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COMBINED CYCLE OFF-DESIGN PERFORMANCE ESTIMATION: A SECOND-LAW PERSPECTIVE

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ABSTRACT

A combined cycle power plant (or any power plant, for that matter) does very rarely –if ever- run at the exact *design point* ambient and loading conditions. Depending on the demand for electricity, market conditions and other considerations of interest to the owner of the plant and the existing ambient conditions, a CC plant will run under boundary conditions that are significantly different from those for which individual components are designed. Accurate calculation of the "off-design" performance of the overall combined cycle system and its key subsystems requires highly detailed and complicated computer models. Such models are crucial to high-fidelity simulation of myriad off-design performance scenarios for control system development to ensure safe and reliable operability in the field.

A viable option in lieu of sophisticated system simulation is making use of the normalized curves that are generated from rigorous model runs and applying the factors read from such curves to a known design performance to calculate the "off-design" performance. This is the common method adopted in the fulfillment of commercial transactions. These curves, however, are highly system-specific and their broad applicability to a wide variety of configurations is limited. Utilizing the key principles of the second law of thermodynamics, this paper describes a simple, physics-based calculation method to estimate the off-design performance of a combined cycle power plant. The method is shown to be quite robust within a wide range of operating regimes for a generic combined cycle system. As such, a second law based approach to off-design performance estimation is a highly viable tool for plant engineers and operators in cases where calculation speed with a small sacrifice in fidelity is of prime importance.

NOMENCLATURE[1]

c_p	= Constant-pressure specific heat, Btu/lb-R (4.1868 kJ/kg-K)
$\dot{E}$	= Total exergy, kW or Btu/s
e	= Specific exergy (availability), Btu/lb (2. 326 kJ/kg)
h	= Specific enthalpy, Btu/lb (kJ/kg)
$\dot{I}$	= Irreversibility, kW or Btu/s
$\dot{m}$	= Mass flow rate, pps (0.4536 kg/s)
N	= Number of GTs (and HRSGs)
P	= Pressure, psia (0.06895 bara)
$\dot{Q}$	= Heat transfer rate, kW (0.947817 Btu/s)
s	= Specific entropy, Btu/lb-R (kJ/kg-K)
T	= Temperature, ºF (ºC = [ºF – 32]/1.8)
$\overline{T}$	= Mean-effective (average) temperature, °C or °K (°F or °R)
$\dot{W}$	= Power (work), kW or Btu/s
w	= Specific power (work), Btu/lb (kJ/kg)

Greek Symbols

α	= Plant auxiliary load as a fraction plant gross output
ε	= Exergetic conversion effectiveness
η	= Efficiency
π	= ST section pressure ratio
ω	= Feed and condensate pump power (fraction of ST output)

Subscripts

i, j, k	= Generic indices used in summation formulas
1,2,3,4	= State points in Figure 3
amb	= Ambient
aux	= Auxiliary
cond	= Condenser
cw	= Cooling water
exh	= GT exhaust
H	= Hot temperature reservoir (heat addition)
L	= Low temperature reservoir (heat rejection)
o	= Reference (usually ambient)
stck	= HRSG stack
stm	= Steam
l, lost	= "Lost" Work

Acronyms

3PRH	= Three-Pressure, Reheat (Rankine Steam Cycle)
ACC	= Air-Cooled (Frame-A) Condenser
BFP	= Boiler Feed Pump (used as *Subscript*)
CAC	= Cooling Air Cooling
CL-CT	= Closed-Loop, Cooling Tower
CV	= Control Volume
EOS	= Equation of State
FPT	= Flow-Pressure-Temperature
CC	= Combined Cycle (also used as *Subscript*)
GT	= Gas Turbine (also used as *Subscript*)
HC	= Heat Consumption (GT)
HDI	= Heavy-Duty Industrial (GT)
HRSG	= Heat Recovery Steam Generator (Boiler)
HUF	= HRSG Heat Utilization Factor

[1] The primary unit system in this paper is the US customary system. Conversions to SI are provided in the text. Common conversion factors are provided in the nomenclature where they first appear.

IGV = Inlet Guide Vane
LHV = Lower Heating Value
MET = Mean-Effective Temperature, $\overline{T}$
OEM = Original Equipment Manufacturer
OT-OL = Once-Through, Open-Loop (Water-Cooled Condenser)
PR = GT Compressor Pressure Ratio
RBC = Rankine Bottoming Cycle (also used as *Subscript*)
SOA = State of the Art
ST = Steam Turbine (also used as *Subscript*)
TEL = Total Exhaust Loss (Btu/lb)
TET = Turbine Exhaust Temperature
TIT = Turbine Inlet Temperature; same as T_3
VAN = ST Exhaust Annulus Velocity (fps)
VSV = Variable Stator Vane (Compressor)

INTRODUCTION

Gas turbine combined cycle power plant is the most efficient system for electric power generation via fossil fuel combustion. At the time of writing this paper, modern CC plants based on advanced natural gas fired gas turbines and three-pressure, reheat steam bottoming cycles are quoted at net efficiencies of 59% to 60% [1]. This is far superior to the state-of-the-art (SOA) fossil-fired (mostly coal) boiler plant, whose thermal efficiencies (measured or *realistically* predicted) hover around the 40-45% (LHV) range [2]. Other advantages of the gas turbine CC power plant are lower specific capital cost, lower emissions per produced kWh and, thus, lower regulatory costs and better incentives, e.g., those associated with greenhouse gas emission caps and cap-and-trade system requirements (e.g., for manufacturers and utilities to purchase pollution permits).

A CC power plant is designed for operation across a wide range of site ambient and loading conditions. Unless one considers the particular requirements of a specific job, the design point is fixed at standard reference conditions. Usually, that standard is ISO ambient conditions with the plant components running at 100% load. A common term for the 100% load condition is "base" or sometimes "full" load. There is a certain ambiguity about the definition of the base or full load. For the best definition of the base load, at least on a conceptual basis, consider that the GT is an air-breathing machine operating at a constant rotational speed (i.e. 3000 or 3600 rpm). At any given ambient, the GT is said to be running at "full" load when the following conditions are met:

1. Compressor inlet guide vanes (IGVs) are at their nominal open position, i.e. the unit is sucking the volumetric flow of air for which it is designed.

2. The combustor is fired exactly to the temperature dictated by the control curve as a function of the measured compressor PR and turbine exhaust temperature.

Furthermore, the GT is said to be running at "base" load when it is operating at a specified, design ambient (in most cases ISO) and at "full" load. At any given time, a GT is said to be running at "part load" when via controlling IGVs, VSVs (if present) and/or firing temperature, the GT is operating at less than "full" load capacity. Note that the part load can be defined in reference to the "base" load (at the design ambient) or the "full" load at the given ambient. Consequently, one should clearly understand the reference load, which a particular part load is referring to. In this paper, the part load is always used in reference to the full load at the given ambient temperature.

In a CC plant, the ST operates in a "sliding pressure" or "valves wide open" mode (until the steam control valve "floor" pressures are reached; thereafter HP and IP inlet pressures at valve inlet are constant). As such, the ST generates the power that it can *naturally* generate with the following inputs:

1. Steam generated in the HRSG that is commensurate with the exhaust energy provided by the GT (assuming an *unfired* system).

2. Condenser pressure that is commensurate with the temperature of circulating cooling water at the particular ambient.

In other words, the ST is a "slave" to the GT, which determines the loading of the CC plant. Consequently, two major (primary) factors determine the "off-design" performance: (1) Ambient conditions, particularly, the temperature and the humidity of the air, and (2) GT loading (i.e., full or part load). Secondary factors such as generator power factor, fuel composition (i.e., LHV), and component degradation also impact the "off-design" performance and can (and should) be accounted for using appropriate corrections.

Engineers resort to simulation via detailed, high fidelity and therefore quite complicated computer models for plant off-design performance calculations due to two key reasons. First, calculation of the performance of "fixed" hardware operating at off-design is much more complicated than the simple enthalpy and mass balances that are sufficient for the design calculations. Second, simulation of the control loops that govern the performance of the plant components is complicated and highly iterative. Examples include but are not limited to attemperation (desuperheating) sprays, low-pressure economizer bypass and/or recirculation, pump flow-head and efficiency curves. Note that these are in addition to myriad control loops that govern the operation of the GT. Thus, for detailed engineering design of plant equipment and control systems, complex model-based simulations are *sine qua non*.

