# 2 Prologue

## 2.1 Note to the Reader

This book is the final installation of a trilogy on gas turbines for electric power generation in simple and combined cycle configurations. Material covered in the first two books (close to 1,500 pages combined) forms the foundation on which the discussion below is based (more on those two books below). This chapter's goal is to equip the reader with the minimum knowledge required to follow the coverage in the rest of the book and also make them aware of the caveats, pitfalls, for example, so that reading the book does not turn into a hassle. Although the reader is expected to be familiar with industry jargon, the lack of a standardized technology and misuse of some terms necessitated this introductory coverage.

The author is not a fan of a large nomenclature section (glossary) in the beginning (or the end) of the book that makes the reader jump back and forth between the narrative and the glossary. All parameters used in the equations are defined where they appear first in the discussion. To the extent possible, acronyms or easy-to-guess alphabetical variables are used, e.g., TAMB for ambient temperature, and subscripts and superscripts are avoided. (For a full list of acronyms see Section 16.1.) Greek letters are used sparsely. Exceptions are those that are widely used in technical literature and textbooks, e.g., $\eta$ for efficiency, $\rho$ for density, and $\sigma$ for stress. The author's hope is that this will make it reasonably comfortable to read sections with quantitative information even if the reader is not thoroughly versed in the subject matter.

Thermodynamic variables commonly used in US textbooks for pressure, temperature, specific volume, enthalpy, entropy, and *availability* (exergy) are retained here as well, i.e., P, T, v, h, s, and *a*, respectively. If the reader is unable to figure out that the parameter $c_p$ designates specific heat (at constant pressure, to be precise), there is a very good chance that this book is too advanced for them.

To the extent possible, SI units will be preferred but not exclusively. The reason for that is the self-consistency of the SI system, which eliminates tedious unit conversion factors from scientific formulae – what you see is what you get. In any event, the counterpart in British units (or vice versa) will be provided in parentheses so that users more familiar with those units will not waste time with making mental conversions. No ink is wasted on unit conversions. At the time of writing (2020–2021), such information is one click away by googling on the reader's computer or smartphone.

Unless otherwise specified (mostly the case in empirical or curve-fit equations), it is assumed that the reader is aware that temperatures used in scientific formulae are in *absolute* temperature scale, i.e., degrees "Rankine" or "Kelvin."

Speaking of temperatures, a few words on the difference between *static* and *total* temperatures are in order. This is primarily a book on turbomachinery, specifically, the two prime movers, i.e., gas and steam turbines. In aero-thermo fluid theory of turbomachines, temperature is not always "temperature," pressure is not always "pressure," and enthalpy is not always "enthalpy." What is the author talking about, one might wonder? Let me explain. Pressure and temperature that one can measure with a barometer and thermometer, respectively, and plug into an *equation of state* (EOS) to calculate density are *static* values. As the designation suggests, this is the case for stagnant fluids or fluids flowing at low velocities. Inside a turbine, for example, fluid velocity can be so high that the Mach number (ratio of fluid velocity to the local sound of speed) can be close to *sonic* (i.e., 1.0) or even *supersonic* ($>$1.0). In that case, one must consider the *total* values of pressure, temperature, and enthalpy, which accounts for the kinetic energy of the fluid. The author covered this subject at length in his earlier book [1] – see section 3.3 therein.

## 2.2 Prerequisites

The leading actor in this story is the gas turbine; specifically, the "heavy duty" variant of the industrial gas turbine family. In essence, these machines are stationary jet engines "on steroids" connected to a synchronous alternating current (ac) generator for electric power generation. This book does not cover the thermo-fluid fundamentals underlying design, analysis, and optimization (performance and cost) of large stationary gas turbines. The author covered that subject in his earlier book [1]. Interested readers are advised to consult that book and references therein for an in-depth look into the gas turbine. In the remainder of the present treatise, information presented in that book, known as **GTFEPG** henceforth, will be referenced frequently. In the present book, the focus is exclusively on the gas turbine already designed, manufactured, and installed in a power plant. The goal is to ensure that it is running smoothly and safely while performing to expectations.

Heavy-duty industrial gas turbines for electric power generation rated at several hundred megawatts are typically deployed in a combined cycle configuration. In this type of power plant, the exhaust gas energy of the gas turbine is utilized in a *bottoming* Rankine cycle to generate additional electric power via a steam turbine. (Not surprisingly, the gas turbine Brayton cycle is referred to as the *topping* cycle.) Not to do that would be tantamount to a thermodynamic crime. A modern 60 Hz gas turbine rated at nearly 400 MWe has an exhaust gas flow of about 1,600 lb/s (725 kg/s) at nearly 1,200°F (650°C). As a rough estimate, this gas stream has an energy content of

$$\mathrm{E} = 1{,}600 \times 0.3 \times (1{,}200 - 200) = 480{,}000\ \mathrm{Btu/s} \times 1.05506 \sim 500{,}000\ \mathrm{kWth}.$$

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(The assumption in the calculation above is that the exhaust gas is cooled to about 200°F [90°C] stack temperature in a heat recovery boiler [HRB] to make steam to be used in a steam turbine.) The typical efficiency of a Rankine bottoming cycle is 40%, so the exhaust gas stream of this gas turbine is worth 200 MWe from a steam turbine generator.

For the thermo-fluid fundamentals underlying design, analysis, and optimization (performance and cost) of gas and steam turbine combined cycle power plants, interested readers should consult another recent book penned by the author and the references listed therein [2]. In the remainder of the discussion here, information presented in that book, **GTCCPP** henceforth, will be referenced frequently. In the present book, the focus will be on the smooth and safe operation of major combined cycle equipment (discussed above) in a seamlessly integrated manner. A brief description of said equipment will be presented in the next chapter to ensure the integrity of the narrative in the main body of the present book. However, the reader is cautioned that the basic premise of the coverage in this book is that he or she is thoroughly acquainted with the fundamental principles governing the thermal design and performance of gas and steam turbine power plants including the HRB (also known as the *heat recovery steam generator* [HRSG], usually pronounced as “her-sig”) and important *balance of plant* (BOP) equipment.

What do we mean by the term thermo-fluid fundamentals? Three subdisciplines of mechanical engineering play an important role in the design, analysis, and optimization of fossil-fired power plant equipment:

- Thermodynamics (including chemical equilibrium)
- Fluid mechanics (including gas-dynamics)
- Heat transfer

The equipment of interest in this book are of three major types:

- Turbomachinery (gas turbine with axial compressor, steam turbine, and myriad pumps)
- Heat exchanger (including the HRSG and steam turbine condenser)
- Flow control (pipes and valves)

To this list, one could also add the combustion equipment, i.e., the gas turbine combustor and HRSG duct burner. In certain cycle variants, large centrifugal process compressors will also be included. Combustion calculations involve chemical (species) balance, which falls under the major discipline of thermodynamics.

As far as design performance calculations are concerned, the only subdiscipline involved is thermodynamics; specifically, the laws of energy and mass conservation. The first one is the well-known first law of thermodynamics. A relatively simple but highly useful perspective can also be gained by applying the less understood but famous (or *infamous*, depending on your disposition) second law. In terms of pure mathematics, one only deals with algebraic equations because the physical processes do not change with time; in other words, they are time independent. In textbooks, this is commonly described as *steady-state, steady flow* (SSSF). Furthermore, strictly speaking, little or no information regarding the actual equipment or hardware is required (e.g., size in terms of geometry, weight, material properties). In fact, one frequently hears the term *rubber* hardware, which implies that the size, cost, mechanical integrity, and similar considerations do not impose a limit on the calculated plant performance.

