ASME PTC PM-2010

(Revision of PTC PM-1993)

Performance Monitoring Guidelines for Power Plants

Performance Test Codes

The American Society of Mechanical Engineers

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Date of Issuance: April 30, 2010

The next edition of this Guide is scheduled for publication in 2015. There will be no addenda or written interpretations of the requirements of this Guide issued to this edition.

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FOREWORD

Since the original publication of these Guidelines in 1993, then limited to steam power plants, the field of performance monitoring (PM) has undergone considerable expansion. PM has gained in importance as the lifetime of equipment and power plants have been lengthened and greater demands on extending it by careful monitoring — rather than its replacement by new equipment — has become the tendency in the power industry. The techniques themselves have also been transformed, largely by the emergence of electronic data acquisition as the dominant, though not exclusive, method of obtaining the necessary information. Manual methods remain but as specialized applications. Based on the realization of the changes that have taken place it was deemed necessary to update the document itself.

The new realities of engineers and other plant personnel concerned with PM are reflected in the revised organization of the new Guide. This consists of three parts which are considered to have equal importance as regards the reader. Part 1 "Fundamental Considerations" stresses, not only by its contents but also by its separate editorial status, the importance of considering the essentials of PM prior to the specifics of the actual application. All too often lack of experience or need for rapid delivery of results has led to implementation without due thought being given to the basic needs, potential benefits and likelihood of tradeoffs of the PM program. The distinction here is in the emphasis given to the underlying importance of basic considerations.

Part 2 "Program Implementation" is a thoroughly revised and updated text of the main body of the 1993 Guide. Readers familiar with the original edition will find some of the material familiar but much that is new. The concepts of PM implementation and diagnostics have been brought into closer conjunction as is the case in contemporary practice rather than as two wholly separate aspects of monitoring activity. Similarly, the importance of cycle interrelationships have now been thoroughly recognized and so the distinction given to it in 1993 was no longer necessary; it has become an accepted part of PM implementation, in practice and in the structure of this revised Guide.

Part 3 "Case Studies/Diagnostic Examples" is wholly new. Since 1993 a large amount of experience and historical data has been accumulated and a selection is here presented. The importance of Part 3 goes beyond the illustrative although the various actual situations briefly described were chosen for their applied significance. In a larger sense, Part 3 illustrates the immense scope and variety of PM and, it is hoped, thereby makes clear the need to carefully consider the specifics of each monitoring situation. There are few general rules and many aspects particular to the plant, equipment and process to be considered. Plant's technical staffs are encouraged to learn from the experience of their predecessors in the field of monitoring and carefully scrutinize these recommendations and details as guidance to establish an optimal PM program.

This edition was approved by the Performance Test Codes Standards Committee on December 8, 2008.

ACKNOWLEDGMENTS

This revision of PTC PM Performance Monitoring Guidelines for Power Plants is dedicated to the memory of Fred H. Kindl, who passed away while this revision was in progress. Mr. Kindl was an outstanding engineer who significantly promoted the importance of power plant performance activities, a faithful member of the Committee, and a major contributor to the content of these Guidelines.

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Proposing Revisions. Revisions are made periodically to the Guide to incorporate changes that appear necessary or desirable, as demonstrated by the experience gained from the application of the Guide. Approved revisions will be published periodically.

The Committee welcomes proposals for revisions to this Guide. Such proposals should be as specific as possible, citing the paragraph number(s), the proposed wording, and a detailed description of the reasons for the proposal including any pertinent documentation.

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INTRODUCTION

This document contains guidelines for performance monitoring and optimization. These guidelines establish procedures for monitoring power plant performance parameters in a routine, ongoing, and practical manner.

These guidelines do not constitute or supersede any of the Performance Test Codes. They constitute a set of nonmandatory guidelines to promote performance monitoring activities.

The guidelines provide methods and procedures to monitor power plant and equipment performance and to validate, process, and analyze the data in order to improve or optimize unit or plant thermal efficiency, capacity, economic dispatch, operator awareness, and cycle component diagnostics, as well as to provide information for engineering studies, preventive or predictive maintenance, and planning purposes concerning equipment maintenance, replacements, or upgrades.