OFF-DESIGN CC MODEL

The aforementioned fact that sophisticated computer models are requisite for reliable CC plant off-design performance calculations does not preclude application of proper engineering judgment to the problem for developing relationships suitable to reasonable performance estimation.

The GT performance is readily amenable to a simplified treatment via correction curves. For a particular unit in the field, curves provided by the OEM are (usually) available. OEM engineers run highly accurate computer models over the entire GT operating envelope and generate these curves. For quick estimates and conceptual studies, generic normalized curves are adequate for most heavy-duty industrial (HDI) gas turbines. Examples of such curves can be found in Brooks [3]. In general, performance variation with ambient conditions (pressure, temperature and humidity) and inlet/exhaust losses for air-cooled HDI machines will not display a significant deviation between different OEMs. However, for advanced machines such as steam-cooled H-System™ [4] or sequential combustion (reheat) GTs [5], product-specific curves should be preferred. GT performance at lower loads is dependent on the particular control philosophy adopted by the OEM. Even then, for the HDI units, two basic approaches can be identified: (1) Constant TIT with inlet flow modulation via IGVs and VSVs (e.g., see Rowen and Van Housen [6]), and (2) constant TET with TIT and inlet flow modulation (e.g., see Jansen et al. [7]). A third method, i.e. TIT variation at constant airflow is possible but it is detrimental to CC performance due to significant reduction in TET and ensuing deterioration in bottoming cycle performance. For the aeroderivative GTs or the reheat HDI GTs, complex control schedules for the former and the existence of a second burner for the latter result in part load characteristics different from the standard HDI machines (especially the exhaust characteristics). A quick assessment of the qualitative differences can be gleaned from the figures in Petek [8]. However, the most reliable source is the OEM supplied curves. The discussion in the remainder of the paper is limited to non-reheat HDI GTs; however, the principles for the bottoming cycle and CC performance estimation can be applied to any GT with known exhaust characteristics (i.e., flow temperature, and composition).

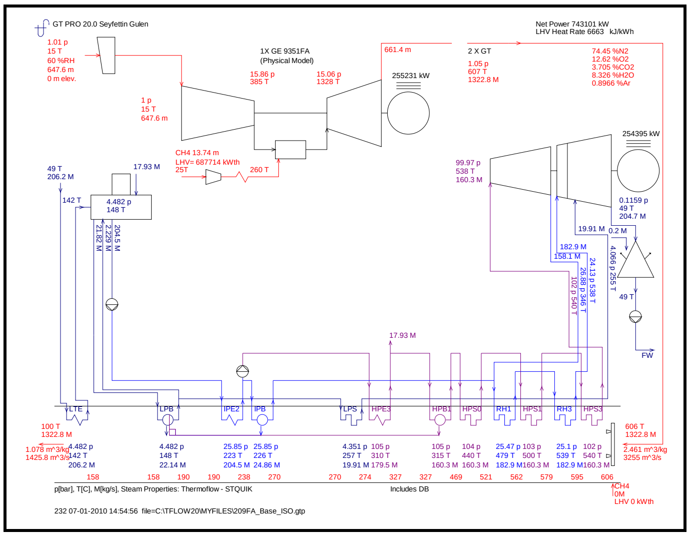

Figure 1 Typical 2×1 GT-CC heat and mass balance diagram from a heat balance simulation tool [10]. (Numbers shown are in SI units.)

A study done by Kim [9] shows the comparative advantage of each part load control approach to GT combined cycle performance. Kim's analysis was based on detailed models of the GT as well as the bottoming cycle. Due to the reasons enumerated earlier, this is not a simple task and requires significant effort in terms of hardware and control model development, model tuning (calibration), and input/output verification. In order to appreciate the magnitude of this undertaking, consider the model of a 209FA CC power plant in Figure 1 (from a commercially available heat balance software tool). The program has over 1600 inputs; going by 80/20 (widely known as *Pareto*) principle (i.e., only 20% of the inputs are used 80% of the time), the user must still tweak at least a few hundred of them for a reasonably accurate system model.

A viable option in lieu of complex heat balance simulations is making use of the normalized curves that are generated from similar exercises and applying the factors read from those curves to a known design performance to calculate the "off-design" performance. An example of such curves is given Figure 2 [11], which shows the CC performance at a particular site ambient (rating point) condition. The horizontal axis is the net plant output and the vertical axis is the heat rate, both as a fraction of their respective rating point values. This type of curve is commonly referred to as a "hockey stick" curve due to the shape of the heat rate vs. output line. As this particular example for a 2×1 CC plant shows, there are *two* distinct ways to achieve the same plant load, e.g. 50%, by either running both GTs at equal loads or by running only one GT and shutting down the other. A better plant heat rate is achieved by the latter option due to the improved heat rate characteristic of the GT running at its "full" load.

As will be explored in detail in the following paragraphs, utilizing the second law of thermodynamics and the exergy concept, accurate conceptual and even practical results can be obtained via simple calculations. However, the reader should recognize that, especially for performance testing and monitoring applications, there really is no substitute for either rigorous GT simulations or OEM-supplied curves for the particular unit. Myriad factors such as inlet bleed heat, VSV schedule and firing temperature changes, which are imposed by the GT control system to ensure that the unit is turned down in an emissions-compliant mode preclude reliable use of simple schemes. For a comprehensive discussion of these aspects of GT part load control at different ambient conditions, the reader is referred to Schmitt and Clement [12]. As such, with the exception of the simple Brayton cycle based conceptual calculations below, the current paper focuses on the bottoming cycle and overall combined cycle. GT performance is assumed to be available (or computable) from GT simulation models, correction curves, or plant's control system and/or historian data. The key information facilitating the exergy-based bottoming cycle calculations comprises only GT exhaust data (primarily, flow and temperature).

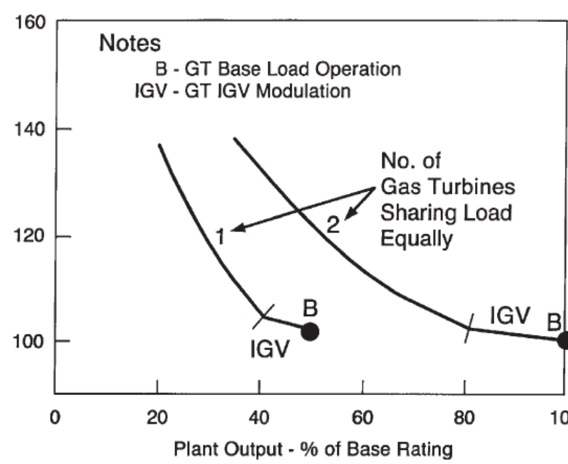

Figure 2 Part-load curve for 2x1 209E CC power plant (vertical axis is heat rate as percent of base rating).

In passing, it should be mentioned that analytical methods to calculate (estimate) off-design performance of the GT-CC power plants are available; e.g., see the treatment by Kehlhofer [13] to calculate the part load operation of the GT-CC plants. However, these methods are relatively cumbersome even in their simplified form and require a substantial investment in computational and other resources. In any event, while they are immensely useful for teaching of governing principles, in terms of practical application, they have been made superfluous by the advance of computers and sophisticated software exploiting their power, e.g. [10].

SECOND LAW ANALYSIS

Simple (Fundamental) Approach

The conceptual development in this section is based on the GT Brayton cycle diagram in Figure 3. For a detailed discussion of the concepts that are underlying the simple formulae below the reader is referred to Hofer and Gülen [14], Elmasri [15] and Smith and Gülen

[16]. In short, the maximum work that can be generated by the bottoming cycle utilizing the (waste) exhaust heat of a heat engine such as a GT is exactly equal to the exergy of the exhaust stream (i.e., state 4 in Figure 3). Exergy (also referred to as *availability* in US textbooks) is a fluid property, and can be calculated using an equation of state (EOS) with known pressure, temperature and composition [14]. In mathematical terms, for the GT Brayton cycle, which is a *cold* air-standard cycle (i.e., pure working fluid, air, modeled as a *calorically perfect* gas), this is equal to (per unit mass of air)

$$w_{BC,Max.} = c_p \cdot (T_4 - T_1) - T_1 \cdot c_p \cdot \ln(T_4/T_1) \qquad [1]$$

Graphically, the specific work given by Eq. [1] is equal to the triangular area (1-4-4c-1) in Figure 3. For a general fluid, Eq. [1] can be written as (e.g., see Chapter 8 in Van Wylen and Sonntag [17])

$$w_{BC,Max.} = e_4 = (h_4 - h_1) - T_1 \cdot (s_4 - s_1) \qquad [2]$$

Enthalpy and entropy terms in Eq. [2] can be calculated explicitly using an appropriate EOS such as JANAF [18].