For *off-design* calculations, the other two subdisciplines, fluid mechanics and heat transfer, come to help. This is where the divergence between computer-based heat and mass balance simulation software and simple hand (or Excel spreadsheet) calculations becomes significant. The ultimate objective is to *size* the hardware requisite for the achievement of the calculated design performance in a feasible (economically and mechanically) manner. The off-design performance is thus limited by the *fixed* hardware with a given size, geometry, and construction materials. The most common example is the sizing of plant heat exchangers, e.g., HRSG sections or the air- or water-cooled steam condenser. Based on the thermodynamic design parameters (i.e., pressure, temperature, and flow rates of the material streams crossing the heat exchanger boundary, commonly known as the *control volume* or CV), the size and number of the tubes, the shell, the materials used for their manufacture (e.g., carbon steel, stainless steel) and their specific arrangement are determined.

Most off-design calculations, e.g., part load performance at varying boundary conditions such as site ambient temperature and humidity, are time independent or SSSF as well. Thus, calculation of the off-design performance also involves the solution of a system of algebraic equations. The problem is, unlike the energy and mass conservation equations constituting the design problem, these equations are highly *nonlinear*. (Note that, while the energy conservation equations are linear at the top level with the mass flow rate and enthalpy terms, ultimately, they are not linear either when one considers the equation of state [JANAF[1] for gases and ASME[2] steam tables for water and steam] used to calculate enthalpies from pressure and temperature.) Therefore, the solution of the system of equations constituting the off-design problem is highly iterative and time consuming. Most of the time, as mentioned earlier, this difficulty can be eschewed by using *correction curves* for quick calculations.

The thermo-fluid fundamentals briefly described above are covered in great depth in earlier books written by the author [1–2]. (Henceforth, as stated earlier, those books will be referred to as **GTFEPG** and **GTCCPP**, respectively.) They will not be repeated here at any length. Of course, to facilitate the flow of narrative in a convenient manner (convenient for the reader, that is) certain basic formulae and/or charts will be reproduced as needed. Otherwise, jumping back and forth between two or three different books to follow the storyline would make reading the present book a chore.

[1] JANAF is the acronym for Joint Army Navy Air Force. The JANAF tables (originally developed for rocket scientists) are maintained by the National Institute of Standards and Technology (NIST) and provide quick access to the thermodynamic data.

[2] American Society of Mechanical Engineers.

## 2.3 Basic Concepts

This is a technical book and thus contains mathematical treatment of physical phenomena. However, equations, graphs, tables, etc., used in the narrative below are not piled into the mix gratuitously. They are intended for the reader to understand the physics underlying operational concepts discussed in the book in the simplest possible (but still rigorous) way. As an example, consider the painstaking process of starting a combined cycle steam turbine while paying close attention to steam flow rates and temperatures. The physical concept lying at the root of a steam turbine start is *thermal stress* induced in the metal body due to a change in temperature. If not controlled properly, thermal stresses can lead to plastic deformation and/or fracture over time.

### 2.3.1 Thermal Stress

There are two mechanisms that lead to thermal stress in a body: (1) restrained thermal expansion or contraction and (2) temperature gradients in thick metal parts. For example, when a metal component (say, the rotor or casing [shell] of the steam turbine) is heated or cooled in the presence of constraints preventing its expansion or contraction, respectively, a compressive or tensile stress, respectively, is produced. The magnitude of the stress resulting from a temperature change from TI (initial) to TF (final) is given by

$$\sigma = E \cdot \alpha \cdot (TI - TF), \tag{2.1}$$

where E is the elastic or Young's modulus of elasticity and α is the linear coefficient of thermal expansion. Let us do a quick calculation. Assume that the low-alloy steel rotor of a steam turbine is initially at 700°F (~370°C). Suddenly, it is subjected to steam flow at 900°F (~480°C). Ignoring the actual rotor construction details and assuming that it is basically a solid cylinder and constrained from both sides, using typical values for E and α, the resulting stress would be

$$\sigma = 200\ \text{GPa} \cdot 12 \cdot 10^{-6} \frac{1}{\text{K}} \cdot (370 - 480)\text{K} = -264\ \text{MPa}.$$

Since TF > TI (heating), σ is a negative value indicating a *compressive* stress. If TF were lower than TI, i.e., cooling, one would obtain a positive value for σ, i.e., *tensile* stress. How bad is this? A typical steam turbine rotor construction material is CrMoV (pronounced "chromolly-vee"). In the 700–900°F range, an average 0.2% yield strength of CrMoV is about 550 MPa. Thus, the magnitude of the compressive stress calculated above is close to 50% of the rotor material's strength. This, of course, is not a desirable condition.

As a result, turbo-generators (steam or gas) have only one thrust bearing. That way, the rotor can expand or contract freely in the axial direction. Turbine casings are supported similarly, i.e., only one end is fixed in the axial direction so that the other end can slide along the guides. This can be seen in Figure 2.1, which depicts an older

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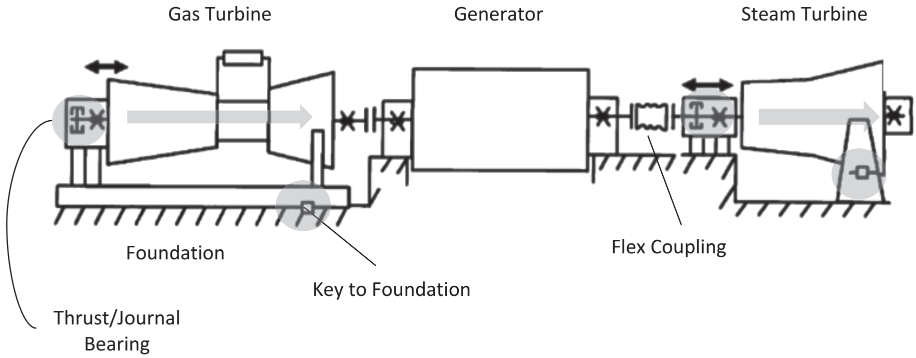


**Figure 2.1** Single-shaft combined cycle configuration.
Source: From "Single-Shaft Combined Cycle Power Generation System," L. O. Tomlinson and S. McCullough, GE Power Systems, Schenectady, NY, GER-3767C.

(E Class) single-shaft gas turbine combined cycle configuration with the generator between the prime movers. The two prime movers and the common generator have their own shafts (rotors). The steam turbine is a simple single casing, axial flow configuration with axial exhaust to the condenser. Note the locations of the two thrust bearings allowing free expansion and contraction of the rotor and where the turbine casings are "keyed" to the foundation. Also noteworthy is the flex coupling between the steam turbine and the generator, which is more forgiving than a rigid coupling in case of slight misalignment. (In modern single-shaft combined cycles with advanced class gas turbines, the steam turbine is connected to the generator through a "triple S" [SSS] *synchro-self shifting* clutch.)

