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GE Power Systems

GE Gas Turbine Performance Characteristics

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Contents

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Introduction

GE offers both heavy-duty and aircraft-derivative gas turbines for power generation and industrial applications. The heavy-duty product line consists of five different model series: MS3002, MS5000, MS6001, MS7001 and MS9001.

The MS5000 is designed in both single- and two-shaft configurations for both generator and mechanical-drive applications. The MS5000 and MS6001 are gear-driven units that can be applied in 50 Hz and 60 Hz markets.

All units larger than the Frame 6 are direct-drive units. The MS7000 series units that are used for 60 Hz applications have rotational speeds of 3600 rpm. The MS9000 series units used for 50 Hz applications have a rotational speed of 3000 rpm. In generator-drive applications the product line covers a range from approximately 35,800 hp to 345,600 hp (26,000 kW to 255,600 kW).

Table 1 provides a complete listing of the available outputs and heat rates of the GE heavy-duty gas turbines. *Table 2* lists the ratings of mechanical-drive units, which range from 14,520 hp to 108,990 hp (10,828 kW to 80,685 kW).

The complete model number designation for each heavy-duty product line machine is provided in both *Tables 1 and 2*. An explanation of the model number is given in *Figure 1*.

This paper reviews some of the basic thermodynamic principles of gas turbine operation and explains some of the factors that affect its performance.

GE Generator Drive Product Line

Model	Fuel	ISO Base Rating (kW)	Heat Rate (Btu/kWh)	Heat Rate (kJ/kWh)	Exhaust Flow (lb/hr) $x10^{-3}$	Exhaust Flow (kg/hr) $x10^{-3}$	Exhaust Temp (degrees F)	Exhaust Temp (degrees C)	Pressure Ratio
PG5371 (PA)	Gas	26,070.	12,060.	12,721	985.	446	905.	485	10.6
	Dist.	25,570.	12,180.	12,847	998.	448	906.	486	10.6
PG6581 (B)	Gas	42,100.	10,640.	11,223	1158.	525	1010.	543	12.2
	Dist.	41,160.	10,730.	11,318	1161.	526	1011.	544	12.1
PG6101 (FA)	Gas	69,430.	10,040.	10,526	1638.	742	1101.	594	14.6
	Dist.	74,090.	10,680.	10,527	1704.	772	1079.	582	15.0
PG7121 (EA)	Gas	84,360.	10,480.	11,054	2361.	1070	998.	536	12.7
	Dist.	87,220.	10,950.	11,550	2413.	1093	993.	537	12.9
PG7241 (FA)	Gas	171,700.	9,360.	9,873	3543.	1605	1119.	604	15.7
	Dist.	183,800.	9,965.	10,511	3691.	1672	1095.	591	16.2
PG7251 (FB)	Gas	184,400.	9,245.	9,752	3561.	1613	1154.	623	18.4
	Dist.	177,700.	9,975.	10,522	3703.	1677	1057.	569	18.7
PG9171 (E)	Gas	122,500.	10,140.	10,696	3275.	1484	1009.	543	12.6
	Dist.	127,300.	10,620.	11,202	3355.	1520	1003.	539	12.9
PG9231 (EC)	Gas	169,200.	9,770.	10,305	4131.	1871	1034.	557	14.4
	Dist.	179,800.	10,360.	10,928	4291.	1944	1017.	547	14.8
PG9351 (FA)	Gas	255,600.	9,250.	9,757	5118.	2318	1127.	608	15.3
	Dist.	268,000.	9,920.	10,464	5337.	2418	1106.	597	15.8

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Table 1. GE gas turbine performance characteristics - Generator drive gas turbine ratings

Mechanical Drive Gas Turbine Ratings

Model	Year	ISO Rating Continuous (kW)	ISO Rating Continuous (hp)	Heat Rate (Btu/shp-hr)	Heat Rate (kJ/kWh)	Mass Flow (lb/sec)	Mass Flow (kg/sec)	Exhaust Temp (degrees F)	Exhaust Temp (degrees C)
M3142 (J)	1952	11,290	15,140	9,500	13,440	117	53	1,008	542
M3142R (J)	1952	10,830	14,520	7,390	10,450	117	53	698	370
M5261 (RA)	1958	19,690	26,400	9,380	13,270	205	92	988	531
M5322R (B)	1972	23,870	32,000	7,070	10,000	253	114	666	352
M5352 (B)	1972	26,110	35,000	8,830	12,490	273	123	915	491
M5352R (C)	1987	26,550	35,600	6,990	9,890	267	121	693	367
M5382 (C)	1987	28,340	38,000	8,700	12,310	278	126	960	515
M6581 (B)	1978	38,290	51,340	7,820	11,060	295	134	1,013	545

Table 2. GE gas turbine performance characteristics - Mechanical drive gas turbine ratings

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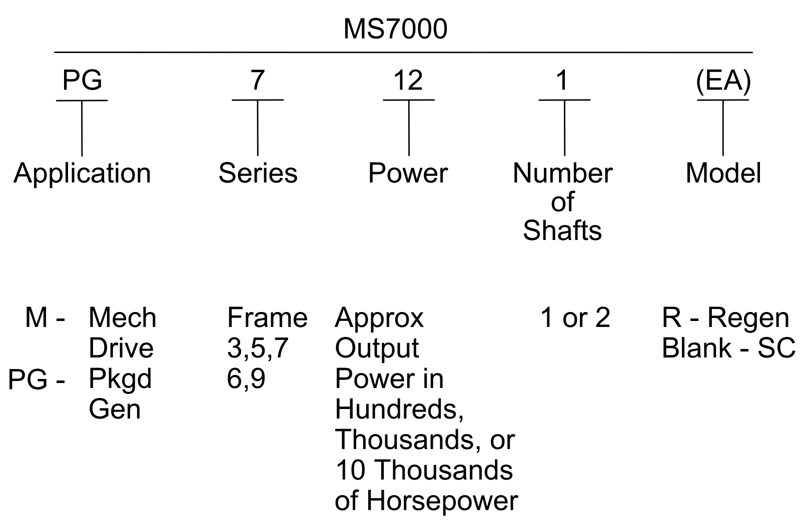

Figure 1. Heavy-duty gas turbine model designation

Thermodynamic Principles

A schematic diagram for a simple-cycle, single-shaft gas turbine is shown in *Figure 2.* Air enters the axial flow compressor at point 1 at ambient conditions. Since these conditions vary from day to day and from location to location, it is convenient to consider some standard conditions for comparative purposes. The standard conditions used by the gas turbine industry are 59 F/15 C, 14.7 psia/1.013 bar and 60% relative humidity, which are established by the International Standards Organization (ISO) and frequently referred to as ISO conditions.