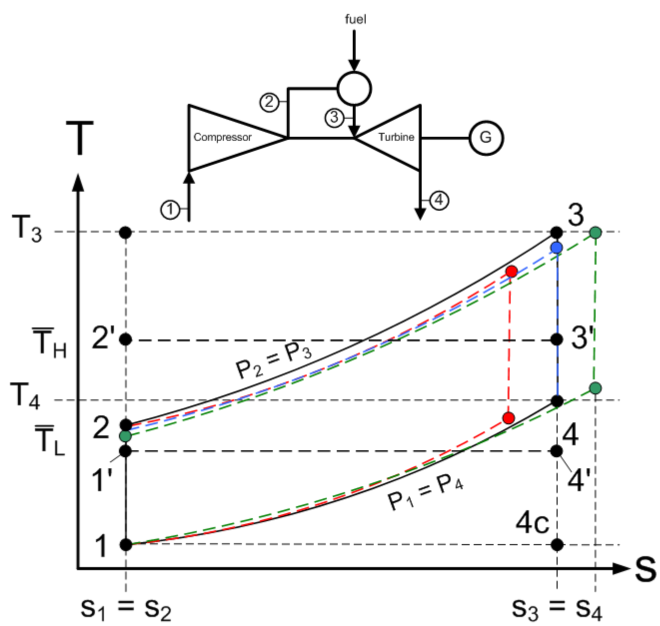

Figure 3 Gas turbine Brayton cycle. Variations in the cycle for three part load controls described in text are also shown (qualitatively): Constant TIT (green dashed), constant TET (blue) and constant airflow (red dashed).

The most straightforward and practical manifestation of the second law of thermodynamics dictates that the *maximum theoretical* work production of a heat engine operating in a thermodynamic cycle is that of a *Carnot* cycle operating between two heat reservoirs with the same (and <u>constant</u>) heat addition and rejection temperatures as the heat engine under consideration. For the bottoming cycle (1-4-4c-1) in Figure 3, those temperatures are $\overline{T}_L$ (the *mean-effective* heat rejection temperature of the Brayton cycle) and T_1. For the combined cycle (1-2-3-4c-1), they are $\overline{T}_H$ (the *mean-effective* heat addition temperature of the Brayton cycle) and T_1. Thus, the maximum combined cycle work is given by

$$w_{CC,Max.} = c_p \cdot (T_3 - T_2) \cdot \left(1 - \frac{T_1}{\overline{T}_H}\right) \qquad [3]$$

These three equations are sufficient to investigate the GT-CC part load operation strategies. (The second term in the parentheses on the RHS of Eq. [3] is the maximum combined cycle efficiency.) Mean-effective heat addition and rejection temperatures in the equations above are logarithmic means and can be easily calculated (see Refs. [14-16]). Their application to off-design CC performance studies is demonstrated below. Consider three well-known GT part-load operation philosophies:

1. **Constant inlet airflow**; TIT modulated to set the load
2. **Constant TIT**; inlet airflow is modulated (via IGVs and VSVs) to set the load (minimum airflow is assumed 70%)
3. **Constant TET**; inlet airflow (via IGVs and VSVs) and TIT are modulated to set the load (minimum airflow is assumed 70%)

The Brayton cycle corresponding to each GT part load control philosophy is shown graphically in Figure 3. Visual examination of the cycle diagrams clearly shows that the constant airflow method is ranked last in terms of CC efficiency and GT specific power (i.e., lowest $\overline{T}_H$, smallest Brayton cycle area and smallest exhaust exergy). The second best is the constant TET method whereas the constant TIT method offers the best CC efficiency and GT specific power (i.e., highest $\overline{T}_H$, largest Brayton cycle area and largest exhaust exergy). This visual (qualitative) ranking is confirmed below quantitatively as well. For the calculations, the following assumptions are made: T_1 of 59ºF, T_3 of 2500ºF, cycle PR of 16 and mass flow of 1000 lb/s. Cycle pressure ratio is scaled using the choked nozzle assumption, i.e.

$$\dot{m} \cdot \frac{\sqrt{T_3}}{P_3} = \text{const.} \qquad [4]$$

The results obtained from Eqs. [1-4] with the assumptions listed above are shown in Figure 4.

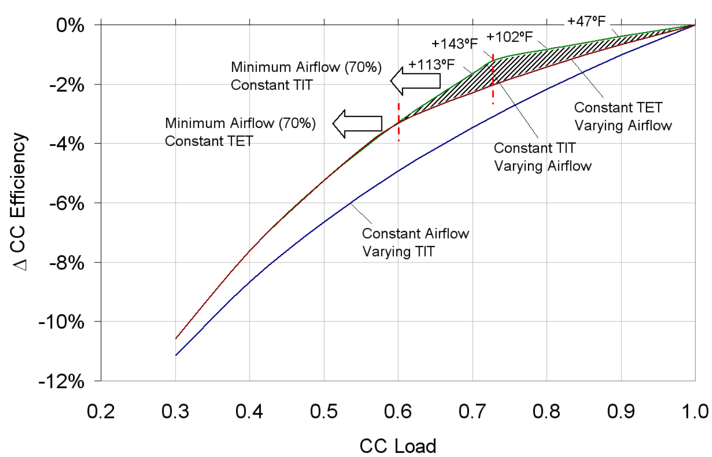

Figure 4 Relative GT-CC part load performance for different control philosophies described in the text. The numbers indicate the delta between TET values of constant TIT control and the base TET.

Qualitatively, the behavior displayed by three different part load control philosophies are identical to those arrived at using more complex models (e.g., see Figure 10 in Ref. [9]). Main takeaways are: (1) Constant airflow control is detrimental to CC performance; (2) constant TIT is the *best* until minimum airflow (i.e. maximum IGV/VSV closure) at the base TET is reached; and (3) thereafter, constant TET and constant TIT controls are equivalent. The extent of the applicability of constant TIT control (i.e., the shaded area between constant TIT and TET curves in Figure 4) is a function of the maximum allowable TET. The lower that limiting value (sometimes referred to the as the *exhaust isotherm*) the smaller is the potential benefit of constant TIT control over constant TET control.

The attributive "best", used above for constant TIT control, is with respect to CC heat rate (or efficiency) at the same CC load only. No claims are made regarding emissions (NOx, CO, etc.), which might render the constant TIT control infeasible, at part or all of the load range where it shows a heat rate advantage. Emissions characteristics are highly dependent on the particular GT combustion technology and cannot be investigated using simple models. Furthermore, limitations imposed by GT emissions on the operating envelope are site-dependent via existing laws and regulations and not subject to a purely technical study.

Exergetic Efficiency Approach (Method I)

At this point, one may legitimately wonder whether this analysis is too simplistic to be of any practical value. Here is another way to approach the problem. The CC efficiency of an (unfired) N×1 CC system (i.e., N GTs and HRSGs and one ST) can be written as

$$\eta_{CC} = \frac{N \cdot \dot{W}_{GT} + \dot{W}_{ST} - \dot{W}_{aux}}{N \cdot HC} \qquad [5]$$

Assume that the GT performance (i.e., power output and efficiency) and exhaust conditions are available via a detailed model or correction curves. The maximum theoretical work that can be obtained from the exhaust of the GT is exactly equal to the exergy of its exhaust stream, which is given by Eq. [1] or Eq. [2]. Since the real bottoming cycle of a GT-CC power plant cannot be a Carnot cycle, one can write

$$\dot{W}_{RBC} = \varepsilon_{RBC} \cdot \dot{E}_{exh} = \varepsilon_{RBC} \cdot \left(N \cdot \dot{m}_{exh} \cdot e_{exh}\right) \qquad [6]$$

where ε_{RBC} is the *Rankine* (steam) bottoming cycle (RBC) *exergetic* efficiency. Note that the *net* RBC power output defined by Eq. [6] is equal to the ST generator output minus power consumption of HRSG feed and condensate pumps. Thus, ST generator output is given as

$$\dot{W}_{ST} = \dot{W}_{RBC} + \dot{W}_{BFP} = \frac{\dot{W}_{RBC}}{1-\omega_{BFP}} \qquad [7]$$

where ω_{BFP} is RBC pump power consumption as a fraction of the ST generator output. A typical *conservative* value for ω_{BFP} is 1.5%; it can be as high as 1.9% [19]. The auxiliary power consumption of the GT-CC power plant, $\dot{W}_{aux}$, in Eq. [5] can be estimated using the guidelines outlined in Ref. [19] for different types of heat rejection systems.