It is not the intent of this document that the instructions it contains be used for acceptance or official testing of new or existing power plants, systems, and components.

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PERFORMANCE MONITORING GUIDELINES FOR POWER PLANTS

Section 1
Fundamental Concepts

1-1 OBJECT AND SCOPE

1-1.1 Object

The object of these guidelines is to provide information to implement and utilize a performance monitoring and optimization program effectively. These guidelines are not intended to become mandatory for power plant performance monitoring, nor do they include all or override any safety considerations.

In performance monitoring of diverse items of power plant equipment, the uncertainty level of results may range from very small to quite large, depending on the given situation. It is important for the engineer to evaluate uncertainty and take appropriate action for meeting goals. Useful references include PTC 19.1 Test Uncertainty and the related Performance Test Codes.

1-1.2 Scope

The scope of these guidelines includes fossil-fueled power plants, gas-turbine power plants operating in combined cycle, and the balance-of-plant portion including interface with the nuclear steam supply system of nuclear power plants. The guidelines include performance monitoring concepts, a description of various methods available, and means for evaluating particular applications.

The guidelines provide procedures for validation and interpretation of data, determination of performance characteristics and trends, determination of sources of performance problems, analysis of the performance in relation to the process, determination of losses due to degradation, possible corrective actions, and performance optimization.

The guidelines provide the necessary information for implementing a performance monitoring program, using either an automated or a manual data acquisition system, or both.

1-2 OVERVIEW

1-2.1 Definition of Performance Monitoring

Performance monitoring is an overall, long-term effort to measure, sustain, and improve the plant and/or unit thermal efficiency, capacity, dispatch cost, emissions control, and maintenance planning. The program can be implemented for multiple reasons such as cost reduction, capacity improvement, and/or reliability improvements. The decision to implement a performance-monitoring program should be based on plant and fleet requirements and available resources. This includes personnel knowledgeable of the process, the instrumentation, the data collection medium, and the required analysis and interpretation techniques.

For the purpose of this document, the term "monitoring" refers to an overall, long-term, continuing program. It can range from periodic testing of individual components to on-line monitoring of all cycle components. The term "testing" refers to a specific part of the performance monitoring program.

These guidelines cover a broad range of performance monitoring techniques oriented toward power plants. They seek to advise plant personnel on how to effectively monitor the efficiency and condition of the equipment throughout its lifetime. They also extend beyond monitoring itself into the areas of information evaluation and application toward corrective action.

The guidelines are intended to meet the user's performance monitoring needs beyond the traditional Performance Test Code function or contract compliance of individual pieces of equipment. The guidelines are intended to be used only to the extent that they are practically feasible in power plant performance monitoring. The value of implementing the guidelines will vary significantly from plant to plant. The remaining life of the plant, size of the plant staff, and other resources already available will influence the degree to which these guidelines can be employed.

The guidelines are arranged by subsection in the logical order of program development and use. Following subsection 1-3, the order of the subsections is as follows:

(a) 2-1, Program Planning

(b) 2-2, Instrumentation

(c) 2-3, Performance Monitoring Implementation

(d) 2-4, Incremental Heat Rate

(e) 2-5, Performance Optimization

These guidelines assume the user has a working knowledge of thermodynamics and plant performance calculations. An overview of the most useful thermodynamic concepts is included in Nonmandatory Appendix A and is intended to provide a targeted thermodynamic review for power plant performance. It is not intended to take the place of a formal course in thermodynamics.

Other available guidance for performance monitoring includes short courses by consultants, universities, professional engineering societies, and industry research firms. Papers and texts recommended for further reading are referenced at the end of most subsections.