Air entering the compressor at point 1 is compressed to some higher pressure. No heat is added; however, compression raises the air temperature so that the air at the discharge of the compressor is at a higher temperature and pressure.

Upon leaving the compressor, air enters the combustion system at point 2, where fuel is injected and combustion occurs. The combustion process occurs at essentially constant pressure. Although high local temperatures are reached within the primary combustion zone (approaching stoichiometric conditions), the

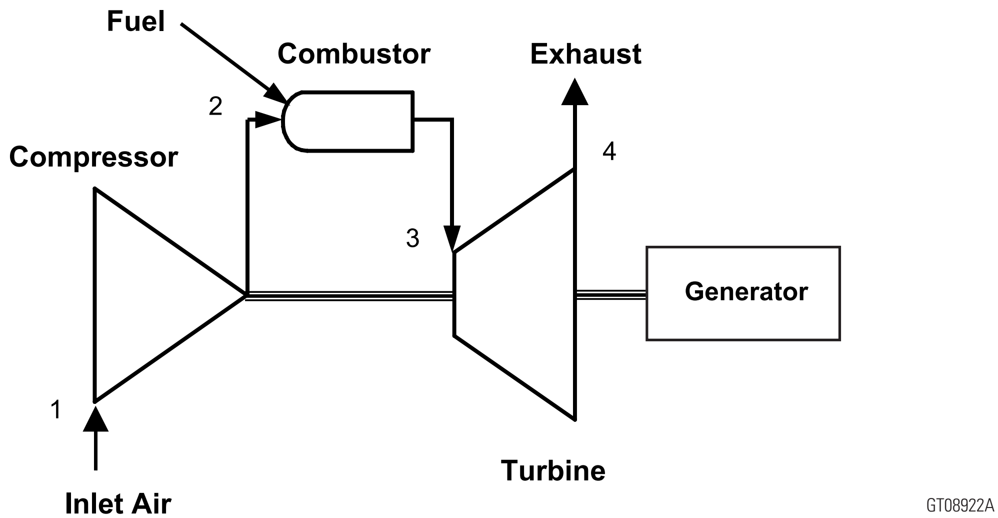

Figure 2. Simple-cycle, single-shaft gas turbine

combustion system is designed to provide mixing, burning, dilution and cooling. Thus, by the time the combustion mixture leaves the combustion system and enters the turbine at point 3, it is at a mixed average temperature.

In the turbine section of the gas turbine, the energy of the hot gases is converted into work. This conversion actually takes place in two steps. In the nozzle section of the turbine, the hot gases are expanded and a portion of the thermal energy is converted into kinetic energy. In the subsequent bucket section of the turbine, a portion of the kinetic energy is transferred to the rotating buckets and converted to work.

Some of the work developed by the turbine is used to drive the compressor, and the remainder is available for useful work at the output flange of the gas turbine. Typically, more than 50% of the work developed by the turbine sections is used to power the axial flow compressor.

As shown in *Figure 2*, single-shaft gas turbines are configured in one continuous shaft and, therefore, all stages operate at the same speed. These units are typically used for generator-drive applications where significant speed variation is not required.

A schematic diagram for a simple-cycle, two-shaft gas turbine is shown in *Figure 3*. The low-pressure or power turbine rotor is mechanically separate from the high-pressure turbine and compressor rotor. The low pressure rotor is said to be aerodynamically coupled. This unique feature allows the power turbine to be operated at a range of speeds and makes two-shaft gas turbines ideally suited for variable-speed applications.

All of the work developed by the power turbine is available to drive the load equipment since the work developed by the high-pressure turbine supplies all the necessary energy to drive the compressor. On two-shaft machines the starting requirements for the gas turbine load train are reduced because the load equipment is mechanically separate from the high-pressure turbine.

The Brayton Cycle

The thermodynamic cycle upon which all gas turbines operate is called the Brayton cycle. *Figure 4* shows the classical pressure-volume (PV) and temperature-entropy (TS) diagrams for this cycle. The numbers on this diagram cor-

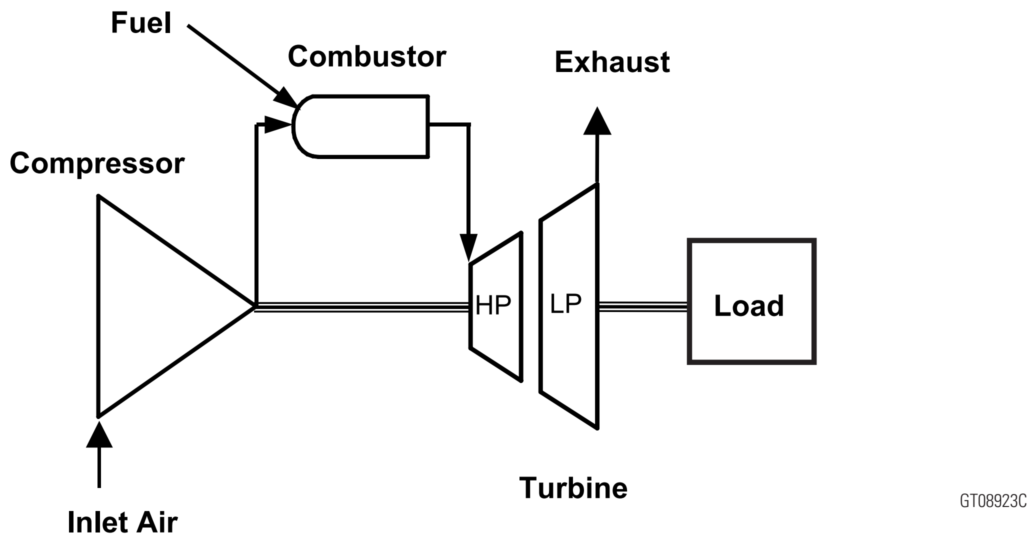

Figure 3. Simple-cycle, two-shaft gas turbine

respond to the numbers also used in *Figure 2.* Path 1 to 2 represents the compression occurring in the compressor, path 2 to 3 represents the constant-pressure addition of heat in the combustion systems, and path 3 to 4 represents the expansion occurring in the turbine.

The path from 4 back to 1 on the Brayton cycle diagrams indicates a constant-pressure cooling process. In the gas turbine, this cooling is done by the atmosphere, which provides fresh, cool air at point 1 on a continuous basis in exchange for the hot gases exhausted to the atmosphere at point 4. The actual cycle is an "open" rather than "closed" cycle, as indicated.