A more comprehensive discussion of thermal stress can be found in chapter 13 of **GTFEPG**. The point to be made herein is that the simple relationship given by Equation (2.1) is the foundation of steam turbine stress control during plant start. For on-line rotor stress monitoring, Equation (2.1) can be reformulated as

$$\sigma = E\frac{\alpha}{1-\nu}K_T(TS - TB), \tag{2.2}$$

where ν is Poisson's ratio, which converts the linear thermal expansion coefficient α to the volume thermal expansion coefficient (for isotropic materials) and $K_T$ is the thermal stress concentration factor. In Equation (2.2), the temperature delta is calculated from the difference between the surface temperature of the rotor, TS, and the bulk or average rotor temperature, TB. For simplification, the rotor can be idealized as a long solid cylinder whereas the actual rotor has step changes in the diameter along its length, e.g., at the wheels, where nominal stress calculated from solid cylinder approximation is amplified or concentrated. This effect is taken care of by the parameter $K_T$. In any event, Equation (2.2) defines the *allowable* temperature difference between the steam flowing outside the rotor and the bulk rotor metal temperature, i.e.,

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$$\Delta T = \frac{1-\nu}{E\alpha K_T}\sigma_{max}, \tag{2.3}$$

where $\sigma_{max}$ is the maximum allowable stress, which is typically determined from an S-N Plot (S for stress and N for number of cycles). For a given material, e.g., CrMoV mentioned above, the S-N plot shows the applied maximum stress versus the number of load-unload cycles it took for the material to fail. The latter is also known as the fatigue life of a material. Each turbine start-shutdown event constitutes a *cycle*, during which major steam turbine components are exposed to thermal stress. The fatigue resulting from this type of cycling is commonly known as *low cycle fatigue* (LCF) because the number of cycles during operational life that can result in fatigue failure is in the range of *thousands*. In comparison, the number of mechanical load-unload cycles caused in rotating parts due to vibration can reach *millions* in a short amount of time. This type of fatigue is referred to as *high cycle fatigue* (HCF). To continue with the example, once $\sigma_{max}$ is determined from S-N data, the *step change* in metal temperature given by Equation (2.3) can be converted into an allowable steam temperature ramp rate (i.e., $\partial TS/\partial t$), via Fourier's law, which describes the heat transfer process from steam to the rotor metal. The three parameters, i.e., (1) metal temperature change (200°F, from 700°F to 900°F), (2) temperature ramp rate, i.e., how fast that amount of $\Delta T$ can happen, and (3) fatigue life of N cycles for the allowable (maximum) stress, are combined in *cyclic life expenditure* (CLE) curves, one example of which is shown in Figure 2.2.

As shown in Figure 2.2, and continuing with our example, steam at 900°F will eventually heat the rotor metal from its initial temperature of 700°F to that temperature. Naturally, this will not happen instantaneously but take time. The heat transfer rate from the steam to the metal is controlled by the convective *heat transfer*

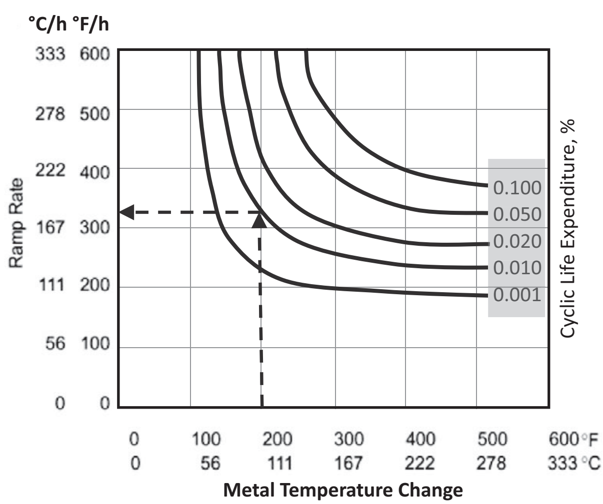


**Figure 2.2** Typical turbine rotor cyclic life expenditure (CLE) curve.

*coefficient* (HTC), which, at a given pressure and temperature, is a function steam flow rate, MS, per

$$\mathrm{HTC} \propto \mathrm{MS}^{0.8}.$$

If ∂TS/∂t is controlled (via HTC, i.e., steam pressure, temperature, and flow rate) to about 330°F/h (i.e., the 200°F temperature rise in rotor metal takes about 200/330 ~ 0.6 hours or 36 minutes), CLE is 0.01%, i.e.,

$$\mathrm{CLE} = 0.01\% = 100\%/\mathrm{N},$$

$$\mathrm{N} = 10{,}000 \text{ cycles}.$$

In other words, the rotor life would be 10,000 cycles if all cycles (start-shutdown, load ramp up-down, etc.) were of this severity.

In conclusion, steam turbine temperature matching and loading control, which are vital components of combined cycle plant start procedure, can be understood and evaluated quite rigorously with the help of a few fundamental relationships and charts. Furthermore, the same principles can be utilized to get an understanding of operability concerns in *first-of-a-kind* (FOAK) technology components, e.g., the casing of a supercritical $CO_2$ turbine operating at 25–30 MPa and 700°C.

### 2.3.2 Load and Torque

While it is not written in the proverbial stone, load and power output are used to describe the same physical quantity, i.e., the rate of useful work production by the prime mover (gas or steam turbine in this book). The difference is that the term load is used on a *relative* basis, i.e., 100% load, 75% load, etc. whereas the term power output is used on an *absolute* basis, i.e., 125 MW, 250,000 kW, etc. In order to have a better feel for this, by no means trivial, distinction, consider how a turbomachine produces work.

*Work* is the product of the *force* acting on a body and the *distance* traveled by that body under the action of the said force. In other words, to produce work, there should be a distance traveled by the said body. In the context of electric power generation, the turbomachine itself, e.g., the gas turbine, does not go anywhere, i.e., no distance is traveled at all. This is where the term *torque* comes into the picture. *Torque* (τ) is the rotational analogue of the force, which acts on a body and causes it to move a distance – linearly. When a torque acts on a body, it causes the body to *rotate*. In the case of a gas or steam turbine, the body in question is the turbine shaft (rotor). The force acting on the body is the *moment* of the force, i.e., the torque. The distance traveled is the *angular* distance, e.g., 360 degrees or 2π radians in one full revolution of the shaft or $\theta = 360° = 2\pi$. Therefore, the work done by the rotating shaft is

$$\mathrm{W} = \tau \cdot \theta. \tag{2.4}$$

Note that work and torque both have the same units, i.e., Nm (Newton-meter) in SI units, which is equal to one Joule (J), i.e., 1 J = 1 Nm. (In passing, 1 Nm is 0.7375621 lbf ft

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in British units; that is why it is much easier to have this discussion in SI units.) *Power* is the time rate of work production, which, for a shaft under the action of a constant torque, $\tau$, is equal to

$$\frac{dW}{dt} = \dot{W} = \tau \frac{d\theta}{dt} = \tau\omega, \tag{2.5}$$

where the angular speed $\omega$ (in radians per second or $s^{-1}$) is related to the physical shaft (rotational) speed in rpm (or *frequency* in *Hertz*, Hz, with 1 Hz = 1 $s^{-1}$) via

$$\omega = \frac{d\theta}{dt} = \frac{2\pi N}{60} = 2\pi f. \tag{2.6}$$

For heavy-duty industrial gas turbines used in electric power generation, N is either 3,000 rpm (or f = 50 Hz) or 3,600 rpm (or f = 60 Hz).

When the gas turbine is first started, it is *cranked* and/or *rolled* from a very low speed (a few rpm via the action of the *turning gear*, TG) to its full speed at a certain rate of angular acceleration, i.e., $\alpha = d\omega/dt$, by the combined action of an external driver and net shaft torque generated by fuel combustion to overcome the combined rotational inertia, I, of the *powertrain* (i.e., gas turbine, synchronous ac generator, and accessories connected to the same shaft). This is described by the following equation

$$\tau_{net} = I \frac{d\omega}{dt} = I\alpha. \tag{2.7}$$

The term *crank* is used when angular acceleration is accomplished by the action of an *external* driver. The term *roll* is used for the entire process from the TG speed to full speed. In all modern, large units, the external driver in question is the synchronous ac generator itself, which is run as a motor by the LCI (*load commutating inverter*, also known as a *static starter*). Cranking to the ignition speed (roughly 15–25% of full speed) is accomplished by the LCI. From ignition speed until the point when the unit becomes self-sustaining (about 50–60% of full speed), the gas turbine is rolled with the assistance of the LCI. Thereafter, the unit is rolled to *full speed, no load* (FSNL) under its own power. The time between ignition and FSNL is typically 8 to 10 minutes. At FSNL, when the torque generated by the turbine is just enough to compensate for torque consumed by the compressor plus all frictional losses, LCI is turned off (i.e., no external assistance) so that $\omega$ = constant, $\alpha = 0$ and the net shaft torque, $\tau_{net} = 0$. Note that the torque generated by the turbine (under the action of hot combustion gas flowing through it), $\tau_{turb}$, is *not* zero. However, it is equal to the torque requisite to compress the air, $\tau_{comp}$ (minus losses).