1-2.2 Purpose of Performance Monitoring

(a) The purpose of performance monitoring is to reduce net production costs and/or increase facility revenues. This can be accomplished by any or all of the following:

(1) improving heat rate

(2) maximizing generation

(3) increasing availability

(4) increasing maximum net capacity

(5) reducing overall net emissions

(6) optimizing maintenance activities

(7) providing information to nuclear power plant operations with respect to maintaining reactor core thermal power within license limits

(8) aiding in analysis of plant information with respect to environmental limitations

(9) aiding in operational decision-making

(b) Performance monitoring programs involve the collection and analysis of process data for various cost-benefit purposes such as

(1) providing instantaneous operator feedback with regard to controllable losses

(2) tracking controllable losses over long-time periods

(3) establishing unit heat rate for fuel accounting, regulatory records, fleet load dispatch, and/or performance comparisons

(4) determining cycle component contribution to total unit performance

(5) troubleshooting air-emissions control equipment

(6) diagnosing component condition for establishing overhaul schedule and scope and to improve ordering of parts requiring long lead times

(7) optimizing individual cycle component operation

(8) determining input/output characteristics for economic/incremental loading

In addition to the above benefits, performance monitoring provides early indication of off-normal plant conditions, allowing performance engineers time to assess and respond to events that, if left unattended or unrecognized, could lead to premature equipment degradation or equipment shutdown. Plants with effective performance monitoring programs identify trends in equipment performance earlier, have more time to plan effective responses to the information collected, and may achieve higher availability, reliability, safety, and lower production costs as a result. Effective programs make use of the engineer's early assessments, the operator's corrective actions, and timely maintenance activities.

1-2.3 Recognizing Safety

(a) When setting up for and implementing a performance monitoring program, site safety policies—including all applicable local, state, and federal laws and regulations in addition to OSHA guidelines—should be followed. Questions to consider during preparations include the following:

(1) Are plant personnel being put in dangerous situations?

(2) Will the test conditions overly stress any portions of the site systems or equipment?

(3) Could the stress caused by test conditions cause damage to equipment or the entire plant?

(4) Are there any impacts to the surrounding community that need to be considered?

(5) Will the plant be put in a situation that could result in a violation of operational safety limits or challenge nuclear safety-related systems for nuclear plants?

Before starting a performance test or monitoring program, all the appropriate site personnel should be notified of any conditions that may impact site equipment. This may include the plant management, operations supervisors, maintenance personnel, and safety manager.

(b) Some performance indicators can also identify potential unsafe conditions before they fully develop. A loss in performance may provide early warning of future safety challenges. Some examples of potential unsafe conditions include

(1) a decrease in sootblower performance, which may indicate tube wall cutting from an incorrect blowing pattern. Improper sootblower operation, nozzle selection, blowing pressures, or blowing patterns may lead to increased tube erosion and if not corrected, can result in tube failures.

(2) a deterioration in finishing superheater performance due to high-temperature creep tube damage prior to a tube failure.

(3) stage deposits in the turbine, which may precede turbine imbalance itself.

(4) seal losses, which precede turbine shaft seal cutting.

(5) cascade drains water flashing, which precedes heater damage.

1-2.4 Periodic Versus Continuous Monitoring

Performance monitoring can be periodic, continuous, or some combination of the two. The additional benefits of continuous or on-line monitoring include

(a) ability to accumulate data over time

(b) knowledge of when changes occur and under what circumstances for early recognition of impact on operation and maintenance

(c) ability to anticipate potentially serious impacts from initial indications

(d) ability to know cost of power as it is generated at all load levels

(e) opportunity to dispatch the unit based on current cost

(f) ability to track controllable losses over time

(g) opportunity for continuous optimization

(h) less labor required for automated systems than for batch systems once implemented

(i) ability to warn nuclear power plant operations of potentially unsafe conditions, such as exceeding licensed reactor thermal power limits or environmental limits

The evaluation of periodic versus continuous monitoring should include a comparison of the higher one-time capital and ongoing maintenance costs of permanently installed instrumentation and data collection equipment versus the repetitive operating costs of setup prior to each periodic test series. A compromise of the two types is to permanently install some or all of the tubing, cabling, or instrumentation used in periodic testing. The joint use of sufficiently accurate existing plant instrumentation is another consideration to be factored into this decision. The joint-use option is most economically beneficial when incorporating a monitoring program into the design of new capacity or major modifications. Periodic monitoring may be the only option available when the data or information required for analysis is not continuously stored in the plant computer data system.