Every Brayton cycle can be characterized by two significant parameters: pressure ratio and firing temperature. The pressure ratio of the cycle is the pressure at point 2 (compressor discharge pressure) divided by the pressure at point 1 (compressor inlet pressure). In an ideal cycle,

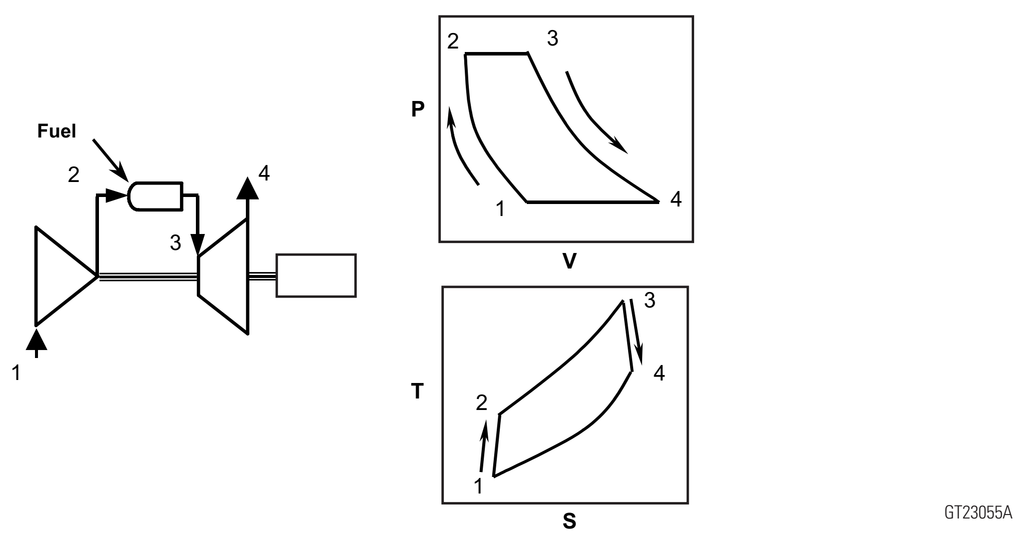

Figure 4. Brayton cycle

this pressure ratio is also equal to the pressure at point 3 divided by the pressure at point 4. However, in an actual cycle there is some slight pressure loss in the combustion system and, hence, the pressure at point 3 is slightly less than at point 2.

The other significant parameter, firing temperature, is thought to be the highest temperature reached in the cycle. GE defines firing temperature as the mass-flow mean total temperature at the stage 1 nozzle trailing edge plane. Currently all first stage nozzles are cooled to keep the temperatures within the operating limits of the materials being used. The two types of cooling currently employed by GE are air and steam.

Air cooling has been used for more than 30 years and has been extensively developed in aircraft engine technology, as well as the latest family of large power generation machines. Air used for cooling the first stage nozzle enters the hot gas stream after cooling down the nozzle and reduces the total temperature immediately downstream. GE uses this temperature since it is more indicative of the cycle temperature represented as firing temperature by point 3 in *Figure 4*.

Steam-cooled first stage nozzles do not reduce the temperature of the gas directly through mixing because the steam is in a closed loop. As shown in *Figure 5*, the firing temperature on a turbine with steam-cooled nozzles (GE's current "H" design) has an increase of 200 degrees without increasing the combustion exit temperature.

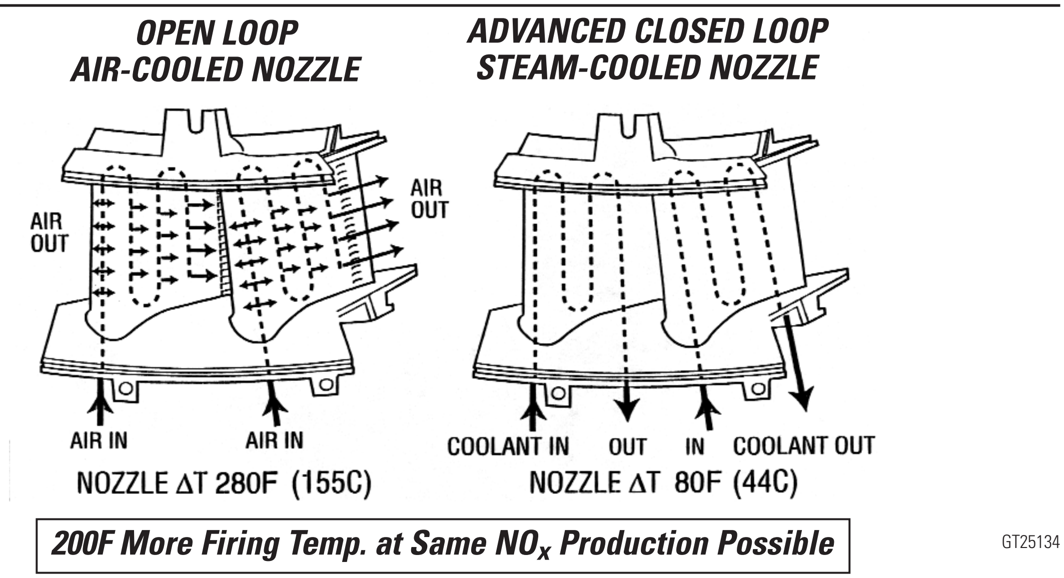

Figure 5. Comparison of air-cooled vs. steam-cooled first stage nozzle

An alternate method of determining firing temperature is defined in ISO document 2314, "Gas Turbines – Acceptance Tests." The firing temperature here is a reference turbine inlet temperature and is not generally a temperature that exists in a gas turbine cycle; it is calculated from a heat balance on the combustion system, using parameters obtained in a field test. This ISO reference temperature will always be less than the true firing temperature as defined by GE, in many cases by 100 F/38 C or more for machines using air extracted from the compressor for internal cooling, which bypasses the combustor. *Figure 6* shows how these various temperatures are defined.

