thermal output in the calculation of output based emissions.

controls and ignition timing. With the addition of SCR after-treatment, described below, some manufacturers tune engines for higher efficiency and allow the SCR system to remove the additional NO_x. Achieving highest efficiency operation results in conditions that generally produce twice the NO_x as low NO_x versions (e.g., 3 lb/MWh versus 1.5 lb/MWh). 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 2-11. Gas Engine Emissions Characteristics with Available Exhaust Control Options

Emissions	System				
	1	2	3	4	5
Nominal Capacity (kW)	100	633	1121	3326	9341
Electrical Efficiency (% HHV)	27.0%	34.5%	36.8%	40.4%	41.6
Engine Combustion	Rich	Rich	Lean	Lean	Lean
Precatalyst Emissions					
NO_x (lb/MWh)		1.77	1.77	1.77	2.64[33]
CO (lb/MWh)		8.12	8.12	8.12	4.18
VOC (lb/MWh)		0.97	0.97	0.97	1.39
Post Catalyst Emissions					
NO_x (lb/MWh)	0.070	0.07	0.07	0.07	.07
CO (lb/MWh)	0.200	0.20	0.20	0.20	.20
VOC, (lb/MWh)	0.1	0.10	0.10	0.10	.10
CO_2 Gross (lb/MWh)	1,479	1,158	1,084	989	988
CO_2 Net (lb/MWh)	499	516	520	520	540

Source: Compiled by ICF from vendor supplied data

2.6 Future Developments

Reciprocating engines have improved significantly over the last two decades in terms of increased efficiency and reduced emissions. Electronic engine control and improved combustion chamber design, including the use of precombustion chambers, allow engines to operate on leaner fuel mixtures. Improvements in materials and design have allowed engines to operate at higher speeds and power densities while still maintaining long life.

These improvements have been the combined results of collaborative research efforts by private industry, universities, and the federal government. Public private partnerships such as the DOE Advanced Reciprocating Engine System (ARES) funded by DOE and the Advanced Reciprocating Internal Combustion Engine (ARICE) program funded by the California Energy Commission (CEC) have focused attention on the development of the next generation reciprocating engine.

[33] The Wartsila engine shown here (System 5) when matched with SCR is tuned for best efficiency at the expense of higher uncontrolled NO_x – letting the SCR remove the NO_x. This strategy results in lower overall operating costs.

The ARES program has been active for more than 10 years. The program has produced and commercialized Phase I and Phase II engines with current work on a Phase III engine to reach the overall design efficiency goals of 0.1 g/bhp NO_x emissions, 50 percent BTE efficiency, 80+ percent CHP efficiency, maintenance costs of $0.01/kWh while maintaining cost competitiveness.[34]

For a technology originally developed in the 19th century, reciprocating internal combustion engines have continually improved and adapted to the needs of the market more than 100 years later. Ongoing improvements in efficiency, cost, and emissions reduction will ensure that reciprocating engines will continue to remain viable and competitive with newer technologies such as fuel cells and microturbines in the distributed generation market. Installations of multiple large engines have proven to be competitive in power generation applications of more than 200 MW.

[34] *Industrial Distributed Energy R&D Portfolio Report: Summary,* U.S. Department of Energy Advanced Manufacturing Office, June 2011.