Practical Experience With Second Law Power Plant Monitoring

Presented By:

Laura M. Douglass

Pacific Gas & Electric Company

PRACTICAL EXPERIENCE WITH

SECOND LAW POWER PLANT MONITORING

by

Fred D. Lang, P.E.
Exergetic Systems, Inc.
Livermore, California 94550 USA

Ken F. Horn, P.E.
Pacific Gas & Electric Company
San Francisco, California 94106 USA

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ABSTRACT

This article discusses the use of an "ultimate" performance monitoring technique derived from Second Law concepts. Other techniques and their methods have been reported.

If electricity is to be produced with the minimum of unproductive consumption of fuel - then fundamental thermodynamic losses must be understood on a system basis. Such understanding cuts across vendor curves, plant design, fuels, etc. Thermal losses in a nuclear unit are comparable at a prima facie level to losses at any other thermal system. They are what we must minimize in the production of electricity, no manner the method of that production. The Second Law offers the only foundation for the study of such losses, and thus affords the basis for a true and ultimate indicator of <u>system</u> performance. From such a foundation, a parameter is needed to tell us specifically what components are

thermodynamically responsible for fuel consumption given either their direct creation of electricity or their contribution to thermodynamic losses. The Fuel Consumption Index, discussed in this article, is this parameter. It can be used for thermodynamic system design, monitoring, diagnosing problems, and economic dispatching. It tells us why fuel is being consumed; consumed by a nuclear plant, trash burner, a 40 year-old fossil plant, etc.

Fuel Consumption Index (FCI) is a dimensionless measure of fuel consumed as assigned thermodynamically to those individual components or processes responsible for fuel consumption, given a system's production of power. It sums to 1000 when considering the entire system. It quantifies the exergy consumption of all components relative to the total exergy and power inputs to the system, the predominant term being fuel exergy. This article defines FCI calculational methods and presents details as to its practical use. ΔHeat rates based on FCI are also discussed.

1.0 INTRODUCTION

Pacific Gas & Electric Company and Exergetic Systems have developed new thermal performance monitoring techniques for power plants. These techniques include both simple parameters, and parameters based on Second Law principals. One simple parameter developed was L^2, which requires no water properties and is monitored with common "plant-level" instrumentation.[1] This article discusses the use of another, "ultimate" performance monitoring parameter derived from Second Law concepts.[3-9] A summary of the specific methodology used by the authors is available.[10,15]

Enthalpy is used in energy balances to analyze the flow of energy through a system's components. Examining the detailed energy flows through components within a system, as opposed to simple summations of heat and power transfers across global boundaries, helps us understand the system and ways of improvement relative to design. In this fashion, enthalpy is the "working variable" for First Law studies and deals with the <u>quantity</u> of energy. Certainly it is useful in this context. However, from a global perspective, the First Law measurement of performance, thermal efficiency or heat rate, quantifies only the exchange of energy from boiler to environment or $(1 - Q_{rej}/Q_{in})$, indeed it fundamentally relates to the <u>utilization of energy flows</u>. Although $Q_{in} - Q_{rej}$ is net system power, one is never advised to assess heat rate changes by addressing changes to power production, but rather by what happens individually to Q_{in} and Q_{rej}.[2] Indeed, for many design situations an increase in efficiency implies lower power (typically Q_{in} is reduced in greater proportion than Q_{rej}).

The professional life of a performance engineer is <u>not</u> devoted to the management of energy flows, nor to the conservation of fuel per se. Our raison d'etrê is the generation of adequate electricity for society using minimum fuel. This two-sided livelihood does not result in nor imply the closing of power stations to conserve fuel. For illustration, heat rate can be improved, most quickly, by doing those things which reduce power production. The increase of turbine extraction flows, the "creation" of steam consuming cogeneration processes, the use of auxiliary turbines for pump drives, the use of steam for space heating - all improve heat rate, but say little of electrical generation. Another point, unit heat rate cannot be used for comparisons between different plant designs. One does not compare a nuclear cycle heat rate to a supercritical fossil-fired cycle heat rate. Such comparisons are needed! For monitoring power plants our industry historically has used differences in heat rate - differences as a function of power level, relative to some test benchmark, and generally based on vendor assumed sensitivities when describing component behavior. **With these perspectives, the parameter "heat rate" can only be thought of as perverse when used to monitor the means of electrical production or when used for absolute comparisons.**

