Environmental
Resources
Management

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January 21, 2013

Mr. Carl E. Edlund, P.E.
Director, Multimedia Planning and Permitting Division
United States Environmental Protection Agency (USEPA)
Region 6
1445 Ross Avenue, Suite 1200
Dallas, Texas 75202-2733

Project No. 0151579

Subject: Response to Application Completeness Determination Greenhouse Gas Prevention of Significant Deterioration Permit - Air Liquide Large Industries, U.S. LP Bayou Cogeneration Plant

Dear Mr. Edlund,

On behalf of our client Air Liquide Large Industries, U.S. LP (Air Liquide), Environmental Resources Management (ERM) submits this letter in response to your letter dated November 27, 2012, requesting additional information regarding the application for a greenhouse gas (GHG) Prevention of Significant Deterioration (PSD) permit currently under review by USEPA, Region 6. A summary of each request for information (RFI) is provided along with Air Liquide's response.

RFI #1: *Please provide supplemental data to the process flow diagram to identify all pieces of equipment and the GHG emission sources with associated emission point numbers (EPN).*

Response: A revised process flow diagram is provided as Attachment 1.

RFI #2a: *On page 17 of the permit application, the "High Efficiency Turbines" section of the BACT provides information on the selected turbines. The applicant should provide comparative benchmark information indicating other similar industry operating or designed units and compare the design efficiency of this process to other similar or alike processes. The applicant should then use this information to rank the available control technologies. A comparison of equipment energy efficiencies is necessary to evaluate the energy efficiency of the proposed equipment and possible control technologies. This information should also detail the basis for your BACT proposal in determining BACT limits for the emission units for which these technologies are applied in Step 5 of the BACT analysis. Did Air Liquide review the BACT determinations for recently issued GHG PSD permits within EPA Region 6, and elsewhere? EPA Region 6 has issued GHG PSD permits to Lower Colorado River Authority (LCRA), Calpine Deer Park, and Calpine Energy Center, all of which have combustion turbines. All these facilities have combustion turbine thermal efficiency that is better than what is proposed for Air Liquide. Please provide additional information to substantiate the proposed efficiency for the GE 7EA units.*

Response: Air Liquide reviewed the BACT determination for the LCRA Thomas Ferguson Plant and Calpine Deer Park as well as numerous other simple cycle and combined cycle units with permits under consideration by Region 6. Further, Air Liquide reviewed BACT determinations from other regions and included the Pio Pico Energy Center in Otay Mesa, California. It should be noted that at the time of this application, a draft BACT determination for the Calpine Energy Center was issued in November 2012 and was not available when this application was submitted. A review of these BACT determinations is provided in Appendix C of the application. An updated review is provided as Attachment 2 that includes the Calpine Energy Center.

As stated in Section 4.2.1.1 of the original application, review of these specific case-by-case BACT determinations for combined cycle units to be from 7,720 to 7,730 Btu (HHV)/kWh. There is only one BACT result for simple cycle units and that resulted in a limit of 9,196 Btu (HHV)/kWh for that particular application. Furthermore, not all units have been assigned thermal efficiency limits and have BACT determinations based on mass emission rates of GHG only. Further, BACT is determined on a case-by-case basis and no one BACT determination prescribes BACT for another. Although previous determinations for similar projects should be considered in determining BACT for a new project, other factors including purpose, energy impacts, and environmental impacts must be considered rather than simple reliance on a result of a similar BACT analysis.

The BACT determinations referenced in your Completeness Determination were for units designed for electrical production. The facility at issue in this permitting action does not function as an electricity production facility, but the turbines will be an integral part of the combine heat and power (CHP) system of the Bayou Cogeneration Plant. The primary purpose of the gas turbines being permitted in this application is to generate commercial steam and the production of electricity is incidental.

As noted in our application, the Bayou Cogeneration Plant is a combined heat and power (CHP) plant. Electricity generating gas turbines units (EGUs) are designed to optimize the conversion of energy to mechanical work rather than transfer energy to a medium such as generating high temperature exhaust gases for steam production. Further, a combined cycle unit uses two thermodynamic cycles, the Brayton cycle and the Rankine cycle, to convert thermal energy into mechanical work. Electricity is produced by expanding exhaust gases or steam through the gas turbine and then a steam turbine to drive a shaft which converts mechanical work into electricity. Energy is consumed in order to drive the turbine mass resulting in mechanical energy losses and a decrease in thermal efficiency. A CHP plant does not generate electricity in a steam turbine and therefore, does not experience the mechanical energy loss resulting from driving the turbine. Instead, the energy in the steam is used through conductive heat transfer in the customers' process. As a result, CHP is an inherently more efficient process than an equivalent combined cycle turbine. For these reasons, comparing thermal efficiency on an energy-to-power basis to either a simple or combined cycle turbine electric generating units (EGUs) to a gas turbine designed for steam production is not appropriate.

Air Liquide conducted an exhaustive search of the USEPA-issued permits and BACT determinations as well as the RACT/BACT/LAER Clearinghouse (RBLC), and could not locate any GHG BACT determinations for a CHP application. In an effort to try to accommodate your request for some type of comparison between the proposed project and those recently permitted by EPA Region 6, Air Liquide proposes to use a combination of combined cycle power production units with a standalone steam generation system for comparison.

For CHP units some of the energy in the fuel is used to generate electricity through the turbine, and some of the energy from the fuel is used to make steam. This use of the residual heat from the turbine is similar to how a combined cycle unit is operated, except the steam generated is left as steam, rather than using it to generate additional electricity in a steam turbine. In CHP processes, because the fuel energy is being used to both generate power and steam, comparing the efficiency of CHP to generate either individually is not an accurate representation of the process efficiency. Instead, we must determine an equivalent measure of useful energy out relative to energy consumed. In a topping unit where the electrical power is generated prior to generation of the steam for heat, the measure of efficiency is Fuel Chargeable to Power (FCP). FCP is defined as the incremental fuel for the generation system relative to the needs of a heat only system divided by the net incremental power produced by the cogeneration system. The FCP is interchangeable to the net heat rate of a plant generating only electrical power; thus FCP is the most appropriate comparison to a combined cycle EGU (See p. 4, *Cogeneration Application Considerations*, General Electric, May 2009, enclosed as Attachment 3). FCP is calculated as the difference between total fuel fired and the fuel used to generate steam divided by the net power output as described in Equation 1.

Equation 1 *Calculation of Fuel Chargeable to Power*

$$FCP = \frac{Q_{GT} - FCS}{P_{NET}}$$

Where: FCP = Fuel Chargeable to Power [Btu (HHV)/kWh]
Q_{GT} = Heat input to gas turbine [MMBtu/hr]
FCS = Fuel Chargeable to Steam [MMBtu/hr]
P_{NET} = Net electrical production [kW]

Fuel Chargeable to Steam (FCS) is the net heat used to generate steam divided by the efficiency of an equivalent boiler. Calculation of FCS is described in Equation 2.

Equation 2 *Calculation of Fuel Chargeable to Steam*

$$FCS = \frac{Q_{HP} + Q_{lP} - Q_{FW}}{e_{boiler}}$$

Where: FCS = Fuel Chargeable to Steam [MMBtu/hr]
Q_{HP} = Heat used to generate high pressure steam [MMBtu/hr]
Q_{LP} = Heat used to generate low pressure steam [MMBtu/hr]
Q_{FW} = Heat used to heat the feedwater [MMBtu/hr]
e_{boiler} = Efficiency of an equivalent boiler [0.84]

The heat required to generate steam of each condition is the product of the change in enthalpy required to convert water to steam of the specified pressure and temperate and the production rate of the steam. The heat used in the feedwater is the change in enthalpy to bring the feedwater to vaporization temperature and mass flow rate as shown in Equation 3.

Equation 3 *Calculation of Heat Consumption for Steam and Feedwater*

$$Q_i = \Delta h_i \cdot m_i$$

Where: Q_i = Heat used for steam or water stream, *i* [MMBtu/hr]
Δh_i = Change in enthalpy, *i* [MMBtu/lb]
m_i = Mass flow of stream *i*

Because the FCP is interchangeable with the net heat rate of an equivalent combined cycle facility, Air Liquide proposes a revised BACT limit of 7,720 Btu [HHV] equivalent to the lowest proposed combined cycle turbine.

In regards to EPA's request to consider alternative turbines for use in the proposed project, Air Liquide reiterates that the business purpose of this project is to replace the existing gas turbines in kind. According to EPA guidance for GHG BACT, *PSD and Title V Permitting Guidance for Greenhouse Gases (EPA 457/B-11-001, pg. 26)*:

> *"While Step 1 [identification of all available control options] is intended to capture a broad array of potential options for pollution control, this step of the process is not without limits. EPA has recognized that a Step 1 list of options need not necessarily include inherently lower polluting processes that would fundamentally redefine the nature of the source proposed by the permit applicant. BACT should generally not be applied to regulate the applicant's purpose or objective for the proposed facility."*

As stated in Section 1.1 of the application, the 79.6 MW GE 7EA turbines are closest in specification to the existing 75.2 MW turbines. The current BACT determinations are for larger units ranging from 100 MW to 195 MW which are inherently more efficient due to scale. Installation of these larger units would require redesign of the heat recovery steam generator (HRSG) and modification to ancillary equipment and infrastructure which would result in a completely different and more expansive construction project that would not be required with the replacement in kind. Installation of larger turbines would require massive modifications throughout the entire facility, which is a project not contemplated by Air Liquide. Therefore, Air Liquide believes that installation of the larger units would constitute a capacity expansion requiring significant modification to the facility and project scope, which does not meet the stated business purpose of this project. However, due to the inherent efficiencies of CHP units, the proposed project is as efficient as these larger turbines, and Air Liquide will meet equivalent thermal efficiency standards proposed for these units as measured in FCP.

Finally, Air Liquide points to the Executive Order - Accelerating Investment in Industrial Energy Efficiency issued by the President of the United States on August 30, 2012. The order was issued "[t]o formalize and support the close interagency coordination that is required to accelerate greater investment in industrial energy efficiency and CHP." The order is clear recognition of the role that CHP plays in the efficient use of energy. The directive in the order is for specific federal agencies, including the USEPA, to:

> "coordinate policies to encourage investment in industrial efficiency in order to reduce costs for industrial users, improve U.S. competitiveness, create jobs, and reduce harmful air pollution. In doing so, they shall engage States, industrial companies, utility companies, and other stakeholders to accelerate this investment."

The Bayou Cogeneration Plant project under consideration is critical to the continuation of a CHP facility that meets the definition of the type of facility encouraged by the August 30, 2012 Executive Order.

RFI #2b: [related to BACT for the gas turbines] *What recordkeeping requirements are you proposing? What will alert on-site personnel to problems?*

Response: Daily thermal efficiency will be calculated as shown in Equations 1 through 3. Air Liquide is proposing daily recordkeeping of the following parameters to calculate thermal efficiency:

- Natural gas consumed;
- Net electricity produced;
- Mass of high pressure steam produced;
- Mass of low pressure steam produced;
- Mass of feedwater used;
- Average daily pressure and temperature of steam produced; and
- Calculated average enthalpy for low and high pressure steam based on average daily steam conditions.

Air Liquide will also maintain monthly records of the fuel heating value provided by the supplier to determine daily heat input. Compliance with the 7,720 Btu (HHV)/kWh limit will be demonstrated by the 365 day rolling average of the calculated daily thermal efficiency.

Air Liquide proposes to demonstrate compliance with the carbon dioxide (CO_2) mass emission limit of 485,112 tons per year (tpy) using a Continuous Emissions Monitoring System (CEMS). Compliance with the CO_2 mass emission rate will be done on a 12-month rolling average basis.

RFI #3: *The application provides a five-step BACT analysis for Carbon Capture and Sequestration (CCS) and concludes that the use of this technology is technically infeasible. A general cost analysis is provided. Please supplement the 5-step top down BACT analysis by supporting your cost analysis on equipment design including any conclusions on a cost per pound CO2e removed basis, total annualized costs, and cost effectiveness for implementing CCS control technology for this project, safety or environmental concerns and any associated energy penalty that may result from the implementation of this add-on control and supports its elimination* *from your BACT consideration. Also, we are requesting a comparison of the cost of CCS to the current project's annualized cost.*

Response: Air Liquide has updated the costs for capture, transport, and long term geologic storage of CO_2 per EPA's request. These updated costs are provided in Attachment 4. The total estimated capital cost for CCS and long-term geologic storage is $859.1MM which is more than four times the estimated annualized capital cost for the proposed project. Including the costs of capture and long-term geologic storage, Air Liquide estimates $75/ton of CO_2 equivalent (CO_2e) controlled. Based on the normalized control cost and comparison of total capital cost of control to project cost, Air Liquide maintains that CCS is not economically feasible.

Although carbon capture technology employed in other industries or on other source types may be transferable to use on a combustion turbine CHP process, there are no such installations in the U.S. In fact, in the power generation industry, carbon capture technologies are in their infancy and all projects are technology demonstration projects, subsidized with government dollars. None of these demonstration projects is on a natural gas-fired combined cycle unit. Presumably, this is due to the inherent challenges of carbon capture in dilute gas streams as opposed to other sources with streams containing high concentrations of CO_2 where it may be more feasible to demonstrate carbon capture - coal combustion sources, pre-combustion gas clean-up (as in the case of Integrated Gasification Combined Cycle), or oxy-fuel technology. At this time, carbon capture is an undemonstrated technology for natural gas streams with a low concentration of CO_2. At your request, our permit application shows that CCS is also economically infeasible at this time.

If you have any questions, please feel free to contact Mr. Eric Hodek of my staff at (512) 374-2261 or at eric.hodek@erm.com.

Sincerely,

Environmental Resources Management

Peter T. Belmonte, P.E.
Partner

Attachments
1 – Revised Process Flow Diagram (Figure 2-3)
2 – Updated Review of Recently Issued Permits and Applications (Appendix C)
3 – *Cogeneration Application Considerations*, May 2009
4 – Conceptual Cost Estimate for Carbon Capture and Sequestration

cc: Mr. Aswath Kalappa, Air Liquide

Revised Process Flow Diagram

Attachment 1

January 21, 2013
Project No. 0151579

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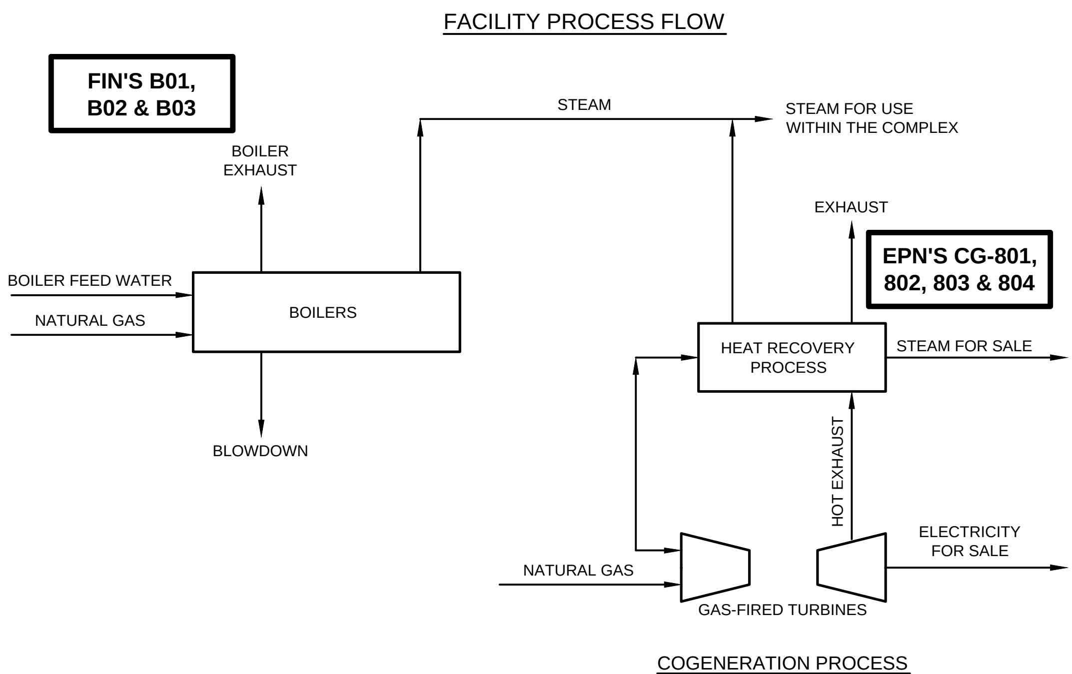

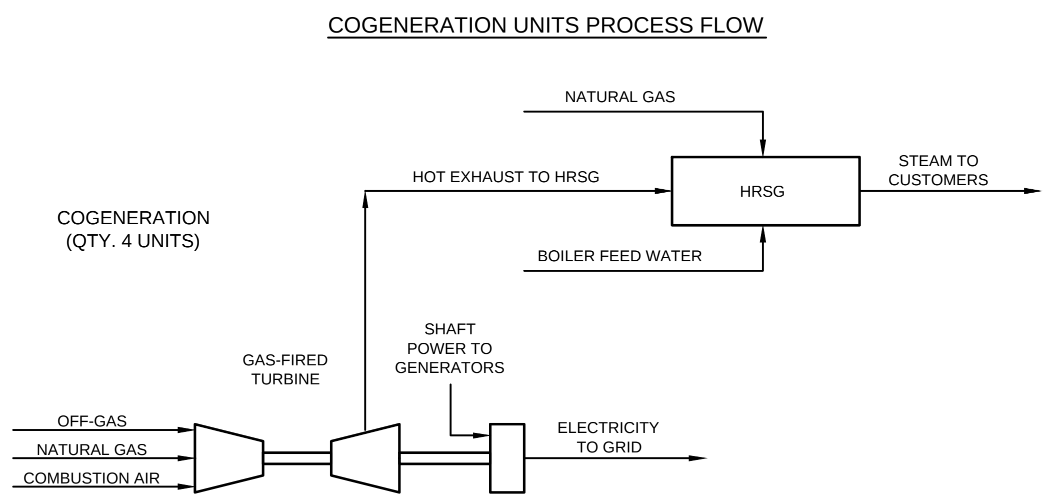

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FIGURE 2-3
PROCESS FLOW DIAGRAM
Air Liquide Bayou Cogeneration Plant
Air Liquide Large Industries U.S., L.P.
11400 Bay Area Boulevard
Pasadena, Texas

ERM

ERM-Southwest, Inc. TX PE Firm No. 2393

Updated Review of Recently Issued Permits and Applications

Attachment 2

January 21, 2013
Project No. 0151579

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Attachment 2
Air Liquide Bayou Cogeneration Plant
GHG BACT Analysis
Recently Issued Permits and Applications Under Review for Greenhouse Gases from Combustion Turbines

No.	Permit Authority	Permit Number	Company Name Facility Name Location	#	Unit Description Model	Capacity		Control Technology	Thermal Efficiency BTU (HHV) per kW-hr (gross)	PTE tpy CO_2e	Proposed BACT Limits Parameter	Units	Monitoring
1	USEPA R6	PSD-TX-1244-GHG	Lower Colorado River Authority Thomas C. Ferguson Power Plant Horseshoe Bay, TX	2	GE 7FA	195	MW	Combined cycle operation Efficient design	N/A	909,833	908,958 16.80 1.70 0.46 7,720 [365 day rolling average]	tpy CO2 tpy CH4 tpy N2O ton CO2/MWh (net) Btu/kWh (HHV)	Fuel monitoring or CEMS
2	USEPA R6	PSD-TX-955-GHG	Calpine Corporation Channel Energy Center Dallas, TX	1	Siemens FD2	168 677	MW MMBtu/hr	Combined cycle operation Efficient design Process monitoring	N/A	985,340	7,730 0.460 985,340 [365 day rolling average]	Btu/kWh (HHV) tons CO_2/MWh tpy CO_2e	Source testing Fuel monitoring CEMS/CMS
3	USEPA R6	PSD-TX-955-GHG	Calpine Corporation Deer Park Energy Center Dallas, TX	3	Siemens 501F	180 725	MW MMBtu/hr	Combined cycle operation Efficient design Process monitoring	N/A	1,045,635	7,730 0.460 1,045,635 [365 day rolling average]	Btu/kWh (HHV) tons CO_2/MWh tpy CO_2e	Source testing Fuel monitoring CEMS/CMS
4	USEPA R9	PSD-SD-11 (draft)	Pio Pico Energy Center, LLC Pio Pico Energy Center Otay Mesa, CA	3	GE LMS100	100 930	MW MMBtu/hr	Simple cycle operation Efficient design	N/A	N/A	1,181 9,196	lb CO2/MWh (net) Btu/kwH (HHV - gross)	Fuel monitoring CEMS, CMS
Applications Pending													
5	USEPA R6	N/A	Calhoun Port Authority ES Joslin Power Station Point Comfort, TX	3	GE 7FA	208	MW	Combined cycle operation Efficient design Evaporative cooling Steam turbine bypass	N/A	N/A	7,730	Btu/kWh (HHV)	N/A
6	USEPA R6	N/A	Copano Processing, LP Houston Central Gas Plant Sheridan, TX	2	Solar Mars 100	15,000	hp	Efficient design Waste heat recovery Process monitoring	N/A	58,672	1.16	ton CO2e/MMscf compressed	Fuel gas flow monitoring AFR monitoring
7	USEPA R6	N/A	DCP Midstream, LP Hardin County NGL Fractionation Plant Hardin County, TX	2	Solar Saturn T-4700	43	MMBtu/hr	Efficient design Waste heat recovery Process monitoring	N/A	24,610	24,610	tpy CO2e	None proposed
8	USEPA R6	N/A	DCP Midstream, LP Jefferson County NGL Fractionation Plant Jefferson County, TX	2	Solar Saturn T-4700	43	MMBtu/hr	Efficient design Waste heat recovery Process monitoring	N/A	24,610	24,610	tpy CO2e	None proposed
9	USEPA R6	N/A	El Paso Electric Company Montana Power Station El Paso, TX	4	GE LMS100	100	MW	Efficient design Evaporative cooling Good operating practices	9,074	227,840	227,840	tpy CO2e	Fuel quality monitoring
10	USEPA R6	N/A	Freeport LNG Development Liquiefaction Plant Freeport, TX	1	GE Frame 7EA	87	MW	Efficient design Waste heat recovery Evaporative cooling	N/A	562,693	562,141 0.03 1.06	tpy CO2 tpy CH4 tpy N2O	Fuel monitoring or CEMS
11	USEPA R6	N/A	La Paloma Energy Center Harlingen, TX	2	GE F7FA	183	MW	Engergy Efficiency, Practices and Designs	7,528	1,300,674	1,299,423 24.10 2.40	tpy CO2 tpy CH4 tpy N2O	Fuel monitoring or CEMS
					Siemens SGT6-5000F(4)	265	MW		7,649	1,451,772	1,450,376 26.80 2.70	tpy CO2 tpy CH4 tpy N2O	
					Siemens SGT6-5000F(5)	271	MW		7,720	1,642,317	1,640,737 30.40 3.00	tpy CO2 tpy CH4 tpy N2O	

Cogeneration Application Considerations
General Electric, May 2009

Attachment 3

January 21, 2013
Project No. 0151579

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GE Energy

Cogeneration
Application Considerations

John A. Jacobs III
Technical Leader
Evaluation & Analysis

Martin Schneider
Senior Marketing Manager

May 2009

Contents:

Cogeneration Application Considerations

Introduction

Cogeneration or CHP (Combined Heat and Power). The terms cogeneration and CHP are used interchangeably in this paper and are defined as the combined simultaneous generation of heat and electrical energy with a common source of fuel. Common examples of cogeneration applications include pulp and paper mills, steel mills, food and chemical processing plants, and District Heating (DH) applications.