1-2.5 Factors Critical to Successful Programs

Some considerations that contribute to the success of a performance monitoring program are discussed in (a) through (g) below.

(a) It is recommended to take the overall approach in specifying scope by planning to monitor all the sensitive areas of a plant rather than concentrate on those that have been historically troublesome. This affords an opportunity for full process optimization and early detection of new areas of degradation.

(b) The more complex levels of performance monitoring may require increased quantities of instrumentation. A major plateau is sufficient instrumentation to allow the calculation of an accurate flow and energy balance around the turbine cycle. The advantages of a flow and energy balance include the ability to calculate reheat steam flow, extraction steam flows, low pressure (LP) turbine shaft work, LP turbine efficiency, turbine cycle heat rate and flow factors, and further cycle analysis. An accurate flow balance may require isolation (shut-off) of all flows not measured or calculated, depending on the design of the facility.

(c) The ability to conduct monitoring at a level of detail and accuracy sufficient to establish component internal condition requires a significant knowledge of and experience with the internal operation of the specific local turbine cycle components. This knowledge and experience and management's confidence in it will result only from demonstrated competence. Competence may be demonstrated by verification of predicted condition by physical inspection, retesting, or improvement of performance as a result of a recommended operating or engineering action.

(d) If the generating units in question are involved in the bulk sale or purchase of power, knowledge of their absolute heat rate (cost) and emissions may be more beneficial than their relative ranking (which may be adequate for dispatching to meet a single company's load).

(e) If the generating units in question are marginal, in that their incremental costs are close to the predominant cost of the system in which they compete, the determination of their incremental heat rate and emissions credits may be more beneficial than if their incremental costs cause them to operate fully loaded (see subsection 2-4).

(f) Optimization programs may often be more effective when plant staff's goals and bonuses are tied to the use and results of the system.

(g) Successful programs involve operations, maintenance, and management personnel and promote awareness of what heat rate is and what affects it.

1-2.6 Typical Plant Energy Distribution

One of the keys to establishing an effective performance monitoring program is to allocate resources to the areas that provide the most benefit. A performance engineer must know the relative magnitudes of losses in power plants before priorities can be established to correct deficiencies.

Figure 1-2.6-1 shows the magnitude of losses for a typical coal-fired power plant. The overall thermal efficiency for a single reheat, supercritical cycle is approximately 36%. Losses due to the boiler, cycle, turbine-generator, and auxiliary power are roughly 11%, 45%, 6%, and 2% of total heat input, respectively. A similar figure for a gas turbine-based combined cycle plant is shown in Fig. 1-2.6-2.

Fig. 1-2.6-1 Typical Plant Losses (Courtesy General Electric Co.)