2.0 PRINCIPLES

All energy flows do not have the same potential for work. Study by Carnot and Gibbs has shown that any material not in equilibrium with the environment has the potential to do work, thus has available energy. In general, the higher the pressure and temperature, the higher the available energy, or quality of energy, and thus more available power per unit mass flow. The direct and immediate measure of this "quality" is exergy[9] (also termed thermodynamic availability). Exergy is the Second Law's "working variable" and deals with the quality of energy. The Second Law fundamentally relates to the utilization of potential power given an operating system. It is ideal for assessing the effective creation of electricity using minimum fuel.

While enthalpy relates to total energy flows, and the transfer of energy from one flow to another in a component; exergy relates to the available energy of a fluid and a component's consumption of available energy given its surroundings. As an example, the ability of a steam turbine to produce useful power is not dependent on a boiler's supply of energy. The boiler could be supplying a great deal of energy to a huge flow of water and not produce a pound of steam. The ability of the turbine to produce power is dependent on the "quality" or available energy of the inlet steam provided by the boiler, as well as quantity.

The following paragraphs discuss the basic concepts needed for proper evaluation of systems. They are involved in their engineering execution. However, for fossil-fired units the analysis has been incorporated into a computer program, called EX-FOSS™, available from Exergetic Systems, which allows the Second Law to be applied quickly and accurately.

2.1 Exergy

Exergy is a measure of the available energy of a substance given its surroundings. It is a state function, thus the change in exergy from one point to another is path independent when considering a closed steady-state system, such as the turbine cycle. Exergy can also be destroyed or converted when considering the total interaction of a system with its environment. For example, the difference in the reheater's steam exergy, from HP exhaust to IP inlet, describes the potential of the turbine cycle to perform work. A turbine cycle is a closed system when considering the confines of the working fluid to be the thermodynamic boundary, within which the component's exergy changes must sum to zero. The total exergy change associated with a reheater, Δexergy of the gas less Δexergy of the working fluid, describes its ultimate thermodynamic losses (irreversibility) relative to the environment. In summary, when viewing total systems interacting with their environments: exergy is "exchanged" for system irreversible losses and power production. Exergy is defined by the following equation:

$$g = \Delta h - T_r \Delta s + \Delta KE/m + \Delta PE/m \qquad (1)$$

Since the initial substance (referring to the actual effluent compounds) may not be present in the reference environment, a thermodynamic path must be provided to reach equilibrium thus to a zero potential for work. The path taken, internal to EX-FOSS™, has three steps:

1) The substance under consideration is brought to the standard state: P^o = 1 bar (14.50382 psia) and T^o = 298.15 K (77.0 F).
2) The substance is then transformed chemically to different reference species which can be found at equilibrium in the environment, with ideal chemical reactions at the standard state condition.
3) The species are brought, via pressure and temperature changes, from standard state to a reference condition.

This path was chosen to make use of published values of the Heat of Formulation, ΔH^o_f, and the Gibbs Free Energy of Formulation, ΔG^o_f, at standard state conditions. It should be noted that the resultant exergy is independent of the chosen path. Using this path Eq.(1) can be written as the following, for a given substance j:

$$g_j = \Delta g_{std,j} + g_{std,j} \qquad (2)$$

In this equation, Δg_{std} is defined as the exergy change from the initial state to the standard state (step 1), and g_{std} is defined as the exergy of the substance at standard state conditions (steps 2 and 3). Once the environment and reference conditions have been defined, g_{std} is constant and only needs to be calculated once. For reference species, i, g_{std} is defined as:

$$g_{std,i} = m_i[(h_{i,To,Po} - h_{i,T_r,P_r}) - T_r(s_{i,To,Po} - s_{i,T_r,P_r})] \tag{3}$$