Since the beginning of the 20th century, cogeneration technology has been utilized by many industrial companies as an eco-friendly means to economically meet a plant's combined heat and power demands. The volatility of fuel costs and electricity prices in deregulated markets—coupled with the need to secure reliable heat and power supplies, along with new environmentally based financial incentives—are driving the evolution of this technology. These key factors are causing many industrial companies, municipalities, developers and utilities to give even more consideration to cogeneration as an eco-friendly, profitable, and reliable means of addressing their specific generation needs while also meeting local environmental regulations.

In the past and certainly prior to 1960, most cogeneration applications were developed based on steam turbine cogeneration systems consisting of conventional fossil-fired boiler(s) in addition to an industrial type steam turbine and/or combinations of industrial type steam turbines. More recent factors have made gas turbine and engine based solutions highly desirable, including:

- Potential economic benefits resulting from higher power-to-heat ratios
- Rising fuel costs
- Operational flexibility
- Emerging environmental policies and incentives
- Increased focus and need for power security
- Availability of a wide range of system integration options coupled with attractive cogeneration system performance levels

These technological advances in the area of fuel flexibility, as well as gas turbine and engine product diversification/adaptation, have served as enablers to make some cogeneration opportunities feasible, while making others even more attractive.

Universal sensitivity to our environment and environmental considerations have led to the development of projects that not only minimize GHG (Green House Gas) emissions, but also help to displace GHG emissions from existing plants as well as other emissions sources. Thus, one of the more significant advantages for gas turbine, combined cycles and gas reciprocating engines is the potential for GHG reductions as compared to less efficient systems. This monetization of GHG reductions serves as a significant driver/incentive for the development of gas-turbine and gas-engine-based cogeneration applications.

Cogeneration applications range from industrial applications such as pulp and paper mills, steel mills, and chemical processing plants to commercial and civic-based applications like hospitals, universities and warehouses—thus encompassing a wide range of unique power-to-heat ratios. The variation of power-to-heat ratio combined with differences in grade/quality of heat (such as water, steam, and process heating/cooling) within the cogeneration application space are dictating both technology selection as well as system and product flexibility requirements.

The primary objectives of this paper are to:

- Review many of the technical considerations and alternative options associated with the development of cogeneration systems.
- Discuss some of the environmental benefits that are potentially available through cogeneration, and to introduce the concept of monetization (primarily surrounding CO_2).
- Illustrate and provide the CHP performance characteristics associated with GE's diverse gas turbine and reciprocating gas engine product portfolios that can ultimately be leveraged for project and technology screening purposes. The technical parameters provided include—but are not limited to—power-to-heat ratio, equipment capacity (thermal/electrical) and efficiency/FCP (Fuel Chargeable to Power), and/or SFC (Specific Fuel Consumption) in the case of reciprocating gas engines.

This paper reviews many of the technical, economical and environmental considerations in the development of cogeneration projects.

Cogeneration

Cogeneration is frequently defined as the sequential production of necessary heat and power (electrical or mechanical) or the recovery of low-level energy for power production. This sequential energy production yields fuel savings relative to separate energy production facilities because both the heat and power requirements are satisfied from a common/single fuel source. The heat that would otherwise be wasted in the power production process is recovered and leveraged to provide process heat requirements (which otherwise would have to be generated with a separate fuel source), thus providing significant fuel savings.

With the recent increases in gas and oil prices, advancements in gas-turbine and gas-reciprocating-engine fuel flexibility—combined with a worldwide desire to reduce GHG (Green House Gas) emissions, increase power security (through localization of power generation), and attractive cogeneration system efficiency levels—have sparked renewed interest in cogeneration applications.

Power can be cogenerated in topping or bottoming cycles. In a topping cycle, power is generated prior to the delivery of thermal energy to the process. Typical topping cycle examples include:

- Non-condensing steam turbine cycles (commonly used in the pulp and paper industry)
- Heat recovery and combined cycles (applied in many chemical plants), where exhaust energy for a gas turbine or heat from gas reciprocating engines provide thermal energy that is ultimately used to satisfy the process requirements
- Central heating/cooling applications that exist in urban locations where electric power stations also supply thermal energy (or similarly on a smaller scale, where heating/cooling requirements are recovered from gas turbine or gas reciprocating engines to satisfy localized, civic or commercial based CHP requirements)

In bottoming cycles, power is produced from the recovery of process thermal energy that would normally be rejected to the heat sink. Typical bottoming cycle examples include:

- Power generation resulting from recovery of excess thermal energy (combined cycle steam turbine output generation)
- Power generation derived from exothermic process reactions, and heat recovery from kilns, process heaters and furnaces.

This paper focuses primarily on application considerations for topping cogeneration cycles.

For comparative purposes *Figure 1* illustrates energy utilization effectiveness (the percent of total energy output from the cycle which is useful heat and/or power) for a typical non-reheat coal-fired utility/industrial plant configuration (three-stage feed water heating with steam conditions of 1450 psig / 950°F [101 bar / 510°C] steam conditions vs. a cogeneration facility utilizing the same fired boiler but with a non-condensing steam turbine generator that supplies steam to process. This diagram suggests that relative to the typical coal-fired power generation application (as previously defined) the energy utilization associated with an equivalent cycle with cogeneration can be improved by as much as 35%. This improvement in energy utilization is made possible because the process demand becomes the heat sink for the cogeneration cycle, thus eliminating energy losses associated primarily with the condenser.

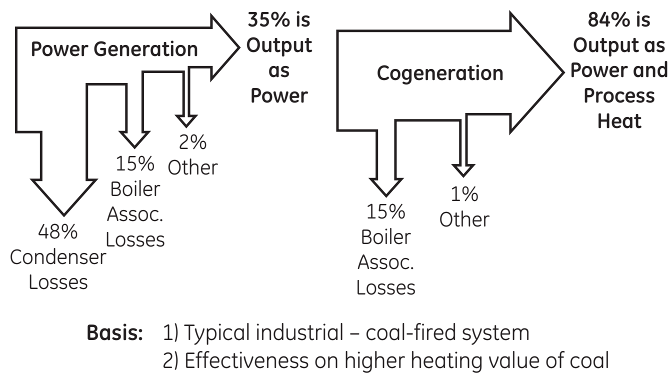
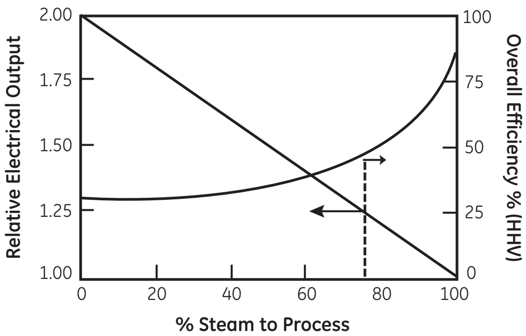


Figure 1. Fuel utilization effectiveness (fossil-fired boiler)

This principal is further illustrated by *Figure 2*, which highlights the influence of decreasing the thermal energy to a process from a steam turbine cycle. As less steam is delivered to process, the electrical output ratio (relative to the electric output at 100% steam-to-process) increases, becoming a maximum of about 2.0 for the steam conditions noted if no steam is delivered to process. The overall efficiency decreases from 84% to 35% as process steam delivery is eliminated.



Basis:

1) Steam conditions 1450 psig, 950°F (101 bars, 510°C), 150 psig (11.4 bars) process, 2 1/2" (63.5mm) HgA condenser pressure
2) Three stages of feedwater heating
3) Boiler efficiency 85% HHV

Figure 2. Steam turbine cycle performance at various process steam demands

Similar performance benefits are also available in gas turbine and reciprocating gas engine cogeneration systems. For example, an F-class technology gas turbine generator with feeding to an HRSG (which in turn provides process steam) can yield overall energy effectiveness levels between 80-85% depending upon process steam conditions. In comparison, the same F-class gas turbine in combined cycle (and producing power only) yields an overall energy effectiveness of between 50-55% depending upon the cycle design. This comparison is illustrated in *Figure 3*.

It is worthy to note that energy effectiveness as previously defined differs from efficiency/CHP efficiency in that CHP Efficiency is defined as the useful energy-out (combined heat and power) divided by energy-in (energy in the fuel), whereas energy effectiveness also accounts for the energy in the air. By comparison the efficiency/CHP efficiency for gas-turbine-based cogeneration plants are 90+% versus 80+% for a conventional steam plant based cogeneration system.

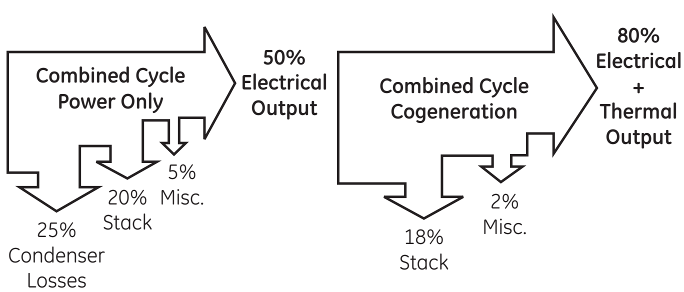


Figure 3. Fuel utilization effectiveness (combined cycle/gas turbine based)

Figure 1 and *Figure 3* clearly illustrate that from a fuel utilization perspective, cogeneration system performances are significantly better than typical steam turbine or gas turbine combined cycles that are designed to only produce power.

Today, across the globe, many local governmental incentives have been established to help promote the development of new cogeneration applications with an objective of driving fuel utilization. One such example is the SPP (Small Power Plant) in Thailand. While such incentives are not new (for example, PURPA in the US), the underlying motivations can be different. More often than not, current incentives are borne out of a want, desire and need to reduce green house gas emissions, whereas the motivations of the past may have focused more on fuel utilization from an energy market perspective (deregulation and market competition). In support of today's market drivers GE not only maintains a position of industry leadership in the areas of gas turbine and gas engine fuel flexibility and emissions capability, but also continues to evolve world-class advanced technology with focused research and development efforts in these areas.

Coincidentally, in the case of the aforementioned regulations a STAG (STeam And Gas) cycle qualification is/was to provide about 6% of its steam generation to process. At this operating condition, the overall performance approaches that of a conventional STAG power generation cycle. Later in this paper, tables are provided that define GE's gas turbine and gas engine product characteristics, which in turn illustrate the wide application range and flexibility of these products to support cogeneration applications.

For purposes of the following discussions, "thermally optimized" cogeneration systems are defined as those developed using non-condensing steam turbine generators or condensing units operated at minimum flow to the condenser for cooling purposes.

Net Heat to Process and Fuel Chargeable to Power

In evaluating and comparing alternative cogeneration cycles, two concepts are key: Net Heat to Process (NHP) and Fuel Chargeable to Power (FCP). Both concepts are "Btu/kJ accounting methods" that can be leveraged to provide normalized performance comparisons between different sized cogeneration systems and different technologies. In turn, the products of these methodologies become the basis of the performance that is used in the economic modeling process.

Net Heat to Process is defined as the net energy supplied by the cogeneration system to the process load, as depicted in *Figure 4*. It is necessary to maintain a constant NHP for all systems being considered, especially when different gas- and steam-turbine configurations export energy to process at different conditions.

Fuel Chargeable to Power (FCP) is a parameter used to define the thermal performance of a topping cogeneration system. The FCP is defined as the incremental fuel for the cogeneration system, relative to the fuel needs of a heat-only system divided by the net incremental power produced by the cogeneration system. Simply put, FCP is the incremental fuel divided by the incremental power (i.e., the incremental heat rate). For a plant generating electric power only (an industrial or a utility), the FCP and net plant heat rate are interchangeable terms commonly expressed in Btu/kWh or kJ/kWh. The FCP concept is illustrated in *Figure 5*.

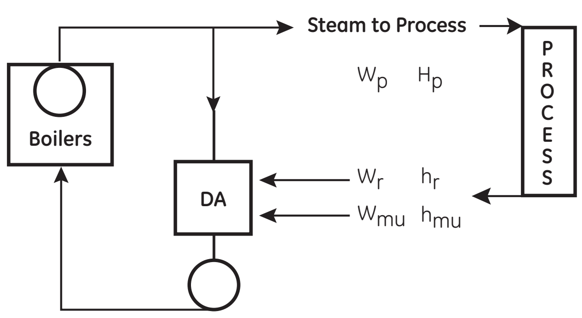
Steam Turbines for Cogeneration

Figure 6 shows several steam turbine configurations that can be used to generate power while satisfying a process need for steam. Steam turbines generally can be designed to meet the specific process heat needs. Unlike gas turbines that are sold in specific sizes or frame sizes, steam turbine generators have traditionally been custom-designed machines and seldom have 100% identical components or capabilities. However, it should be noted that many OEMs (including GE) continue to push more and more toward product and as a minimum component/hardware standardization wherever possible.

Configurations 1, 3 and 4 (illustrated in *Figure 6*) provide steam at a "controlled" pressure, consistent with the process header requirements. Configuration 5 includes two uncontrolled extraction openings in the steam turbine generator and provides steam that would be taken to a common line and pressure-reduced if necessary to meet the pressure requirements in the process. The higher uncontrolled opening would be used during lighter load operation of the turbine, when the pressure at the lower opening is too low for process use. Uncontrolled-extraction turbines of this type are typically used when process extractions are small compared to total turbine flow—or when process needs are fairly constant except during start up, shut down or emergency situations.

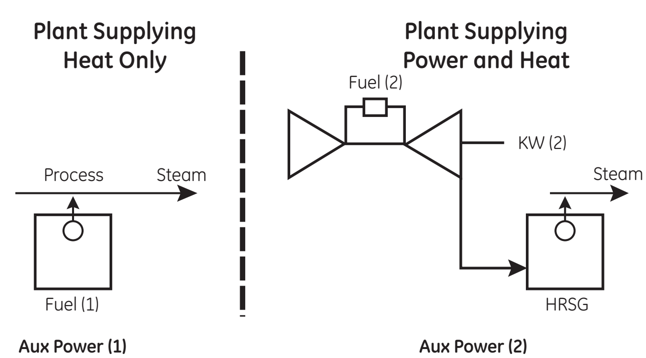
Turbines represented in Configurations 1 and 3 will yield power dependent directly on process demands, since no condensing section capability exists. Their power production depends on the rise and fall of the steam demand. The addition of condensing capability (Configurations 2, 4 and 5) provides added power-generating flexibility. When a condenser is used, power can be generated independently from the process steam demand (assuming that the steam turbine is sized accordingly).

In "thermally optimized" steam turbine cogeneration cycles, steam is expanded in non-condensing or automatic-extraction non-condensing steam turbine-generators that extract and/or exhaust into the process-steam header(s). The FCP for these systems is typically in the 4000 to 4500 Btu/kWh HHV (4220 to 4750 kJ/kWh) range. The influence of initial steam conditions and process steam



$$NHP = W_pH_p - W_rH_r - W_{mu}h_{mu}$$

Figure 4. Net Heat to Process (NHP)



$$FCP = \frac{\text{Fuel (2)} - \text{Fuel (1)}}{\text{KW (2)} - \text{Aux Power (2)} + \text{Aux Power (1)}}$$

Figure 5. Fuel Chargeable to Power (FCP)

pressure on the amount of cogenerated power per 100 million Btu/h (105.5 GJ/h) NHP is illustrated in *Figure 7*. The increase in cogenerated power through the use of higher initial steam conditions, and lower process pressures, is readily apparent.

Studies have shown that higher steam conditions can be economically justified more easily in industrial plants with relatively large process steam demands. Data given in *Figure 8* provide guidance with regard to the initial steam conditions that are normally considered for industrial cogeneration applications.

It should be noted, and it may even be obvious, that there is a correlation between fuel price and/or energy prices and the initial steam condition selection for a given application. Specifically, this illustrates that higher fuel prices and/or energy prices favor the upper portion of the bands shown in *Figure 8*.

Even when utilizing the most effective thermally optimized steam turbine cogeneration systems, the amount of power that can be cogenerated without a condensing section to the steam turbine, per unit of heat energy delivered to process, will usually not exceed

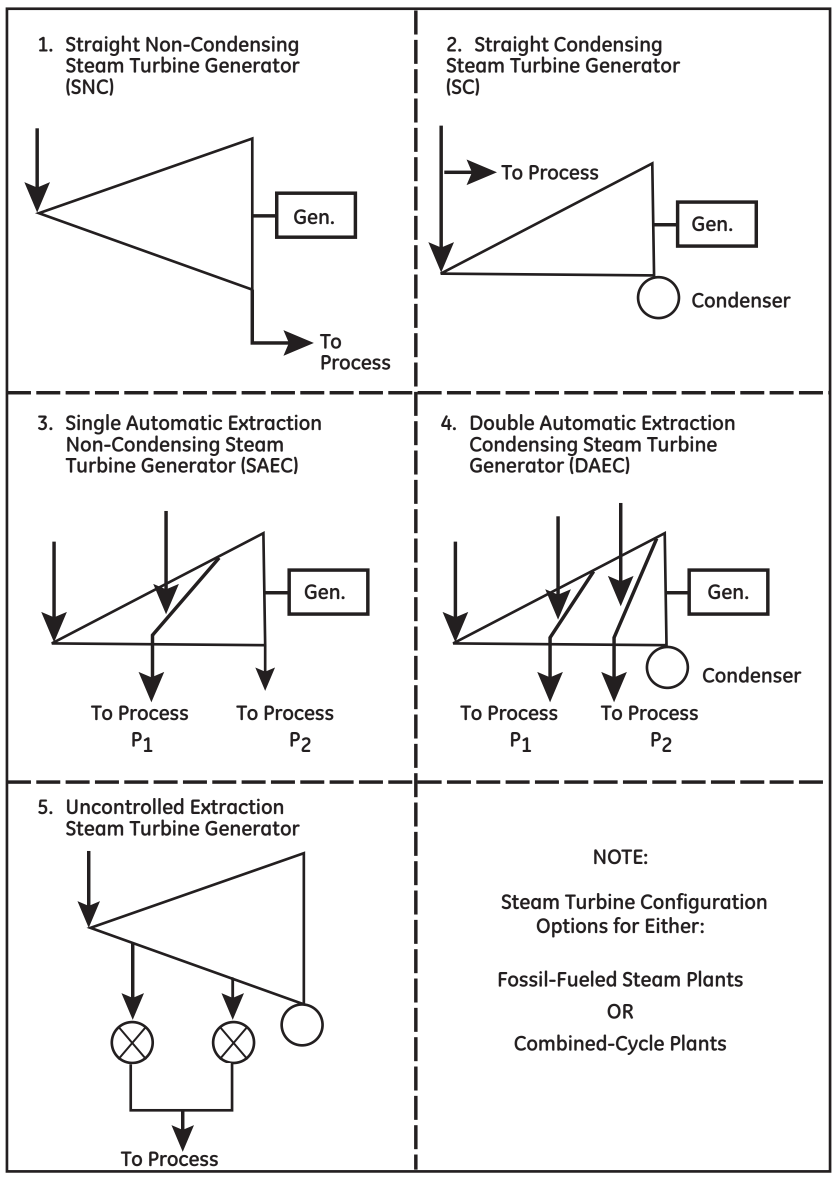
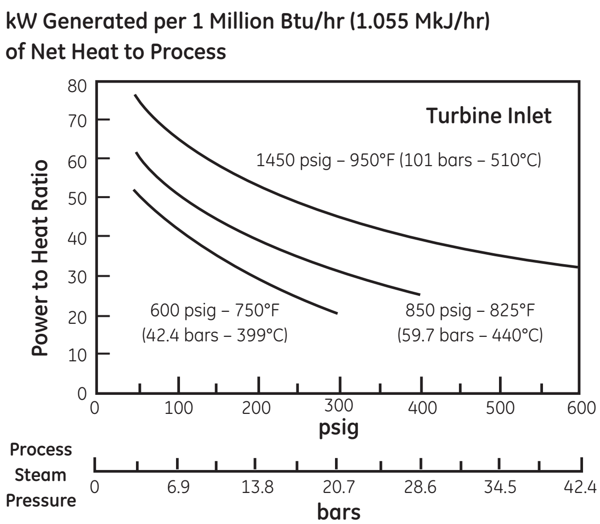


Figure 6. Steam turbine configurations for power generation and process needs

about 85 kW/MBtu (0.6 kW/GJ) net heat supplied. This is generally less power than that required to satisfy most industrial plant electrical energy needs. Thus, with thermally-optimized steam turbine cogeneration systems, a purchased power tie or additional



Basis:

1) Average temperature of process returns and makeup is 165°F (74°C)
2) Power cycle credited for feedwater heating to:
455°F (235°C) for 1450 psig (101 bars)
400°F (204°C) for 850 psig (59.7 bars), and
370°F (188°C) for 600 psig (42.4 bars) systems
3) Turbine efficiency 75%

Figure 7. Cogeneration power with steam turbines

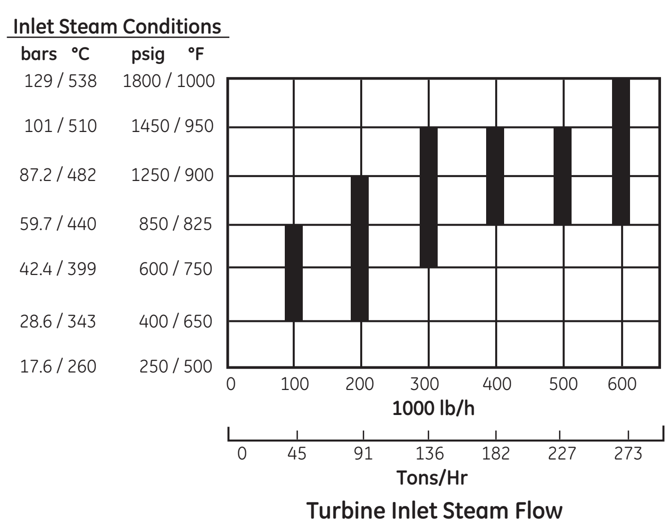


Figure 8. Range of initial steam conditions normally selected for industrial steam turbines

condensing steam turbine is likely necessary to provide the balance of the industrial plant power needs.