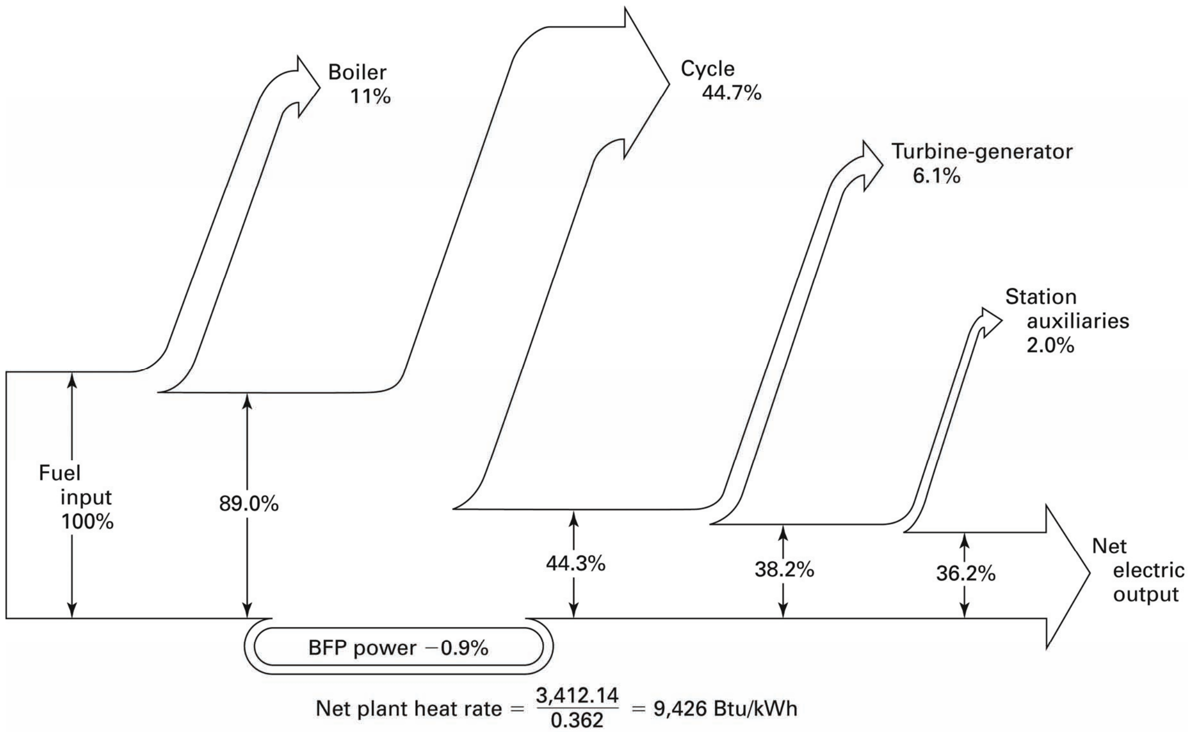
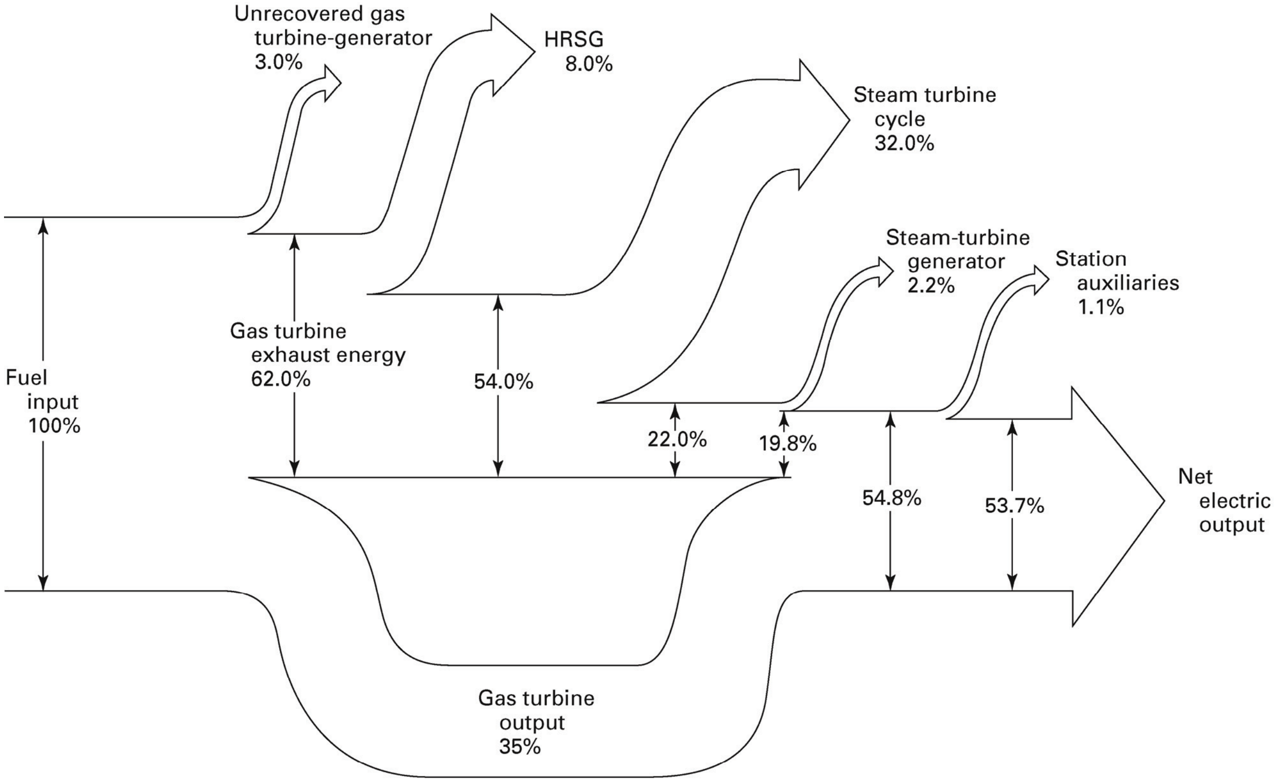