For substances (j), not defined by a reference specie (i), g_{std} is defined as:

$$g_{std,j} = \sum g_{std,i} - \Delta H^o_{c,j} - \frac{T_r}{T^o}(-\Delta H^o_{c,j} + \Delta G^o) \tag{4}$$

Eq.(2) is a general equation which calculates the absolute exergy of <u>any</u> substance and can be shown to equal Eq.(1) ($\Delta KE = \Delta PE = 0$). The exergy of fuels is calculated using the same method described above with the Gibbs Function of the combustion reaction (for fuels with undefined Gibbs Free Energy of Formation) estimated using a method developed by Ikumi, Lou, Wen.[12]

Reference conditions are defined by the conditions existing at the plant, and are used to define the thermodynamic **reference environment.** EX-FOSS™ defines this environment in an unique but practical manner, as the conditions which would exist if the actual environment was allowed to reach thermodynamic equilibrium with the thermal system.[13] This is established by performing mass and energy balances of the air and water entering the system, and then assuming the resultant mixed thermodynamic state is that at which the air and water are in equilibrium. Typical results yield a reference temperature approximately equal to inlet cooling water or wet bulb temperature for plants using a cooling tower.

The maximum power which can be produced or the minimum power consumed by the working fluid in any process or component is measured by its associated change in exergy. The net exergy change across any component is simply the difference in the exergy flows between the outlet and inlet streams:

$$\Delta G = \sum m g_{outlet} - \sum m g_{inlet} \tag{6}$$

Therefore, exergy audits permit performance engineers to quickly evaluate the system and determine which components should be consuming or producing the most power. Another important quality of exergy is that, unlike First Law energy flows, total exergy flows are destroyed/converted when viewing an active system interfaced with its environment. In other words, in the process of power production the exergy bound in the fuel is destroyed/ converted (i.e., its potential for power is manifested in system losses <u>and</u> electricity production). However, since exergy is a thermodynamic property, within the confines of a closed, steady-state system, the summation of all stream exergy changes must be zero (e.g., working fluid in a closed turbine cycle). These subtleties are important: the rate of exergy destruction/ conversion, and concomitant creation of either thermodynamic losses (i.e., irreversibilities) or shaft power, when viewed from a systems standpoint, allows qualitative assessment as to where in the system the fuel's exergy is dissipated. However, within a closed system, the distribution of exergy changes allows the tracking of potential sources of lost power at a component by component level.

2.2 Irreversibility

Irreversibility is the unrecoverable thermodynamic loss associated with any process. Irreversibility is synonymous with exergy destruction. Power production arises from converted exergy. For a process assumed interfaced with its environment, irreversibility is the measure of exergy destruction associated with the system. It is defined as:

$$I = \int (1.0 - T_r/T)\partial Q - \int \partial W - \int m dg \tag{7}$$

In Eq.(7) the term $\int (1.0 - T_r/T)\partial Q$ is the Carnot conversion of heat flow to work. It is the exergy resultant from heat transfer to or from the process, i.e., the heat transport of exergy. For all EX-FOSS™ heat exchangers which have heat losses (via the L_b term of ASME PTC 4.1), such losses are assumed transferred directly to the environment. By definition of the thermodynamic environment no positive work can occur within the environment. Mathematically, the Carnot conversion term <u>from</u> the component, is exactly offset by a similar term (operating within the environment) of opposite sign. Thus the irreversibility listed in the EX-FOSS™ output is a total irreversibility, for heat exchangers defined as Δexergy.

Irreversibility is a measure of the exergy <u>destroyed</u> and is directly proportional to fuel consumption. Y.M. El-Sayed has suggested to the authors an index based on a ratio of irreversibility to total fuel energy flow.[11] From this suggestion the authors[10] developed a "Fuel Consumption Index" or FCI, based on the total exergy and actual power supplied <u>to</u> the system. A simular concept has been proposed by Kotas.[17] Recall that exergy <u>conversion</u> and power consumption are, of course, directly related to power production. Thus, when weighed against power production <u>and</u> irreversible losses, FCI sums to unity when considering global systems.