Condensing power generation (although not necessarily energy efficient) has proven economic in many industrial applications. Favorable economics are often associated with systems where:

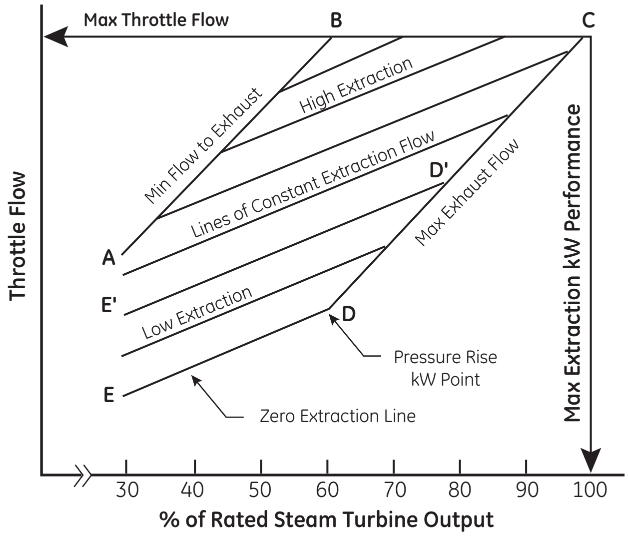
- Condensing power is used to control purchased power demand
- Low-cost fuels or process by-product fuels are available
- Adequate low-level process energy is available for a bottoming cogeneration system
- Condensing provides the continuity of service in critical plant operations where loss of the electric power can cause a major disruption in process operations and/or plant safety
- Utility-specific situations favoring power sales, particularly if low cost fuels are available

Steam Turbine Performance Flexibility

Significant flexibility is achieved when combining a non-condensing turbine with a condensing steam turbine, or when a steam turbine supplies controlled pressure steam to more than one process header. This is accomplished with a single- or double-auto extraction condensing steam turbine generator. *(See Figure 6.)*

Figure 9 illustrates a performance map (flow vs. kilowatt output) for a single auto extraction steam turbine generator. This generic performance map applies equally to single-auto non-condensing and to single-auto condensing steam turbine generators. The maximum throttle flow line (B-C) defines the maximum guarantee steam flow that can be admitted to the high-pressure inlet of the steam turbine, whereas the zero extraction line (E-D) shows the performance of the steam turbine with zero extraction. The line on the far left (A-B) defines the performance of the steam turbine with minimum flow to exhaust. This portion of the curve denotes a turbine operating with only cooling steam being sent to the exhaust of the steam turbine and the balance of steam is extracted. In this area of the curve, the steam turbine is essentially operating as a non-condensing turbine. The sloping lines in the center of the performance map (E'-D') are lines of constant extraction flow.

The performance map (or envelope) flows and kilowatt production accurately define the flexibility of the steam turbine, and in the



Auto-Extracting Condensing Steam Turbines Provide a Wide Range of Power and Heat to Process Control, Independently

Figure 9. Typical single-automatic extraction turbine-performance map

case of a combined cycle, defines much of the flexibility of that cycle as well. It is possible to design the steam turbine for higher maximum throttle flow. In doing so, the high-pressure section of the steam turbine is enlarged and the flow that can be admitted to that section of the turbine is increased. Likewise, the maximum throttle flow line may be lowered, which makes the inlet capability less. A similar change is possible by extending the zero extraction line to the right, allowing the turbine to produce additional kilowatts with zero extraction flow. In this case, the exhaust section of the steam turbine is enlarged.

This tailoring of steam turbine capability to the needs of the industrial process steam user is critical for maximizing the flexibility of the cogeneration project—as well as optimizing the efficiency of the cogeneration system.

Cogeneration and Reheat Steam Turbine Cycles

In most instances, thermal energy in the form of steam is utilized in industrial plants by condensing steam in process heat exchangers. Since most processes require heat transfer at a constant temperature, high degrees of steam superheat are not desirable and de-super heating (steam attemperation) stations are commonly applied to control steam temperatures. In a steam turbine cogeneration cycle, considerable de-super heater spray water would be required if reheat was considered. In fact, in most instances the amount of "thermally optimized" cogenerated power would be less in a reheat cycle compared to a non-reheat cycle, assuming inlet steam conditions are held constant. For example, assuming a 500,000 lb/hr (227 metric ton/hr) process steam demand at 150 psig (10.3 bars) saturated, a non-reheat cycle with 1450 psig/950°F (100 bars/510°C) initial steam conditions would deliver about 28 MW. A reheat cycle with 1450 psig/950°F/950°F (100 bars/510°C/510°C) would generate about 27.3 MW, or 2.5% less power. In addition, the cycle complexity due to reheat would increase the cost of the turbine, boiler and associated systems relative to the non-reheat case. The economics of reheat steam turbines are enhanced in cogeneration when most of the steam is expanded to the condenser to produce electric power, i.e., for applications requiring high power to heat ratios.

Cogeneration with Gas Reciprocating Engines

Gas Engines

Reciprocating engine generator sets and cogeneration systems are well suited to fulfil many decentralized energy supply needs. Some key features of our products include:

- High electrical efficiencies up to 43%
- Overall efficiencies (electrical and thermal) over 90%
- Minimum NO_x emissions through the patented LEANOX®
- Lean mixture combustion
- Specially designed gas engines for utilization of alternative, renewable energy sources (e.g., biogas or landfill gas) and special gases (e.g., coal mine gas or coke gas)
- Maximum operational safety and availability
- High power density

Through supply of energy directly at the load source, it is also possible to reduce or avoid altogether transport and distribution losses.

Cogeneration – Overall Efficiencies

With combined power and heat generation (cogeneration) the waste heat incurred during engine operation is recovered and utilized to satisfy thermal system process requirements for low-grade steam and/or hot water. In many cases this utilization of waste heat results in overall systems of efficiencies of up to 90+% (thermal + electrical). This efficient form of energy conversion is able to achieve primary energy savings of about 40% using gas engine cogeneration systems, compared with conventional separate power and heat generation. Figure 10 represents a typical Sankey diagram illustrating the Electrical/Thermal energy utilization from a typical Jenbacher gas reciprocating engine. In addition, the Sankey diagram also identifies the various heat sources from the engine (from which useful thermal energy can be extracted).

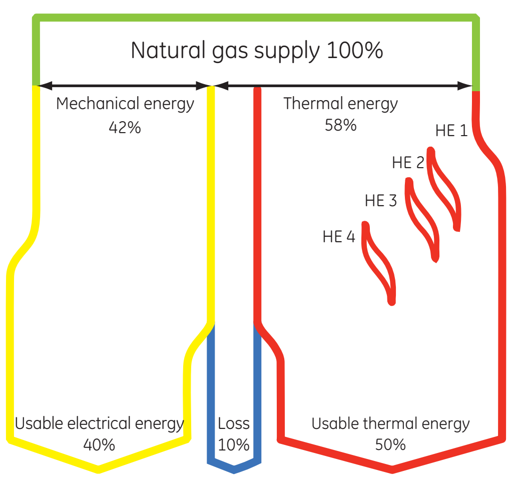
Power and Heat Utilization

In general, for typical cogeneration applications utilizing gas reciprocating engines, the power generated is for localized applications of heat and power. The combined heat and power loads are primarily based upon the consumption requirements of an individual facility (e.g., hospitals) with excess electrical and/or thermal energy fed into the public power grids and/or district heating systems respectively. It should be noted that the thermal energy can be used for either (or both) the generation or hot water and/or steam production—as well as for various types of process heat. Gas engine cogeneration systems are also used for CO_2 fertilization in greenhouses and trigeneration systems (combined generation of power, heat, and cooling).

Generation of heating water. Cogeneration systems capture excess heat from the engine, which can be used to generate heating water that can then be utilized by local or district heating systems to cover their basic heat requirements. Peak heat demand can be covered through the combined use of a buffer and a peak boiler. Due to varying heat demands during the year, multi-engine-installations are the preferred solution for district heating systems.

Steam production and drying processes. Roughly 50% of the thermal energy generated in a gas engine consists of exhaust gas heat with a temperature of approximately 400°to 500°C and can be utilized for the production of steam. The remaining waste heat contained in the engine cooling water, oil, or air/fuel gas mixture, can be utilized for feeding water preheating. Applications include processed steam for industrial operations; hospitals to meet their requirement for sterilization steam; and foodstuff processing operations. The exhaust gas from the gas engines can also be utilized directly or indirectly for drying processes (e.g., in brick works, the ceramic industry, and animal feed drying). Overall efficiencies of more than 98% can be achieved through the recovery of the heat discharged from the cogeneration plant by way of heat exchangers and the exhaust and radiation heat.

CHP systems utilize the waste heat incurred during engine operation to generate overall plant efficiencies of more than 90%.



HE 1
Mixture intercooler ~ 7-10%

HE 2
Oil exchange heater ~ 4-5%

HE 3
Engine jacket water heat exchange ~ 8-13%

HE 4
Exhaust gas heat exchanger ~ 20–25%

Figure 10. Sankey diagram of typical CHP application with typical gas reciprocating engine

CO_2 fertilization in greenhouses. Heat, light and CO_2 promote the growth of plants in greenhouses. With artificial lighting, plants absorb even more CO_2. If the greenhouse atmosphere is enriched with CO_2, plant growth and consequently the harvest yield can be increased by up to 40%. This process—also called CO_2 fertilization—is able to make use of the CO_2 contained in the exhaust gas of a gas engine through catalytic converter purification. As a result, greenhouses utilizing gas engine cogeneration systems can cover the power and heat requirement for the artificial lighting and heating in an economical manner, while effectively utilizing CO_2 of the engine exhaust gas.

Trigeneration. The combination of gas engines with absorption chillers is an optimal solution for generating air conditioning and/or refrigeration. The waste heat from the mixture intercooler, the engine oil, the engine cooling water, and the exhaust gas serves as drive energy for the chillers. Combining a cogeneration plant unit with an absorption refrigeration system allows utilization of seasonal excess heat for cooling. Using trigeneration, it is possible to achieve overall efficiencies (power and air conditioning and/or refrigeration) of up to 75%, increasing both annual capacity and overall plant efficiency.

Figures 11 and 12 are graphical representations of both the electrical and thermal energy available from GE's 50 and 60 Hz Jenbacher gas reciprocating engines as a function of engine model. As identified on the graphs the thermal output is based upon heating an incoming water source from 70°C to a discharge temperature of 90°C.

Tables 1a, 1b, 2a and 2b are performance summary tables for GE's 50 and 60 Hz gas reciprocating engines. For convenience data has been provided in both SI and English units.

Fuel Flexibility and Gas Reciprocating Engines

One of the more significant characteristics/values of GE's Jenbacher gas engines is its fuel flexibility. Not only are standard engines capable of burning natural gas, but specialized engines also are available for burning unique gases such as flare gas and other specialized gases (such as biogas, landfill, sewage, coal mine and coke gases). The value of the standard engines is that they are well suited for production of reliable, decentralized energy as supported by a well-established, reliable natural gas supply system infrastructures. The value of the gas engines revolves around the fact that they are capable of generating useful energy (both heat and power) from fuel sources that otherwise would have been wasted and which otherwise would have served as an ecological detriment. To this end, biogas, landfill gas, and coal mine gas-fueled engines have been certified as GE "ecomagination" products by an independent agency. It is also worthy to note that in addition to having a wide range of fuel flexibility in terms of fuel composition and heat characteristics, another value which reciprocating engines afford is that they are capable of burning low pressure fuels. Thus, they require little or no fuel gas compression (as compared to alternative technology solutions).

Flare gas. This is an associated gas obtained during crude oil exploration, largely consisting of methane and higher hydrocarbons. The use of flare gas—which is generally available free of charge as a waste product—ensures a fuel source for on-site power generation and, if required, the engines can also provide a heat supply for surrounding facilities. Consequently this problem gas, instead of flaring it off while causing ecological exposure, can be used economically and practically.

Biogas. For a wide range of organic substances from agriculture, foodstuff, and feed industries, anaerobic fermentation is a superior alternative to composting. Biogas—a mixture of methane and carbon dioxide—is formed in the fermentation process of a wide range of organic substances from the agriculture, foodstuff, and feed industries. It is a high-energy fuel with a calorific value of 5–6 kWh/m^3N that can substitute fossil fuel energy. Due to the organic nature of the components of biogas, burning it in a gas engine for power generation emits the same amount of CO_2 into the atmosphere as was originally absorbed during the process of photosynthesis in the natural CO_2 cycle. Using biogas in gas engines promotes proper waste disposal, and allows the use of the end products from the fermentation process as fertilizer.

Landfill gas. This is biogas in that it is formed as a result of the decomposition of organic substances. It consists of methane (CH_4), carbon dioxide (CO_2), nitrogen (N_2), and a small amount of oxygen (O_2). To allow for proper re-cultivation of landfills, prevent offensive smells, smouldering fires, or the migration of landfill gas into the waterways these gases must be continuously extracted under controlled conditions. With a calorific value of about 4 to 5 kWh/m^3N, landfill gas constitutes a high-value fuel for gas engines and can therefore be economically utilized for power generation.

GE Jenbacher, Gas Reciprocating Engine, Generator Drive - Electrical & Thermal Performance Summary (50 Hz, SI Units)

Technical Data		**J 208 GS**	**J 312 GS**	**J 316 GS**
Expression		C05	C05	C05
Fuel gas type		Natural Gas	Natural Gas	Natural Gas
Fuel gas pressure	mbar	80-200	80-200	80-200
Based on methane number	MN	70	70	70
Max. inlet cooling water temp. (intercooler)	°C	40	40	40
Specific lube oil consumption	g/kWh	0.3	0.3	0.3
Mean efficiency pressure at stand. power and nom. speed	bar	16.50	17.70	17.70
ISO standard fuel stop power ICFN	kW	342	646	861
Speed	1/min	1,500	1,500	1,500
Electrical output (cos phi =1.0)	kW el.	330	625	834
Electrical output (cos phi =0.8)	kW el.	327	617	825
Recoverable thermal output (hot water 70/90°C)	kW	363	735	994
Electrical efficiency (cos phi =1.0)	%	38.78	39.86	39.94
Thermal efficiency (hot water 70/90°C)	%	42.63	46.89	47.58
Total efficiency	%	81.41	86.75	87.52
Technical Data of Engine				
Jacket-water temperature max.	°C	90	90	90
Energy input [LHV]	kW	851	1,567	2,089
Intercooler 2nd stage	kW	64	47	43
Intercooler 1st stage	kW	~	90	125
Oil – heat	kW	39	68	90
Engine jacket water – heat	kW	117	193	267
Exhaust gas 120°C	kW	207	384	512
Surface heat	kW	21	30	42
Balance heat	kW	9	16	21
Exhaust gas temperature at full load	°C	478	485	485
Exhaust gas mass flow rate, wet	kg/h	1,843	3,355	4,473
NO_x	mg/Nm³@5%O_2	500	500	500
Technical Data of Generator		50 Hz/400 V	50 Hz/400 V	50 Hz/400 V
Technical Data of Hydraulic				
Return temperature	°C	70	70	70
Forward temperature	°C	90	90	90
Hot water flow rate	m³/h	15.6	31.6	42.7

Note:
- All energy values in kWh are based on 25°C and 1013.25 mbar according DIN ISO 3046/ DIN 51850
- All volume flows in Nm³ are based on 0°C and 1013.25 mbar according SI standard
- Emission values in mg/Nm³ based on dry exhaust; 5%O_2

Table 1a. Gas Reciprocating Engine Performance Summary Table – 50 HZ – SI Units

GE Jenbacher, Gas Reciprocating Engine, Generator Drive - Electrical & Thermal Performance Summary (50 Hz, SI Units)

J 320 GS	J 412 GS	J 416 GS	J 420 GS	J 612 GS	J 616 GS	J 620 GS
C05	A05	A05	A05	F11	F11	F11
Natural Gas	Natural Gas	Natural Gas	Natural Gas	Natural Gas	Natural Gas	Natural Gas
80-200	80-200	80-200	80-200	80-200	80-200	80-200
70	70	70	70	70	70	70
40	40	40	40	35	40	40
0.3	0.3	0.3	0.3	0.3	0.3	0.3
18.00	19.00	19.00	19.00	22.00	20.00	20.00
1,095	871	1,161	1,451	2,058	2,495	3,119
1,500	1,500	1,500	1,500	1,500	1,500	1,500
1,063	844	1,131	1,415	1,998	2,423	3,045
1,051	834	1,122	1,402	1,980	2,398	3,025
1,208	865	1,155	1,442	1,943	2,340	2,961
40.82	42.69	42.90	42.94	43.91	43.90	44.15
46.36	43.75	43.81	43.76	42.69	42.40	42.92
87.18	86.44	86.71	86.70	86.60	86.30	87.08
90	90	90	90	95	95	95
2,605	1,977	2,636	3,295	4,551	5,518	6,897
65	53	71	88	94	116	145
196	164	219	274	470	527	695
118	110	147	183	184	237	296
352	217	290	362	337	419	524
542	374	499	623	952	1,157	1,446
51	42	56	71	109	133	166
26	20	26	33	46	55	69
427	390	390	390	405	400	400
5,675	4,494	5,993	7,491	10,851	13,444	16,806
500	500	500	500	500	500	500
50 Hz/400 V	50 Hz/400 V	50 Hz/400 V	50 Hz/400 V	50 Hz/10,5 kV	50 Hz/10,5 kV	50 Hz/6,3 kV
70	70	70	70	70	70	70
90	90	90	90	90	90	90
51.9	37.2	49.7	62.0	83.5	100.6	127.3

GE Jenbacher, Gas Reciprocating Engine, Generator Drive - Electrical & Thermal Performance Summary (50 Hz, English Units)

Technical Data		J 208 GS	J 312 GS	J 316 GS
Expression		C05	C05	C05
Fuel gas type		Natural gas	Natural gas	Natural gas
Fuel gas pressure	psi	1,2 - 3,0	1,2 - 3,1	1,2 - 3,2
Based on methane number	MN	70	70	70
Max. inlet cooling water temp. (intercooler)	°F	104	104	104
Specific lube oil consumption	g/bhp.hr	0.2	0.2	0.2
Mean efficiency pressure at stand. power and nom. speed	psi	239	257	257
ISO standard fuel stop power ICFN	bhp	459	866	1,155
Speed	rpm	1,500	1,500	1,500
Electrical output (cos phi =1.0)	kW el.	330	625	834
Electrical output (cos phi =0.8)	kW el.	327	617	825
Recoverable thermal output (hot water 158/194°C)	MBTU/hr	1,238	2,507	3,392
Electrical efficiency (cos phi =1.0)	%	38.78	39.86	39.94
Thermal efficiency (hot water 158/194°C)	%	42.63	46.89	47.58
Total efficiency	%	81.41	86.75	87.52
Technical Data of Engine				
Jacket-water temperature max.	°F	194	194	194
Energy input (LHV)	MBTU/hr	2,904	5,347	7,128
Intercooler 1st stage	MBTU/hr	218	160	147
Intercooler 2nd stage	MBTU/hr	0	307	427
Oil – heat	MBTU/hr	133	232	307
Engine jacket water – heat	MBTU/hr	399	659	911
Exhaust gas 248°F	MBTU/hr	706	1,310	1,747
Surface heat	MBTU/hr	72	102	143
Balance heat	MBTU/hr	31	55	72
Exhaust gas temperature at full load	°F	892	905	905
Exhaust gas mass flow rate, wet	lbs/hr	4,064	7,398	9,863
NO_x	g/bhp.hr	1	2	3
Technical Data of Generator		50 Hz/400 V	50 Hz/400 V	50 Hz/400 V
Technical Data of Hot Water Heat Recovery				
Return temperature	°F	158	158	158
Forward temperature	°F	194	194	194
Hot water flow rate	GPM	68.6	139.0	188.1
Technical data of gearbox:		no	no	no
Type		~	~	~
Recoverable thermal output	MBTU/hr	~	~	~
Efficiency	%	~	~	~
Mass	lbs	~	~	~

Note:
- All energy values in kWh are based on 25°C and 1013.25 mbar according DIN ISO 3046/ DIN 51850
- All volume flows in Nm³ are based on 0°C and 1013.25 mbar according SI standard
- Emission values in mg/Nm³ based on dry exhaust; 5%O_2

Table 1b. Gas Reciprocating Engine Performance Summary Table – 50 HZ – English Units

GE Jenbacher, Gas Reciprocating Engine, Generator Drive - Electrical & Thermal Performance Summary (50 Hz, English Units)