Once the gas or steam turbine generator reaches FSNL, it is ready to be synchronized to the electric network, i.e., the electric grid. After the prime mover generator (the *turbogenerator*) is synchronized to the grid, i.e., the generator breakers are closed, it is ready to be loaded. The load in question is the electric power generated by the unit. The gas turbine is loaded by increasing the airflow and fuel flow through the machine in accordance with the algorithm imposed by the control system. This can be in the form of control curves or a model representing the gas turbine in the control system computer (i.e., *model-based control*). The airflow is controlled by the compressor *inlet guide vanes* (IGV). When the IGVs are at their nominal 100% open position and the fuel flow is controlled to its value corresponding to the base load turbine inlet temperature at the prevailing cycle pressure ratio, the gas turbine is said to be running at *full load* or, more precisely, at *full speed, full load* (FSFL). At any point between FSNL and FSFL, the gas turbine is running at the synchronous speed of 3,000/3,600 depending on the grid frequency, 50/60 Hz, respectively. At each such point the net shaft/mechanical torque generated by the turbogenerator is countered by the electrical torque imposed by the grid, which is of equal magnitude and in the opposite direction, so that $\omega$ = constant, $\alpha = 0$. In the case of the steam turbine, power generation is controlled by the steam flow. This will be discussed in Section 3.3.

### 2.3.3 Simple vs. Combined Cycle

If the power plant in question has only gas turbine(s) as a prime mover, the installation and performance is commonly referred to as a *simple cycle*. To be precise, this is a misnomer because the actual gas turbine itself does not operate in a cycle per se. Air is ingested, fuel is added, and hot gas is ejected. The loop is not closed, and the working fluid is not constant. Therefore, it is also referred to as *open cycle* (yes, it *is* an *oxymoron*). Indeed there are *closed cycle* (yes, it *is* a *tautology*) gas turbines, where the term cycle is indeed apt (thermodynamically speaking), but they are not the subject of this book. (See chapter 22 of **GTFEPG** for detailed coverage.) However, closed, or semi-closed cycle systems, e.g., supercritical $CO_2$ technology in Chapter 10, will be covered in depth.

The term *combined* cycle refers to a power plant with a *topping* cycle and a *bottoming* cycle. The terms refer to the relative positions of the cycles on a temperature-entropy (T-s) diagram (see the next section). Heat rejected by the topping cycle is heat added to the bottoming cycle. This book's focus is primarily on a special case of combined cycle, i.e., with a gas turbine topping cycle and steam turbine bottoming cycle. On an ideal basis, on the T-s diagram, the gas turbine is described by the Brayton cycle and the steam turbine is described by the Rankine cycle. As described above, the real gas turbine does not operate in a true cycle. However, the steam turbine does. The Rankine cycle is indeed a cyclic process with a closed loop and single working fluid ($H_2O$).

The "official" term is *gas turbine combined cycle* (GTCC) power plant, which differentiates from *gas turbine simple cycle* power plant. For brevity, the term combined cycle (CC) is widely used in the literature, as well as in this book, and should be understood to mean GTCC.

The connection between the topping cycle and the bottoming cycle is the *heat recovery steam generator* (HRSG), pronounced as "hersig." In the author's opinion, the correct term should be *heat recovery boiler* (HRB) because the term generator also refers to the synchronous ac generator, which converts the mechanical (shaft) power of the prime movers into electric power. Another term encountered in the technical literature is *waste heat recovery boiler*, which is rarely used for electric power generation applications.

### 2.3.4 Performance

The sticker performance of a gas turbine is referred to as the *rating* performance. Almost without exception, it is the performance of the gas turbine at *full* load at ISO ambient conditions (59°F/15°C, 1 atm [i.e., zero altitude], and 60% relative humidity). Typically, rating performance is quoted with zero inlet and exit pressure losses and no performance fuel (100% methane, $CH_4$) heating (but not always). The reader is advised to check the fine print. Rating performances can be found in OEM brochures or in trade publications (e.g., *Gas Turbine World* or *Turbomachinery International*).

There is no sticker performance for a steam turbine. Its performance is dependent on myriad factors, first and foremost the gas turbine exhaust energy. Other factors include steam conditions (pressure and temperature), steam cycle (e.g., two or three pressure levels, with or without reheat), and condenser vacuum (steam turbine *backpressure*). Steam turbines are characterized by their casing configuration, last stage bucket size, and maximum steam pressure/temperature ratings. A comprehensive coverage can be found in **GTCCPP**. Key aspects are covered in depth in Section 3.3.

In a combined cycle context, the term "performance" refers to the net electric power output of the power plant, which is found by subtracting the *auxiliary power consumption* of the power plant equipment and facilities from the power output of the prime mover generators (the *gross* power output), and net efficiency. Net efficiency is the ratio of net power output to total fuel input. The latter is also referred to as *fuel* or *heat consumption*. There are two fuel consumers in a GTCC power plant: gas turbine combustors and HRSG duct burners. Duct burners increase the temperature of exhaust gas from the gas turbine by burning fuel (utilizing the approximately 11% by volume of $O_2$ in the exhaust gas) to increase steam production in the HRSG and thus steam turbine generator output. The technology is referred to as *supplementary firing* and is a widely used (especially in the USA) method of power augmentation on hot days (see Chapter 1). The goal is to compensate for the reduced power output of the gas turbines at hot ambient temperatures (via reduced air density and inlet airflow) by generating more power in the bottoming cycle. Supplementary firing is detrimental to GTCC efficiency via increased fuel burn (i.e., more money spent by the operator). However, increased power output at times of high demand for electric power (i.e., residential and commercial users' air conditioners going full blast) when electricity prices skyrocket more than make up for increased fuel expenditure (especially between 2010 and 2020 in the USA when natural gas prices were at historical). Combined cycle power plants equipped with duct burners are commonly referred to as *fired* power plants vis-à-vis *unfired* power plants with no supplementary firing capability.

Thermal efficiencies are expressed on a *lower heating value* (LHV) basis. As an example, natural gas, which is assumed to be 100% $CH_4$ (methane) has an LHV of

21,515 Btu/lb at 77°F (about 50 MJ/kg at 25°C). This is not a capricious choice. The *higher heating value* (HHV) is the *true* energy content of the fuel, which includes the latent heat of vaporization that is released by the gaseous $H_2O$ in the combustion products when they are cooled to the room temperature. In other words, HHV is the value measured in a *calorimeter*. In a real application (e.g., a gas turbine), the combustion products, by the time they reach the exhaust gas stack (e.g., at around 180°F [~82°C] for a modern combined cycle power plant), are not cooled to a temperature to facilitate condensation, which, depending on the amount of $H_2O$ vapor in the gas mixture, is around 110°F (~43°C). Thus, the latent heat of vaporization is *not* recovered and utilized. (It *can* be done by adding a condensing heat exchanger before the heat recovery steam generator [HRSG] stack but it would be highly uneconomical.)