2.3 Fuel Consumption Index

Fuel Consumption Index (FCI) is a dimensionless measure of fuel consumed as assigned thermodynamically to those individual components or processes responsible for fuel consumption, given a system's production of power. It quantifies the exergy consumption of all components relative to the total exergy and power inputs to the system, by far the predominant term being the fuel's exergy. FCI is defined as:

$$FCI_i = 1000 \frac{(I_i \text{ or } W_{output})}{(m_{AF} g_{Fuel} + m_{Air} g_{Air} + \sum G_{Misc} + \sum W_{input})} \tag{8}$$

As used in Eq.(8) all total exergy, irreversibility and power terms employ units of Btu/hr, thus FCI is dimensionless. We chose to multiply the index by 1000. Some typical values of the FCI include: 402 for direct electrical production (gross power available at the generator terminals), 271 for a fossil combustion process, 202 for boiler heat exchangers, 40 for the main turbines, etc. The FCI relates that 40.2% of the supply exergy results in electricity, 27.1% is destroyed via combustion losses, 20.2% is destroyed via boiler heat exchanger losses, etc. Again, one important characteristic of the FCI is that it always sums to 1000, therefore a decrease in the FCI of one component means an increase in the FCI of another component. Thus systems can be compared relative to their fuel usage. For example, if the FCI of direct electrical production decreases from 402 to 395 and the FCI of the boiler heat exchangers increases from 202 to 209, with no other changes, one can attribute the loss in electrical production (less of the fuel's exergy is converted to electricity) to problems in the boiler heat exchangers (more of the fuel's exergy is destroyed in the heat exchangers).

2.4 Differential Heat Rate

Differential heat rate as defined for this work, is determined for individual components and processes consistent with <u>both</u> First and Second Law concepts. Differential heat rate, termed HR_i in units of ΔBtu/kWh, is defined from <u>Second Law</u> concepts through the FCI by the following:

$$HR_i = FCI_i \; m_{AF} \; (HHV + HHBC) \; / \; (1000 \; W_{output}) \tag{11}$$

The summation of HR_i for all components and processes is the First Law's definition of unit heat rate, termed HR. This is more than a mathematical nicety. This must be the case since $\sum HR_i$ involves an inherent consideration of <u>all</u> thermodynamic losses and power production processes, i.e., the entire system.

$$HR = \sum HR_i = m_{AF} \; (HHV + HHBC) \; / \; W_{output} \tag{12}$$

Classical unit heat rate, employing an "In/Out" approach <u>does nothing for understanding individual component thermodynamics or processes</u>. **Second Law appraisal of heat rate, through FCI, is based on a rational evaluation of a system's response to fuel consumption. Given certain electrical production, the FCI concept considers the entire system through individual components. The Second Law's FCI approaches the system from a distribution of losses viewpoint, given a certain electrical production - not merely the trivial and perverse minimization of heat rejection.** Nuclear power cycles are comparable to fossil: one finds their turbine cycles are generally more effective than fossil. The FCI concept is of critical importance to a power plant engineer. It allows the breakdown of heat rate, component by component, thus allows the monitoring of degraded equipment and the search for improved operation. It also noteworthy that, with these concepts, the power plant engineer does not require "vendor curves", etc. to evaluate a particular component's effects on the system; he needs only an operating system, knowledge of the fuel, a few key measurements, and a Second Law analysis.

2.5 Comments on Unit and Turbine Cycle Heat Rates

It is obvious that employment of the Fuel Consumption Index requires no redefinition of unit heat rate - a rethinking in terms of differential heat rates, yes.[14] As discussed, the summation of differential heat rates, $\sum HR_i$, determined through FCIs, indeed results in the classical definition of gross unit heat rate.