For the SOA 3PRH bottoming cycles of advanced F-Class and H-Class GTs, published data by the OEMs show that, to a very good approximation, ε_{RBC} is about 72% [19]. For more reliable estimates, the following formula can be used with very good accuracy to estimate the *design point* value of ε_{RBC} [20]:

$$\varepsilon_{RBC} = 0.2441 + 0.0746 \cdot \left(\frac{T_{exh}}{100}\right) - 0.00279 \cdot \left(\frac{T_{exh}}{100}\right)^2 \qquad [8]$$

Equation [8] (with T_{exh} in ºF) represents the SOA in Rankine steam bottoming cycle technology with 3PRH steam cycle and advanced ST utilizing the exhaust gas energy of F, G and H-Class HDI gas turbines with exhaust temperatures exceeding 1100ºF. Once a design point value of ε_{RBC} is obtained from Eq. [8], it can be used for any part load point. (It will be shown later in the paper that this is indeed a very good approximation.) CC off-design performance calculations are carried out using Eqs. [5-8] along with a more detailed GT performance calculation approach. The results are shown in Figure 5.

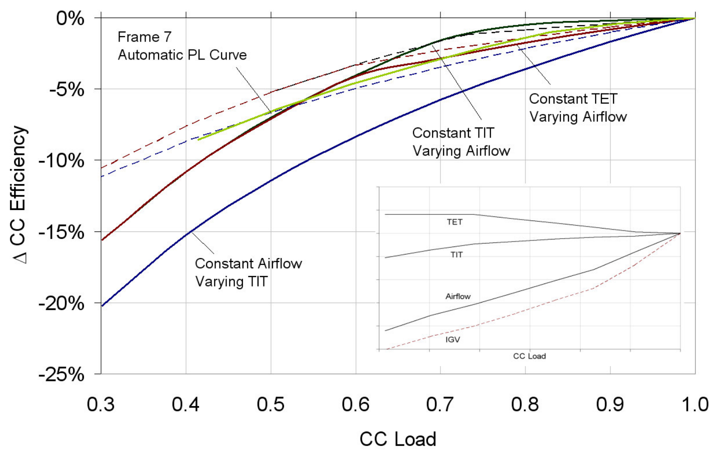

Figure 5 Relative GT-CC part load performance for different control philosophies. (The dashed lines correspond to the ideal model calculations in Figure 4.)

The GT model (represented by solid lines in Figure 5) is taken from Ref. [16], which is essentially an adoption of the model in Chapter 2 of Saravanamuttoo et al. [22]. Once again, the cycle pressure ratio is scaled using the choked nozzle assumption. Auxiliary power estimate is from Ref. [19] for a CL-CT heat rejection system. Furthermore, one set of calculations is carried using a rigorous model (commonly referred to as a *cycle deck*) of a Frame 7 GT with the built-in automatic GT part load control (solid green line). The inset in Figure 5 shows the GT characteristics for the particular load control scheme. Quantitatively, rigorous GT-CC calculations differ from the ideal calculations in Figure 4, although not significantly for CC loads above 0.6. Qualitatively, however, they are essentially identical and enable the design engineer to draw the same conceptual conclusions. Furthermore, they are also in quite close *quantitative* agreement with the much more detailed calculations of Ref. [9] by using only four simple equations, which can be programmed into an Excel spreadsheet in a few minutes.

Control Volume Approach (Method II)

Now that the key concepts of second law analysis of GT-CC off-design performance are outlined, a closer look at the RBC is warranted. This will lead one to additional governing principles, which are equally simple to understand and evaluate. These principles provide powerful insights to a variety of engineering tasks such as performance monitoring, performance analysis and performance prediction at GT-CC plant operating conditions appreciably different from those at the design point.

For the control volume (CV) encompassing the RBC of a GT-CC power plant, the steady-state exergy rate balance can be written as follows [20]:

$$\dot{m}_{exh} \cdot (e_{exh} - e_{stck}) + \sum_j \left(1 - \frac{T_o}{\overline{T}_j}\right) \cdot \dot{Q}_j = \dot{W}_{RBC} + \dot{W}_{lost} \quad [9]$$

Equation [9] provides a concise description of the RBC from a second law point of view. The power generation potential represented by the LHS of Eq. [9] is due to the net exergy transfer from the GT exhaust gas and other heat transfer (if any) into the RBC. The term $\dot{Q}_j$ is the time rate of heat transfer at a given location on the CV boundary where the instantaneous (mean-effective) temperature is $\overline{T}_j$. In unfired systems, there are two major RBC heat inputs (other than the GT exhaust gas): (1) GT hot gas path cooling steam in G-Class units and H-System™ [4,23] and (2) cooling-air cooling (CAC) heat transfer in H-System™ and sequential combustion (reheat) turbines [24]. For advanced F-Class air-cooled GTs such as GE's Frame 7 and 9 machines, the second term on the LHS of Eq. [9] is zero. To accommodate duct firing, one can recalculate $\dot{m}_{exh}$ and T_{exh} to reflect the duct burner fuel flow and exit temperature.

The first term on the RHS of Eq. [9] is the net power output of the RBC, $\dot{W}_{RBC}$, which is defined by Eq. [6]. The second term on the RHS of Eq. [9] represents the "lost" opportunity of generating useful shaft/generator work due to (i) exergy transfer out of the RBC and (ii) exergy destruction in the RBC. They are lumped into a single term that will be referred to as the "lost work", which is commonly used in the academic literature:

$$\dot{W}_{lost} = \dot{I} + \sum_j \left(1 - \frac{T_o}{\overline{T}_j}\right) \cdot \dot{Q}_j \quad [10]$$

The second term on the RHS of Eq. [10] represents the exergy transfer out of the RBC via heat transfer. The most significant heat transfer rate for the RBC is the heat rejection from the condensing steam to the environment via the cooling water in the ST condenser. The other two heat transfer rates that are readily identifiable albeit much smaller than cycle heat rejection in magnitude are the HRSG heat loss and the heat transfer from the HRSG feed water to the GT fuel gas in the performance heater. The remaining heat transfer rates are associated with miscellaneous pipe, valve and component heat losses that are too small and numerous to merit individual treatment.

The first term on the RHS of Eq. [10] is the total *irreversibility* (i.e., the rate of *exergy destruction* or the rate of *entropy generation*) of the RBC, which can be expressed as the sum of individual component irreversibilities:

$$\dot{I} = \dot{I}_{HRSG} + \dot{I}_{ST} + \dot{I}_{COND} + \dot{I}_{MISC} + \dot{I}_{MECL} + \dot{I}_{ELEC} \quad [11]$$

The first three terms on the RHS of Eq. [11] are readily amenable to calculation from the first principles as shown in Ref. [20]. The fourth term is a lumped representation of many small thermodynamic loss mechanisms. The fifth term is the mechanical losses due to friction between turbine and pump shafts and their bearings and –if any- gear losses. The last term is the electrical loss in the ST generator and pump motors.

Identifying the mechanisms contributing to the lost work is another way to calculate the net RBC work, which will provide additional insight about what can or cannot be done to improve it practically and/or feasibly. Combining Eqs. [10-11] and rearranging the terms, one can write the following formula describing the breakdown of RBC lost work contributors:

$$\dot{W}_{lost} = \dot{W}_{l,HRSG} + \dot{W}_{l,ST} + \dot{W}_{l,COND} + \dot{W}_{l,MISC} \quad [12]$$

In Eq. [12], the first three terms on the RHS of the equality designate the RBC lost work associated with the HRSG, ST, and condenser, respectively. (The losses represented by the last two terms on the RHS of Eq. [11] are usually lumped into the lost work term for the ST.) The fourth term is a lumped representation of miscellaneous small exergy destruction and transfer mechanisms. For SOA RBC systems it is adequately accounted for by assuming 1.5% of the total GT exhaust exergy. If there is sufficient flow-pressure-temperature (FPT) data to evaluate the stream exergies for all material streams crossing the system CV, the three major lost work items (HRSG, ST and condenser) can be "exactly" calculated, i.e.:

$$\dot{W}_{lost} = \sum_i \dot{m}_i \cdot e_i(P_i, T_i) - \sum_k \dot{m}_k \cdot e_k(P_k, T_k) + \dot{E}_Q - \dot{E}_W \quad [13]$$

The four terms on the RHS of Eq. [13], from left to right, are:

1. Total material stream exergy (availability) entering the system CV using stream pressure and temperature in Eq. [2] via a suitable EOS such as ASME steam tables [25];
2. Total material stream exergy exiting the system CV;
3. Net exergy added to the system via heat transfer crossing the CV boundary;
4. Net exergy taken from the system via shaft work crossing the CV boundary.