Note that LHV is not measurable; it can only be calculated from the laboratory analysis of the fuel by subtracting the latent heat of vaporization. For 100% $CH_4$, the ratio of HHV to LHV is 1.109. Many handy formulae can be found in handbooks to estimate LHV and HHV for various fuels and fuel gas compositions.

Heat consumption is the product of fuel mass flow rate into the combustor of the gas turbine (plus fuel supplied into the duct burners of the HRSG in a combined cycle power plant, if applicable) and fuel LHV. For example, a gas turbine burning 100% $CH_4$ fuel at a rate of 30 lb/s has a heat consumption of 30 × 21,515 = 645,450 Btu/s, which is equal to about 681 MWth. If the gas turbine in question generates 275 MWe of power, its efficiency is 275/681 = 40.4% (again, net or gross).

Another unit commonly used for heat consumption is MMBtu/h (million Btus per hour). For the example above, 645,450 Btu/s is 2,324 MMBtu/h. For a 40% gas turbine at different ratings between 275 and 375 MWe, heat consumption ranges between ~2,300 and 3,200 MMBtu/h.

Frequently, thermal efficiency is expressed as a *heat rate*, which is given by 3,412 Btu/kWh divided by thermal efficiency. In SI units, heat rate is measured by kJ/kWh and it is found by dividing 3,600 kJ/kWh by the thermal efficiency. The ratio 3,600/3,412 = 1.0551 is the conversion factor for British and SI units of heat rate. Heat rate is the land-based counterpart of *specific fuel consumption*, which is a widely used metric for aircraft engines. For example, the heat rate of a gas turbine with 40% efficiency is 3,412/0.4 = 8,530 Btu/kWh. Typically, large "frame" gas turbine efficiencies range between 36 and 42% (ISO base load rating). Thus, their heat rate range is ~8,000 to 9,500 Btu/kWh (~8,500 to 10,000 in SI units).

For a gas turbine combined cycle power plant, the difference between net and gross power output can be anywhere from 1.6% to more than 3% – mainly dictated by the steam turbine heat rejection system. This will be discussed in detail in Chapter 3.

For quoting performances of new and emerging technologies, there simply are no rules. There is no well-established rating performance criteria. On top of that, scant attention is paid to the differences (maybe out of ignorance or, maybe, deliberately for marketing hyperbole) between cycle performance and plant net and gross performance. Eye-catching numbers are liberally thrown around and they do not pass a critical examination of the underlying assumptions and details (sometimes hidden, again, whether intentionally, or unintentionally, it is hard to tell). In this book, dear reader, such hyperbole is smashed with the help of the second law of thermodynamics (and with irrepressible, scientific glee on the part of the author). Read on.

#### 2.3.4.1 Cycle versus Plant Net Efficiency

The author would like to let the readers know that he is truly embarrassed that he felt obligated to pen this section. However, outrageous performance claims made by new technology developers and equipment manufacturers seem to be taken at their face value not only by the lay audience but also by industry practitioners and academic researchers. Unfortunately, marketing hyperbole has replaced rigorous engineering analysis and scientific truth. The author has harped on about this sad situation in his technical papers and articles [11–13]. Chapter 10 in this book contains rigorous thermodynamic analysis, busting the performance myth of two popular technologies widely touted in the trade publications and archival journals. (The same critical approach is also used in other chapters. The reader will surely recognize them when they see them.)

By far the biggest culprit in this deliberate manipulation or inadvertent (or inept) misrepresentation of the true performance of a given heat engine technology is the blurring of the line between the cycle efficiency and plant net efficiency. Cycle efficiency is a theoretical number that determines how well a given heat engine cycle approximates the ultimate *theoretical* value that is set by the *Carnot efficiency*. Plant net efficiency is a *commercial* number that measures the bottom line based on two directly and accurately *measurable* and *monetizable* quantities:

1. Amount of fuel burned in MMBtu/h or MWth (HHV)
2. Amount of net power supplied to the grid in MWh or kWh

The owner/operator pays money for the former and generates revenue from the latter. The difference between the two either sinks the ship (if negative) or floats it (if positive). This is the bottom line. Period.

Cycle efficiency is neither directly measurable nor monetizable. It can be inferred from measured parameters within an error band depending on the cycle complexity as well as precision and accuracy of available transducers. The gap between the two is primarily a function of the following:

- Cycle heat addition equipment (e.g., fired boiler, fired or unfired heat exchanger)
- Cycle heat rejection equipment (e.g., water- or air-cooled condenser and cooling tower)
- Balance of plant (auxiliary) equipment power consumption
- Power consumed by *add-on* process blocks (e.g., air separation unit to generate oxygen used in the oxy-combustor)

It is worth noting that these factors will be present and highly impactful even if the cycle performance is evaluated with the utmost care in attending to the design parameters such as key heat exchange equipment pinch points (i.e., heat exchanger *effectiveness*), component polytropic/isentropic efficiencies, and parasitic loss causes such as secondary flows for hot gas path cooling. In many cases, these cycle design intricacies are either completely ignored or optimistically set with little attention paid to size, cost, and manufacturability considerations.

Finally, one must realize that, especially in the present state of power grid mix of generating assets and operating rhythms, a gas/steam turbine power plant operates like a typical car during a normal day. It starts, stops, accelerates, runs at constant speed for a while, decelerates, stops, restarts, etc. During the normal transients encountered within this operating rhythm, the fuel consumption of the power plant is significantly different than the calculated (and published) rating performance at a prescribed site ambient and loading condition. For conventional GTCC power plants, this information is readily available (and calculable using established models, correction curves, etc.). Actual US plant data can be obtained from the statistics published by the US Energy Information Administration (EIA) – see Form EIA-923 data available online at www.eia.gov/electricity/data/eia923/. For a representative selection of US combined cycle power plants, the results for the period January–September 2018 are summarized in Table 2.1. Note that all six power plants in the table are equipped with supplementary-fired HRSGs (heavily deployed during summertime). Unfortunately, the EIA data files do not include the design performance data so back-calculating the load factor is not possible (probably somewhere around 70% or so). In any event, the disconnect between the ISO base load ratings and the reality in the field (more than 60% LHV for H and H+ Class units) is clear.

This is why pretentious efforts to make new and emerging technologies look much better than they really are eventually end up in failure. What is more, this is also why this book places great importance on RAM and operability. A doctored number at a single design point with highly optimistic assumptions and/or omissions does not a viable technology make. How to start and shut down a system, how it performs at low loads and/or extreme ambient temperatures, and whether it has a realistic RAM assessment are all vital pieces of information.

**Table 2.1** Selected Form EIA-923 data (January–September 2018).

MWH_TOT: Total power generated, HC_TOT: Total fuel consumed, MEE: Mean-effective (average) efficiency.

| | | | MWH_TOT | HC_TOT | Heat Rate | Efficiency, MEE | |
|---|---|---|---|---|---|---|---|
| OEM | Class | Configuration | MWh | MMBtu HHV | Btu/kWh HHV | % HHV | % LHV |
| A | F | 2 (2×2×1) | 2,315,620 | 16,954,109 | 7,322 | 46.60 | 51.68 |
| A | F+ | 2×2×1 | 2,159,244 | 15,275,314 | 7,074 | 48.23 | 53.49 |
| A | H | 3×3×1 | 5,482,523 | 36,275,531 | 6,617 | 51.57 | 57.19 |
| A | H | 3×3×1 | 5,413,559 | 36,020,550 | 6,654 | 51.28 | 56.87 |
| B | H+ | 2×2×1 | 2,315,067 | 16,682,609 | 7,206 | 47.35 | 52.51 |
| B | H+ | 2×2×1 | 1,983,351 | 13,060,540 | 6,585 | 51.82 | 57.46 |

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A very apt cautionary lesson in this sense is provided by the saga of the integrated gasification gas turbine (IGCC) technology (see Chapter 13). The underlying core technology, gasification, is more than a century old. Key subsystems on the syngas side have been successfully operational in chemical process and refinery industries for decades. The same is true for the power block, the GTCC. Nevertheless, despite a significant number of demonstration IGCC plants deployed in the last three decades of the twentieth century in the USA (Wabash, TECO Polk), Europe (Buggenum in Netherlands, Puertellano in Spain), and Japan (Negishi), the IGCC technology failed to make a commercial breakthrough mainly due to (unexpectedly) high costs, complexity of operation, and/or low reliability and availability. Even the inherent capability of pre-combustion capture of $CO_2$ failed to help the IGCC technology. Several such demonstration projects, e.g., the Southern Company's Kemper County project among them (an especially sobering failure), were ultimately dropped or repurposed. The interested reader can find the details of that colossal failure by simply googling the name on the internet.