As will be seen, there are computed FCIs associated with miscellaneous turbine cycle components, electrical power production, and the boiler's heat exchangers. The summation of these terms **is not** the commonly used "turbine cycle heat rate". As practiced in the past we speak of differential heat rates, HR_i, associated with turbine cycle components. For example, if hot reheat temperature is degraded by 10 ΔF, the computed turbine cycle heat rate is said to be affected by $\approx$10 ΔBtu/kWh, the unit heat rate is said to be affected by $\approx 10/\eta_{boiler}$ ΔBtu/kWh (typically assuming the boiler's efficiency is constant). Such studies give no consideration of **why** the reheater's temperature is degraded; and without the **why**, the result is coarse. The degradation could be caused by any one or more of the following: degraded heat transfer on the gas side, increases in environmental losses (boiler casing), lower ambient temperatures, lower fuel heating value, lower fuel flow, improved flue gas heat transfer upstream of the reheater, changes in gas flows via baffling in the convective gas path (if used), and/or degraded heat transfer on the working fluid side of the reheater. Many of these operational conditions might not significantly impact combustion efficiency, or the ratio of useful energy supplied to fuel flow might be constant - thus the assumption of $\approx 10/\eta_{boiler}$ for Δheat rate, in the conventional sense, might be valid! However, the point is that all of these situations effect system losses and thus the use of fuel for a given power production. The tracking of losses, through FCIs, adds needed sensitivity and addresses the problem directly.

It is obvious to the authors that turbine cycle differential heat rates, computed in isolation, as associated with a component interfaced with the thermodynamic boundary, is preposterous in any boiler-follow-turbine scenario. If the boiler intrinsically adjusts to any changes in fuel, electrical generation, thermal loading, the environment, heat transfer and/or degraded machinery then differential heat rates can not be assessed in isolation. However, common on-line performance calculations typically examine "controllable parameters" in isolation; for the turbine cycle these include throttle pressure, throttle temperature, hot reheat temperature and condenser pressure. For the boiler, "controllable parameters" include stack temperature and excess air. From a thermodynamicist's viewpoint, turbine cycle differential heat rates or boiler Δefficiencies computed for these parameters, in isolation, are meaningless terms. Obviously the Fuel Consumption Index is needed!

2.6 Calculational Procedure

The following procedure is recommended for on-line monitoring of power plants when using the Fuel Consumption Index:

1) Start with data from Input/Output tests, or some set of reference (bogey) heat balances. It is suggested that at least three load points be obtained. Both boiler and turbine cycle data must be acquired and **must be consistent.** These data sets are referred to as bogey data.
2) Correct the bogey data to an appropriate reference condition. This might include reference condenser pressure, throttle conditions, reheat temperature, etc. The specifics of such correction methods are described in Ref.(16).
3) Using the corrected bogey data, compute reference Fuel Consumption Indexes, termed $(FCI_i)_{B-Ref}$, for each load point using the EX-FOSS program for the boiler, and either PEPSE or EX-SITE for the turbine cycle. The PEPLUS program (available from PG&E) can be used to reduce data from PEPSE in combination with the EX-FOSS program. Arrange the data as a function of power, for later interpolation.
4) Monitor the actual plant conditions, on-line, producing $(FCI_i)_A$ and $(HR_i)_A$. This data is reduced using the same boiler and turbine cycle simulators employed for the bogey.
5) Interpolate within the reference $(FCI_i)_{B-Ref}$ bogey data, using the actual monitored power to determine $(FCI_i)_B$ for all components and processes. Calculate the respective $(HR_i)_B$ values. It is critical to employ linear interpolation such that for any monitored load $\sum(FCI_i)_B = 1000$. Of course for the actual condition, $\sum(FCI_i)_A$ will always be 1000.
6) Calculate ΔFCI_i and ΔHR_i for each modelled component and process, more fully discussed in Section 3.0.
7) When ΔFCI_{Power} is negative, the system is producing the same power for <u>less</u> fuel consumed (improved performance relative to the bogey. When all $\Delta FCI_{non\text{-}Power}$ terms (i.e., irreversible terms) are negative degraded performance is at hand. This, of course, implies increased unrecoverable losses for the respective i-th component or process.