J 320 GS	J 412 GS	J 416 GS	J 420 GS	J 612 GS	J 616 GS	J 620 GS
C05	A05	A05	A05	F11	F11	F11
Natural gas	Natural gas	Natural gas	Natural gas	Natural gas	Natural gas	Natural gas
1,2 - 3,3	1,2 - 3,4	1,2 - 3,5	1,2 - 3,6	1,2 - 3,7	1,2 - 3,8	1,2 - 3,9
70	70	70	70	70	70	70
104	104	104	104	95	104	104
0.2	0.2	0.2	0.2	0.2	0.2	0.2
261	276	276	276	319	290	290
1,468	1,168	1,557	1,946	2,760	3,346	4,183
1,500	1,500	1,500	1,500	1,500	1,500	1,500
1,063	844	1,131	1,415	1,998	2,423	3,045
1,051	834	1,122	1,402	1,980	2,398	3,025
4,121	2,951	3,940	4,920	6,630	7,982	10,101
40.82	42.69	42.90	42.94	43.91	43.90	44.15
46.36	43.75	43.81	43.76	42.69	42.40	42.92
87.18	86.44	86.71	86.70	86.60	86.30	87.08
194	194	194	194	203	203	203
8,888	6,746	8,994	11,243	15,528	18,827	23,533
222	181	242	300	321	396	495
669	560	747	935	1,604	1,798	2,371
403	375	502	624	628	809	1,010
1,201	740	989	1,235	1,150	1,430	1,788
1,849	1,276	1,703	2,126	3,248	3,948	4,934
174	143	191	242	372	454	566
89	68	89	113	157	188	235
801	734	734	734	761	752	752
12,513	9,909	13,215	16,518	23,926	29,644	37,057
4	5	6	7	8	9	10
50 Hz/400 V	50 Hz/400 V	50 Hz/400 V	50 Hz/400 V	50 Hz/10,5 kV	50 Hz/10,5 kV	50 Hz/6,3 kV
158	158	158	158	158	158	158
194	194	194	194	194	194	194
228.5	163.7	218.5	272.8	367.6	442.6	560.1
no	no	no	no	no	no	no
~	~	~	~	~	~	~
~	~	~	~	~	~	~
~	~	~	~	~	~	~
~	~	~	~	~	~	~

GE Jenbacher, Gas Reciprocating Engine, Generator Drive - Electrical & Thermal Performance Summary (60 Hz, SI Units)

Technical Data		**J 208 GS**	**J 312 GS**
Expression		C85	C85
Fuel gas type		Natural Gas	Natural Gas
Fuel gas pressure	mbar	80-200	80-200
Based on methane number	MZ	75	75
Max. inlet cooling water temp. (intercooler)	°C	50	50
Specific lube oil consumption	g/kWh	0.3	0.3
Mean efficiency pressure at stand. power and nom. speed	bar	14.00	15.00
ISO standard fuel stop power ICFN	kW	349	657
Speed	1/min	1,800	1,800
Electrical output (cos phi =1.0)	kW el.	335	633
Electrical output (cos phi =0.8)	kW el.	332	629
Recoverable thermal output (hot water 70/90°C)	kW	402	814
Mechanical efficiency	%	38.78	39.53
Electrical efficiency (cos phi =1.0)	%	37.23	38.11
Thermal efficiency (hot water 70/90°C)	%	44.64	48.99
Total efficiency	%	81.87	87.10
Technical Data of Engine			
Jacket-water temperature max.	°C	90	90
Energy input (LHV)	kW	900	1,662
Intercooler 2nd stage	kW	53	35
Intercooler 1st stage	kW	~	90
Oil – heat	kW	44	80
Engine jacket water – heat	kW	128	211
Exhaust gas 120°C	kW	230	433
Surface heat	kW	20	39
Balance heat	kW	19	17
Exhaust gas temperature at full load	°C	510	505
Exhaust gas mass flow rate, wet	kg/h	1,916	3,579
NO_x	mg/Nm³@5%O_2	500	500
Technical Data of Generator		60 Hz/480 V	60 Hz/480 V
Technical Data of Hot Water Heat Recovery			
Return temperature	°C	70	70
Forward temperature	°C	90	90
Hot water flow rate	m³/h	17.3	35.0
Technical Data of Gearbox		no	no
Type		~	~
Recoverable thermal output	kW	~	~
Efficiency	%	~	~
Mass	kg	~	~

Note:
- All energy values in kWh are based on 25°C and 1013.25 mbar according DIN ISO 3046/ DIN 51850.
- All volume flows in Nm³ are based on 0°C and 1013.25 mbar according SI standard.
- Emission values in mg/Nm³ based on dry exhaust; 5%O_2.

Table 2a. Gas Reciprocating Engine Performance Summary Table – 60 HZ – SI Units

GE Jenbacher, Gas Reciprocating Engine, Generator Drive - Electrical & Thermal Performance Summary (60 Hz, SI Units)

J 316 GS	J 320 GS	J 420 GS	J 612 GS	J 616 GS	J 620 GS
C85	C85	A85	F11	F11	F11
Natural Gas	Natural Gas	Natural Gas	Natural Gas	Natural Gas	Natural Gas
80-200	80-200	80-200	80-200	80-200	80-200
75	75	70	70	70	70
50	50	50	35	40	40
0.3	0.3	0.3	0.3	0.3	0.3
15.00	15.00	16.00	22.00	20.00	20.00
876	1,095	1,466	2,058	2,495	3,119
1,800	1,800	1,800	1,500	1,500	1,500
848	1059	1426	1951	2390	2994
840	1049	1416	1951	2368	2974
1,089	1,324	1,610	1,963	2,372	3,002
39.53	40.33	41.96	45.22	45.22	45.22
38.27	39.00	40.82	42.86	43.32	43.42
49.15	48.78	46.07	43.13	42.99	43.53
87.42	87.78	86.90	85.99	86.31	86.94
90	90	90	95	95	95
2,216	2,715	3,494	4,551	5,518	6,897
47	50	74	94	116	145
120	176	206	470	527	695
108	129	129	184	237	296
284	356	468	337	419	524
577	663	807	952	1,157	1,446
47	51	71	109	133	166
22	27	35	46	55	69
505	487	451	405	400	400
4,772	5,767	7,828	10,851	13,444	16,806
500	500	500	500	500	500
60 Hz/480 V	60 Hz/480 V	60 Hz/480 V	60 Hz/480 V	60 Hz/480 V	60 Hz/4,16 kV
70	70	70	70	70	70
90	90	90	90	90	90
46.8	56.9	69.2	84.4	102.0	129.1
no	no	no	yes	yes	yes
~	~	~	ANO - 090	ANO - 110	ANO - 110
~	~	~	20	33	42
~	~	~	98.7	98.6	98.6
~	~	~	1035	2500	2500

GE Jenbacher, Gas Reciprocating Engine, Generator Drive - Electrical & Thermal Performance Summary (60 Hz, English Units)

Technical Data		**J208**	**J312**
Expression		C85	C85
Fuel gas type		Natural gas	Natural gas
Fuel gas pressure	psi	1,2 - 2,9	1,2 - 2,9
Based on methane number	MN	75	75
Max. inlet cooling water temp. (intercooler)	°F	122	122
Specific lube oil consumption	g/bhp.hr	0.2	0.2
Mean efficiency pressure at stand. power and nom. speed	psi	203	218
ISO standard fuel stop power ICFN	bhp	468	881
Speed	rpm	1,800	1,800
Electrical output (cos phi =1.0)	kW el.	335	633
Electrical output (cos phi =0.8)	kW el.	332	629
Recoverable thermal output (hot water 158/194°C)	MBTU/hr	1,372	2,777
Electrical efficiency (cos phi =1.0)	%	37.23	38.11
Thermal efficiency (hot water 158/194°C)	%	44.67	48.98
Total efficiency	%	81.87	87.10
Technical Data of Engine			
Jacket-water temperature max.	°F	194	194
Energy input (LHV)	MBTU/hr	3,071	5,671
Intercooler 1st stage	MBTU/hr	~	307
Intercooler 2nd stage	MBTU/hr	181	119
Oil – heat	MBTU/hr	150	273
Engine jacket water – heat	MBTU/hr	437	720
Exhaust gas 248°F	MBTU/hr	785	1477
Surface heat	MBTU/hr	68	133
Balance heat	MBTU/hr	65	58
Exhaust gas temperature at full load	°F	950	941
Exhaust gas mass flow rate, wet	lbs/hr	4,224	7,890
NO_x	g/bhp.hr	1.1	1.1
Technical Data of Generator:		60 Hz/480 V	60 Hz/480 V
Technical Data of Hot Water Heat Recovery			
Return temperature	°F	158	158
Forward temperature	°F	194	194
Hot water flow rate	GPM	76.2	154.4
Technical Data of Gearbox		no	no
Type		~	~
Recoverable thermal output	MBTU/hr	~	~
Efficiency	%	~	~
Mass	lbs	~	~

Note:

- All energy values in kWh are based on 25°C and 1013.25 mbar according DIN ISO 3046/ DIN 51850.
- All volume flows in Nm³ are based on 0°C and 1013.25 mbar according SI standard.
- Emission values in mg/Nm³ based on dry exhaust; 5%O_2.

Table 2b. Gas Reciprocating Engine Performance Summary Table – 60 HZ – English Units

GE Jenbacher, Gas Reciprocating Engine, Generator Drive - Electrical & Thermal Performance Summary (60 Hz, English Units)

J316	J320	J420	J612	J616	J620
C85	C85	A85	F11	F11	F11
Natural gas	Natural gas	Natural gas	Natural gas	Natural gas	Natural gas
1,2 - 2,9	1,2 - 2,9	1,2 - 2,9	1,2 - 2,9	1,2 - 2,9	1,2 - 2,9
75	75	70	70	70	70
122	122	122	95	104	104
0.2	0.2	0.2	0.2	0.2	0.2
218	218	232	319	290	290
1,175	1,468	1,966	2,760	3,346	4,183
1,800	1,800	1,800	1,500	1,500	1,500
848	1059	1426	1951	2390	2994
840	1049	1416	1951	2368	2974
3,716	4,518	5,493	6,698	8,095	10,245
38.27	39.00	40.82	42.86	43.32	43.42
49.14	48.77	46.08	43.13	43.00	43.53
87.42	87.78	86.90	85.99	86.31	86.94
194	194	194	203	203	203
7,561	9,264	11,922	15,529	18,828	23,534
409	601	703	1,604	1,798	2,371
160	171	252	321	396	495
369	440	440	628	809	1,010
969	1,215	1,597	1,150	1,430	1,788
1969	2262	2753	3248	3948	4934
160	174	242	372	454	566
75	92	119	157	188	235
941	909	844	761	752	752
10,520	12,714	17,258	23,922	29,639	37,051
1.1	1.1	1.1	1.1	1.1	1.1
60 Hz/480 V	60 Hz/480 V	60 Hz/480 V	60 Hz/480 V	60 Hz/480 V	60 Hz/4,16 kV
158	158	158	158	158	158
194	194	194	194	194	194
206.5	251.0	305.1	372.1	449.7	569.1
no	no	no	yes	yes	yes
~	~	~	ANO - 090	ANO - 110	ANO - 110
~	~	~	68	111	142
~	~	~	98.7	98.6	98.6
~	~	~	2282	5512	5512

Sewage gas. This is a by-product of the fermentation of sewage sludge, typically consisting of 60–70% methane and 30–40% carbon dioxide. This composition makes sewage gas highly suitable for combustion in gas engines. The electrical energy produced by the gas engine can be utilized for the treatment plant as well as for feeding into the public power grid. The thermal energy can be used to heat the sewage sludge or to offset the treatment plant's other heat requirements.

Gas Turbine and Combined Cycles

Gas turbine combined cycle plants are often suitable for both large scale/centralized cogeneration applications as well as decentralized CHP/cogeneration requirements. Some of the key features of our products include:

- A large range of electrical and thermal output, as well as the flexibility to tune the ratio of electrical-to-thermal output (as a

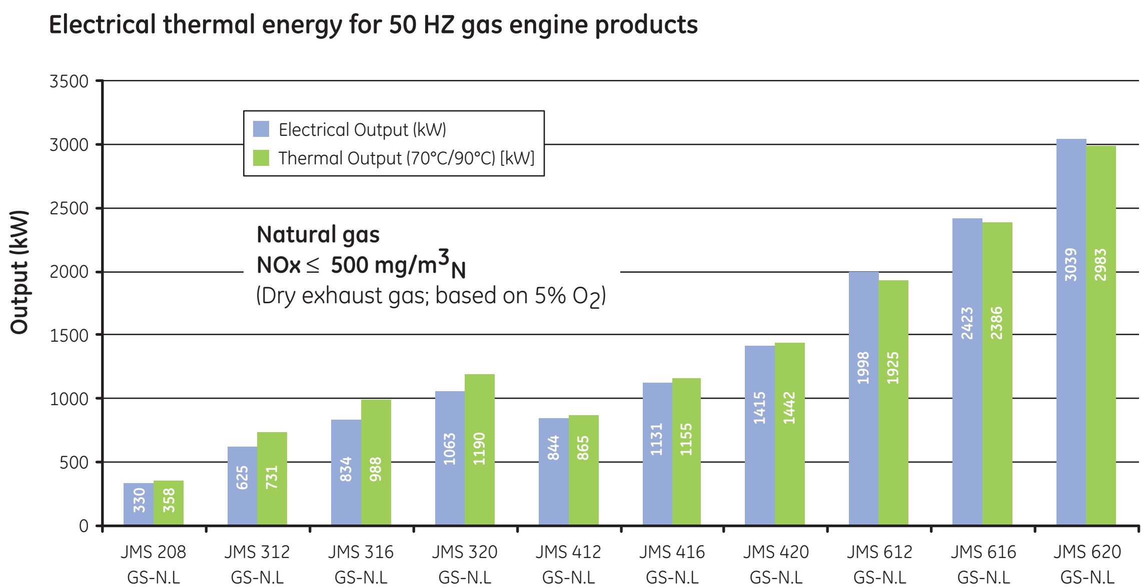


Figure 11. Electrical thermal energy for 50 HZ gas engine products

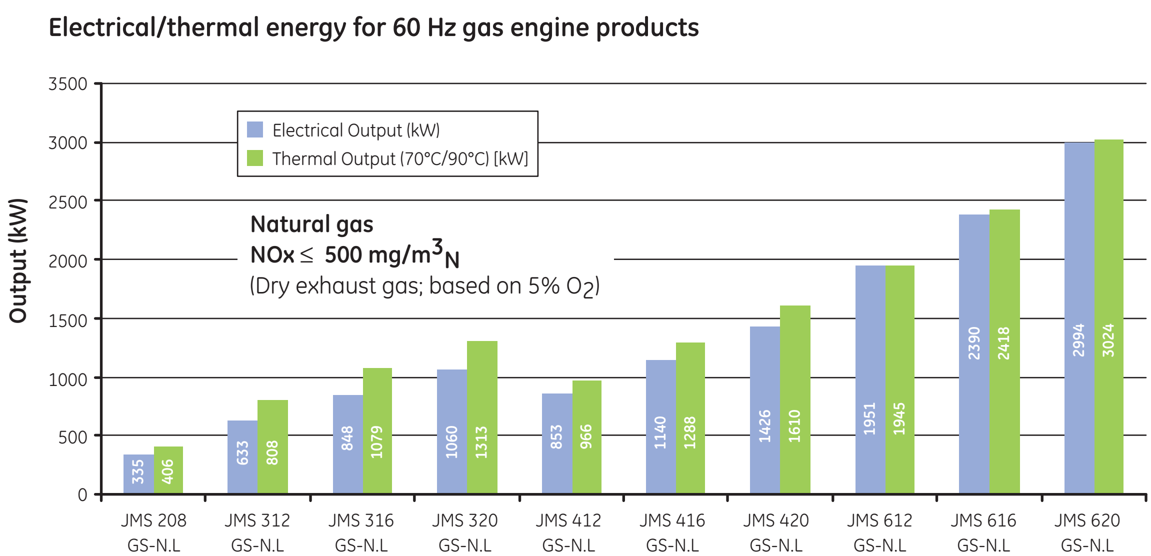


Figure 12. Electrical/thermal energy for 60 Hz gas engine products

result of a significant portfolio of heavy duty and aeroderivative gas turbines, coupled with cycle flexibility that is inherent to combined cycle cogeneration)

- Overall efficiencies (electrical and thermal) of 90-95% for unfired and fired applications and higher yet for fully fired combined cycle, cogeneration applications
- A wide range of liquid and gas fuel capability/flexibility
- Industry leadership NO_x capability via DLN/DLE combustion systems
- Industry leadership in gas turbine experience, reliability and availability

Gas turbine cycles provide an opportunity to generate two-to-four times more power output per unit of heat required in process, relative to the "thermally optimized" steam turbine cogeneration systems as defined in the Cogeneration section of this document. Historically it has been this characteristic, combined with a favorable FCP and proven reliability, that has made this technology widely accepted in applications where suitable fuels are economically available.

Figures 13 and 14 represent a characterization of the nominal electrical and thermal output capability of both our heavy-duty gas turbine products as well as our aeroderivative units respectively. It should be noted that the thermal output capability reflected by this graphic is relative to ISO ambient conditions versus a pre-defined steam and/or hot water requirement.

Another significant driver—in a world now more environmentally conscious than ever—is a global thrust to develop projects which not only minimize GHG emissions but also help to displace GHG emissions from other industrial sources. Thus, one of the more significant advantages of gas-turbine and engine-based cogeneration is the potential for significant GHG reductions. Further, it is the monetization of GHG reductions and other possible financial incentives that are helping to encourage and facilitate the development of cogeneration applications.

Gas Turbine Power Enhancements

The gas turbine is an air-breathing engine that responds to the mass flow entering its compressor. For constant speed units, the gas turbine output will generally vary in proportion to the inlet air temperature (density) as shown for the MS6001B in *Figure 15*. GE's aeroderivative, multi-shaft designs use a variety of parameters in their control logic and therefore can have a different operating profiles as illustrated in *Figure 15*.

The gas turbine output may be enhanced at high ambient temperatures and low humidity levels by application of an

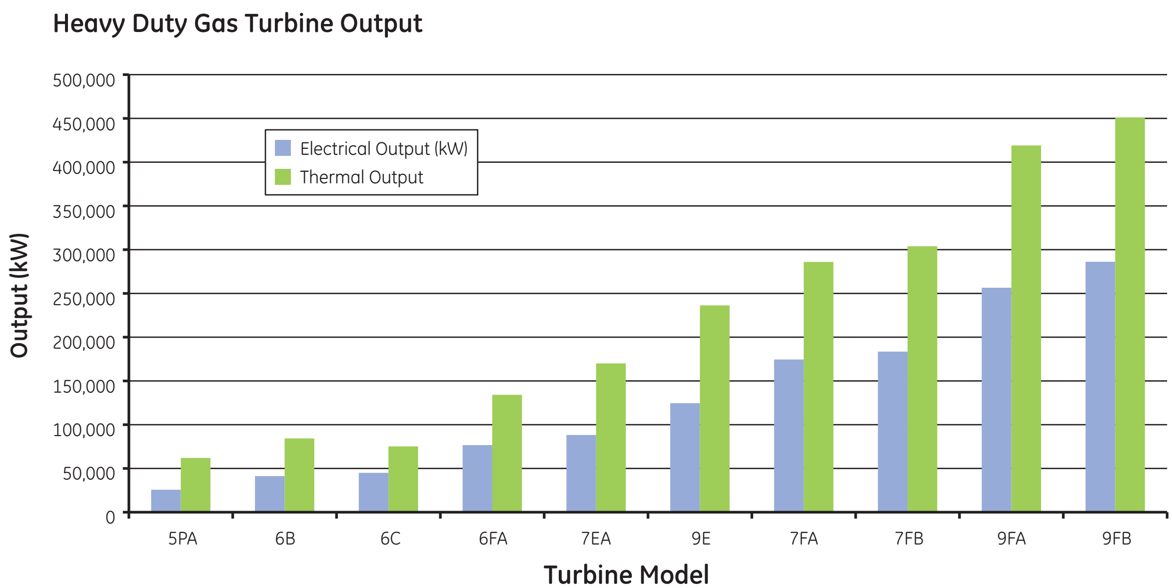


Figure 13. Electrical thermal energy for 50/60 Hz heavy-duty gas turbines

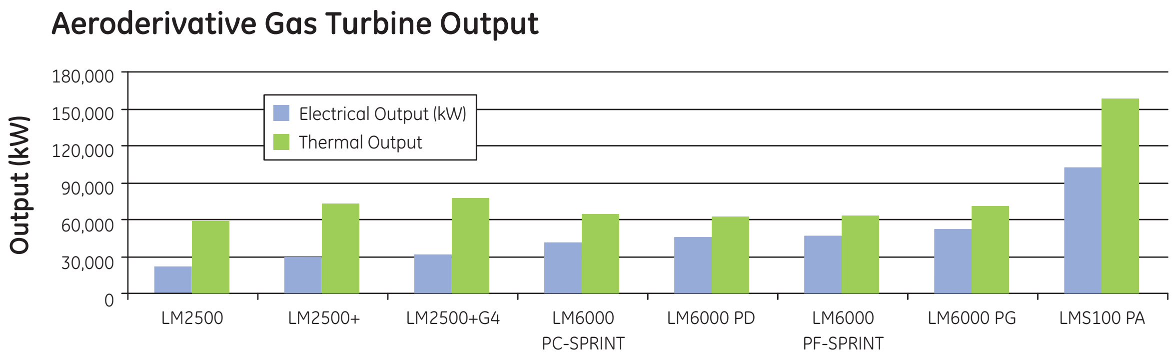
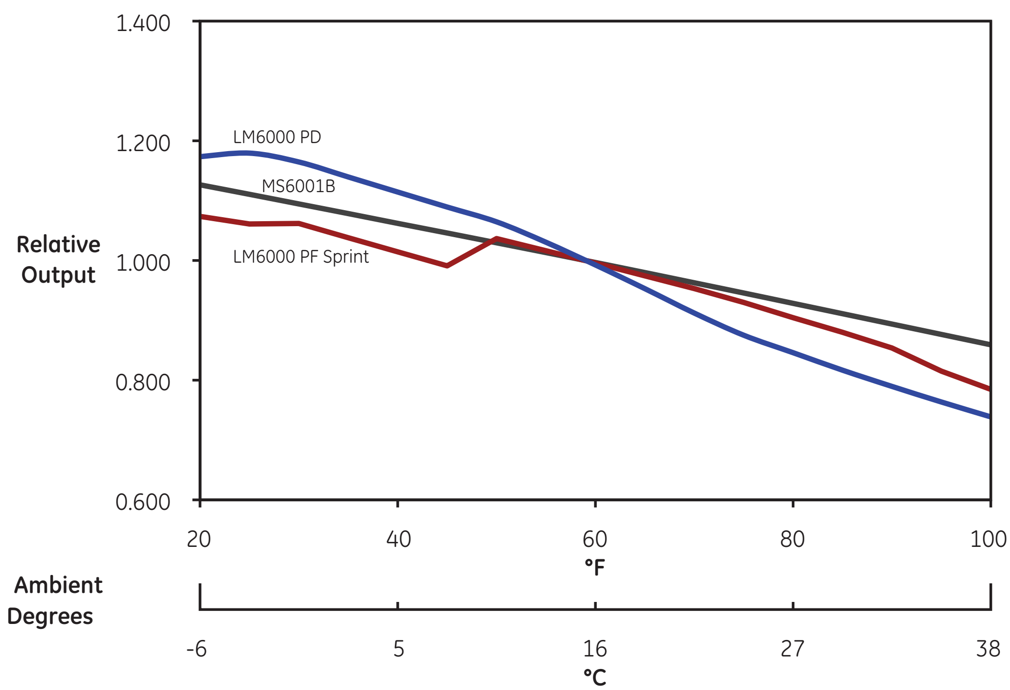


Figure 14. Electrical/thermal energy for 50/60 Hz aeroderivative gas turbines

evaporative cooler. This system decreases the compressor inlet temperature by evaporating water introduced into the inlet airflow upstream of the compressor. This approach frequently can be economically justified for MS and LM units in both base load and peaking applications. Output increases of about 9% can be experienced on heavy-duty (MS) units at a 90°F/32°C ambient temperature at a relative humidity of 20%. For the LM6000, the use of an 85% effective evaporative cooler will increase its output about 22% at a 90°F/32°C temperature and 20% relative humidity ambient condition.