### 2.3.5 Technology Factor

The most powerful concept in performance claim assessment is the concept of *technology factor*. The basic premise of this concept is that any practical heat engine cycle (no exceptions!) is an attempt (in most cases, quite unsuccessful) to approximate the underlying Carnot cycle. What does the qualifier "underlying" mean? It means that any heat engine cycle (no exceptions!) can be identified by two temperatures characteristic of two major heat transfer processes:

- Mean-effective temperature during cycle heat addition (METH)
- Mean-effective temperature during cycle heat rejection (METL)

This fact is a direct corollary of the Kelvin–Planck statement of the second law of thermodynamics. Note, however, that,

- METH is not necessarily the same as maximum cycle temperature; in fact, in almost all cases, METH $<$ TMAX
- METL is not necessarily the same as minimum cycle temperature; in general, METL $\geq$ TMIN (see Figure 2.3 for the terminology used in the discussion of Brayton simple and combined cycle thermodynamics)

Identifying and calculating METH and METL will be described in the context of the gas turbine Brayton cycle in Section 3.1. The reader can also refer to chapter 5 of **GTFEPG**. For an in-depth look at the technology factor concept, refer to chapter 6 of **GTFEPG**. Briefly, once METH and METL are identified, the underlying or "equivalent" Carnot efficiency of the heat engine cycle in question can be found as

$$\mathrm{ECEFF} = 1 - \frac{\mathrm{METL}}{\mathrm{METH}}, \tag{2.8}$$

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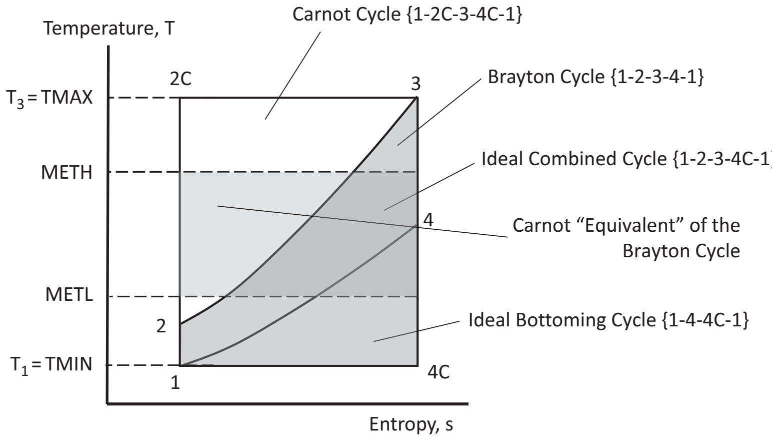


**Figure 2.3** Temperature-entropy diagram of Carnot and air-standard (ideal) Brayton cycles.

which is lower than the "ultimate" Carnot efficiency given by

$$\mathrm{UCEFF} = 1 - \frac{\mathrm{TMIN}}{\mathrm{TMAX}}. \tag{2.9}$$

The ratio of the two is the *cycle factor*, CF, i.e.,

$$\mathrm{CF} = \frac{\mathrm{ECEFF}}{\mathrm{UCEFF}}. \tag{2.10}$$

The actual cycle efficiency can be found from rigorous heat and mass balance simulation calculations and verified in the field by conducting performance tests governed by applicable test codes. The ratio of the actual cycle efficiency to that of the equivalent Carnot efficiency is the *technology factor*, i.e.,

$$\mathrm{TF} = \frac{\mathrm{ACTEFF}}{\mathrm{ECEFF}}. \tag{2.11}$$

For modern heavy-duty industrial gas turbines, the cycle factor is around 0.70. The cycle efficiency as a function of turbine inlet temperature (TIT) is plotted in Figure 2.4. For the advanced class (i.e., HA or J Class) gas turbines with inlet temperatures of 1,700°C (cycle pressure ratio [PR] of 24:1), the TF is 0.73. At the origin of the jet age, the gas turbine of the Jumo 004 engine (which powered the first operational jet Messerschmitt Me 262 in 1944–1945), TF was 0.54 at a TIT of 775°C (cycle PR of 3:1).

A correct understanding of the technology factor concept or method by the reader is of utmost importance. It is by no means a "fudge factor." Its connection to the key cycle design parameters via fundamental thermodynamic principles and correlations has been demonstrated step by step in chapter 6 of **GTFEPG**. Furthermore, the concept has been successfully used in the past by German scientists and engineers in the design of turbomachinery, e.g., turbo-superchargers under the name of *Gütezahl* or *Gütegrad*. The technology factor quantitatively identifies the prevailing state of the

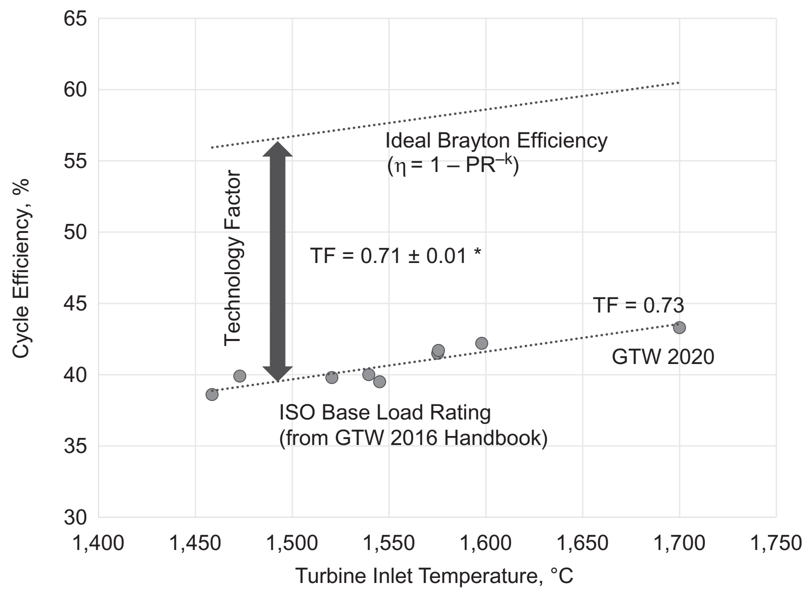


**Figure 2.4** Gas turbine Brayton cycle efficiencies (ISO base load rating) as a function of turbine inlet temperature (TIT).
*: Jumo 004B (1943–1945) TF = 0.54 (TIT= 775°C)

art in heat engine design in a manner not subject to obfuscating discussion or interpretation.