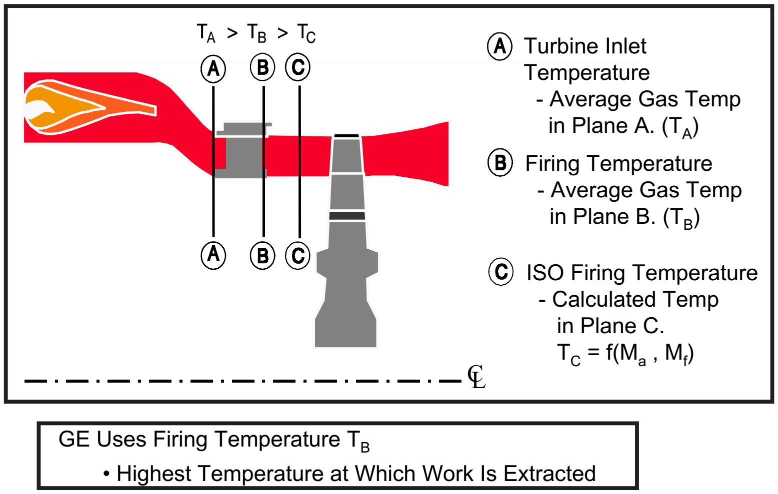

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Figure 6. Definition of firing temperature

Thermodynamic Analysis

Classical thermodynamics permit evaluation of the Brayton cycle using such parameters as pressure, temperature, specific heat, efficiency factors and the adiabatic compression exponent. If such an analysis is applied to the Brayton cycle, the results can be displayed as a plot of cycle efficiency vs. specific output of the cycle.

Figure 7 shows such a plot of output and efficiency for different firing temperatures and various pressure ratios. Output per pound of airflow is important since the higher this value, the smaller the gas turbine required for the same output power. Thermal efficiency is important because it directly affects the operating fuel costs.

Figure 7 illustrates a number of significant points. In simple-cycle applications (the top curve), pressure ratio increases translate into efficiency gains at a given firing temperature.

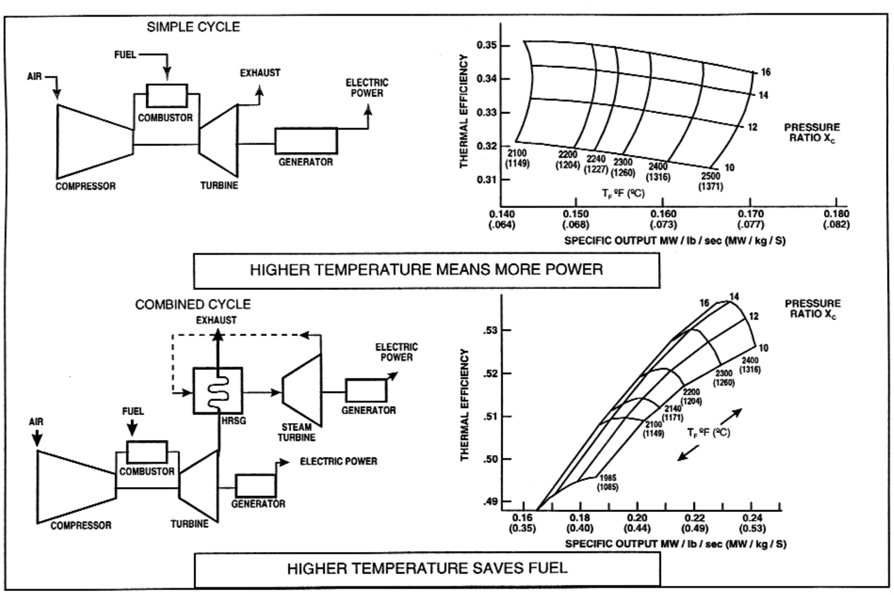

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Figure 7. Gas turbine thermodynamics

The pressure ratio resulting in maximum output and maximum efficiency change with firing temperature, and the higher the pressure ratio, the greater the benefits from increased firing temperature. Increases in firing temperature provide power increases at a given pressure ratio, although there is a sacrifice of efficiency due to the increase in cooling air losses required to maintain parts lives.

In combined-cycle applications (as shown in the bottom graph in *Figure 7*), pressure ratio increases have a less pronounced effect on efficiency. Note also that as pressure ratio increases, specific power decreases. Increases in firing temperature result in increased thermal efficiency. The significant differences in the slope of the two curves indicate that the optimum cycle parameters are not the same for simple and combined cycles.

Simple-cycle efficiency is achieved with high pressure ratios. Combined-cycle efficiency is obtained with more modest pressure ratios and greater firing temperatures. For example, the MS7001FA design parameters are 2420 F/1316 C firing temperature and 15.7:1 pressure ratio; while simple-cycle efficiency is not maximized, combined-cycle efficiency is at its peak. Combined cycle is the expected application for the MS7001FA.

Combined Cycle

A typical simple-cycle gas turbine will convert 30% to 40% of the fuel input into shaft output. All but 1% to 2% of the remainder is in the form of exhaust heat. The combined cycle is generally defined as one or more gas turbines with heat-recovery steam generators in the exhaust, producing steam for a steam turbine generator, heat-to-process, or a combination thereof.

Figure 8 shows a combined cycle in its simplest form. High utilization of the fuel input to the gas turbine can be achieved with some of the more complex heat-recovery cycles, involving multiple-pressure boilers, extraction or topping steam turbines, and avoidance of steam flow to a condenser to preserve the latent heat content. Attaining more than 80% utilization of the fuel input by a combination of electrical power generation and process heat is not unusual.

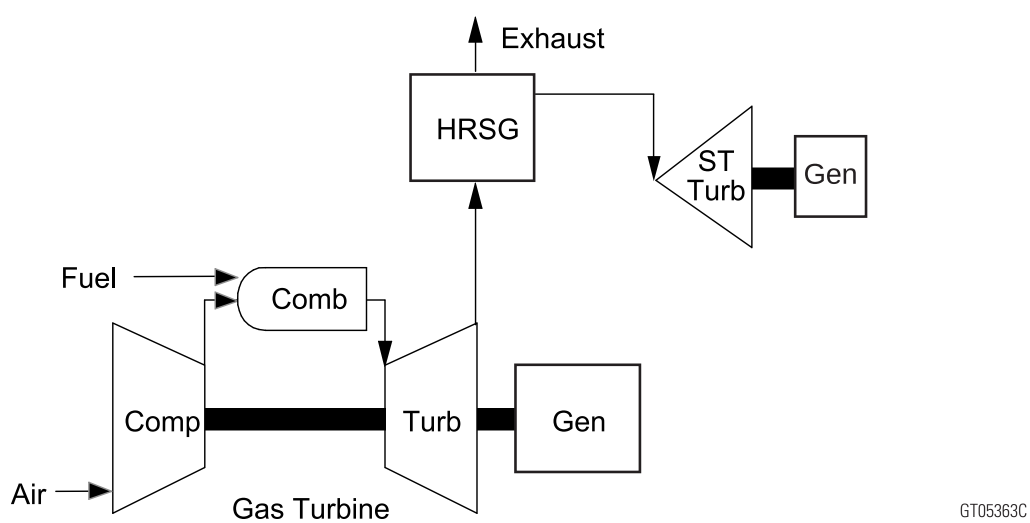

Figure 8. Combined cycle

Combined cycles producing only electrical power are in the 50% to 60% thermal efficiency range using the more advanced gas turbines.