Basis:

1) Sea Level
2) Natural Gas
3) Standard Inlet/Exhaust Loss
4) DLN / DLE NO_x Control

Figure 15. Gas turbine ambient output characteristics

Another alternative often considered for power augmentation and/or to simply minimize the impact of ambient temperature effects is the use of inlet air chillers. Depending upon power plant economics—in conjunction with ambient temperature and plant load profiles—chillers can afford substantial economic value. This alternative cools the incoming air, thus increasing the output relative to the gain available with an evaporative cooler. Frequently, the energy for cooling can be supplied by a mechanical or absorption refrigeration system that receives its steam from a low-pressure section in the gas turbine heat recovery steam generator (HRSG).

For the diluent-injected LM6000, the normal decrease in power output at ambient temperatures less than about 50°F (10°C) can be mitigated through inlet air heating to the maximum power output temperature. Low-level energy recovery from the HRSG can accomplish this task. The net effect is to drive the performance characteristics for the LM6000 flat over the ambient temperature range. (See *Figure 16.)*

The example gas turbine output enhancements are not limited to LM units only, and should be evaluated for all gas turbines to ensure that the maximum economic benefits are realized.

The greater the output change (lapse rate) with changing ambient temperature, the larger the economic potential associated with various power enhancement alternatives.

Fuel Flexibility and Gas Turbines

Over the years, significant strides have been made in the progression of combustion system technology. This technology progression includes advancements in fuel flexibility (the ability to use traditional fuels over a wider range and a steadily increasing ability to burn more non-traditional fuels), reductions in emissions levels, as well and the ability to meet lower emissions level over an extended range of turn down (larger output load range).

Almost everyone knows that gas turbines can burn natural gas, but many of them can also burn alternative fuels either with or without water injection, depending upon NO_x/CO level requirements. Typical examples of the aforementioned alternative fuels include:

- Gases such as synthetic gas (resulting from petcoke or coal gasification), steel mill gases, and petrochemical process gases
- Liquids such as light distillate, naphtha, and heavy fuel oils

Dry Low NO_x combustors—originally developed in the 1990s to

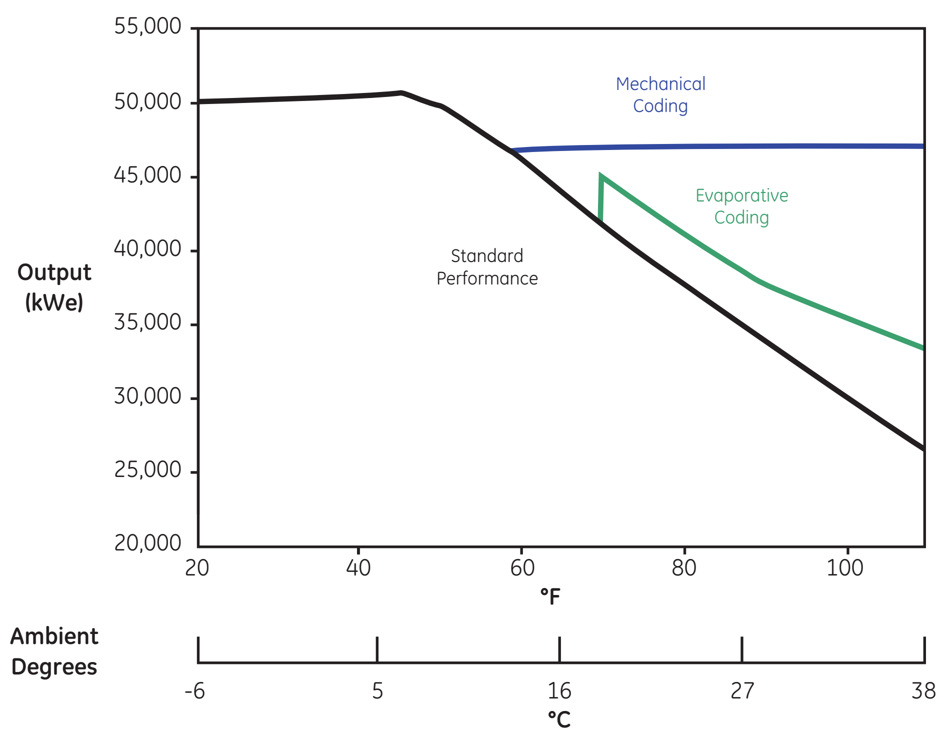


Figure 16. LM6000 PC inlet conditioning output enhancements

meet 25 ppm @ 15% 02 without water injection and mainly with methane—have recently improved their capabilities to burn gases with heavier hydrocarbons or more hydrogen to levels down to below 5 ppm @ 15% 02 on the 6B, 7E and 9E.

Combustion systems have also progressed significantly to accept gas composition variations and to meet emission levels on a large range of output.

Multi-Nozzle Quiet Combustors (MNQC) and single nozzle diffusion combustors provide the capability to some gas turbines to burn liquid fuels and gases with high CO_2, CO or H_2 content. In the case of these systems, water and steam are often injected into the combustor as a means of reducing/controlling emissions levels. However, it should also be noted that water or steam injection can also be used as a means of gas turbine power augmentation.

Table 3 illustrates a classification of the main alternative fuels and their "parent" primary energy sector: oil, gas, coal, residual and renewable, accessible to gas turbines

Gas Turbine Exhaust Heat Recovery

The economics of gas turbines in process applications usually depend on effective use of the exhaust energy, which generally represents 60% to 70% of the inlet fuel energy. The increase in overall system efficiency as the exhaust temperature is decreased through use of effective heat recovery is illustrated in *Figure 17*. The most common use of this energy is for steam generation in HRSGs, with unfired as well as fired designs. However, the gas turbine exhaust gases can also be used as a source of direct energy, for unfired and fired process fluid process fluid heaters, as well as preheated combustion air for power boilers.

Heat Recovery Steam Generators

The overall FCP in a gas-turbine HRSG system is a function of the amount of energy recovered from the turbine exhaust gas. The greater the amount of energy recovered, the lower the HRSG stack temperature, and the better the FCP. Thus, gas-turbine HRSG cycles should use the lowest practical feedwater temperature to the economizing section of the HRSG, within the constraints imposed due to gas-side corrosion considerations. The typical feedwater temperature is 230°F (110°C) if corrosion is not a problem. With an integral de-aerating section or de-aerating condenser, the inlet water temperatures can be much lower. For applications using sulfur-bearing fuels, a feedwater temperature of about 270–290°F (132–143°C) should be used to ensure metal temperatures remain above the condensation temperature of the sulfurous products of combustion. These feedwater temperatures are in contrast to steam turbine cycles, which provide increased cogenerated power as more regenerative feedwater heating (higher feedwater temperature to the boiler) is incorporated into the cycle.

HRSG units are available in unfired, supplementary-fired, and fully fired designs. The appropriate selection is established through economic evaluations of various potential configurations for the application.

Unfired HRSG. An unfired unit is the simplest HRSG configuration. For industrial type applications, steam conditions characteristically range from 150 psig (10.3 bar) saturated to approximately 1450 psig/950°F (100 bar/510°C). The steam temperature is typically set somewhere around 50°F (28°C) or more below the turbine exhaust gas temperature. Thus for applications leveraging F-class technology gas turbines, exhaust conditions will permit superheated steam temperatures of 1000°F–1050°F/538°C–566°C and for large F-class gas turbine units, reheat steam cycles will be permitted where project economics warrant this approach.

Generally speaking, unfired units can be economically designed to

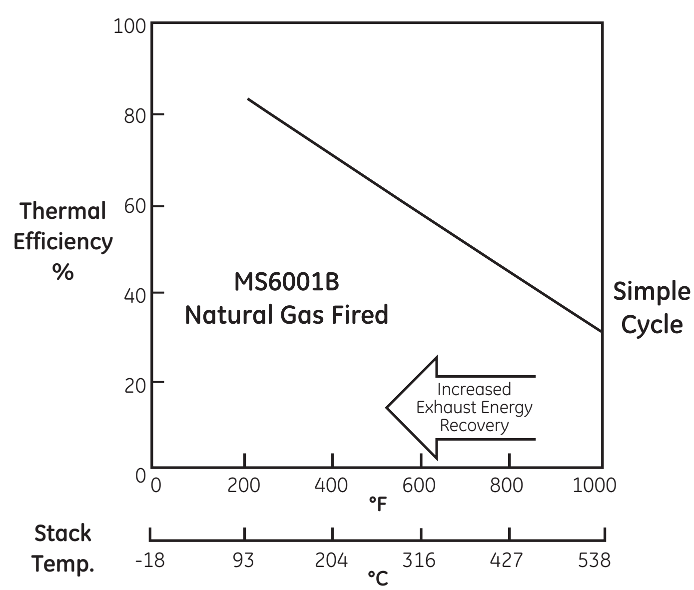


Figure 17. Thermal efficiency vs. stack temperature

Origin Process	Fuel Name	Liquid/Gas (L/G)	Characteristics Range	Ashless (AL)/ Ash Forming (AF)	Parent Primary Energy
Oil extraction	Crude oil	L	Light to heavy	AL	
Oil distillation	LPG: propane, butane	L/G	Variable C3/C4		
	Naphtha, kerosene,	L		AL	**Oil**
	Gas oils	L	Light to heavy	AL	
	Heavy oils	L	Atm. & vac. Resides	AF	
Catalytic cracking	Light cycle oil	L	Highly aromatic	AL	
NG extraction	Natural gas	G	- Rich to weak	AL	
			- Soft to sour		
NG extr. /treatment	Gas condensates	L	Light to heavy	AL to AF	**Natural Gas (NG)**
NG reforming	NGL	G	Variable CO_2 content	AL	
	H_2			AL	
Coal pyrolysis	Coke oven gas	G	Medium BTU	AF	
Iron production	Blast furnace gas (BFG)	G	Low BTU	AF	
	Finex				**Steel**
	Corex	G	Low BTU		
		G	Medium BTU		
Naphtha cracking	Olefins	G	Variable olefin %	AL	
Aromatics synthesis	H_2-rich gas	G	Variable H_2 %	AL	**Petro Chemical Industry**
Butadiene unit, etc.	C3/C4-rich gas	G	Variable C3/C4 ratio	AL	
Fermentation	Biogas: $CH4\text{-}N_2\text{-}CO_2$	G	Medium to low BTU	AL (purified))	
			Medium to low BTU	AL (purified)	**Residuals**
Gasification	Syngas	G			
Coal extraction	Coalbed gas	G	Low BTU gas	AL	
Coal liquefaction	Synfuels	L	Highly aromatic	AL	
	Methanol	L	Medium BTU liquid	AL	**Coal & Lignite**
Coal gasification	Syngas (CO/H_2)	G	Medium to Low BTU	AL (purified)	
	SNG	G			
Vegetable processing	Biofuels from farming	L	# 2 DO substitute	AL	**Renewables**

Table 3. Fuel Sources/Origins

recover approximately 95% of the energy in the turbine exhaust gas that is available for steam generation. Higher performance levels are possible; however, the increased cost of the heat transfer surface and possible larger gas side pressure drop must be evaluated vs. the additional energy recovered to establish whether the higher costs are warranted.

When unfired units are designed with higher steam conditions for a combined cycle, multiple-pressure units are usually applied to increase exhaust heat recovery and enhance system performance. The intermediate level may be that required for steam injection for NO_x control and/or a process level. In applications using natural gas, a third pressure level will further enhance overall system performance. Typical design practice is that unfired HRSGs are convective heat exchangers that respond to the exhaust conditions of the gas turbine. Thus, the performance of unfired HRSG units are driven by the gas turbine operating mode and cannot easily provide steam flow control.

Supplementary-Fired HRSG. Since gas turbines generally consume very little of the available oxygen within the gas turbine air flow, the oxygen content of the gas turbine exhaust generally permits supplementary fuel firing ahead of the HRSG to increase steam production rates relative to an unfired unit. A supplementary-fired unit is defined as a HRSG fired to an average temperature not exceeding about 1800°F/982°C.

Since the turbine exhaust gas is essentially preheated combustion air, the supplementary-fired HRSG fuel consumption is less than that required for a power boiler providing the same incremental increase in steam generation. Characteristically, the incremental steam production from supplementary firing above that of an unfired HRSG will be achieved at 100% efficiency, based on the lower heat value of the fuel fired. The amount of incremental fuel will be about 10% to 20% less than for a natural-gas-fired power boiler providing the same incremental increase in steam produced.

As previously stated, the unfired HRSG with higher steam conditions is often designed with multiple pressure levels to recover as much energy as possible from the gas turbine exhaust. This adds cost to the unfired HRSG, but the economics are often enhanced for the cycle. In the case of the supplementary-fired HRSG, if the HRSG is to be fired during most of its operating hours to the 1400°F to 1800°F/760°C to 982°C range, then a suitably low stack temperature can usually be achieved with a single pressure level unit. This is the result of increased economizer duty as compared to the unfired HRSG.

A supplementary-fired HRSG is basically a convective unit with a design quite similar to an unfired HRSG. However the firing capability provides the ability to control the HRSG steam production—within the capability of the burner system—independent of the normal gas turbine operating mode.

Fully-Fired HRSG. A few industrials have used the exhaust of the gas turbine as preheated combustion air for a fully-fired HRSG. A fully fired HRSG is defined as a unit having the same amount of oxygen in its stack gases as an ambient-air-fired power boiler. The HRSG is essentially a power boiler for which the gas turbine exhaust serves as the source of preheated air supply.

Steam production from fully-fired HRSGs (10% excess air) may range up to six or seven times the unfired HRSG steam production rate. The actual increase is a function of the oxygen remaining for combustion and the gas turbine exhaust temperature. Because of the use of preheated combustion air, fuel requirements for fully-fired units will usually range between 7.5% and 8% less than those of an ambient-air-fired boiler providing the same incremental steam generating capacity. With the more efficient gas turbines (higher firing temperatures resulting in lower oxygen content in the exhaust gases), the ability to ignite and maintain stable combustion in the HRSG should be confirmed with the HRSG manufacturer.

Even though fully-fired units can provide a significant amount of steam, few applications of this type can be found in industry. Evaluations show that the higher power-to-heat ratio available using unfired or supplementary-fired HRSGs is usually economically preferable over fully-fired HRSGs and lower amount of power generated.

HRSG Steam Production Rates

The amount of steam that can be generated using the exhaust gas from various GE gas turbine-generators frequently considered in industrial cogeneration systems is given in *Tables 3–5*.

In addition, the FCP is shown for the combination of the gas turbine and HRSG. This data is useful in performing gas-turbine cogeneration feasibility studies to obtain a rough estimation of the cycle's overall FCP. To do this, simply take the gas turbine kilowatts generated and the tabulated FCP from *Table 4* through *Table 6*. Then, add the non-condensing steam turbine kilowatts generated at the previously mentioned 4000–4500 Btu/kWh / 4219– 4747 kJ/kWh and the condensing steam turbine kilowatts generated (if there is a condenser in the cycle being considered) at 12,000–14,000 Btu/kWh /12,658–14,767 kJ/kWh. The weighted average of the FCP for the amount of power produced in the above three modes (gas turbine, non-condensing steam turbine, and condensing steam turbine) will be a close estimate of the overall FCP for the system being considered.

Cycle Configurations

The most simple gas turbine cogeneration cycle is one where the exhaust energy is used to generate steam at conditions suitable for the process steam header. (See *Figure 18*.)

The generation of steam at higher initial steam conditions than those required in process will allow the use of a steam turbine in addition to the gas turbine in the cogeneration cycle. *(See Figure 19.)* This configuration derives the benefits of both gas and steam turbine cogeneration and yields a higher power-to-heat ratio than the arrangement given in *Figure 18*.

A multi-pressure HRSG system is illustrated in *Figure 19*. This arrangement is common for unfired and moderately fired (~1200°F/654°C) HRSG systems. The multi-pressure HRSG provides increased recovery of the gas turbine exhaust energy, and thus contributes to the favorable FCP associated with these cycles. For example, an unfired multi-pressure HRSG used in conjunction with an MS7001EA combined cycle supplying steam to process at 150 psig (10.3 bars) will yield about 5150 Btu/kWh HHV (5430 kJ/kWh HHV) FCP—whereas a single-pressure, unfired HRSG used in a combined cycle, with the same gas turbine, would have a FCP of 6030 Btu/kWh HHV / 6360 kJ/kWh HHV.

The steam turbine design schematic in *Figure 19* provides considerable cycle flexibility in cogeneration applications. The condenser provides a heat sink for HRSG steam generating capability in excess of that extracted from the turbine for process use. Furthermore, the admission capability will permit the introduction of lower-pressure steam into the turbine for expansion to the condenser during periods when excess HRSG steam at the process pressure level is available.

Combined-Cycle Design Flexibility

One method of displaying the many options available using a gas turbine in a cogeneration application is illustrated in *Figure 20*. This diagram has been developed for the GE MS6001FA gas turbine-generator (75,000 kW ISO, natural-gas-fired). A summary of the performance used to develop the envelope given in Figure 20 is presented in *Table 6*.

Point A represents the MS6001FA gas turbine-generator exhausting into an unfired low-pressure HRSG. Point C is a combined-cycle configuration based on use of a two-pressure-level unfired HRSG. The steam turbine in the C cycle is a non-condensing unit expanding the HP HRSG steam to the 150 psig/10.3 bar process steam header.

Points B and D in *Figure 20* represent operation of the HRSG with supplementary firing to a 1600°F/871°C average exhaust-gas-temperature entering the heat transfer surface.