Once a new heat engine technology, say, the currently (i.e., at the time of writing this book, 2020 and 2021) very much in fashion, supercritical $CO_2$ cycle, is identified by its temperature-entropy (T-s) diagram, its performance entitlement is easily calculable as demonstrated above (and later, in Section 3.1, for the gas turbine) for the Brayton cycle. If the implied technology factor of a published performance claim is comparable to or beyond the value established by a 75 years-old technology at the pinnacle of its development stage (i.e., a J Class gas turbine), it should be rejected out of hand. To gauge the difficulty in pushing the envelope in terms of the technology associated with translating a heat engine cycle from the T-s diagram on paper to the actual hardware in the field, simply spend a few minutes on the data plotted in Figure 2.4. It took nearly half a century to bring the TF from 0.54 (TIT of 775°C and cycle PR of 3:1) to around 0.70 (roughly representative of vintage F and introductory H Class TIT of 1,500°C with PR of around 20:1). In the following quarter of a century, a significant leap was made by the gas turbine OEMs in pushing the TIT from 1,500°C to 1,700°C but the proverbial needle barely budged. (This push was primarily driven by combined cycle efficiency, which will be discussed in Section 3.1.)

The correlation between the cycle and technology factors is illustrated in the chart in Figure 2.5. As one would expect, the higher cycle factor is an enabler of higher technology factor. This can be best explained by a financial analogy (admittedly not a perfect one but still useful). If one goes to a credit institution with say, $10K, that

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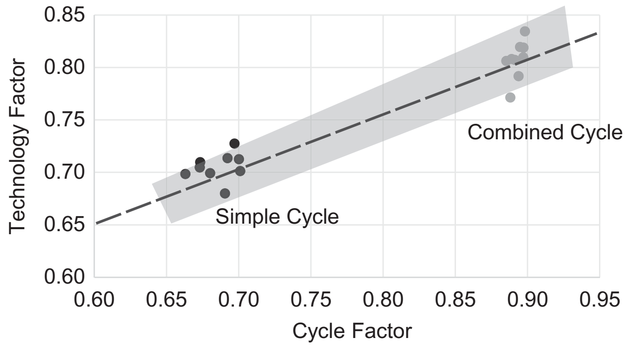


**Figure 2.5** Gas turbine Brayton simple and combined cycle technology and cycle factors.

person will get a much lower interest rate than someone who brings $10 million to the same institution.

There is no question that one must acknowledge that the starting point in TF for a new technology in the first quarter of the twenty-first century does not have to be as low as 0.54. Present knowledge and experience base in materials (e.g., superalloys used in manufacturing and construction of hot gas path components), advances in aerodynamics, heat transfer and fluid mechanics, 3D computational fluid dynamics (CFD) tools, and full-scale test facilities certainly make it possible for a new technology to start its life cycle at a higher TF. Thus, if the cycle factor of the technology in question is around 0.7, a value between 0.6 and 0.7 can be taken at its face value with certain caveats in place. For a higher cycle factor, a higher technology factor can be acceptable. The evaluation should be made for any new technology on a case-by-case basis.

### 2.3.6 Power Augmentation

As mentioned above, on hot days, gas turbine outputs drop drastically due to reduced air density and air flow. The resulting loss in electric power generation capacity is compensated for by burning extra fuel in the HRSG to increase steam production and thus steam turbine generator output. This is the most common method of hot day power augmentation, especially in the USA with low natural gas prices.

Another commonly utilized hot day power augmentation is compressor inlet air conditioning, which refers to cooling of airflow sucked into the compressor by various means. The most common method is *evaporative cooling* (colloquially referred to as *evap cooling*), which consists of water addition in a porous media into the incoming air, which is cooled by the mechanism of evaporation of sprayed water with latent heat of evaporation supplied by incoming air. Evap coolers are especially effective in dry climates with low ambient air humidity so that the amounts of evaporation and saturation are maximized for maximum cooling effect. A reduction in air temperature leads to increased density and mass flow rate to compensate for the hot ambient temperature's detrimental effect.

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A variant of evap cooling is *inlet fogging*, where high-pressure water is sprayed into the incoming air in the form of microscopic droplets (i.e., the "fog"). Cooling takes place in two steps: (1) evaporation of water droplets to the point of saturation (i.e., 100% relative humidity) and (2) evaporation of the remaining droplets carried into the compressor with airflow during the compression process. The latter effectively acts as a continuous intercooling and reduces parasitic power consumption of the compressor.

The third, less commonly deployed, inlet air conditioning method is *inlet chilling*, where cooling of inlet airflow is accomplished via sensible heat transfer. For detailed discussion of gas turbine inlet conditioning options and quantitative information, refer to section 18.1.1 in **GTFEPG**.

### 2.3.7 Key Parameters

A gas turbine can be fully defined by a few parameters, e.g., shaft speed, airflow, "firing" temperature and cycle (compressor) pressure ratio (PR). Airflow and firing temperature set the power output of the gas turbine (or its size). Firing temperature sets its fuel or "heat" consumption whereas cycle PR sets its exhaust temperature (and its suitability to combined cycle application). Let us look at them quickly.

Electric grids in the world are either 60 Hz, e.g., in the USA, or 50 Hz, e.g., in Europe. In some countries, e.g., Saudi Arabia and Japan, both are present. When a prime mover generator, i.e., gas or steam turbine generator, is *synchronized* to the grid, it runs either at 3,600 rpm (60 Hz grid) or at 3,000 rpm (50 Hz), where rpm denotes *revolutions per minute*. In angular terms,

- 3,600 rpm is 120π *radians per second* (rad/s)
- 3,000 rpm is 100π rad/s

Since connecting a heavy-duty industrial gas turbine rated at, say, 300 MWe and has a speed other than 3,000 or 3,600 rpm to a generator running at those speeds would require a very large, expensive, and parasitic power consuming gearbox, large *frame* machines are designed to run at 3,000 or 3,600 rpm. There are, however, some notable exceptions. In particular:

1. Smaller aero-derivative gas turbines with self-contained gas generator and power turbine, e.g., some of General Electric's (GE's) LM-2500 units, which run at 6,100 rpm.
2. Smaller industrial gas turbines such as GE's Frame 6, which runs at 5,100 rpm.
3. Alstom (now owned by GE) GT11N2 at 3,600 rpm for both 50 *and* 60 Hz versions (this is a not-so-small 115 MWe gas turbine).

Other than size, whether a gas turbine is a 50 Hz (3,000 rpm) or 60 Hz (3,600 rpm) unit is immaterial to the discussion at hand – unless, of course, the discussion is on compressor or turbine aerothermodynamics. One difference, which can be seen in combined cycle applications, is due to the larger steam flow generated in the HRSG of a larger 50 Hz gas turbine (roughly the same exhaust temperature but ~40% higher

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mass flow rate). Since steam turbine efficiency is a function of volumetric flow rate of steam, combined cycles with 50 Hz gas turbines are slightly more efficient than their 60 Hz counterparts (everything else being the same, of course). By the same token, at given speed, multi-gas turbine units are slightly more efficient than single-gas turbine units.

A significant source of ambiguity is associated with the definition of highest cycle temperature. In many references, terms like *firing temperature* and *turbine inlet temperature* (TIT) are used interchangeably and erroneously. This subject is covered in depth in Section 3.1.