Equation [13] is the general exergy balance formulation for control volumes to be found in the literature (e.g., see Eq. (3.10a) in Bejan et al. [26]). Please refer to the Appendix for its application to the systems herein and its correspondence to Eq. [10]. Lost work calculation using Eq. [13] is the preferred method for on-line performance monitoring systems with a well-designed and instrumented distributed control system and plant data historian as well as analysis of data obtained from performance tests conducted per applicable ASME standards. However, as will be discussed below, major losses can also be accurately captured using only a few key measurements in a few simple relationships.

Approximate Control Volume Approach (Method III)

In order to fully describe the ST, HRSG and condenser CVs using a complete exergy balance via Eq. [13], for the steam bottoming cycle of a modern 3PRH CC system with advanced GTs, one needs about 25 independent (measured) parameters: FPT data for key material streams (i.e., main steam, cold and hot reheat steam, LP admission steam, condensate, GT fuel gas heater hot water extraction, GT exhaust and HRSG stack gas), condenser pressure, ST generator and boiler feed and condensate pump kilowatts. These three RBC subsystems contribute about 75% to 85% of the total RBC lost work. In the absence of sufficient (and reliable) FPT data, the total lost work contribution of these subsystems can be estimated in a reasonably accurate manner using *six* key parameters, i.e., GT exhaust gas flow and temperature, HRSG stack temperature, condenser pressure, main steam temperature and pressure (i.e., about 25% of the total parameters). In yet another confirmation of the Pareto principle, about 80% of the total RBC losses are explained or predicted by only about 20% of the key cycle parameters. Thus, the CC design or plant engineer is able to focus on the *vital few*, which may be quite difficult when one has to make sense of plant data and/or models with a significant overload of information.

For rigorous development of the principal equations for the approximate CV method and some of the terms therein (e.g. mean effective exhaust gas and HRSG steam temperatures), as well as typical distribution of the individual lost work contributions, the reader is referred to Gülen and Smith [20]. For convenience and immediate application of the principles discussed herein, Eqs. [A-1]-[A-3] in the Appendix provide a concise summary.

Conceptual Analysis Application

For conceptual analysis of design and off-design performance, it is adequate to use Eqs. [5-8]. For the off-design calculations, the key piece of information is the variation in ε_{RBC} obtained from Eq. [8] with site ambient and loading conditions. Examining data generated using rigorous GT models with built-in part load control and a CC heat balance model similar to that shown in Figure 1 has shown that this can be indeed accomplished by using a few rules of thumb. The ambient lapse and part load CC bottoming cycle performance data generated by detailed model runs is shown in Figure 6. Equation [6] can be rewritten as follows

$$\dot{W}_{RBC} = (\varepsilon_{BC} + \Delta\varepsilon_{amb} + \Delta\varepsilon_{PL}) \cdot (N \cdot \dot{m}_{exh} \cdot e_{exh}) \qquad [14]$$

where $\Delta\varepsilon_{amb}$ and $\Delta\varepsilon_{PL}$ represent change in ε_{RBC} with site ambient and loading, respectively. Thus, for preliminary studies one can use the trends displayed in Figure 6 as a rough guideline to estimate the respective deltas and apply them to ISO base design point ε_{RBC} to estimate the off-design RBC performance.

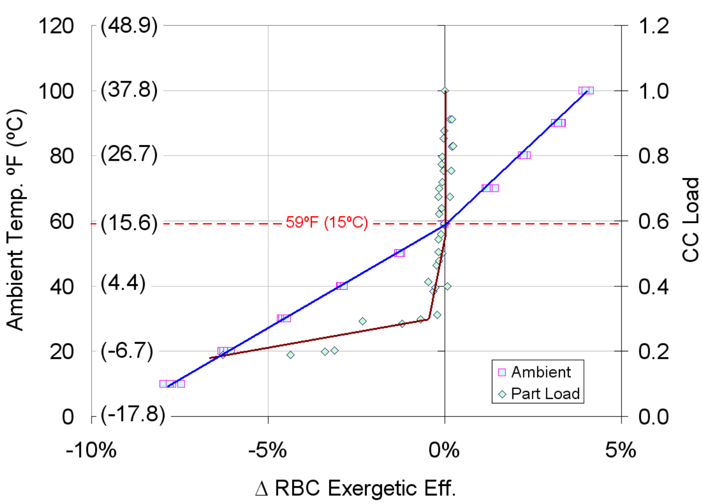

Figure 6 Ambient and part load variation of RBC exergetic efficiency. For a 1×1 CC system with 50-Hz and 60-Hz frame GTs and OT-OL heat rejection system. Design point is ISO base load.

The driving mechanisms for the RBC exergetic efficiency variation across the ambient temperature and CC load range as displayed in Figure 6 are explained using the plots in Figures 7-8. For the ambient performance, the key cycle parameter is the condenser pressure, P_{cond}.

At hot ambient operation, exergetic performance increases due to the decrease in work potential via higher *dead state* exergy and relatively constant performance of plant hardware. At cold ambient operation, ST lost work (via much higher VAN and TEL) and increase in work potential via lower dead state exergy (because the already designed plant hardware cannot take advantage of it) take over.

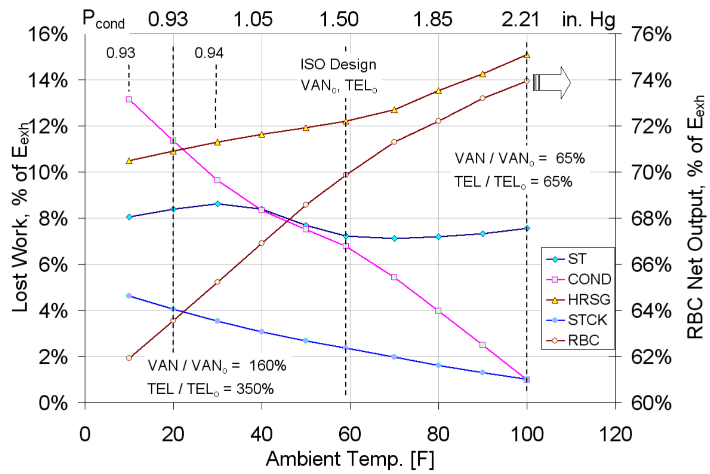

Figure 7 Individual lost work contributors and RBC net output as a fraction of total GT exhaust exergy; dependence on ambient temperature.

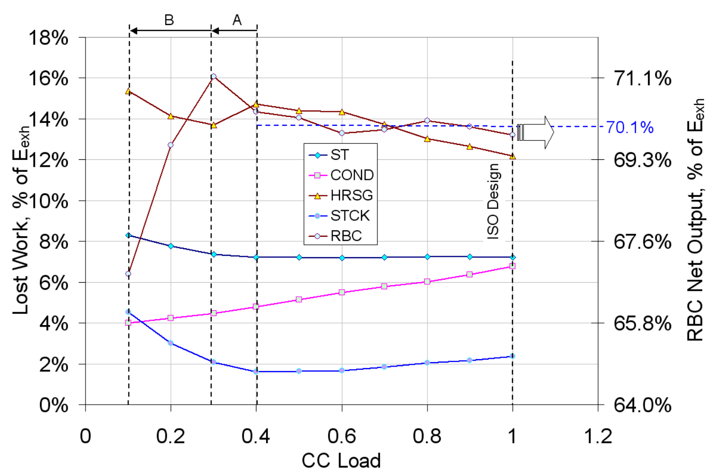

Figure 8 Individual lost work contributors and RBC net output as a fraction of total GT exhaust exergy; dependence on CC load.