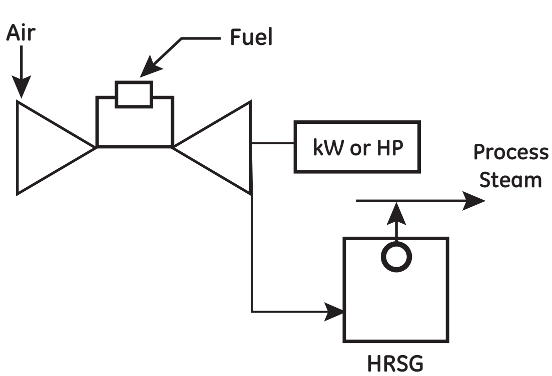


Figure 18. Gas turbine with LP HRSG

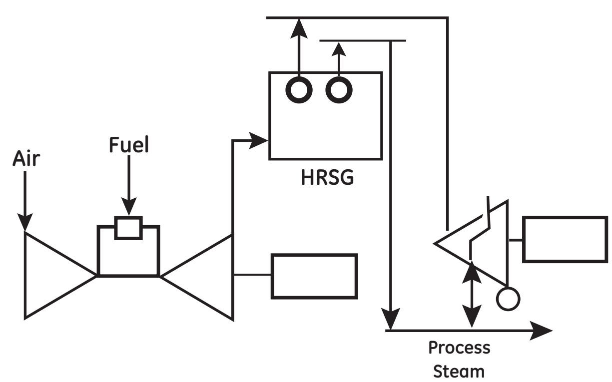


Figure 19. Typical industrial gas turbine cycle

Generator Drives - Natural Gas Fuel - Dry Performance - English Units

Gas Turbine Type		MS5001(PA)		MS6001(B)		MS6001(C)		MS6001(FA)	
Gas Turbine Model		**PG5371(PA)**		**PG6581(B)**		**PG6591(FA)**		**PG6111(FA)**	
ISO Base Rating (KW)		26,060		41,600		45,250		77,060	
Performance at 59 F, Sea Level, Natural Gas Fuel		50 / 60 Hz		50 / 60 Hz		50 / 60 Hz		50 / 60 Hz	
Output - kW									
Unfired, 1 PL		25,690		41,250		44,990		76,660	
Unfired, 2 PL		25,590		41,160		44,860		76,490	
Supp Fired		25,490		41,070		44,720		76,310	
Fully Fired		25,190		40,780		44,410		75,770	
Power Turbine Speed - rpm		5,100		5,163		5,250		5,231	
Fuel - MBtu/h (HHV)		348.9		445.3		469.1		822.2	
Exhaust Flow - lb/h		985,000		1,168,000		973,000		1,679,000	
Exhaust Temp - F									
Unfired, 1 PL		910		1018		1080		1112	
Unfired, 2 PL		911		1022		1082		1113	
Supp Fired		912		1023		1083		1114	
Fully Fired		916		1024		1088		1118	
HRSG Performance Fuel - MBtu/h (HHV)									
Supp Fired		225.9		228.4		277.7		280.5	
Fully Fired		845.9		910.6		1182.4		1188.2	
Steam Conditions (Psig / F)		**HRSG Steam 1000 lb/h**	**FCP GT Btu/kWh**	**HRSG Steam 1000 lb/h**	**FCP GT Btu/kWh**	**HRSG Steam 1000 lb/h**	**FCP GT Btu/kWh**	**HRSG Steam 1000 lb/h**	**FCP GT Btu/kWh**
Unfired									
150 / 365	1 PL	155	5810	224.7	3790	347	530	363	4630
400 / 650	1 PL	124	6560	185.4	4260	296	890	314.1	4770
600 / 750	1 PL	113	6940	172.3	4500	278	1140	295.3	4920
850 / 825	1 PL	104	7320	162.2	4720	264	1390	281.5	5050
850 / 825	2 PL	104	6060	162.5	4020	265	700	281.9	4650
150 / 365		25.7	-	22.44	-	23.0	-	24.1	-
1250 / 900	2 PL	-	-	152	4040	252	720	268.5	4810
150 / 365		-	-	30.79	-	31.6	-	24.05	-
1450 / 950	2 PL	-	-	145.8	4040	244	720	259.6	4820
150 / 365		-	-	35.03	-	36.2	-	28.9	-
Supp Fired									
150 / 365		349	4930	415	3410	347	6700	598.3	4360
400 / 650		310	4870	367.9	3400	307.5	6690	530.2	4350
600 / 750		299	4870	355.1	3390	296.9	6680	511.9	4340
850 / 825		292	4860	346.9	3380	290	6670	500.1	4330
1250 / 900		286	4850	340	3360	284.3	6660	490.3	4320
1450 / 950		282	4800	334.5	3350	279.7	6650	482.2	4320
Fully Fired									
400 / 650		763	3440	866.5	2410	1173	-1050	1194.4	3640
600 / 750		737	3380	836.7	2380	1133	-1100	1153.4	3610
850 / 825		720	3340	817.4	2350	1107	-1140	1126.9	3590
1250 / 900		705	3330	801.3	2310	1085	-1190	1104.9	3550
1450 / 950		694	3270	788.2	2290	1067	-1200	1087	3540

- Gas turbines and boilers fueled with natural gas and all fuel data based on higher heating value (HHV)
- Gas Turbines equipped with DLN combustors
- Fuel chargeable to gas turbine power assumes GT credit with PH auxiliaries and equivalent boiler fuel required to generate steam in an 84% efficient boiler (HHV)
- Standard inlet losses; exhaust losses 10 "H2O for unfired 1PL, 12" H2O for unfired 2 PL, 14" H_2O for supplementary fired, 20" H_2O for fully fired
- Assumes 0% exhaust bypass stack damper leakage, 0% blowdown, and 150 F condensate return for all cases

Table 4a. Steam generation and fuel chargeable to power with gas turbine and heat recovery boilers – Heavy Duty (MS series) – English units

Generator Drives - Natural Gas Fuel - Dry Performance - English Units

MS7001(EA)	MS9001(E)	MS7001(FA)	MS7001(FB)	MS9001(FA)	MS9001(FB)
PG7121(EA)	PG9171(E)	PG7241(FA)	PG7251(FB)	PG9351(FA)	PG9371(FB)
84,440	125,300	175,400	184,400	257,200	287,400
60 Hz	50 Hz	60 Hz	60 Hz	50 Hz	50 Hz
84,040	124,600	170,800	183,500	256,100	286,200
83,870	124,400	170,400	183,100	255,600	285,600
83,690	124,100	170,100	182,600	255,100	285,100
83,140	123,300	169,000	181,300	253,500	283,300
3,600	3,000	3,600	3,000	3,000	3,000
885.0	1271.0	1783.0	1886.1	2620.9	2865.5
2,378,000	3,313,000	3,612,000	3,597,000	5,235,000	5,229,000
1002	1010	1113	1165	1113	1185
1003	1011	1114	1166	1114	1186
1003	1012	1115	1167	1115	1187
1006	1015	1119	1172	1119	1191
480.9	661.2	602.6	539.7	874.0	751.1
1866.0	2573.2	2517.9	2423.2	3673.4	3390.6

HRSG Steam 1000 lb/h	FCP GT Btu/kWh	HRSG Steam 1000 lb/h	FCP GT Btu/kWh	HRSG Steam 1000 lb/h	FCP GT Btu/kWh	HRSG Steam 1000 lb/h	FCP GT Btu/kWh	HRSG Steam 1000 lb/h	FCP GT Btu/kWh	HRSG Steam 1000 lb/h	FCP GT Btu/kWh
442.1	3760	625	3750	788.7	4500	836.4	4410	1143.8	4490	1246.1	4410
362.8	4260	514	4210	677	4680	733.6	4470	981.8	4660	1101.2	4420
336.2	4500	476.9	4430	636.7	4820	693.6	4580	923.4	4800	1043	4520
315.3	4740	449	4640	606.8	4950	664.8	4680	880.1	4930	1001.6	4610
316	3960	448.9	4080	607.5	4570	665.6	4410	881.4	4550	1000.4	4400
50.7	-	55.0	-	51.5	-	39.15	-	74.7	-	50.04	-
294.1	3960	418.7	4080	578.9	4580	639.1	4420	839.7	4560	965.4	4400
69.17	-	80.45	-	70.15	-	53.19	-	101.7	-	67.96	-
281.3	3970	400.9	4080	559.9	4590	620.3	4430	812	4570	938.1	4410
78.41	-	93.33	-	80.36	-	61.53	-	116.5	-	79.07	-
845.3	3330	1178.6	3360	1287.7	4280	-	-	-	-	1867.8	-
749.1	3320	1044.5	3350	1141.1	4280	1137	4240	1655	4270	1655.2	4250
723.2	3310	1008.4	3340	1101.7	4270	1097.8	4230	1597.8	4270	1598.1	4240
706.5	3300	985.1	3320	1076.2	4260	1072.4	4220	1560.9	4260	1561.1	4230
692.7	3280	965.8	3310	1055.2	4240	1051.4	4210	1530.3	4240	1530.6	4220
681.4	3270	950.1	3300	1038	4240	1034.3	4200	1505.4	4240	1505.7	4220
1761.8	2340	2442.6	2430	2540.5	3620	-	-	-	-	3585	-
1701.3	2310	2358.7	2400	2455.4	3570	2428.7	3600	3575	3600	3462	3680
1662.2	2270	2304.5	2370	2398.9	3550	2390.6	3430	3493	3570	3383	3660
1629.9	2220	2259.7	2320	2352.2	3520	2326.7	3550	3424	3550	3317	3630
1603.4	2200	1464.7	12020	2314	3500	2288.9	3530	3368	3530	3263	3620

- Unfired boiler design based on a 15 F pinch point / 15 F subcool approach temperature, with criteria to limit the stack temperature to a minimum of 220 F for all cases
- Supplementary firing based on average gas temperature of 1600 F
- Methane fuel with Lower heating value (LHV) - 21515 Btu/lb, HHV = LHV x 1.11
- Power Factor = 0.8 lagging

Generator Drive - Natural Gas Fuel - Dry Performance - SI Units

Gas Turbine Type	MS5001(PA)	MS6001(B)	MS6001(C)	MS6001(FA)
Gas Turbine Model	**PG5371(PA)**	**PG6581(B)**	**PG6591(FA)**	**PG6111(FA)**
ISO Base Rating (kW)	26060	41600	45250	77060
Performance at 15 C, Sea Level, Natural Gas Fuel	50 / 60 Hz	50 / 60 Hz	50 / 60 Hz	50 / 60 Hz
Output - kW				
Unfired, 1 PL	25690	41250	44990	76660
Unfired, 2 PL	25590	41160	44860	76490
Supp Fired	25490	41070	44720	76310
Fully Fired	25190	40780	44410	75770
Power Turbine Speed - rpm	5100	5163	5250	5231
Fuel - MKJ/h (HHV)	368.1	469.8	494.9	867.4
Exhaust Flow - Tons/h	447	530	441	761
Exhaust Temp - C				
Unfired, 1 PL	488	548	582	600
Unfired, 2 PL	488	550	583	601
Supp Fired	489	551	584	601
Fully Fired	491	551	587	603
HRSG Performance Fuel - MKJ/h (HHV)				
Supp Fired	238.3	241.0	293.0	296.0
Fully Fired	892.4	960.6	1247.4	1253.5

Steam Conditions (bara/C)		HRSG Steam Tons/h	FCP GT KJ/kWh	HRSG Steam Tons/h	FCP GT KJ/kWh	HRSG Steam Tons/h	FCP GT KJ/kWh	HRSG Steam Tons/h	FCP GT KJ/kWh
Unfired									
11.4 / 185	1 PL	70.3	6130	101.9	4000	157.4	560	164.6	4880
28.6 / 343	1 PL	56.2	6920	84.1	4490	134.2	940	142.4	5030
42.4 / 399	1 PL	51.2	7320	78.1	4750	126.1	1200	133.9	5190
59.7 / 441	1 PL	47.2	7720	73.6	4980	119.7	1470	127.7	5330
59.7 / 441	2 PL	47.2	6390	73.7	4240	120.2	740	127.8	4910
11.4 / 185		11.7	-	10.2	-	10.4	-	10.9	-
87.2 / 482	2 PL	-	-	68.9	4260	114.3	740	121.8	4910
11.4 / 185		-	-	14.0	-	14.3	-	10.9	-
101 / 510	2 PL	-	-	66.1	4260	110.7	760	117.7	5070
11.4 / 185		-	-	14.0	-	16.4	-	13.1	-
Supp Fired									
11.4 / 185		158.3	5200	188.2	3600	157.4	7070	271.3	4600
28.6 / 343		140.6	5140	166.8	3590	139.5	7060	240.5	4590
42.4 / 399		135.6	5140	161.0	3580	134.6	7050	232.2	4580
59.7 / 441		132.4	5130	157.3	3570	131.5	7040	226.8	4570
87.2 / 482		129.7	5120	154.2	3540	128.9	7030	222.4	4560
101 / 510		127.9	5060	151.7	3530	126.8	7020	218.7	4560
Fully Fired									
28.6 / 343		346.0	3630	393.0	2540	532.0	-1110	541.7	3840
42.4 / 399		334.2	3570	379.5	2510	513.8	-1160	523.1	3810
59.7 / 441		326.5	3520	370.7	2480	502.0	-1200	511.1	3790
87.2 / 482		319.7	3510	363.4	2440	492.1	-1260	501.1	3750
101 / 510		314.7	3450	357.5	2420	483.9	-1270	493.0	3730

- Gas turbines and boilers fueled with natural gas and all fuel data based on higher heating value (HHV)
- Gas turbines equipped with DLN combustors
- Fuel chargeable to gas turbine power assumes GT credit with PH auxiliaries and equivalent boiler fuel required to generate steam in an 84% efficient boiler (HHV)
- Standard inlet losses; exhaust losses 254 mm H_2O for unfired 1PL, 305 mm H_2O for unfired 2 PL, 356 mm H_2O for supplementary fired, 508 mm H_2O for fully fired

Table 4b. Steam generation and fuel chargeable to power with gas turbine and heat recovery boilers – Heavy Duty (MS series) – SI units

Generator Drive - Natural Gas Fuel - Dry Performance - SI Units

MS7001(EA)	MS9001(E)	MS7001(FA)	MS7001(FB)	MS9001(FA)	MS9001(FB)
PG7121(EA)	PG9171(E)	PG7241(FA)	PG7251(FB)	PG9351(FA)	PG9371(FB)
84440	125300	175400	184400	257200	287400
60 Hz	50 Hz	60 Hz	60 Hz	50 Hz	50 Hz
84040	124600	170800	183500	256100	286200
83870	124400	170400	183100	255600	285600
83690	124100	170100	182600	255100	285100
83140	123300	169000	181300	253500	283300
3600	3000	3600	3000	3000	3000
933.7	1340.9	1881.1	1989.8	2765.1	3023.1
1,078	1,502	1,638	1,631	2,374	2,371
539	543	601	629	601	641
539	544	601	630	601	641
539	544	602	631	602	642
541	546	604	633	604	644
507.4	697.5	635.7	569.4	922.1	792.5
1968.6	2714.8	2656.4	2556.5	3875.5	3577.1

HRSG Steam Tons/h	FCP GT KJ/kWh	HRSG Steam Tons/h	FCP GT KJ/kWh	HRSG Steam Tons/h	FCP GT KJ/kWh	HRSG Steam Tons/h	FCP GT KJ/kWh	HRSG Steam Tons/h	FCP GT KJ/kWh	HRSG Steam Tons/h	FCP GT KJ/kWh
200.5	3970	283.4	3960	357.7	4750	379.3	4650	518.7	4740	565.1	4650
164.5	4490	233.1	4440	307.0	4940	332.7	4720	445.3	4920	499.4	4660
152.5	4750	216.3	4670	288.8	5090	314.6	4830	418.8	5060	473.0	4770
143.0	5000	203.6	4900	275.2	5220	301.5	4940	399.1	5200	454.2	4860
143.3	4180	203.6	4300	275.5	4820	301.9	4650	399.7	4800	453.7	4640
23.0	-	24.9	-	23.4	-	17.8	-	33.9	-	22.7	-
133.4	4180	189.9	4300	262.5	4820	289.8	4650	380.8	4800	437.8	4640
31.4	-	36.5	-	31.8	-	24.1	-	46.1	-	30.8	-
127.6	4180	181.8	4300	253.9	4830	281.3	4660	368.3	4810	425.4	4640
35.6	-	42.3	-	36.4	-	27.9	-	52.8	-	35.9	-
383.4	3510	534.5	3540	584.0	4520	-	-	-	-	-	-
339.7	3500	473.7	3530	517.5	4520	515.6	4470	750.6	4500	750.7	4480
328.0	3490	457.3	3520	499.6	4500	497.9	4460	724.6	4500	724.8	4470
320.4	3480	446.8	3500	488.1	4490	486.3	4450	707.9	4490	708.0	4460
314.1	3460	438.0	3490	478.5	4470	476.8	4440	694.0	4470	694.1	4450
309.0	3450	430.9	3480	470.7	4470	469.1	4430	682.7	4470	682.9	4450
799.0	2470	1107.8	2560	1152.2	3820	-	-	-	-	-	-
771.6	2440	1069.7	2530	1113.6	3770	1101.5	3800	1621.3	3800	1570.1	3880
753.8	2390	1045.1	2500	1087.9	3750	1084.2	3620	1584.1	3770	1534.2	3860
739.2	2340	1024.8	2450	1066.8	3710	1055.2	3750	1552.8	3750	1504.3	3830
727.2	2320	664.3	12680	1049.4	3690	1038.0	3720	1527.4	3720	1479.8	3820

- Assumes 0% exhaust bypass stack damper leakage, 0% blowdown, and 65.6 C condensate return for all cases
- Unfired boiler design based on a 8.3 C pinch point / 8.3 C subcool approach temperature, with criteria to limit the stack temperature to a minimum of 104.4 C for all cases
- Supplementary firing based on average gas temperature of 871 C
- Methane fuel with Lower heating value (LHV) - 50031 kJ/kg, HHV = LHV x 1.11
- Power Factor = 0.8 lagging

Generator Drives - Natural Gas Fuel - Dry Performance - English Units

Gas Turbine Type		LM2500 PJ		LM2500+ PR		LM6000 PF		LM6000 PF SPRINT		LMS100 PA	
Gas Turbine Model		PGLM2500 (PJ)		PGLM2500 (+)		PGLM6000 (PF)		PGLM6000 (PF SPRINT)		PGLMS100 (PA)	
ISO Base Rating (kW)		22,719		33,165		43,068		48,092		103,045	
Performance at 59 F, Sea Level, Natural Gas Fuel		60 Hz		60 Hz		60 Hz		60 Hz		60 Hz	
Output - kW											
Unfired, 1 PL		22,032		32,106		41,763		46,915		102,680	
Unfired, 2 PL		21,974		32,019		41,655		46,814		102,507	
Supp Fired		21,915		31,927		41,547		46,714		102,335	
Fully Fired		21,736		31,658		41,225		46,410		101,823	
Power Turbine Speed - rpm		3,600		3,600		3,600		3,600		3,600	
Fuel - MBtu/h (HHV)		240.0		323.5		391.6		439.8		929.5	
Exhaust Flow - lb/h		536,500		713,300		975,700		1,034,700		1,685,400	
Exhaust Temp - F											
Unfired	1 PL	996.4		988.0		861.0		855.4		777.8	
Unfired	2 PL	997.7		989.5		862.3		856.6		779.1	
Supp Fired		999.0		990.9		863.7		857.9		780.4	
Fully Fired		1003.0		995.2		867.5		861.5		784.2	
HRSG Performance Fuel - MBtu/h (HHV)											
Supp Fired		110.0		148.2		241.7		261.1		478.1	
Fully Fired		408.0		537.3		789.1		798.2		1049.9	
Steam Conditions (Psig / F)		**HRSG Steam 1000 lb/h**	**FCP GT Btu/kWh**	**HRSG Steam 1000 lb/h**	**FCP GT Btu/kWh**	**HRSG Steam 1000 lb/h**	**FCP GT Btu/kWh**	**HRSG Steam 1000 lb/h**	**FCP GT Btu/kWh**	**HRSG Steam 1000 lb/h**	**FCP GT Btu/kWh**
Unfired											
150 / 365	1 PL	98.69	5130	129.4	4890	137.5	5140	145.5	5380	200.5	6540
400 / 650	1 PL	80.76	5560	105.6	5290	107.1	5650	113	5870	148.5	6950
600 / 750	1 PL	74.72	5780	97.59	5490	96.01	5910	101.1	6120		
850 / 825	1 PL	69.96	5990	91.22	5690						
1250 / 900	1 PL	64.93	5670	84.45	5390						
850 / 825	2 PL	70.43	5180	91.89	4940	87.82	5170				
150 / 365		11.68	-	15.9	-	29.3	-				
1250 / 900	2 PL	65.45	5290	85.2	5030						
150 / 365		15.96	-	21.7	-						
1450 / 950	2 PL	62..55	9860	81.31	5110						
150 / 365		18.09	-	24.61	-						
Supp Fired											
150 / 365		191.3	4740	254.5	4520	348.4	4450	373.5	4710	623.8	5900
400 / 650		169.7	4720	225.7	4500	309	4440	331.2	4700	553.2	5890
600 / 750		163.9	4700	218	4490	298.4	4420	320	4680	534.3	5880
850 / 825		160.2	4680	213.1	4470	291.7	4410	312.7	4670	522.3	5870
1250 / 900		157.2	4660	209.1	4450	286.3	4380	306.9	4640	512.5	5850
1450 / 950		154.7	4650	205.8	4430	281.7	4370	302	4630	504.4	5840
1800 / 1000		152.9	4780	203.4	4550	278.4	4500	298.4	4760	498.4	5930
Fully Fired											
400 / 650		387.7	3900	510.5	3770	709.4	3650	724.2	4010	972.2	5550
600 / 750		374.6	3860	493.2	3730	685.3	3610	699.7	3970	939.2	5530
850 / 825		366.2	3820	482.1	3700	670	3570	684	3930	918.1	5510
1250 / 900		359.4	3750	473.1	3640	657.5	3510	671.2	3880	901	5480
1450 / 950		353.7	3720	465.7	3610	647.1	3480	660.6	3850	886.7	5460

- Gas turbines and boilers fueled with natural gas and all fuel data based on higher heating value (HHV)
- Gas Turbines equipped with DLN combustors
- Fuel chargeable to gas turbine power assumes GT credit with PH auxiliaries and equivalent boiler fuel required to generate steam in an 84% efficient boiler (HHV)
- Standard inlet losses; exhaust losses 10 "H_2O for unfired 1PL, 12" H2O for unfired 2 PL, 14" H_2O for supplementary fired, 20" H_2O for fully fired

Table 5a. Steam generation and fuel chargeable to power with gas turbine and heat recovery boilers – LM series – English units