For practical purposes, the cycle maximum temperature is the temperature of hot combustion gas at the inlet of the turbine stage 1 rotor buckets (or blades) before it starts producing useful turbine work. This temperature is commonly known as the *firing temperature* (TFIRE) and it is several hundred degrees Fahrenheit *lower* than the hot gas temperature at the combustor exit (at the inlet of turbine stage 1 stator nozzle vanes, i.e., the "true" TIT in essence). The reason for this reduction is dilution with cooling flow used for cooling the nozzle vanes and wheel spaces. Thus, another term for it is *rotor inlet temperature* (RIT) where the rotor in question is the stage 1 rotor. RIT is essentially what is defined in the ANSI Standard B 133.1 (1978),[3] which is referred to as an ISO-rated *cycle* temperature in §3.15 of API Standard 616.[4]

Firing temperature (or RIT) is about 100°C *higher* than the fictitious temperature as defined in ISO-2314 and adopted by European OEMs.[5] Based on the latter definition, all compressor airflow, including *hot gas path* cooling flows are assumed to enter reaction with the combustor fuel. Note that ISO-2314 defines *and* outlines a calculation methodology with requisite equations whereas API 616 and ANSI B 133.1 do *not*.

The thermodynamic cycle of the gas turbine (i.e., the Brayton cycle) is fully defined by TIT and cycle PR. The underlying theory is covered in great depth in **GTFEPG** and in Section 3.1. Heavy-duty industrial gas turbines are classified by their TIT, e.g., E Class (TIT = 1,300°C), F Class (TIT = 1,400°C). Fundamental thermodynamics dictate the optimal cycle PR for given a TIT as the one that maximizes specific power output of the gas turbine. Refer to Section 3.1 for details.

As mentioned earlier, the steam turbine is not amenable to a similar classification. Steam turbines are defined by their steam cycle parameters. In particular,

- Main or high pressure (HP) steam *admission* pressure and temperature (at the main stop/control valve inlet)
- Reheat steam temperature
- Intermediate (IP) steam admission pressure (at the intercept/control valve inlet)

[3] ANSI B113.1: Gas Turbine Terminology (1978), American Society of Mechanical Engineers, New York, USA.
[4] API Standard 616: Gas Turbines for Refinery Services, 5th ed. (2011), American Petroleum Institute, Washington, DC, USA.
[5] ISO-2314:2009: Gas Turbines – Acceptance Tests (2013), International Organization for Standardization, Geneva, Switzerland.

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- Low (LP) steam admission pressure (at the LP admission valve inlet)
- Condenser vacuum (backpressure)

Modern *three-pressure reheat* (3PRH) steam cycle parameters are

- 170 bar (2,500 psi) or higher main (HP) steam pressure
- Up to 600°C (1,112°F) main and reheat steam temperature
- IP steam pressure subject to cycle optimization (usually around 25–30 bar)
- LP steam pressure subject to cycle optimization (usually around 4–5 bar)
- Condenser vacuum subject to myriad regulatory and economic factors (see Section 3.3)

HRSG design is characterized by key temperature deltas, i.e.,

- HP, IP, and LP steam evaporator pinch
- HP, IP, and LP economizer approach subcool
- HP, IP, and LP superheater approach temperature
- Condensate heater (last economizer section before the stack) terminal temperature difference

See Section 3.2 for a description of these parameters. Smaller temperature deltas result in increased steam production but at the expense of heat transfer surface area, i.e., HRSG size and cost.

## 2.4 Suggestions for Further Reading

### 2.4.1 Books by the Author

1. Gülen, S. C. 2019. *Gas Turbines for Electric Power Generation*. Cambridge: Cambridge University Press.
2. Gülen, S. C. 2019. *Gas Turbine Combined Cycle Power Plants*. Boca Raton, FL: CRC Press.

Henceforth, the first book will be referred to as **GTFEPG** and the second book will be referred to as **GTCCPP**. As mentioned in the preceding paragraphs, relevant items covered in depth in these two books will be cited frequently in the remainder of the current treatise. There are many other valuable references out there covering similar topics. An exhaustive list can be found in **GTFEPG** accompanied by a short discussion on why a particular reference book should be present in the library of someone working in this field. I will cite only three that I value highly. For the best coverage of gas turbine fundamentals, the reader is pointed to

3. Saravanamuttoo, H. I. H., Rogers, G. F. C., Cohen, H., and Straznicky, P. V. 2009. *Gas Turbine Theory*, 6th ed. Harlow, Essex, England: Pearson Education.

Older editions of this book (used, of course) can be found on-line for a low price and will serve the purpose equally well. This author is not a fan of "handbooks," which he calls "travel guides," as practically useful references. There are, however, two exceptions, one in English and one in German, which are well worth having a copy of. (They are not too expensive either.)

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4. Jansohn, P. (ed.) 2013. *Modern Gas Turbine Systems: High Efficiency, Low Emission, Fuel Flexible Power Generation* (Woodhead Publishing Series in Energy) 1st ed. Cambridge, UK: Woodhead Publishing.
5. Lechner, C. and Seume, J. 2010. *Stationäre Gasturbinen, 2. neue bearbeitete Auflage*. Heidelberg: Springer.

### 2.4.2 Book Chapters by the Author

6. Gülen, S. C. 2017. Advanced Fossil Fuel Power Systems. In D. Y. Goswami and F. Kreith (eds.), *Energy Conversion*, 2nd ed. Boca Raton, FL: CRC Press.
7. Smith, R. W. and Gülen, S. C. 2012. Natural Gas Power. In Robert A. Meyers (ed.), *Encyclopedia of Sustainability Science and Technology*. New York: Springer.

The first of these two, with more than 160 pages, is a small book in and of itself. It is a concise guide to technologies pertinent to sustainable deployment of fossil fuel resources. Suffice it to say that the author himself uses it as a reference for a quick understanding of, say, “oxy-combustion” and “steam methane reforming.” It also includes an extensive bibliography.

The second one is effectively a condensed treatise on gas turbines – also with an extensive bibliography. It was written around 2010 and updated in 2016.

### 2.4.3 Papers and Articles by the Author

Over the course of his career, the author has written numerous papers and articles published in prestigious archival journals such as *ASME Journal of Engineering for Gas Turbines and Power (J. Eng. Gas Turbines Power)* and trade publications such as *Gas Turbine World*. Some papers were presented at landmark industry gatherings such as *ASME IGTI Turbo Expo* and *POWERGEN International* and frequently won “Best Paper” awards. Readers interested in details of certain subjects covered in the current treatise may want to check out some of them for more in-depth information.

8. Gülen, S. C. and Lu, H. 2019. Coal-fired direct injection carbon engine (DICE) and gas turbine compound-reheat combined cycle (GT CRCC), International Pittsburgh Coal Conference, September 3–6, 2019, Pittsburgh, PA.
9. Gülen, S. C., Rogers, J., and Sprengel, M. 2019. DICE-Gas turbine compound reheat combined cycle, Presentation at: 9th International Conference on Clean Coal Technologies, June 3–7, 2019, Houston, TX (also published in the journal *Fuel*).
10. Gülen, S. C. 2019. Combined cycle technology trends, *Turbomachinery International*, 60 (May/June), 26–28.
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12. Gülen, S. C. 2019. Disappearing thermo-economic sanity in gas turbine combined cycle ratings – a critique, ASME Paper GT2019–90883, ASME Turbo Expo 2019, June 17–21, Phoenix, AZ.
13. Gülen, S. C. 2018. Combined cycle performance ratings undergo thermodynamic reality check, *Gas Turbine World*, November/December, 23–32.
14. Gülen, S. C. 2018. HRSG duct firing revisited, ASME Paper GT2018–75768, ASME Turbo Expo 2018, June 11–15, Lillestrøm (Oslo), Norway.

15. Elliott, W. R. and Gülen, S. C. 2017. Cost-effective post-combustion carbon capture from coal-fired power plants, International Pittsburgh Coal Conference, September 5–8, Pittsburgh, PA.
16. Gülen, S. C., Adams, S. S., Haley, R. M., and Carlton, C. 2017. Compressed gas energy storage, *Power Engineering*, 121(8), 26–38.
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