Papers dealing with combined-cycle applications in the GE Reference Library include: GER-3574F, “GE Combined-Cycle Product Line and Performance”; GER-3767, “Single-Shaft Combined-Cycle Power Generation Systems”; and GER-3430F, “Cogeneration Application Considerations.”

Factors Affecting Gas Turbine Performance

Air Temperature and Site Elevation

Since the gas turbine is an air-breathing engine, its performance is changed by anything that affects the density and/or mass flow of the air intake to the compressor. Ambient weather conditions are the most obvious changes from the reference conditions of 59 F/15 C and 14.7 psia/1.013 bar. *Figure 9* shows how ambient temperature affects the output, heat rate, heat consumption, and exhaust flow of a single-shaft MS7001. Each turbine model has its own temperature-effect curve, as it depends on the cycle parameters and component efficiencies as well as air mass flow.

Correction for altitude or barometric pressure is more straightforward. The air density reduces as the site elevation increases. While the resulting airflow and output decrease proportionately, the heat rate and other cycle parameters are not affected. A standard altitude correction curve is presented in *Figure 10*.

Humidity

Similarly, humid air, which is less dense than dry air, also affects output and heat rate, as shown in *Figure 11*. In the past, this effect was thought to be too small to be considered. However, with the increasing size of gas turbines and the utilization of humidity to bias water and steam injection for NO_x control, this effect has greater significance.

It should be noted that this humidity effect is a result of the control system approximation of firing temperature used on GE heavy-duty gas turbines. Single-shaft turbines that use turbine exhaust temperature biased by the compressor pressure ratio to the approximate firing temperature will reduce power as a result of

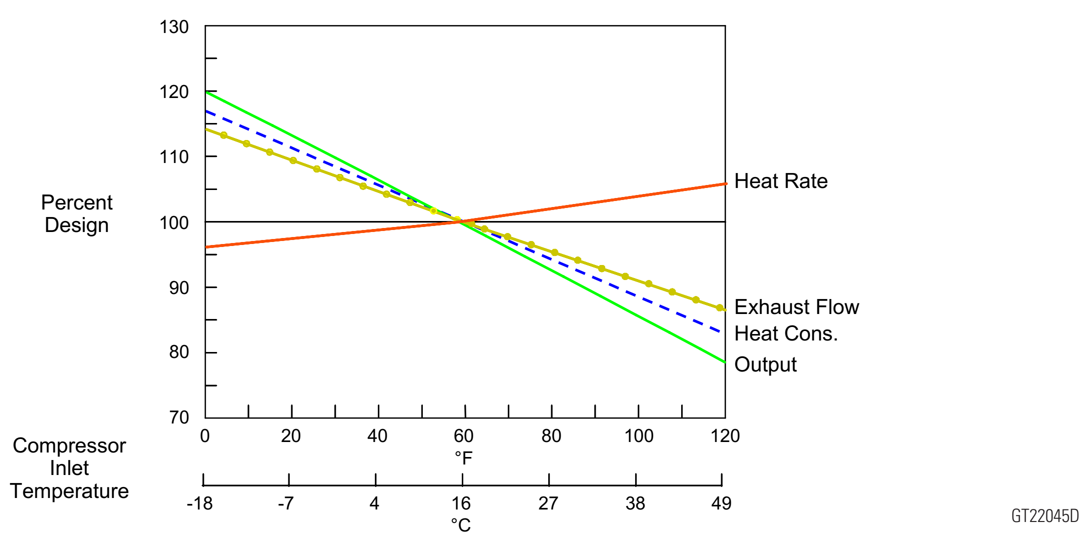

Figure 9. Effect of ambient temperature

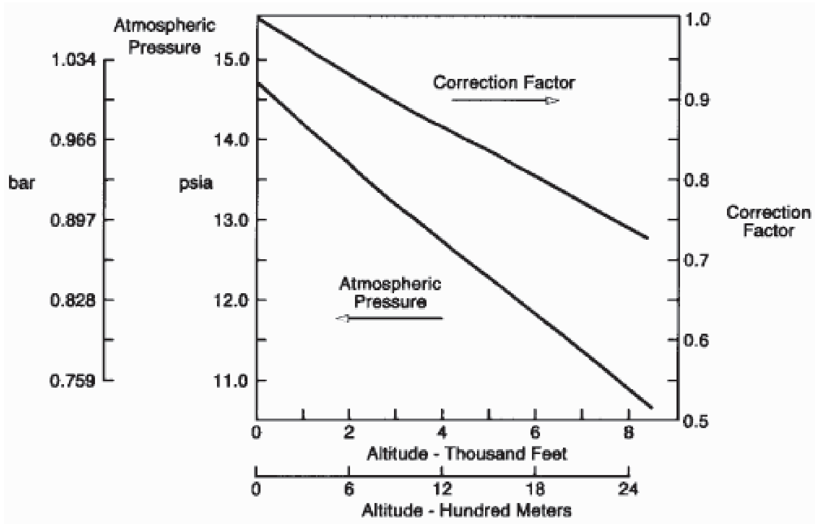

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Figure 10. Altitude correction curve

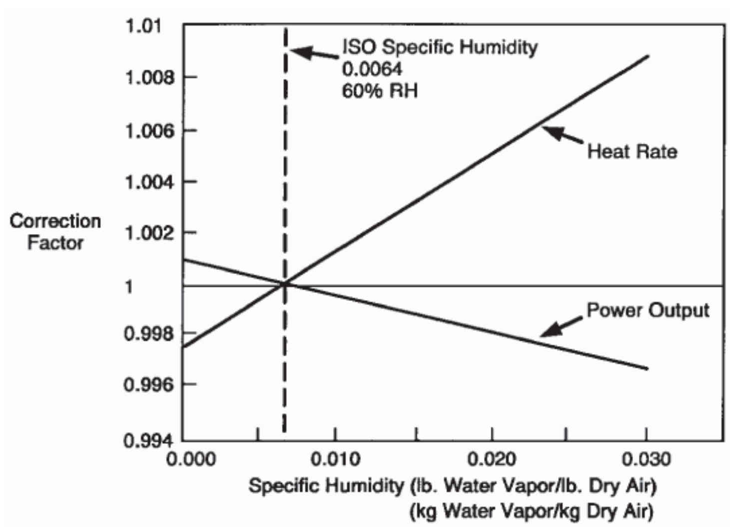

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Figure 11. Humidity effect curve

increased ambient humidity. This occurs because the density loss to the air from humidity is less than the density loss due to temperature. The control system is set to follow the inlet air temperature function.