3.0 RESULTS

Figure 1 presents a typical Second Law output page produced by EX-FOSS™ for a 790 MWe coal-fired unit. Note that "Total Exergy Out" is the exergy out of the combustion process (the exergy of the flue gases at the actual flame temperature, before interfacing with the first heat exchanger). The "Stack Loss" is listed near the bottom and is simply the exergy of the flue gases (stack side, without air pre-heater leakage) exiting the system; air leakage is accounted in the air pre-heater component. The FCI of direct electrical production, termed "Elec. Power", is based on the gross power produced. The

"Misc. TG Cycle" terms include the exergy added to the working fluid of the turbine cycle by the boiler (listed in the flue gas column), and the irreversibility which is the exergy into the cycle minus the exergy out (the exergy added to the working fluid in the boiler, plus pump power, minus gross power out). The heat rate term for "Misc. TG Cycle" must not be confused with a vendor's turbine cycle heat rate (as discussed). "Misc. TG Cycle" is simply relating miscellaneous losses within the turbine cycle, <u>excluding</u> gas/water heat exchangers and electrical production. The "Sys. Totals" terms include the sums of the various columns, except for effectiveness which is gross power out divided by total exergy in.

Much can be written, and studied, concerning the Second Law balance presented by EX-FOSS™. The following points summarize the highlights and a few calculational over-checks which are possible:

- The total exergy and shaft power supplied to the system, termed G_{in}, is determined as:

  $$G_{in} = m_{AF}g_{Fuel} + m_{Air}g_{Air} + \sum G_{Misc} + \sum W_{Pump} + \sum W_{Fan} \tag{13}$$

  The fuel's exergy is listed under "Exergy in Fuel". The air's exergy is listed <u>either</u> as the input to the first air component, <u>or</u> (if no air component is modelled) is listed under the "Exergy in Air" (reflecting the air state condition). The $\sum G_{Misc}$ term accounts for miscellaneous exergy flows, for example exergy supplied to a steam air heater.

- The differential heat rate for direct electrical production, listed as "Elec. Power", is constant:

  $$HR_{Power} = 3412.1416 \tag{14}$$

  the units conversion factor. Note that it is possible to "tag" that portion of the fuel flow which produces power. This "Elec. Power" Δheat rate, since it is a constant, is an obvious illustration of why <u>unit</u> heat rate is typically not viewed from a power change view point - but only from a losses viewpoint, that is, changes to heat supplied or rejected.

- The differential heat rate which can be charged directly to the environment (as a positive <u>or</u> negative quantity), is determined by the following relationship:

  $$HR_{Envir} = -\ 3412.1416\ [G_{in} - (m_{AF}HHV + m_{AF}HHBC)]/G_{in} \tag{15}$$

  More correctly stated, HR_{Envir} relates to the exergy associated with combustion air conditions, with fuel conditions, and with bringing unique combustion products to equilibrium conditions in the environment. The term in parenthesis, the difference between the heating value & boiler credits and the total exergy supply, relates to the absolute contribution the environment plays on the system. Of interest to a thermo-dynamicist: if pure graphite is burned with theoretical O_2 at 77 F (no excess air, and $g_{Air} = 0$); and $\sum G_{Misc} = \sum W_{Pump} = \sum W_{Fan} = 0$; and a "thought environment" consists only of CO_2 at 77F - then HR_{Envir} is identically zero, HHBC = 0 and g_{Fuel} = HHV.

- The total unit heat rate, determined from Second Law principles, will always equal the classical definition (energy flow supplied, m_{AF}(HHV + HHBC), divided by gross power out).

- Boiler effectiveness (listed next to the "+" symbol in the "Stack Loss" row) times turbine cycle effectiveness (listed in the "Misc. TG Cycle" row) will equal system effectiveness.

- The combustion's irreversibility is the numeric summation of the "Exergy in Fuel", "Exergy in Air" and "Total Exergy Out".

- The FCI of direct electrical production (divided by 10) will equal the plant's percentage effectiveness.

- The "Mixing Loss" is the summation of irreversible losses associated with mixing gases from two gas path streams and/or the irreversible loss of inserting gas recirculation at the point of combustion.