For the part load performance, the driver is the particular GT load control and HRSG steam attemperation. Above 40% load, HRSG lost work increase (high attemperation) is balanced by the reduction in condenser lost work (lower duty and steam temperature) so that variation in ε_{RBC} is minimal. Below 40% CC load, IGV closure reaches its maximum and firing/exhaust temperatures go down to reduce the GT/CC load. Initially the sudden drop in attemperation reduces HRSG lost work and improves ε_{RBC} (Region A). Further reduction, however, severely hampers the HRSG heat transfer effectiveness, which is reflected in increases in HRSG and stack lost works and decrease in ε_{RBC} (Region B). For most of the CC load range, the assumption of constant ε_{RBC} is adequate for performance estimation. The exact breakpoint is dependent on the particular GT and its load control. In most cases, a severe drop should be expected once the maximum IGV closure is reached.

Thus, a detailed model with 1600 inputs is replaced by four simple formulas, whose only inputs are GT exhaust flow and temperature. This is sufficient for many GT-CC feasibility studies, where the focus is on the GT off-design control and performance. One can either use Eq. [6] with a constant ε_{RBC} (at the ISO base design value) or Eq. [14] with the aid of Figure 6. Either option is adequate for most CC conceptual off-design performance studies. It should be pointed out that the approximate CV approach is also a viable option for conceptual analysis. (In a conceptual study, FPT data requisite for the full-blown CV approach would only be available from a *bona fide* heat balance model, which would obviate the need for a simple method in the first place.) In this case, since measurements are not available, key parameters such as stack temperature and condenser pressure have to be estimated using suitable approximations, which are listed in the Appendix.

Performance Evaluation Application

An important use of plant modeling is to assess the deterioration of major Rankine steam bottoming cycle equipment in terms of its impact on CC performance. The analysis in Gülen and Smith [20] clearly shows that three key systems, namely, HRSG, ST and the condenser, account for almost all of the RBC losses. Data from the two performance tests of a 2×1 CC plant with a frame GT are analyzed by applying Eq. [13] to each subsystem CV using the FPT measurements. The results are shown in the bar chart in Figure 9. Also shown in the chart are key system temperatures.

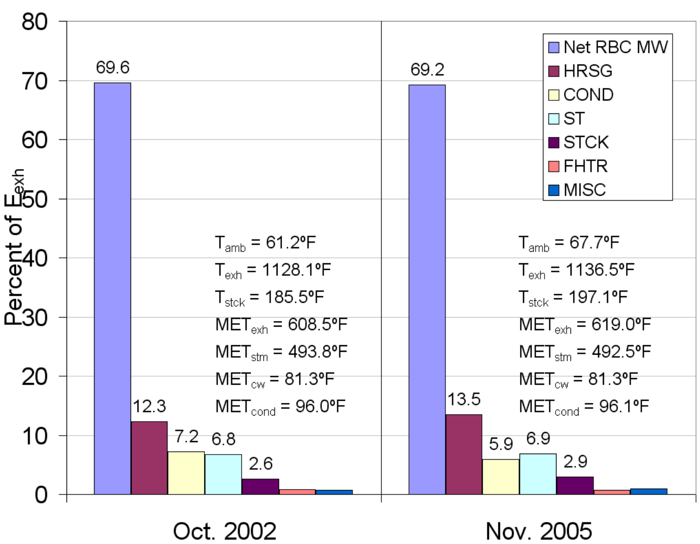

Figure 9 Exergy analysis of data from two different performance tests of a 2×1 CC power plant. Listed GT exhaust and HRSG stack gas temperatures are for the mixed flow of two units. HRSG lost work is inclusive of transition duct and casing heat losses.

The exergy analysis seems to point to a problem with the heat recovery process, which results in higher HRSG irreversibility and stack exergy loss (even with 10ºF higher GT exhaust temperature). In gas-fired advanced CC systems, HRSG performance degradation of the observed magnitude via internal (water or steam-side) tube fouling is atypical. HRSGs normally undergo inspection and preventive maintenance during schedule GT outages and the water chemistry is closely monitored. External (gas-side) fouling is even more unlikely to be a cause due to the relative impact of the gas-side heat transfer on the overall heat transfer. Mean-effective cooling water and steam temperatures indicate that condenser irreversibility is the same so that the reduction in the condenser lost work difference is a result of the higher ambient temperature. HRSG irreversibility difference can be traced back to the relative difference in mean-effective exhaust gas and steam temperatures. In essence, the HRSG in the more recent test cannot seem to take advantage of the higher exergy/energy input from the GT exhaust gas. This is most likely a result of heat exchanger sizing and/or excess attemperation rather than system degradation or fouling. Note that this is only a finger pointed in the right direction. Determination of the exact location (e.g., individual component or system) and the physical mechanism constituting the root cause of the problem requires more detailed data analysis.

Table 1 Key measurements for bottoming cycle lost work calculation. See Gülen and Smith for more information [27].

Parameter	Measurement Method	Error
ST generator output	Precision watt-hour meter meeting ASME PTC 6 code requirement	±0.25%
GT exhaust flow	From GT fuel gas and inlet airflow measurements	±1.5-2.5%
GT exhaust temperature	GT exhaust thermocouple rakes	±7-10°F
HRSG stack temperature	Stack thermocouple rakes	±10°F
Condenser pressure	Precision-calibrated or station	±0.1-0.5%
Steam cycle pressures	Precision-calibrated or station	±0.1-0.5%
Steam cycle flows (water)	Precision-calibrated or station	±0.8-1.25%
Steam cycle flows (steam)	Precision-calibrated or station	±2.0-2.5%
Steam cycle temperatures	Precision-calibrated or station	±1-7°F

Using the available measurements (see Table 1 below) and the principles developed herein, for performance monitoring and/or test performance assessment purposes, two major strategies can be identified:

Multi-shaft CC (separate GT and ST generators): Evaluate HRSG and condenser lost works (Eq. [13] or Eq. [A-1] and Eq. [A-2]); evaluate RBC power output from measured ST generator and pump power consumption; and evaluate ST lost work: (i) Directly and in detail from Eq. [13] or Eq. [A-3], or (ii) indirectly, as a lumped value, from Eqs. [9-12].

Single-shaft CC (common CC generator): Evaluate HRSG and condenser lost works (Eq. [13] or Eq. [A-1] and Eq. [A-2]); evaluate ST lost work from Eq. [A-3] using ST section data; and evaluate RBC power output by combining Eqs. [9-12]. In this case, ST output is calculated from Eq. [6] and the GT output is inferred from the balance of the measured generator output. The reader can consult Ref. [27] for details of a rigorous reconciliation process to obtain the most accurate value.

Note that fuel heater and miscellaneous exergy losses show very little variation and can be assumed to be constant at their nominal design values (see the Appendix). In either case, one key item requiring attention is the LP section of the ST. The two-phase (*wet*) exhaust steam flow to the condenser is not amenable to a reliable "measurement" of the stream enthalpy and/or the LP section efficiency. As such, a key piece of information is missing whether one is using the full-blown or approximate CV approach. The only feasible option is to use all information available to come up with a reasonable assessment of η_{LP} (η_3 in Eq. [A-3] below) and use it either directly in Eq. [A-3] or to evaluate the exhaust stream enthalpy in Eq. [13] with an appropriate uncertainty ascribed. Typically, ±1 percentage point for the assumed LP efficiency is a good starting point. Each percentage point in LP efficiency translates into 3.5 Btu/lb (~8 kJ/kg) in exhaust steam enthalpy.

ST throttle steam pressure and temperature (assumed to be the same as *hot reheat* steam temperature) are the key steam cycle parameters. Other ST section inlet and exhaust pressure and temperatures can be utilized for evaluation of section efficiencies and pressure ratios. Boiler feed and condensate pump power consumption (if available) can be used to obtain the net RBC power output. (They can also be expressed as a simple percentage of the ST output.) With known fuel gas composition, GT exhaust gas composition can be calculated using the stoichiometric balance and GT exhaust and stack gas exergy can be calculated from an appropriate EOS. Alternately, simplified equations (see the Appendix) are quite adequate as well for natural gas fired systems. The parameter uncertainties in Table 1 are used in a *Monte Carlo* simulation based on Eq. [13] to find the uncertainty of three major lost work contributions (Table 2).

Table 2 Uncertainty of RBC lost work for HRSG, ST and the condenser using FPT data in Eq. [13] with parameter uncertainties in Table 1 along with ±1% (points) in assumed LP section efficiency.