By contrast, the control system on aeroderivatives uses unbiased gas generator discharge temperature to approximate firing temperature. The gas generator can operate at different speeds from the power turbine, and the power will actually increase as fuel is added to raise the moist air (due to humidity) to the allowable temperature. This fuel increase will increase the gas generator speed and compensate for the loss in air density.

Inlet and Exhaust Losses

Inserting air filtration, silencing, evaporative coolers or chillers into the inlet or heat recovery devices in the exhaust causes pressure losses in the system. The effects of these pressure losses are unique to each design. *Figure 12* shows

4 Inches (10 mbar) H_2O Inlet Drop Produces:

1.42% Power Output Loss

0.45% Heat Rate Increase

1.9 F (1.1 C) Exhaust Temperature Increase

4 Inches (10 mbar) H_2O Exhaust Drop Produces:

0.42% Power Output Loss

0.42% Heat Rate Increase

1.9 F (1.1 C) Exhaust Temperature Increase

GT18238C

Figure 12. Pressure drop effects (MS7001EA)

the effects on the MS7001EA, which are typical for the E technology family of scaled machines (MS6001B, 7001EA, 9001E).

Fuels

Work from a gas turbine can be defined as the product of mass flow, heat energy in the combusted gas (Cp), and temperature differential across the turbine. The mass flow in this equation is the sum of compressor airflow and fuel flow. The heat energy is a function of the elements in the fuel and the products of combustion.

Tables 1 and 2 show that natural gas (methane) produces nearly 2% more output than does distillate oil. This is due to the higher specific heat in the combustion products of natural gas, resulting from the higher water vapor content produced by the higher hydrogen/carbon ratio of methane. This effect is noted even though the mass flow (lb/h) of methane is lower than the mass flow of distillate fuel. Here the effects of specific heat were greater than and in opposition to the effects of mass flow.

Figure 13 shows the total effect of various fuels on turbine output. This curve uses methane as the base fuel.

Although there is no clear relationship between fuel lower heating value (LHV) and output, it is possible to make some general assumptions. If the fuel consists only of hydrocarbons with no inert gases and no oxygen atoms, output increases as LHV increases. Here the effects of Cp are greater than the effects of mass flow. Also, as the amount of inert gases is increased, the decrease in LHV will provide an increase in output. This is the major impact of IGCC type fuels that have large amounts of inert gas in the fuel. This mass flow addition, which is not compressed by the gas turbine's compressor, increases the turbine output. Compressor power is essentially unchanged. Several side effects must be considered when burning this kind of lower heating value fuels:

- Increased turbine mass flow drives up compressor pressure ratio, which eventually encroaches on the compressor surge limit
- The higher turbine power may exceed fault torque limits. In many cases, a larger generator and other accessory equipment may be needed
- High fuel volumes increase fuel piping and valve sizes (and costs). Low- or medium-Btu coal gases are frequently supplied at high temperatures, which further increases their volume flow

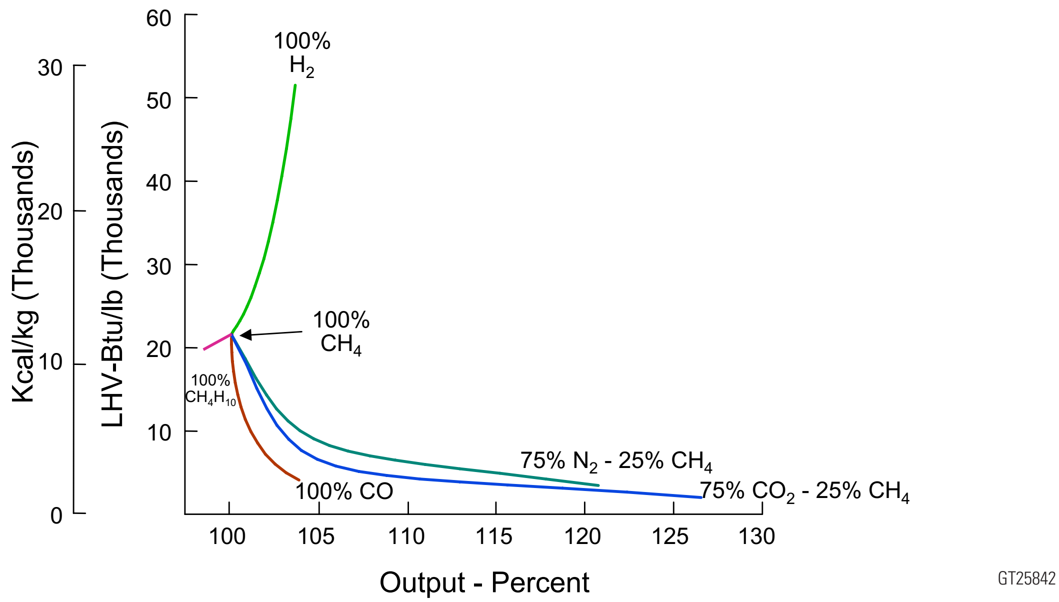

Figure 13. Effect of fuel heating value on output

- Lower-Btu gases are frequently saturated with water prior to delivery to the turbine. This increases the combustion products heat transfer coefficients and raises the metal temperatures in the turbine section which may require lower operating firing temperature to preserve parts lives
- As the Btu value drops, more air is required to burn the fuel. Machines with high firing temperatures may not be able to burn low Btu gases
- Most air-blown gasifiers use air supplied from the gas turbine compressor discharge
- The ability to extract air must be evaluated and factored into the overall heat and material balances

As a result of these influences, each turbine model will have some application guidelines on flows, temperatures and shaft output to preserve its design life. In most cases of operation with lower heating value fuels, it can be assumed that output and efficiency will be equal to or higher than that obtained on natural gas. In the case of higher heating value fuels, such as refinery gases, output and efficiency may be equal to or lower than that obtained on natural gas.

Fuel Heating

Most of the combined cycle turbine installations are designed for maximum efficiency. These plants often utilize integrated fuel gas heaters. Heated fuel results in higher turbine efficiency due to the reduced fuel flow required to raise the total gas temperature to firing temperature. Fuel heating will result in slightly lower gas turbine output because of the incremental volume flow decrease. The source of heat for the fuel typically is the IP feedwater. Since use of this energy in the gas turbine fuel heating system is thermodynamically advantageous, the combined cycle efficiency is improved by approximately 0.6%.