Figure 2 represents differences between bogey (target) and actual ΔFCI_i, for each major component and process. Figure 2 is taken from an on-line monitoring system, called MAPS. Again, since FCI sums to unity, the operator can quickly learn where in the system problems exist. Differential heat rate displays are used in a similar manner. These displays are perfectly consistent with classical heat rate deviation monitoring schemes for the overall unit: the difference between a "bogey unit heat rate" assigned to the unit (e.g., from In/Out testing), and that computed via $\sum \Delta HR_i$ is identical.

As discussed, when ΔFCI_{Power} is negative, as developed in Section 2.6 and defined above, the system is producing the same power for less fuel consumed (improved performance relative to the bogey). When all $\Delta FCI_{non\text{-}Power}$ terms (i.e., irreversibility terms) are negative degraded performance is at hand. This implies increased unrecoverable losses for the respective i-th component or process.

Since the heat rate assigned to direct power production is constant, at 3412 Btu/kWh, the difference between the bogey and actual is always zero, $\Delta HR_{Power} = 0$. Thus, if a system's electrical generation is considered the independent variable, then obviously differences in unit heat rate relative to some bogey, at the same reference power, will be entirely due to non-power effects in the system. For example, changes in component I_i imply changes in fuel energy flow, m_{AF}(HHV + HHBC) if the same power is to be produced. If $\Delta FCI_{Power} = 0$ for the same power, then $(G_{in})_A = (G_{in})_B$ and the bogey unit heat rate and the actual will be the same, $\sum \Delta HR_i = 0$. This does not imply that the individual processes are identical (if high losses occur in some processes they must be offset by gains in other processes). If $\Delta FCI_{Power} \neq 0$ for the same power, and $(HHV+HHBC)_A \approx (HHV+HHBC)_B$, then any difference in unit heat rate is approximately $\Delta FCI_{Power} m_{AF}(HHV+HHBC)/W_{output}$.

For on-line monitoring systems the bogey values, $(FCI_i)_B$, should be computed from linear interpretation of several In/Out test results as reduced by EX-FOSS™. The independent parameter in the interpretation should be power. Linear interpolation is required such that $\sum FCI_i = 1000$ for all interpreted points (i.e., power levels) between the I/O reference test data.

4.0 NOMENCLATURE

FCI_i Fuel Consumption Index for components and power production, dimensionless, $\sum FCI_i = 1000$.
g Specific exergy composed of physical, chemical, and thermal contributions, see Ref.(10).
g_{Air} Specific exergy of moist air inlet to the system.
g_{Fuel} Specific exergy of as-fired fuel, see Ref.(10).
$\sum G_{Misc}$ Summation of miscellaneous exergy flows into the system, such as a steam-air heater.
Δh Enthalpy change from initial to reference, Btu/lbm.
HHBC Boiler energy credits, defined by PTC 4.1, Btu/lbm.
HHV Fuel's higher heating value, Btu/lbm(as-fired).
HR Heat rate, Btu/wKh.
I_i Irreversibility for components, see Ref.(10).
$\Delta KE/m$ Relative specific kinetic energy, taken as zero.
m_{AF} Mass flow rate of as-fired fuel.
$\Delta PE/m$ Relative specific potential energy, taken as zero.
Q_{in} Heat transfer to the working fluid from boiler.
Q_{rej} Heat transfer from condenser to circulatory water.
Δs Entropy change from initial to reference, Btu/R-lbm.
T Temperature, degree F.
T_r Temperature of reference conditions, see Ref.(10).
$\sum W_{input}$ Summation of net shaft power input to the system.
W_{output} Gross electrical generated.

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15 F. D. Lang and K. F. Horn. "Make Fuel-Consumption Index Basis of Performance Monitoring", Power, Oct. 1990.

16 F. D. Lang. "Methodology for Testing and Evaluating Power Plants Using Computer Simulators", Proceedings of the 1990 Performance Software (PEPSE) User's Group Meeting, sponsored by EI International, Inc., May 1-4, 1990, St. Louis, USA.