		Uncertainty	
	Mean		% of Mean
GT Exhaust Exergy (% of HC)	29.3%	±0.7%	2.3%
HRSG Lost Work (% of GT Exhaust Exergy)	12.1%	±2.4%	19.5%
ST Lost Work (% of GT Exhaust Exergy)	7.3%	±0.8%	11.5%
Condenser Lost Work (% of GT Exhaust Exergy)	6.8%	±0.1%	1.9%

CONCLUSIONS

A simple but powerful method that draws upon the key concept of exergy (directly from the second law of thermodynamics) is developed and demonstrated. In particular,

1. In its purely theoretical, fundamental form, i.e., Eqs. [1-4], the method leads to the same conceptual conclusions via simple formulas that an engineer can program in Excel in less than 30 minutes.
2. **Method I**: In its most basic embodiment, i.e., Eqs. [5-8], the method is shown to describe the GT-CC performance at off-design ambient and loading conditions with fidelity commensurate with conceptual studies in the early CC system design stage (using Eq. [14] with Figure 6 in lieu of Eq. [6] is an option).
3. **Method II**: When sufficient information is available (e.g., FPT data from performance tests or plant data historian), it is shown that the method can be implemented in a more rigorous manner to pinpoint the key source(s) of observed system performance loss (i.e., Eqs. [9-13]).
4. **Method III**: Approximate relationships are provided to make this more exact form of the method applicable in the absence of complete FPT information (i.e., Eqs. [9-12] with Eqs. [A-1,2,3]).

The predictive qualities of the three variants of the second law method for off-design performance calculation are summarized in Tables 3 and 4 (±1% (points) is assumed LP section efficiency for Methods II and III).

Table 3 Average (absolute) error of the methods in predictive mode. (Based on comparison with model data for ST generator power output of four (4) 1x1 GTCC plant systems with 50-Hz and 60-Hz advanced F-Class GTs.)

	Ambient 10-100ºF	Part Load 10-100%
METHOD I – Using Eq. [16] and Figure 6	0.25%	0.75%
METHOD II	0.35%	0.50%
METHOD III	0.30%	1.35%

Table 4 Uncertainty of the calculated ST generator power output using Methods II and III in (on-line) performance monitoring mode. (For a single-shaft 1×1 GTCC based on the uncertainties in Table 1.)

	% Error
METHOD II (~25 FPT Measurements)	±1.70%
METHOD III (6 FPT Measurements)	±2.50%

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APPENDIX

As pointed out in the main body of the paper, in the presence of sufficient FPT data, component lost work can be exactly calculated using the generic Eq. [13] for the particular system CV. This is the recommended application of the second-law principles in performance test data analysis and on-line monitoring applications. For conceptual studies, especially in the early stages when detailed plant performance simulation models are not available, simple estimates of three major lost work contributions (from the HRSG, the ST and the condenser) can be obtained using the following equations. For detailed description, including their derivation and application, the reader is referred to Ref. [20].

$$\dot{W}_{l,HRSG} = \dot{Q}_{HRSG} \cdot \left\{1 - HUF + \frac{T_o}{\overline{T}_{stm}} \cdot \left(HUF - \frac{\overline{T}_{stm}}{\overline{T}_{exh}}\right)\right\} \quad \text{[A-1]}$$

$$\dot{W}_{l,COND} = \dot{Q}_{cond} \cdot \left(1 - \frac{T_o}{\overline{T}_{cond}}\right) \quad \text{[A-2]}$$

$$\dot{W}_{l,ST} = \sum_{i=1}^{3} \left\{ c \cdot \dot{m}_{s,i} \cdot T_o \cdot (1-\eta_i) \cdot \frac{(\pi_i - 1)}{k_i \cdot (\pi_i + 1)} \right\} \cdot (1 + 1.5\%) \quad \text{[A-3]}$$

$$\dot{W}_{l,MISC} = 1.5\% \cdot \dot{E}_{exh} \quad \text{[A-4]}$$

Equation [A-4] accounts for miscellaneous losses and GT fuel gas heating. If a more recent and/or accurate study is available for a particular system, it can be revised accordingly. These equations can also be used in lieu of the simple exergy-based model for off-design RBC performance calculation. (The last term in parentheses in Eq. [A-3] accounts for ST bearing and generator losses.) Substituting them into Eq. [9] via Eq. [12] one obtains $\dot{W}_{RBC}$. (Assuming no additional RBC exergy input via GT cooling steam, CAC, etc. These can be incorporated into the mix easily; e.g., see Ref. [21].)

The relationships described below are reasonably accurate for 3PRH CC systems with advanced F-Class (i.e., air-cooled) GTs. They are based on the premise that a suitable GT simulation model (or information) and, therefore, GT exhaust stream data (i.e., flow, temperature, and, optionally, composition) are available to the engineer over the intended operability envelope of the particular unit.

The mean-effective GT exhaust gas temperature is as follows:

$$\overline{T}_{exh} = \frac{(h_{exh} - h_{stck})}{(s_{exh} - s_{stck})} \approx T_{exh} \cdot \frac{\tau - 1}{\ln(\tau)} \quad \text{[A-5]}$$

where τ is the ratio of the stack temperature (in absolute scale) to the exhaust gas temperature. The enthalpies and entropies can be calculated using a suitable property package (e.g., JANAF [18]) with known GT exhaust gas temperature, composition and the HRSG stack temperature. The simpler form is adequate for quick estimates.

HRSG stack temperature can be estimated quite reasonably using Eqs. [A-7] and [A-8] in Ref. [20]. A simplified variant suitable to conceptual studies, with wet or dry heat rejection systems, is given as:

$$T_{stck} = 201.51 + 7.7168 \cdot x - 1.0475 \cdot x^2 \qquad \text{[A-6.1]}$$

$$x = \frac{T_{exh}}{1000} \cdot (10 - P_{cond}) \qquad \text{[A-6.2]}$$

In Eq. [A-6], temperatures are in degrees-F and the condenser pressure is in inches of mercury. However, noting that the variation in the stack temperature over the ambient range is typically small (a few degrees), especially for systems where the low-pressure economizer tube temperature is maintained via feed water recirculation, using a constant value specified at the design point is adequate for full-load, ambient temperature predictions.

Table 5 Absolute average error of Eq. [A-6] for predicting the HRSG stack temperature.

	OT-OL	CL-CT	ACC
$\overline{\varepsilon}_{abs}$ **[ºF]**	**1.1±1.5**	**2.3±0.9**	**0.5±0.3**

If enough FPT data is available for the steam and water streams crossing the HRSG CV, $\overline{T}_{stm}$ can be calculated exactly using ASME steam tables [25], i.e.

$$\overline{T}_{stm} = \frac{\sum_k \dot{m}_k \cdot h_k(P_k, T_k) - \sum_i \dot{m}_i \cdot h_i(P_i, T_i)}{\sum_k \dot{m}_k \cdot s_k(P_k, T_k) - \sum_i \dot{m}_i \cdot s_i(P_i, T_i)} \qquad \text{[A-7]}$$

The numerator of the term on the RHS of Eq. [A-7] is the net enthalpy transfer across the HRSG CV via steam/water flows. The term in the denominator is the net entropy transfer. (Note that the number obtained from Eq. [A-7] is in absolute temperature scale.)

In the absence of extensive FPT data, Eq. [A-3] in Ref. [20] can be used for estimating the value of $\overline{T}_{stm}$ at ISO base load. For quick estimates, use the following rules of thumb:

For variation with ambient temperature: Each 1ºF change (increase or decrease) in $\overline{T}_{exh}$ is accompanied by 1ºF change in $\overline{T}_{stm}$ in the same direction until about 70°F ambient. At higher ambient temperatures, the variation ratio can be assumed as ⅓ºF to 1ºF.

For variation with CC load: $\overline{T}_{stm}$ can be considered constant at the same value as in ISO base load down to about 30% load. At lower loads, it can be assumed to vary in lockstep with $\overline{T}_{exh}$.

Note that these guidelines are rough approximations for quick but reasonably accurate estimates. Variations due to different GT control and exhaust characteristics can and will result in deviations. They should be evaluated on a case-by-case basis.