Diluent Injection

Since the early 1970s, GE has used water or steam injection for NO_X control to meet applicable state and federal regulations. This is accomplished by admitting water or steam in the cap area or "head-end" of the combustion liner. Each machine and combustor configuration has limits on water or steam injection levels to protect the combustion system and turbine section. Depending on the amount of water or steam injection needed to achieve the desired NO_X level, output will increase because of the additional mass flow. *Figure 14* shows the effect of steam injection on output and heat rate for an MS7001EA. These curves assume that steam is free to the gas turbine cycle, therefore heat rate improves. Since it takes more fuel to raise water to combustor conditions than steam, water injection does not provide an improvement in heat rate.

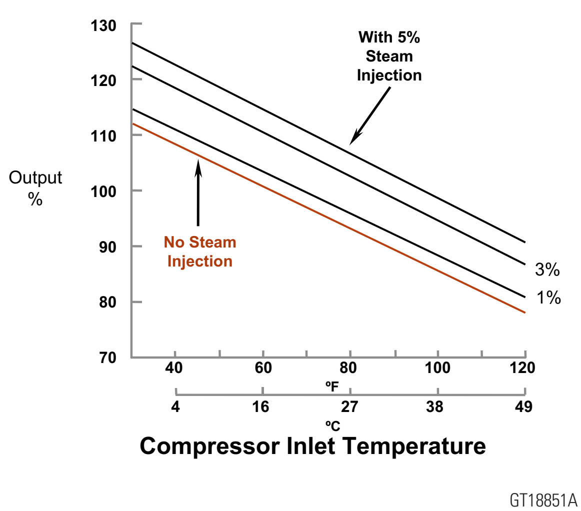

Figure 14. Effect of steam injection on output and heat rate

Air Extraction

In some gas turbine applications, it may be desirable to extract air from the compressor. Generally, up to 5% of the compressor airflow can be extracted from the compressor discharge casing without modification to casings or on-base piping. Pressure and air temperature will depend on the type of machine and site conditions. Air extraction between 6% and 20% may be possible, depending on the machine and combustor configuration, with some modifications to the casings, piping and controls. Such applications need to be reviewed on a case-by-case basis. Air extractions above 20% will require extensive modification to the turbine casing and unit configuration. *Figure 15* shows the effect of air extraction on output and heat rate. As a "rule of thumb," every 1% in air extraction results in a 2% loss in power.

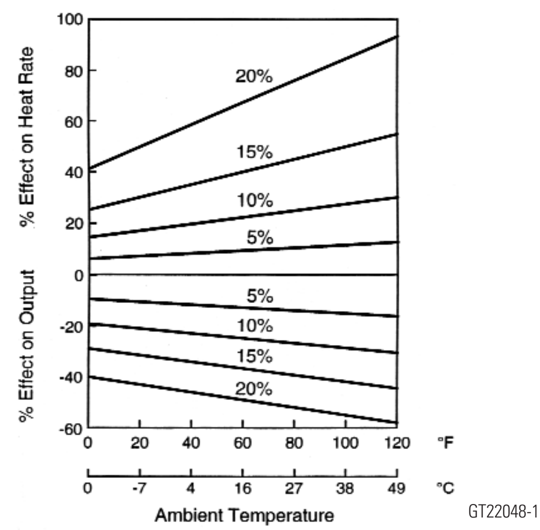

Figure 15. Effect of air extraction on output and heat rate

Performance Enhancements

Generally, controlling some of the factors that affect gas turbine performance is not possible. The planned site location and the plant configuration (such as simple- or combined-cycle) determine most of these factors. In the event additional output is needed, several possibilities to enhance performance may be considered.

Inlet Cooling

The ambient effect curve (see *Figure 9*) clearly shows that turbine output and heat rate are improved as compressor inlet temperature decreases. Lowering the compressor inlet temperature can be accomplished by installing an evaporative cooler or inlet chiller in the inlet ducting downstream of the inlet filters. Careful application of these systems is necessary, as condensation or carryover of water can exacerbate compressor fouling and degrade performance. These systems generally are followed by moisture separators or coalescing pads to reduce the possibility of moisture carryover.

As *Figure 16* shows, the biggest gains from evaporative cooling are realized in hot, low-humidity climates. It should be noted that evaporative cooling is limited to ambient temperatures of 59 F/15 C and above (compressor inlet temperature >45 F/7.2 C) because of the potential for icing the compressor. Information contained in *Figure 16* is based on an 85% effective evaporative cooler. Effectiveness is a measure of how close the cooler exit temperature approaches the ambient wet bulb temperature. For most applications, coolers having an effectiveness of 85% or 90% provide the most economic benefit.

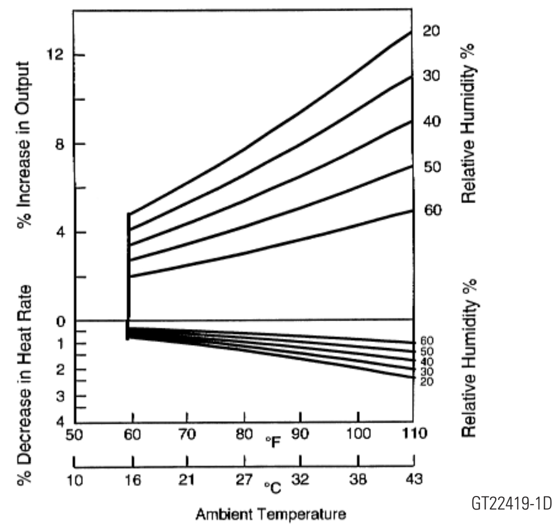

Figure 16. Effect of evaporative cooling on output and heat rate

Chillers, unlike evaporative coolers, are not limited by the ambient wet bulb temperature. The achievable temperature is limited only by the capacity of the chilling device to produce coolant and the ability of the coils to transfer heat. Cooling initially follows a line of constant