Generator Drives - Natural Gas Fuel - Dry Performance - English Units

LMS100 PB		LM2500 PJ		LM2500+ PR		LM6000 PF		LM6000 PF SPRINT		LMS100 PA		LMS100 PB	
PGLMS100 (PB)		PGLM2500 (PJ)		PGLM2500 (+)		PGLM6000 (PF)		PGLM6000 (PF SPRINT)		PGLMS100 (PA)		PGLMS100 (PB)	
99,012		21,818		32,881		42,732		48,040		102,995		99,044	
60 Hz		50 Hz		50 Hz		50 Hz		50 Hz		50 Hz		50 Hz	
95,752		21,145		31,831		41,685		46,932		102,600		95,868	
95,581		21,086		31,744		41,586		46,831		102,439		95,708	
95,410		21,028		31,653		41,489		46,729		102,276		95,548	
94,903		20,852		31,386		41,195		46,427		101,792		95,069	
3,600		3,000		3,600		3,627		3,627		3,000		3,000	
840.6		238.1		323.5		392.6		441.9		929.6		840.5	
1,615,932		538,700		713,300		986,700		1,045,100		1,685,400		1,615,824	
792.1		1004.7		988.0		854.3		850.8		779.8		792.8	
793.5		1006.0		989.5		855.7		852.1		781.0		794.0	
794.8		1007.4		990.9		857.0		853.3		782.2		795.3	
798.8		1011.3		995.2		861.3		857.0		785.8		799.0	
441.3		108.9		148.2		246.5		265.2		477.1		441.0	
1105.6		412.6		537.3		801.5		808.6		1050.0		1105.7	
HRSG Steam 1000 lb/h	FCP GT Btu/kWh	HRSG Steam 1000 lb/h	FCP GT Btu/kWh	HRSG Steam 1000 lb/h	FCP GT Btu/kWh	HRSG Steam 1000 lb/h	FCP GT Btu/kWh	HRSG Steam 1000 lb/h	FCP GT Btu/kWh	HRSG Steam 1000 lb/h	FCP GT Btu/kWh	HRSG Steam 1000 lb/h	FCP GT Btu/kWh
195.6	6150	99.33	5210	128.6	4960	137	5190	145.4	5430	201.6	6530	196	6130
146.4	6550	82.38	5600	105.6	5340	106.3	5710	112.7	5920	149.5	6940	146.7	6540
127.7	6770	76.33	5820	97.59	5540	95.11	5980	100.7	6180			128	6750
		71.57	6030	91.22	5740								
		66.58	5720	84.45	5440								
		72.04	5210	91.89	4980								
		11.4	-	15.89	-								
		67.09	5320	85.2	5080								
		15.64	-	21.7	-								
		64.19	5410	81.34	5150								
		17.73	-	24.61	-								
585.6	5530	191.9	4760	254.5	4560	352.4	4470	377.2	4740	623.7	4890	585.5	4510
519.3	5520	170.2	4740	225.7	4540	312.5	4460	334.5	4730	553.1	5890	519.2	5510
501.6	5510	164.4	4720	218	4530	301.9	4440	323.1	4710	534.3	5870	501.6	5500
490.3	5500	160.7	4700	213.1	4510	295	4430	315.9	4690	522.2	5860	490.2	5490
481.1	5480	157.7	4680	209.1	4490	289.5	4400	310	4670	512.5	5840	481.1	5470
473.5	5470	155.2	4660	205.8	4470	284.9	4390	305	4660	504.3	5840	473.4	5460
467.9	5560	153.3	4810	203.4	4590	281.6	4520	301	4780	498.4	5930	467.8	5550
1005.9	5100	392.4	3870	510.5	3800	718.7	3650	732.2	4020	973	5040	1006.1	4530
971.7	5080	379.1	3830	493.2	3770	694.3	3610	707.3	3990	939.9	5530	971.9	5060
949.9	5050	370.6	3780	482.1	3730	678.7	3570	691.5	3950	918.8	5500	950.1	5040
932.2	5010	363.7	3720	473.1	3670	666.1	3510	678.6	3900	901.7	5470	932.4	5000
917.4	5000	357.9	3690	465.7	3640	655.6	3480	667.9	3870	887.4	5450	917.6	4980

- Assumes 0% exhaust bypass stack damper leakage, 0% blowdown, and 150 F condensate return for all cases
- Unfired boiler design based on a 15 F pinch point / 15 F subcool approach temperature, with criteria to limit the stack temperature to a minimum of 220 F for all cases
- Supplementary firing based on average gas temperature of 1600 F
- Lower heating value (LHV) - 21515 Btu/lb, HHV = LHV x 1.11

Generator Drive - Natural Gas Fuel - Dry Performance - SI Units

Gas Turbine Type		LM2500 PJ		LM2500+ PR		LM6000 PF		LM6000 PF SPRINT		LMS100 PA	
Gas Turbine Model		PGLM2500 (PJ)		PGLM2500 (+)		PGLM6000 (PF)		PGLM6000 (PF SPRINT)		PGLMS100 (PA)	
ISO Base Rating (kW)		26,060		41,610		45,250		45,250		77,650	
Performance at 15 C, Sea Level, Natural Gas Fuel		60 Hz		60 Hz		60 Hz		60 Hz		60 Hz	
Output - kW											
Unfired, 1 PL		22,032		32,106		41,763		46,915		102,680	
Unfired, 2 PL		21,974		32,019		41,655		46,814		102,507	
Supp Fired		21,915		31,927		41,547		46,714		102,335	
Fully Fired		21,736		31,658		41,225		46,410		101,823	
Power Turbine Speed - rpm		3,600		3,600		3,600		3,600		3,600	
Fuel - MKJ/h (HHV)		396.0		533.8		646.2		725.6		1533.6	
Exhaust Flow - Tons/h		243		323		442		469		764	
Exhaust Temp - C											
Unfired, 1 PL		535.8		531.1		460.6		457.4		414.3	
Unfired, 2 PL		536.5		531.9		461.3		458.1		415.1	
Supp Fired		537.2		532.7		462.1		458.8		415.8	
Fully Fired		539.4		535.1		464.2		460.8		417.9	
HRSG Performance Fuel - MKJ/h (HHV)											
Supp Fired		116.0		156.3		255.0		275.5		504.3	
Fully Fired		430.5		566.9		832.5		842.1		1107.6	
Steam Conditions (bara / C)		**HRSG Steam Tons/h**	**FCP GT KJ/kWh**	**HRSG Steam Tons/h**	**FCP GT KJ/kWh**	**HRSG Steam Tons/h**	**FCP GT KJ/kWh**	**HRSG Steam Tons/h**	**FCP GT KJ/kWh**	**HRSG Steam Tons/h**	**FCP GT KJ/kWh**
Unfired											
11.4 / 185	1 PL	44.76	5410	58.68	5160	62.36	5420	65.99	5680	90.93	6900
28.6 / 343	1 PL	36.63	5870	47.89	5580	48.57	5960	51.25	6190	67.35	7330
42.4 / 399	1 PL	33.89	6100	44.26	5790	43.54	6240	45.85	6460		
59.7 / 441	1 PL	31.73	6320	41.37	6000						
87.2 / 482		29.45	5980	38.30	5690						
59.7 / 441	2 PL	31.94	5460	41.67	5210						
11.4 / 185		5.30	-	7.21	-						
87.2 / 482	2 PL	29.68	5580	38.64	5310						
11.4 / 185		7.24	-	9.85	-						
101 / 510	2 PL										
11.4 / 185											
Supp Fired											
11.4 / 185		86.76	5000	115.42	4770	158.00	4690	169.39	4970	282.90	6220
28.6 / 343		76.96	4980	102.36	4750	140.14	4680	150.20	4960	250.88	6210
42.4 / 399		74.33	4960	98.87	4740	135.33	4660	145.12	4940	242.31	6200
59.7 / 441		72.65	4940	96.64	4720	132.29	4650	141.81	4930	236.87	6190
87.2 / 482		71.29	4920	94.83	4690	129.84	4620	139.18	4900	232.43	6170
101 / 510		70.16	4910	93.33	4670	127.76	4610	136.96	4880	228.75	6160
125 / 538		69.34	5040	92.24	4800	126.26	4750	135.33	5020	226.03	6260
Fully Fired											
28.6 / 343		175.83	4110	231.52	3980	321.72	3850	328.44	4230	440.91	5860
42.4 / 399		169.89	4070	223.67	3940	310.79	3810	317.32	4190	425.94	5830
59.7 / 441		166.08	4030	218.64	3900	303.85	3770	310.20	4150	416.37	5810
87.2 / 482		162.99	3960	214.56	3840	298.19	3700	304.40	4090	408.62	5780
101 / 510		160.41	3920	211.20	3810	293.47	3670	299.59	4060	402.13	5760

- Gas turbines and boilers fueled with natural gas and all fuel data based on higher heating value (HHV)
- Gas turbines equipped with DLN combustors
- Fuel chargeable to gas turbine power assumes GT credit with PH auxiliaries and equivalent boiler fuel required to generate steam in an 84% efficient boiler (HHV)
- Standard inlet losses; exhaust losses 254 mm H_2O for unfired 1PL, 305 mm H_2O for unfired 2 PL, 356 mm H_2O for supplementary fired, 508 mm H_2O for fully fired

Table 5b. Steam generation and fuel chargeable to power with gas turbine and heat recovery boilers – LM series – SI units

Generator Drive - Natural Gas Fuel - Dry Performance - SI Units

LMS100 PB	LM2500 PJ	LM2500+ PR	LM6000 PF	LM6000 PF SPRINT	LMS100 PA	LMS100 PB
PGLMS100 (PB)	PGLM2500 (PJ)	PGLM2500 (+)	PGLM6000 (PF)	PGLM6000 (PF SPRINT)	PGLMS100 (PA)	PGLMS100 (PB)
255,600	21,818	32,881	42,732	48,040	102,995	255,600
60 Hz	50 Hz	50 Hz	50 Hz	50 Hz	50 Hz	50 Hz
95,752	21,145	31,831	41,685	46,932	102,600	95,868
95,581	21,086	31,744	41,586	46,831	102,439	95,708
95,410	21,028	31,653	41,489	46,729	102,276	95,548
94,903	20,852	31,386	41,195	46,427	101,792	95,069
3,600	3,000	3,600	3,627	3,627	3,000	3,000
1387.0	392.9	533.8	647.7	729.1	1533.8	1386.9
733	244	323	447	474	764	733
422.3	540.4	531.1	456.8	454.9	415.4	422.7
423.1	541.1	531.9	457.6	455.6	416.1	423.3
423.8	541.9	532.7	458.3	456.3	416.8	424.1
426.0	544.1	535.1	460.7	458.3	418.8	426.1
465.5	114.9	156.3	260.1	279.8	503.3	465.2
1166.4	435.3	566.9	845.6	853.1	1107.8	1166.6

HRSG Steam Tons/h	FCP GT KJ/kWh	HRSG Steam Tons/h	FCP GT KJ/kWh	HRSG Steam Tons/h	FCP GT KJ/kWh	HRSG Steam Tons/h	FCP GT KJ/kWh	HRSG Steam Tons/h	FCP GT KJ/kWh	HRSG Steam Tons/h	FCP GT KJ/kWh	HRSG Steam Tons/h	FCP GT KJ/kWh
88.71	6490	45.05	5500	58.32	5230	62.13	5480	65.94	5730	91.43	6890	88.89	6470
66.39	6910	37.36	5910	47.89	5630	48.21	6020	51.11	6250	67.80	7320	66.53	6900
57.91	7140	34.62	6140	44.26	5840	43.13	6310	45.67	6520			58.05	7120
		32.46	6360	41.37	6060								
		30.20	6030	38.30	5740								
		32.67	5500	41.67	5250								
		5.19	-	7.21	-								
		30.43	5610	38.64	5360								
		7.09	-	9.85	-								
265.58	5830	87.03	5020	115.42	4810	159.82	4720	171.07	5000	282.86	5160	265.53	4760
235.51	5820	77.19	5000	102.36	4790	141.72	4710	151.70	4990	250.84	6210	235.46	5810
227.48	5810	74.56	4980	98.87	4780	136.92	4680	146.53	4970	242.31	6190	227.48	5800
222.36	5800	72.88	4960	96.64	4760	133.79	4670	143.27	4950	236.83	6180	222.31	5790
218.19	5780	71.52	4940	94.83	4740	131.29	4640	140.59	4930	232.43	6160	218.19	5770
214.74	5770	70.39	4920	93.33	4720	129.21	4630	138.32	4920	228.71	6160	214.69	5760
212.20	5870	69.52	5070	92.24	4840	127.71	4770	136.69	5040	226.03	6260	212.15	5860
456.19	5380	177.96	4080	231.52	4010	325.94	3850	332.06	4240	441.27	5320	456.28	4780
440.68	5360	171.93	4040	223.67	3980	314.88	3810	320.77	4210	426.26	5830	440.77	5340
430.79	5330	168.07	3990	218.64	3940	307.80	3770	313.61	4170	416.69	5800	430.88	5320
422.77	5290	164.94	3920	214.56	3870	302.09	3700	307.76	4110	408.93	5770	422.86	5280
416.05	5280	162.31	3890	211.20	3840	297.32	3670	302.90	4080	402.45	5750	416.15	5250

- Assumes 0% exhaust bypass stack damper leakage, 0% blowdown, and 65.6 C condensate return for all cases
- Unfired boiler design based on a 8.3 C pinch point / 8.3 C subcool approach temperature, with criteria to limit the stack temperature to a minimum of 104.4 C for all cases
- Supplementary firing based on average gas temperature of 871 C
- Lower heating value (LHV) - 50031 kJ/kg, HHV = LHV x 1.11

Performance of MS6001FA Gas Turbine Cycles

Cycle		A	B	C	D	E	F
Net Output	MW	75.5	74.9	88.6	100.5	110.8	138.9
NHP	MBtu/hr	466	768	347	648	0	0
	GJ/hr	491	810	366	684	0	0
FCP	Btu/kWh, HHV	4716	4462	5364	4517	7418	7944
	kJ/kWh, HHV	4975	4707	5659	4766	7826	8381

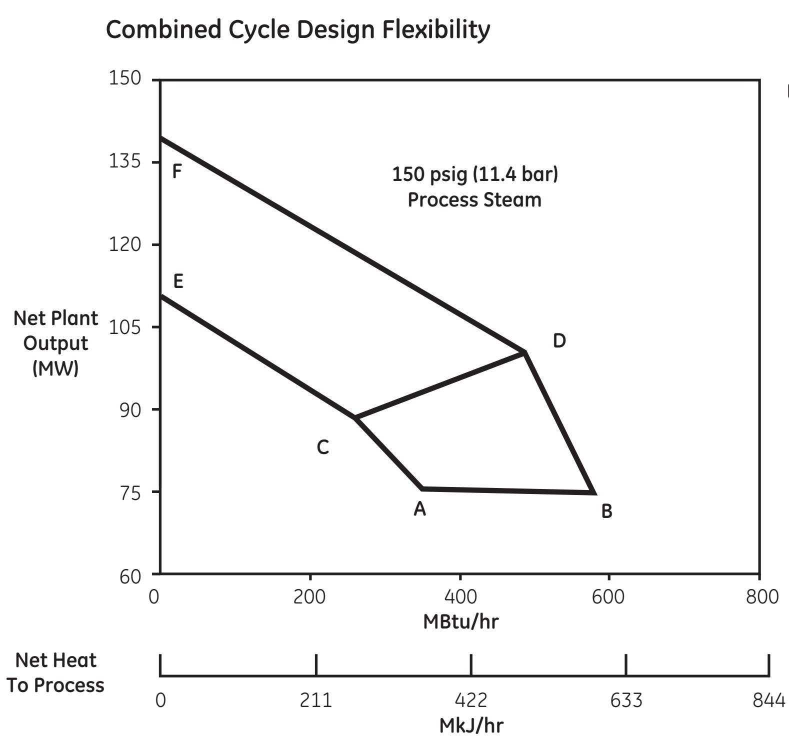
Basis:
1. Cycle definition as given in Figure 20.
2. Net output is the total power credited to the cogen cycle.
3. Net fuel includes credit for Net Heat to Process (NHP) at an 84% process boiler efficiency.

Table 6. Performance of MS6001FA gas turbine cycles

The temperature used for the HRSG firing in *Figure 20* has been arbitrarily limited to 1600°F/871°C, even though higher firing temperatures (and thus higher steam production rates) are possible in the exhaust of this unit.

The envelope defined by A, B, C, D in *Figure 20* represents the most thermally optimized use of a gas turbine in a cogeneration application (i.e., it provides the lowest FCP). Operation along the line CE, DF or any intermediate point to the left of line CD represents use of condensing steam turbine power generation with the E and F points applicable for combined-cycle operation without any heat supplied to process. Thus, the cycles along line EF are combined cycles providing power alone.

Performance envelopes for many of the gas turbines included in *Tables 3–5* are presented in *Figures 21–23*. These data are on the same basis as *Figure 20*, except for point C. Point C for all units, except the various MS7001 models, is based on



Basis:

1) Gas turbine operating at its 59°F (15°C) capability, sea level site, natural gas fuel, DLNto for 25 ppmvd (1.45g/gc) No_X emission level
2) Cycle A – Unfired HRSG (LP process steam)
   Cycle B – Supplementary fired (1600°F/871°C) HRSG, LP process steam
   Cycle C – Combined cycle, unfired, two-pressure level HRSG, HP at 1250 psig, 900 F (87.2 bars, 482 C), LP at 150 psig saturated (11.4 bars saturated), noncondensing steam turbine-generator
   Cycle D – Combined cycle, supplementary fired HRSG, steam at 1250 psig, 900°F (87.2 bars, 482°C), noncondensing steam turbine generator
   Cycle E – Same as Cycle C, but with admission condensing steam turbine-generator
   Cycle F – Same as Cycle D, but with condensing steam turbine-generator
3) Process returns and makeup enter the integral 44 psia (3 bars deaerating heater at a mixed temperature of 150°F (66°C)

Figure 20. Performance envelope for MS6001FA gas turbine cogeneration system

850 psig, 825°F/59 bars, 440°C initial steam conditions to the non-condensing steam turbine. Furthermore, the only condensing power illustrated is based on unfired, two-pressure-level HRSG designs.

The per unit cost of power generation for cycles A through F as defined/identified in *Figure 20* are illustrated in *Figure 24*. The per unit costs are based on a 16% fixed charge rate for invested capital and other operating costs such as fuel, operating labor and maintenance. The plant costs, not given, are based on separate stand-alone facilities, i.e., no "investment credit" applied to any of the cogeneration cases (Cases A through D).

Per unit costs given in *Figure 24* define two distinct performance levels. The "thermally optimized" cogen cases—Cases A through D—result in per unit costs that are about 20-30% lower than the unfired power generation case, Case E. Further, if the thermally optimized cogen cases were considered as additions to an existing facility, part of a major plant expansion, or used to displace new boilers which are intended to replace aging equipment, the comparisons would be more dramatic. That is, the incremental capital costs for the cogen systems might be 25% to 40% less than those used for *Figure 24* due to significant savings represented by the use of existing infrastructure. Even so, site-specific fuel and power costs, or power sales opportunities may dictate cycles with considerable condensing power as the appropriate economic choice.

An example illustrating the performance and economics of various MS6001FA gas turbine cogeneration cycles is given in *Table 7* and *Figure 25*. Cycles range from the "thermal match" examples (Cases 1 and 2) to configurations including considerable steam turbine condensing power (Cases 3, 4 and 5).

The evaluation results tabulated in *Table 7* illustrate the need for a project developer to have a good handle on the combined heat and power requirements at the onset of project development. This example indicates that Cases 1 and 5 are equivalent on the basis of Discounted Rate of Return (DRR). The economics for

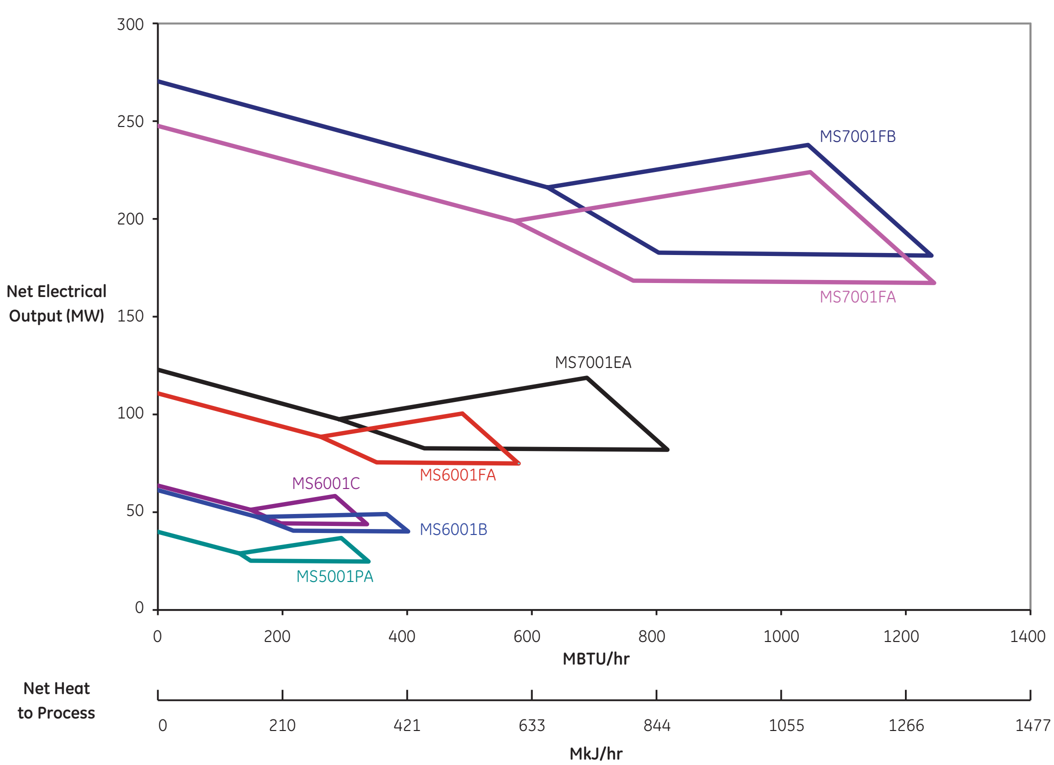


Figure 21. Gas turbine cogeneration systems (MS options, 60 Hz)

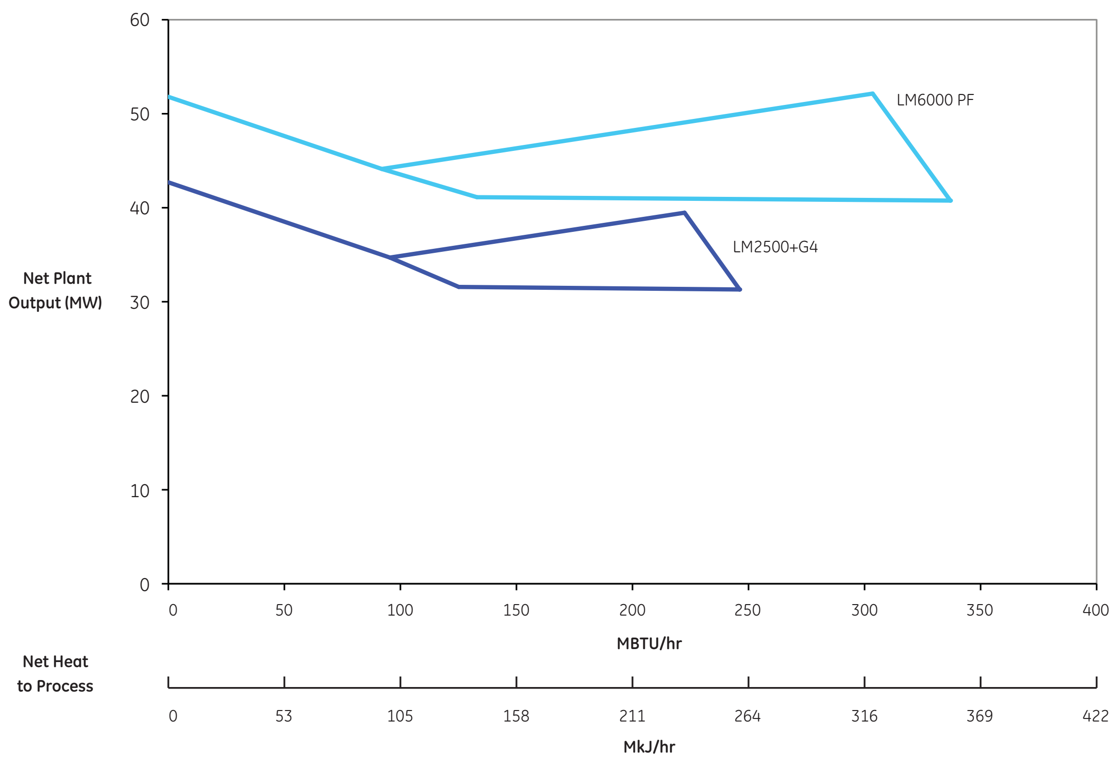


Figure 22. Gas turbine cogeneration systems (LM options, 50/60 Hz)