Condenser heat duty in Eq. [A-2] can be readily calculated using Eqs. [A-6] and [A-7] in Ref. [20] or Eqs. [A-1]-[A-3] in Ref. [19]. The mean-effective condenser temperature is simply the saturated steam temperature at the condenser pressure and can be readily obtained from the ASME steam tables [25]. HRSG duty in Eq. [A-1] is the total enthalpy difference between GT exhaust gas and HRSG stack gas, which can be directly calculated using a proper EOS or can be estimated as follows (for one GT)

$$\dot{Q}_{HRSG} = \dot{m}_{exh} \cdot \overline{c}_p \cdot (T_{exh} - T_{stack}) \qquad \text{[A-8.1]}$$

$$\overline{c}_p = 0.20982 + 9.73325 \cdot 10^{-5} \cdot \overline{T}_{exh} \qquad \text{[A-8.2]}$$

It is no exaggeration to state that the condenser pressure is probably the most important RBC parameter. It is a closely monitored plant measurement and readily available for performance monitoring applications. For the conceptual studies, the following simple relationship can be used for three different types of heat rejection systems:

$$\frac{P_{cond}}{P_{cond,d}} = c_1 \cdot \exp\{c_2 \cdot x\} \qquad \text{[A-9]}$$

The subscript $_d$ in Eq. [A-9] denotes the ISO base load design value and x is T_{amb} in degrees-F or CC load as a fraction. See Table 6 for values of c_1 and c_2 appropriate to the particular heat rejection system.

Table 6 Parameters in Eq. [A-9] for different heat rejection systems (load is expressed as a fraction on CC basis).

x		OT-OL	CL-CT	ACC
T_{amb}	c_1	**0.4868**	**0.4536**	**0.1939**
	c_2	**0.0122**	**0.0134**	**0.0278**
Load	c_1	**0.5011**	**0.4464**	**0.3209**
	c_2	**0.691**	**0.8065**	**1.1366**

For the ST irreversibility formula, Eq. [A-3], and the default values of the parameters therein, the reader is referred to Eqs. [A-9]-[A-12] and the accompanying discussion in Ref. [20]. The default value for the parameter *c* is ⅛ but it can be treated as a calibration factor to bring the calculated value in line with a known ST product.

Briefly, k_i in Eq. [A-3] is ⅓ for i=1, 2,and 0.2 for i=3; $\dot{m}_{s,i}$ is the steam flow through the ST sections. Simplified ST exhaust loss calculation can be found in the Appendix of Ref. [19] or the references listed therein. Obviously, a user can also refer to available information for a particular unit to substitute for parameters such as η_i and π_i in Eq. [A-3]. The variation in those parameters from their design value is mostly negligible across the operability range (e.g., see Ref. [13]). One exception is η_3, which can be corrected for the exhaust loss using the method suggested in Ref. [20].

Calculation of Exergy

The exhaust gas exergy in Eq. [6] is a fluid property, which is a function of gas temperature and composition. It can be readily calculated using Eq. [2] with an appropriate EOS for gases such as JANAF. (Exhaust gas composition is determined for a known fuel using the stoichiometric relationships.) A reasonable approximation for natural gas (100% methane) fired GTs, valid between 900ºF and 1600ºF, is given below [21]:

$$e_{exh} = 0.001628 \cdot T_{exh}^{1.60877} \qquad \text{[A-10]}$$

This equation represents a curve-fit to detailed stoichiometric calculations using JANAF. A relationship similar to Eq. [A-10] can be used to calculate the HRSG stack gas exergy, which is valid between 165ºF and 225ºF, i.e.,

$$e_{stck} = 2.15479 \cdot 10^{-6} \cdot T_{stck}^{2.72872} \quad \text{[A-11]}$$

GT exhaust and HRSG stack gas exergy formulas, Eq. [A-10] and Eq. [A-11], respectively, are based on a reference state of 14.7 psia and 59ºF. (As explained in detail in Ref. [20], exergy associated with the latent heat of water vapor in the GT exhaust gas is ignored.) This is also known as the *dead state*, at which the conditions of mechanical, thermal, and chemical equilibrium between the system and the environment are satisfied [26]. In order to be able to use those formulas for different ambient temperatures, one should correct for the temperature and composition effect so that they can represent the proper dead state for the exergy calculations. (Note that the formulas and the corrections below are for 100% CH_4 natural gas fired systems.) Multiplicative correction factors in Figure 10 can be used for that purpose.

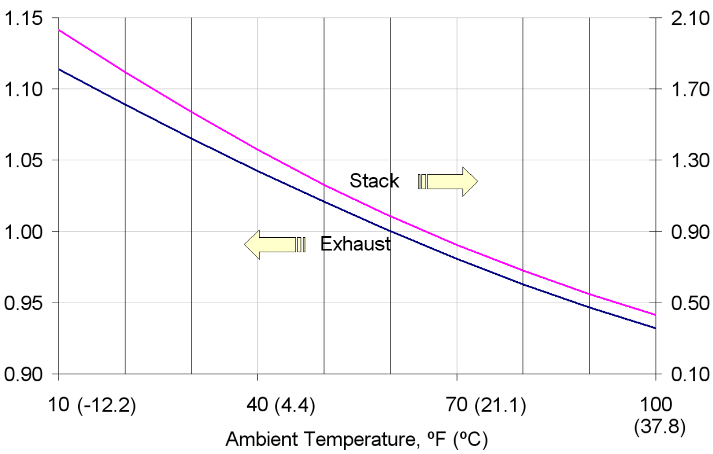

Figure 10 Gas exergy formula multiplicative correction factors (to be applied to Eqs. [A-10,A-11]) for ambient (i.e. reference) temperature.

Note on Auxiliary Power Calculation

For the calculation of CC net power output and heat rate, reliable estimates for plant auxiliary power consumption are required [19]. In general, the auxiliary power consumption is expressed as a fraction of the CC plant gross power output (i.e., the sum total of prime mover generator outputs):

$$\dot{W}_{aux} = \alpha \cdot \dot{W}_{Gross} \quad \text{[A-12]}$$

For the off-design calculations, the following scaling is adequate

$$\frac{\alpha}{\alpha_d} = \left(\frac{\dot{W}_{Gross}}{\dot{W}_{Gross,d}} \right)^{\beta} \quad \text{[A-13]}$$

Reasonable default values for α_d can be obtained from Table 1 in Ref. [19] for different plant heat rejection options. The term in the parentheses on the RHS is essentially the CC plant load referenced to the ISO base design value. The recommended values of β are listed in Table 7.

Table 7 Parameter β in Eq. [A-13] for different heat rejection systems

	OT-OL	CL-CT	ACC
Ambient Full Load	**0.61**	**0.66**	**0.70**
ISO Part Load	**0.67**	**0.75**	**0.68**

Note on Eq. [13]

Equation [13] is the more rigorous formulation of Eq. [10] in a generic form. For an adiabatic turbine (e.g., the ST in the RBC) or pump, $\dot{E}_Q$ is zero and $\dot{E}_W$ is the shaft work generated or absorbed, respectively, by these components. Combining with the first law (i.e., the energy balance for the CV), Eq. [13] reduces to

$$\dot{W}_{lost} = \dot{I} = \sum_k \dot{m}_k \cdot T_o \cdot s_k(P_k, T_k) - \sum_i \dot{m}_i \cdot T_o \cdot s_i(P_i, T_i) \quad \text{[A-14]}$$

For a general heat exchanger, both $\dot{E}_Q$ and $\dot{E}_W$ are typically zero. For the HRSG, however, it is sometimes more convenient to define the CV such that it encompasses the feed pumps as well. Thus, $\dot{E}_W$ is the power consumption of those pumps (a negative value by definition) and it is effectively <u>added</u> to the flow exergy entering the HRSG CV.

For the water-cooled condenser, both $\dot{E}_Q$ and $\dot{E}_W$ are indeed zero. Thus, Eq. [13] reduces to Eq. [A-14] and describes only the exergy *destruction* in the condenser CV. This result is correct but incomplete in the sense of fully describing the loss mechanisms associated with cycle heat rejection. Thus, it is more convenient to regroup the terms in Eq. [13] so that

$$\dot{W}_{lost} = \dot{I}_{cond} + \dot{E}_{cond} = \sum_k \dot{m}_k \cdot e_k(P_k, T_k) - \sum_i \dot{m}_i \cdot e_i(P_i, T_i) \quad \text{[A-15]}$$

where the two terms on the RHS account for the net *exergy* transfer associated with the cycle working fluid (i.e., steam or water) <u>only</u>. $\dot{I}_{cond}$ is the exergy destruction per Eq. [A-14] and $\dot{E}_{cond}$ is the exergy transfer associated with the energy carried out of the condenser (also out of the RBC) CV with the cooling water:

$$\dot{E}_{cond} = \dot{Q}_{cond} \cdot \left(1 - \frac{T_o}{\overline{T}_{cw}} \right) \quad \text{[A-16]}$$

In Eq. [A-16], $\overline{T}_{cw}$ is the mean-effective cooling water temperature, which is the logarithmic mean of the condenser cooling water inlet and outlet temperatures.