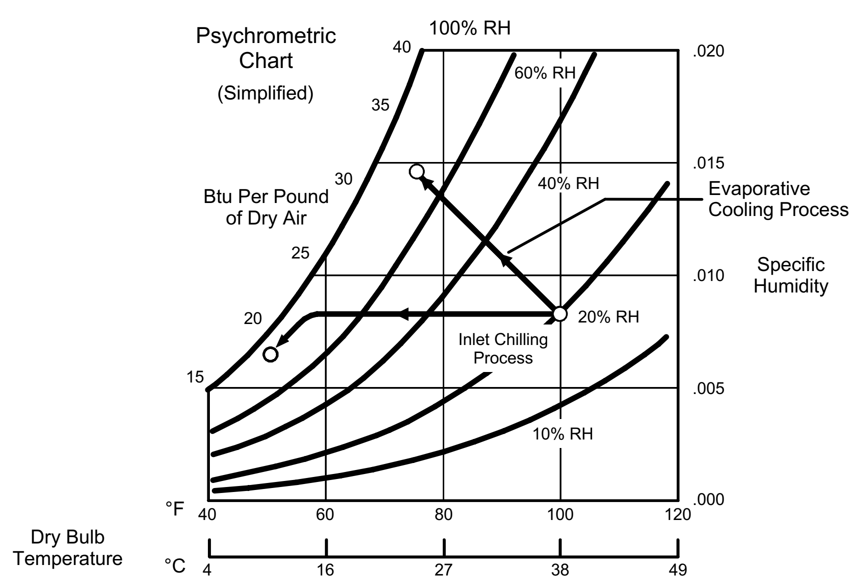

Figure 17. Inlet chilling process

specific humidity, as shown in *Figure 17*. As saturation is approached, water begins to condense from the air, and mist eliminators are used. Further heat transfer cools the condensate and air, and causes more condensation. Because of the relatively high heat of vaporization of water, most of the cooling energy in this regime goes to condensation and little to temperature reduction.

Steam and Water Injection for Power Augmentation

Injecting steam or water into the head end of the combustor for NO_X abatement increases mass flow and, therefore, output. Generally, the amount of water is limited to the amount required to meet the NO_X requirement in order to minimize operating cost and impact on inspection intervals.

Steam injection for power augmentation has been an available option on GE gas turbines for over 30 years. When steam is injected for power augmentation, it can be introduced into the compressor discharge casing of the gas turbine as well as the combustor. The effect on output and heat rate is the same as that shown in *Figure 14*. GE gas turbines are designed to allow up to 5% of the compressor airflow for steam injection to the combustor and compressor discharge. Steam must contain 50 F/28 C superheat and be at pressures comparable to fuel gas pressures.

When either steam or water is used for power augmentation, the control system is normally designed to allow only the amount needed for NO_X abatement until the machine reaches base (full) load. At that point, additional steam or water can be admitted via the governor control.

Peak Rating

The performance values listed in *Table 1* are base load ratings. ANSI B133.6 Ratings and Performance defines base load as operation at 8,000 hours per year with 800 hours per start. It also defines peak load as operation at 1250 hours per year with five hours per start.

In recognition of shorter operating hours, it is possible to increase firing temperature to generate more output. The penalty for this type of operation is shorter inspection intervals. Despite this, running an MS5001, MS6001 or MS7001 at peak may be a cost-effective way to obtain more kilowatts without the need for additional peripheral equipment.

Generators used with gas turbines likewise have peak ratings that are obtained by operating at higher power factors or temperature rises. Peak cycle ratings are ratings that are customized to the mission of the turbine considering both starts and hours of operation. Firing temperatures between base and peak can be selected to maximize the power capabilities of the turbine while staying within the starts limit envelope of the turbine hot section repair interval. For instance, the 7EA can operate for 24,000 hours on gas fuel at base load, as defined. The starts limit to hot section repair interval is 800 starts.

For peaking cycle of five hours per start, the hot section repair interval would occur at 4,000 hours, which corresponds to operation at peak firing temperatures. Turbine missions between five hours per start and 800 hours per start may allow firing temperatures to increase above base but below peak without sacrificing hours to hot section repair. Water injection for power augmentation may be factored into the peak cycle rating to further maximize output.

Performance Degradation

All turbomachinery experiences losses in performance with time. Gas turbine performance degradation can be classified as recoverable or non-recoverable loss. Recoverable loss is usually associated with compressor fouling and can be partially rectified by water washing or, more thoroughly, by mechanically cleaning the compressor blades and vanes after opening the unit. Non-recoverable loss is due primarily to increased turbine and compressor clearances and changes in surface finish and airfoil contour. Because this loss is caused by reduction in component efficiencies, it cannot be recovered by operational procedures, external maintenance or compressor cleaning, but only through replacement of affected parts at recommended inspection intervals.

Quantifying performance degradation is difficult because consistent, valid field data is hard to obtain. Correlation between various sites is impacted by variables such as mode of operation, contaminants in the air, humidity, fuel and dilutent injection levels for NO_X. Another problem is that test instruments and procedures vary widely, often with large tolerances.

Typically, performance degradation during the first 24,000 hours of operation (the normally recommended interval for a hot gas path inspection) is 2% to 6% from the performance test measurements when corrected to guaranteed conditions. This assumes degraded parts are not replaced. If replaced, the expected performance degradation is 1% to 1.5%. Recent field experience indicates that frequent off-line water washing is not only effective in reducing recoverable loss, but also reduces the rate of non-recoverable loss.

One generalization that can be made from the data is that machines located in dry, hot climates typically degrade less than those in humid climates.

Verifying Gas Turbine Performance

Once the gas turbine is installed, a performance test is usually conducted to determine power plant performance. Power, fuel, heat consumption and sufficient supporting data should be recorded to enable as-tested performance to be corrected to the condition of the guarantee. Preferably, this test should be done as soon as practical, with the unit in new and clean condition. In general, a machine is considered to be in new and clean condition if it has less than 200 fired hours of operation.

Testing procedures and calculation methods are patterned after those described in the ASME Performance Test Code PTC-22-1997, "Gas Turbine Power Plants." Prior to testing, all station instruments used for primary data collection must be inspected and calibrated. The test should consist of sufficient test points to ensure validity of the test set-up. Each test point should consist of a minimum of four complete sets of readings taken over a 30-minute time period when operating at base load. Per ASME PTC-22-1997, the methodology of correcting test results to guarantee conditions and measurement uncertainties (approximately 1% on output and heat rate when testing on gas fuel) shall be agreed upon by the parties prior to the test.

Summary

This paper reviewed the thermodynamic principles of both one- and two-shaft gas turbines and discussed cycle characteristics of the several models of gas turbines offered by GE. Ratings of the product line were presented, and factors affecting performance were discussed along with methods to enhance gas turbine output.

GE heavy-duty gas turbines serving industrial, utility and cogeneration users have a proven history of sustained performance and reliability. GE is committed to providing its customers with the latest in equipment designs and advancements to meet power needs at high thermal efficiency.

List of Figures

List of Tables