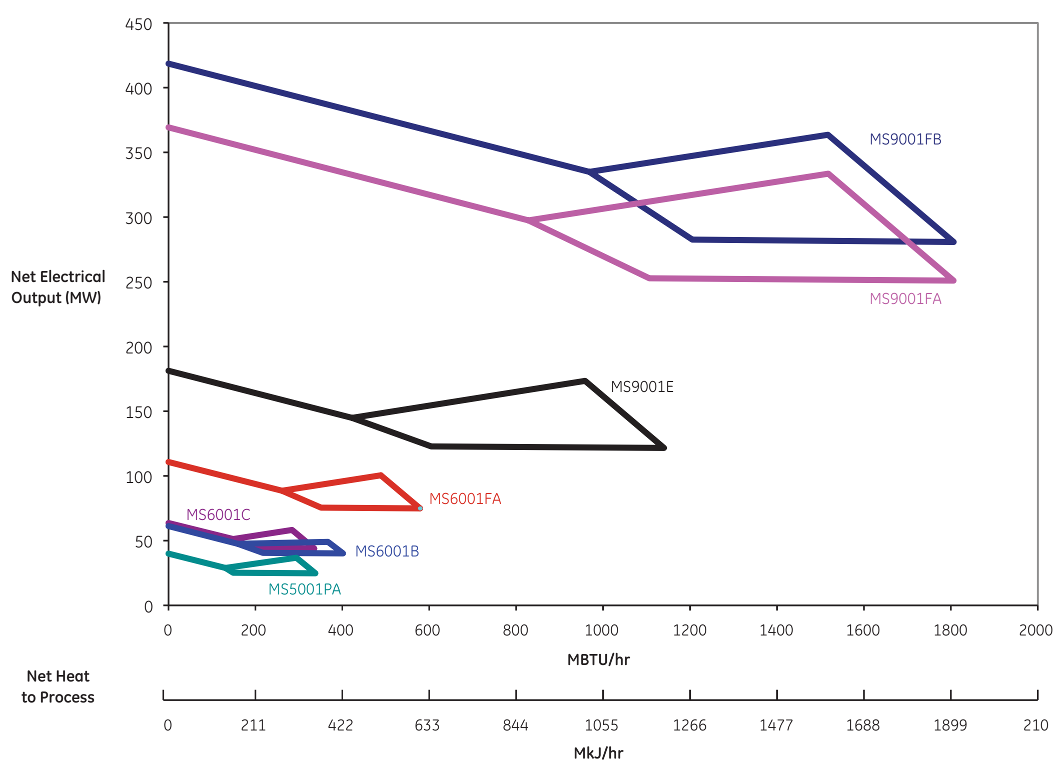
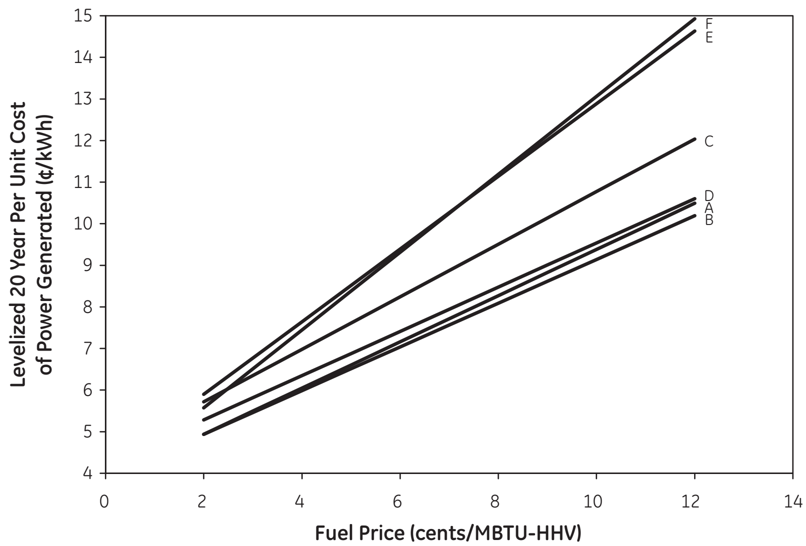


Figure 23. Gas turbine cogeneration systems option (50 Hz)



Basis:
1) Grass Roots MS6001FA Cogeneration Plant
2) Cycles as Defined in Figure 15 and Table 4
3) Operation 8400

Figure 24. Per unit cost of power generation (MS60001FA cycles)

Case 1 – 1GT + HRSG (a thermally matched case) is primarily driven by FCP (has best cogeneration efficiency) while Case 5 benefits from improved capital recovery (lower $/kW) and is driven primarily by power sales. *Figure 25* is a plot of Discounted Rate of Return as a function of fuel price at two different power sale price levels. It is included here to illustrate the sensitivity of each of the five plant configuration cases as a function of the aforementioned parameters—which in turn highlight the need to evaluate the configurations over a range of economic conditions vs. a single point. *Figure 25* illustrates that while Case 1 (1GT + HRSG) and Case 2 (1GT + NC ST) (thermally matched cases) may not have the best DRR over the range considered, they do have the shallowest slope—which means they are the least sensitive to fuel and power sale prices. Another way of looking at this is that these cases have the best cogeneration efficiencies – lowest FCP.

Additional inspection of *Figure 25* reveals that at the low end of the fuel price range the economics are largely driven by capital investment – $/kW. In other words the DRR case *Rank order* (highest to lowest) of 5, 4, 1, 3, 2 is exactly the same rank order (lowest to highest) of capital investment as identified in *Table 7*. In the high fuel price regime, power plant economics are largely driven by the combination of efficiency and power sale margin.

Cogeneration Opportunities

Circumstances under which cogeneration should be considered include:

- Development of new industrial or commercial facilities
- Major expansions to existing industrial facilities
- Expansion of large commercial and educational institutions (such as universities, hospitals and shopping malls that need power, heat and/or cooling)
- Replacement of aging steam generation equipment
- Significant changes in energy costs (fuel and power)
- Power sales opportunities

New industrial plants or major expansions to existing facilities that have large process heat demands and continuous process operations provide ideal opportunities to evaluate cogeneration. In these instances, cogeneration is compared to a Base Case where process heat is produced on-site with power requirements purchased from the utility. Cogeneration represents an incremental investment relative to the Base Case with significant infrastructure savings. Thus, the capital cost on a $/kW basis is less than for a grass roots Base Case facility without this “investment credit.” For example, assuming that a new facility requires 360,000 lb/hr (163,290 tons/hr) of gas-fired boiler capacity at 150 psig/11.4 bars, and 75 MW, the incremental investment for an MS6001FA with an unfired HRSG providing a portion of the required steam may be about 860 $/kW, whereas, installation of a separate facility with the MS6001FA and supplementary-fired HRSG system may approach 1230 $/kW—making a potential project more difficult to economically justify. *(See Table 8.)*

Replacement of old low-pressure process steam boilers, or even boilers with higher steam conditions used to support a steam turbine cogeneration system often provides an attractive cogeneration opportunity. Boiler steam capacity can be replaced by a gas turbine/HRSG system significantly increasing the system power-to- heat ratio at an attractive FCP. In addition, the "investment credit" for the replacement boiler generally assures that the $/kW cost can be reasonable.

When a facility anticipates a significant change in energy costs, the economic potential of cogeneration should be examined. This is particularly true in locations where purchased power costs may be increasing much faster than fuel costs. A cogeneration evaluation may suggest attractive economics even if there are no offsetting investments. Furthermore, if the cogeneration system results in an attractive FCP, the profitability may increase as fuel costs increase.

Many projects have been developed as a result of favorable power sales opportunities. Some projects are of a size that could have simple displaced power purchases. Others are based on circumstances where large process heat demands permit generation of electric power significantly in excess of plant power needs, such as the enhanced oil recovery projects using steam injection.

Case	1	2	3	4	5
Case Name	1GT + HRSG	1GT + NC ST	2GT + Cond ST	3GT + Cond ST	4GT + Cond ST
Gas Turbine Units	1	1	2	3	4
HRSG Pressure Levels	1	2	2	2	2
Steam Turbine	None	Noncondensing	Extraction condensing	Extraction condensing	Extraction condensing
Net Fuel					
MBTU/hr (HHV)	821.7	902.1	1642.9	2464.4	3285.9
GJ/hr (HHV)	866.9	951.7	1733.3	2599.9	3466.6
Net Power					
MW	75	93	198	311	425
Fuel Chargeable to Power					
BTU/kWh (HHV)	5654	5435	6290	6654	6798
GJ/kWh (HHV)	5965	5734	6636	7020	7172
Estimated Installed Cost					
$ / kW (2008)	1210	1479	1280	1102	1012
Discounted Rate of Return					
Percent	15.0	12.8	13.2	14.1	15.0

Basis:

1. Process steam demand at 150 psig saturated (10.3 barg saturated) is 366,200 lb/hr (166.1 tonnes/hr).
2. Case 2 has supplimentary firing. All others are unfired.
3. Net fuel includes credit for process steam delivered at 84% boiler efficiency (HHV).
4. All comparisons with existing facility generating steam for direct use in process, 8000 hr/yr operation assumed.
5. Fuel cost 7.0 $/MBTU (HHV); power value 0.0755 $/kWh.
6. Incremental costs for operating labor, water and maintenance are included.
7. DRR based on 50% equity financing, 100% accelerated depreciation, 20 year economic life, 1.8% local property taxes and insurance, 38% income taxes and 3% annual escalation.

Table 7. MS6001FA Cogeneration Example

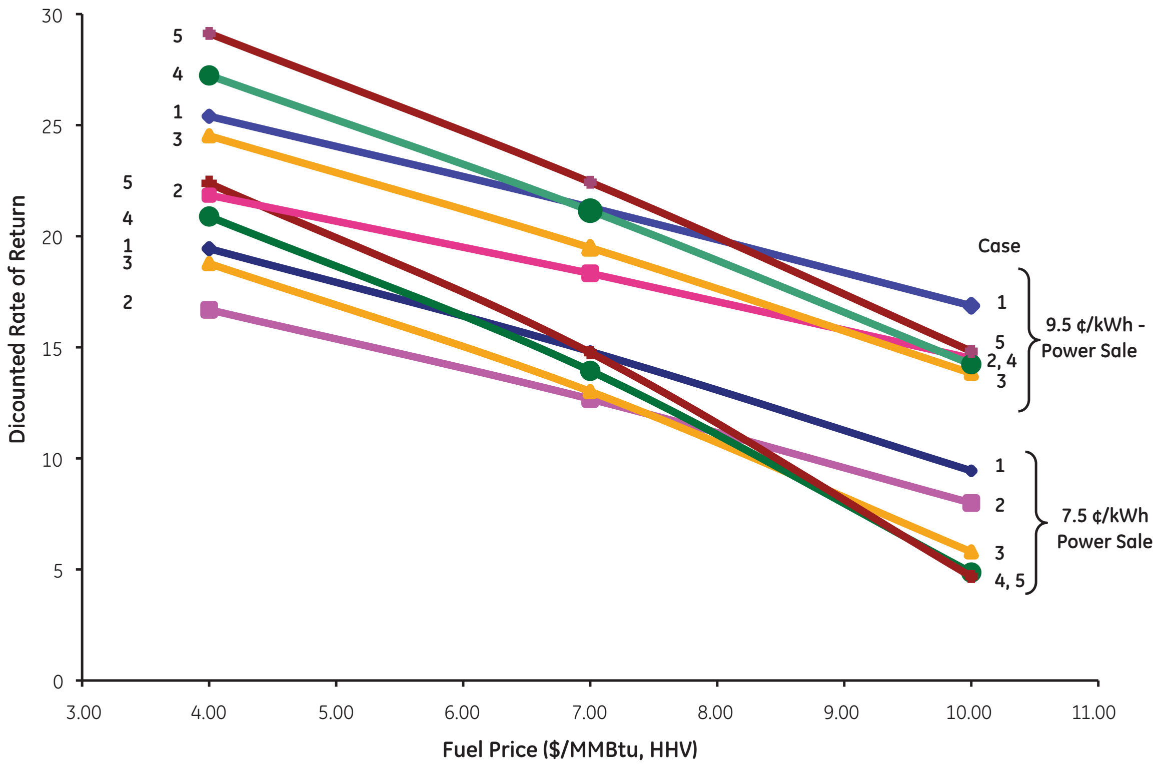


Figure 25. Economic performance at various fuel and power costs (MS6001FA cogeneration example)

Many cogeneration projects have been developed where the value is driven by the revenues from power sales to the utility grid. Frequently, the steam host is simply the mechanism for qualification and revenue from steam sales incidental to the financial success of the project. Through the use of financial leverage, projects can be developed yielding returns that, on a 100% equity basis, are lower than that considered acceptable by many industrials for discretionary investments—yet based on the leverage, they are quite attractive as independent investments.

Grass Roots Facility		
Alternative	Base	MS6001FA
Generator - MW	NA	75.5
Estimated Total Installed Cost - $ Millions	28	92.9
Incremental Investment $ Millions	Base	64.9
Unit Cost - $/kW	NA	1230
Incremental Unit Cost - $/kW	Base	860

Basis

1 Plant requires 363,000 lb/hr (164,625 kg/hr) of 150 psig (11.4 bara) saturated steam, 75 MW electric power

2 Gas turbine performance based on sea level site, 59F (15C) ambient temperature 60% relative humidity, natural gas fuel, DLN for NOx control to 15 ppm (30mg/Nm3)

3 Costs are feasibility grade values that do not include escalation, interest during construction, spares, or project soft costs.

Table 8. Feasibility Grade Installed Cost Comparison

Conclusion

Cogeneration continues to play an important role in controlling industrial or commercial energy costs through the effective integration of power generation options into the planned energy supply system. The overall performance and application flexibility of the cogeneration equipment and system is critical to the success of these ventures. The use of automatic extraction steam turbines to control process pressures, integration of gas turbine exhaust energy for process steam generation, process fluid heating and preheated combustion air for fired process heaters are a few examples of the many options available.

As more and more industrials, commercial/educational establishments, developers and utilities around the world search for low-cost electric energy and process heat, cogeneration is found to offer high efficiency and possibly environmental benefits as well. The industrial steam host is one important key to success. The host provides the thermal energy demands that can be leveraged to highly efficient cogeneration systems as well as land for utilities and developers to site new generation facilities.

This paper has shown the large array of choices available to those configuring a future power system. Optimizing a cogeneration system is a complicated process that is usually most satisfactorily addressed when the turbine supplier, permitting engineer, steam host, system owner and utility work hand in hand. Application engineering decisions should be made with as much knowledge as possible. Each project has its own unique drivers such as redundancy, maximum kilowatt capability, pollution issues, reliability of steam or kilowatt supply, or part load operational flexibility. To respond to these issues, the application engineering team must "know" the project.

GE remains committed to the development of effective and efficient cogeneration systems that provide the user with the operational and service characteristics necessary for successful applications. It is vital to these projects that the envisioned cycles will be viewed as reliable steam supplies by the industrial hosts and, at the same time, provide solid reliable capacity in the eyes of the industrial or utility that is utilizing that electric power. We offer our application resources to develop potential alternatives and identify those systems that most economically satisfy specified energy requirements.

Acknowledgement

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Conceptual Cost Estimate for Carbon Capture and Sequestration

Attachment 4

January 21, 2013
Project No. 0151579

Environmental Resources Management
15810 Park Ten Place, Suite 300
Houston, Texas 77084-5140
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The capital cost estimates for carbon capture and compression are based on work presented in *Cost and Performance Baseline For Fossil Energy Plants*, Volume 1: Bituminous and Natural Gas to Electricity, DOE/2010/1397 (Revision 2, November 2010). The carbon capture cost estimates in that USDOE publication are based on natural gas combined cycle (NGCC) power production using a GE 7FA turbine. It is assumed that the carbon capture cost factors for a 7EA are similar to those for a 7FA machine. Because the factors are based on power production, the GE website was used to determine what the typical 7EA power production is in simple and combined cycle mode as compared to the Bayou Cogeneration Plant. On the website, the simple cycle electrical output is listed as 89 MW and the combined cycle output is listed at 263 MW. The Bayou Cogeneration Plant simple cycle rating is at 80 MW; therefore the combined cycle rating is assumed to be equal to 80 x 263/89, or 236 MW of electrical production. The capital cost factor found in the DOE document was adjusted using the Engineering News Record Construction Cost Index to 2012 dollars. The operation and maintenance (O&M) costs are not adjusted or escalated. The cost of the parasitic load imposed by the carbon capture system is monetized by determining the fuel cost to compensate for the loss in output (14.7% parasitic load) at $4/MMBtu cost for natural gas.

The cost estimates for the pipeline transportation and geologic sequestration are based on information presented in the USDOE document, *Estimating Carbon Dioxide Transport and Storage Costs*, DOE/NETL-2010/1447 (March 2010). The formulas presented in that document are followed directly. The pipeline diameter and length are assumed. In addition, it is assumed that 2 injection wells are necessary for redundancy.

The annual project costs are determined by adding the annual O&M costs to the annualized cost of capital. Capital costs are annualized over 20 years at 10% interest. It is assumed that the CCS system provides 90% control of carbon dioxide that would otherwise go to the atmosphere. The cost of control is determined by dividing the annual cost by the amount of carbon captured.

Attachment 4
Air Liquide Bayou Cogeneration Plant
GHG BACT Analysis
Conceptual Cost Estimate for Carbon Capture and Sequestration

Post-Combustion CO_2 Capture and Compression		
Capital [1]	$758/kW	$716,058,722
Annual O&M [1]	$0.00124/kWh	$10,254,106
Annual Fuel [2]	14.7% fuel use at $4/MMBtu	$4,883,034

Pipeline Cost Breakdown [3]		
L, Pipeline Length (miles)		100
D, Pipeline Diameter (inches)		20
	Pipeline Costs	
Materials	$\$64,632 + \$1.85 \times L \times (330.5 \times D^2 + 686.7 \times D20 + 26,960)$	$32,050,022
Labor	$\$341,627 + \$1.85 \times L \times (343.2 \times D^2 + 2074 \times D + 170,013)$	$64,864,632
Miscellaneous	$150,166 + $1.58 x L x (8,417 x D + 7,234)	$27,890,858
Right of Way	$48,037 + $1.2 x L x (577 x D + 29,788)	$5,007,397
	Other Capital	
CO_2 Surge Tank	Fixed	$1,150,636
Pipeline Control System	Fixed	$110,632
	O&M	
Fixed O&M ($/year)	$8,632 x L	$863,200

Geologic Storage Costs [3]		
Number of Injection Wells		2
Well Depth (m)		2,134
CO_2 Captured (tons)		1,746,403
	Capital	
Site Screening and Evaluation	Fixed	$4,738,488
Injection Wells	$\$240,714 \times e^{0.0008 \times \text{Well Depth}}$	$1,327,177
Injection Equipment	$\$94,029 \times (7,839/(280 \times \text{Number of Injection Wells}))^{0.5}$	$351,802
Liability Bond	Fixed	$5,000,000
	Declining Capital Funds	
Pore Space Acquisition	$0.334/short ton CO_2	$583,299
	O&M	
Normal Daily Expenses	$11,566/Injection Well	$23,132
Consumables	$2,995/yr/ton CO_2/day	$14,330,076
Surface Maintenance	$\$23,478 \times (7,839/(280 \times \text{Number of Injection Wells}))^{0.5}$	$87,841
Subsurface Maintenance	$7.08/ft-depth/Injection Well	$30,217

Annualized Cost Estimate	
Economic Life, years	20
Interest Rate (%)	10
Capital Costs	$859,133,664
O&M Costs (Annual)	$30,471,606
Capital Recovery	$100,913,520
Total Annualized Cost	$131,385,126
Total CO_2 Controlled (tpy)	1,746,403
CO_2 Cost Effectiveness ($/ton)	75

[1] Adapted from Cost and Performance Baseline For Fossil Energy Plants, Volume 1: Bituminous and Natural Gas to Electricity, DOE/2010/1397 (Revision 2, November 2010). Plant output converted from CHP to equivalent Frame 7EA combined cycle output to enable use of cost information (www.ge-energy.com/products and services/products/gas turbines heavy duty/7ea heavy duty gas turbine.jsp). Capital costs adjusted using the ENR Construction Cost Index to 2012 dollars. O&M costs not adjusted.
[2] Fuel costs represent the additional fuel necessary to compensate for parasitic load caused by the addition of CCS. Based on review of review of the plant heat rates used in Case 13 and 14 presented in Cost and Performance Baseline For Fossil Energy Plants, Volume 1: Bituminous and Natural Gas to Electricity, DOE/2010/1397 (Revision 2, November 2010), CCS imposes a 14.7% increase in the plant heat rate; therefore, 14.7% more fuel is necessary to meet plant output. That amount of output need to come from somewhere, and
[3] Pipeline and Geologic Storage cost estimates based on National Energy Technology Laboratory (US DOE) document, *Estimating Carbon Dioxide Transport and Storage Costs* , DOE/NETL-2010/1447 (March 2010).