17 T. J. Kotas, op. cit. Refer to pp. 73ff in which the ratio of exergy transfer making up the output ($\sum \Delta E_{out}$) is divided by the exergy tranfers making up the input ($\sum \Delta E_{in}$) to form a "rational efficiency". This leads to $\sum \Delta E_{in} = \sum \Delta E_{out} + \sum I_i$, conceptuallity identical to Eq.(8) writen as $G_{in} = W_{output} + \sum I_i$ used in this work. Differences between works lies with the use of irreversibilities within the FCI definition, the use of Δheat rates to present results and calculational techniques used to determine fuel exergy. Use of the FCI concept makes no discrimination between irreversibilities and power output thus the FCI relates to why fuel is comsumed for all system components.

Figure 1:

Example of EX-FOSS Second Law Output

S E C O N D   L A W   A N A L Y S I S — Page 6A of 6B

ID of Heat Exchanger (Type)		Working Fluid Exergy Btu/hr	Flue Gas Exergy Btu/hr	IRR Btu/hr	EFFECT PCent	Relative IRR PCent	Heat Rate Btu/kWh	Fuel Consumption Index ---
	In:							
	Out:							
FAN	In:	6185397.3	80419107.*	6184656.	92.31	.0984	7.447	.6885
	Out:	-80419848.	N/A					
AIR	In:	80419848.	964919013	.19859E9	62.69	3.159	239.1	22.10
	Out:	-414042815	-432705970					
ECON	In:	707649150	.32126E10	.54564E9	75.72	8.679	657.0	60.74
	Out:	-.24097E10	-964919013					
sprays	In:	29661044.	.35887E10	.16600E9	55.87	2.641	199.9	18.48
	Out:	-239825774	-.32126E10					
RHT-RR	In:	.23060E10	.38878E10	.13028E9	56.43	2.072	156.8	14.50
	Out:	-.24747E10	-.35887E10					
Combustion:				.2507E10	72.87	39.88	3018.	279.1
Exergy In Fuel:			.88273E10					
Exergy In Air:			414042815					
Total Exergy Out:			-.67343E10					

Legend:
IRR = Irreversibility; EFFECT = Effectiveness;
* = Shaft Power; # = Working Fluid; + = Boiler Effectiveness.

Ref. State:
Pres = 14.61000
Temp = 78.52662
Enth = 46.65643

S E C O N D   L A W   A N A L Y S I S — Page 6B of 6B

ID of Heat Exchanger (Type)		Working Fluid Exergy Btu/hr	Flue Gas Exergy Btu/hr	IRR Btu/hr	EFFECT PCent	Relative IRR PCent	Heat Rate Btu/kWh	Fuel Consumption Index ---
SHT-FN	In:	.34969E10	.39956E10	33244081	69.16	.5288	40.03	3.701
	Out:	-.35715E10	-.38878E10					
RHT-FT	In:	.24747E10	.49068E10	.32825E9	63.98	5.221	395.2	36.54
	Out:	-.30577E10	-.39956E10					
SHT-PP	In:	.34191E10	.50283E10	43693529	64.04	.6950	52.61	4.864
	Out:	-.34969E10	-.49068E10					
SHT-PD	In:	.32555E10	.52844E10	92524247	63.87	1.472	111.4	10.30
	Out:	-.34191E10	-.50283E10					
WW-1	In:	.24097E10	.60192E10	.30741E9	58.16	4.890	370.2	34.22
	Out:	-.28370E10	-.52844E10					
WW-23	In:	.28370E10	.67343E10	.29661E9	58.53	4.718	357.1	33.02
	Out:	-.32555E10	-.60192E10					
Mixing Loss:		N/A	N/A	.0000000	N/A	.0000	.0000	.0000
Stack Loss:		N/A	432705970	.43270E9	42.59+	6.883	521.0	48.17
Environment:		N/A	N/A	.0000000	N/A	.0000	-166.	.0000
Elec. Power:		-.26955E10	68242832.*	N/A	N/A	N/A	3412.	300.1
Misc. TG Cycle:		N/A	.38257E10	.1198E10	70.46	19.06	1443.	133.4
Sys. Totals:		-.69292E10	.13215E11	.6286E10	30.01	100.0	10816	1000.

Figure 2:

FCI Differences Between Bogey and Actual