Fig. 1-2.6-2 Typical Losses for a Gas-Turbine–Based Combined Cycle Plant (Courtesy General Physics Corp.)



$$\text{Net plant heat rate} = \frac{3{,}412.14}{0.537} = 6{,}354 \text{ Btu/kWh (LHV)}$$

Figure 1-2.6-3 shows the heat balance diagram for the turbine cycle of a typical pressurized water reactor nuclear plant. The associated mass flows for the steam and feedwater system are shown in Fig. 1-2.6-4. For this plant, the gross turbine cycle heat rate is 10,256 Btu/kWh and the turbine cycle efficiency is 3,412.14/10,256 = 33.3%. Figure 1-2.6-5 shows the energy distribution.

Fig. 1-2.6-3 Heat Balance for Turbine Cycle of Typical Pressurized Water Reactor Nuclear Plant (Courtesy of Power and Energy Systems Services)

NSSS thermal power = 2,775.41 MWT
Generator output = 923.34 MW
Gross turbine heat rate = 10,256 Btu-kWh

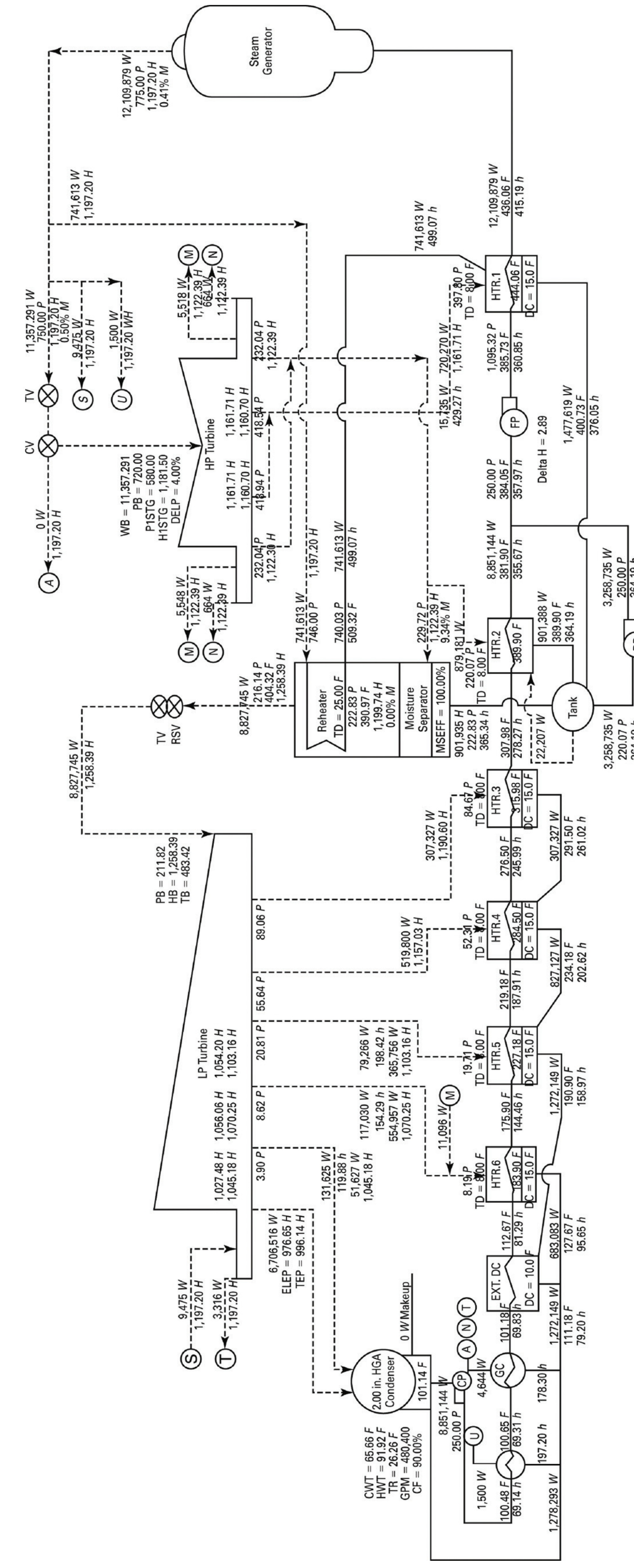


Fig. 1-2.6-4 Mass Flows Through Steam and Feedwater System for Typical Pressurized Water Reactor Plant (Courtesy Power & Energy Systems Services)

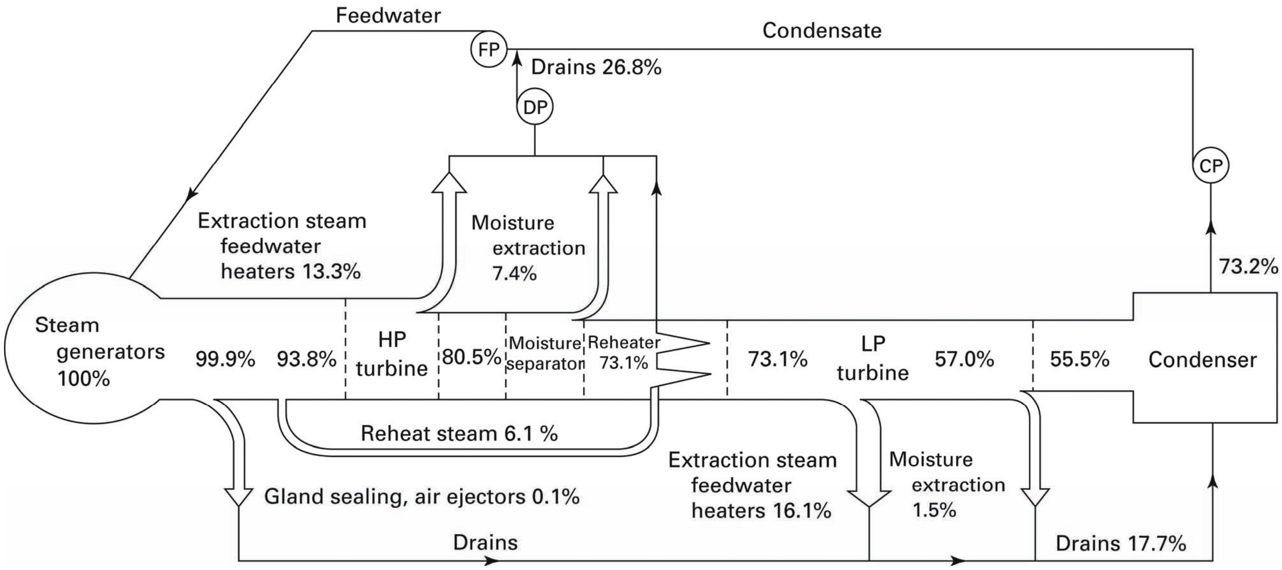


Fig. 1-2.6-5 Energy Distribution for a Typical Pressurized Water Reactor Nuclear Plant (Courtesy Power & Energy Systems Services)

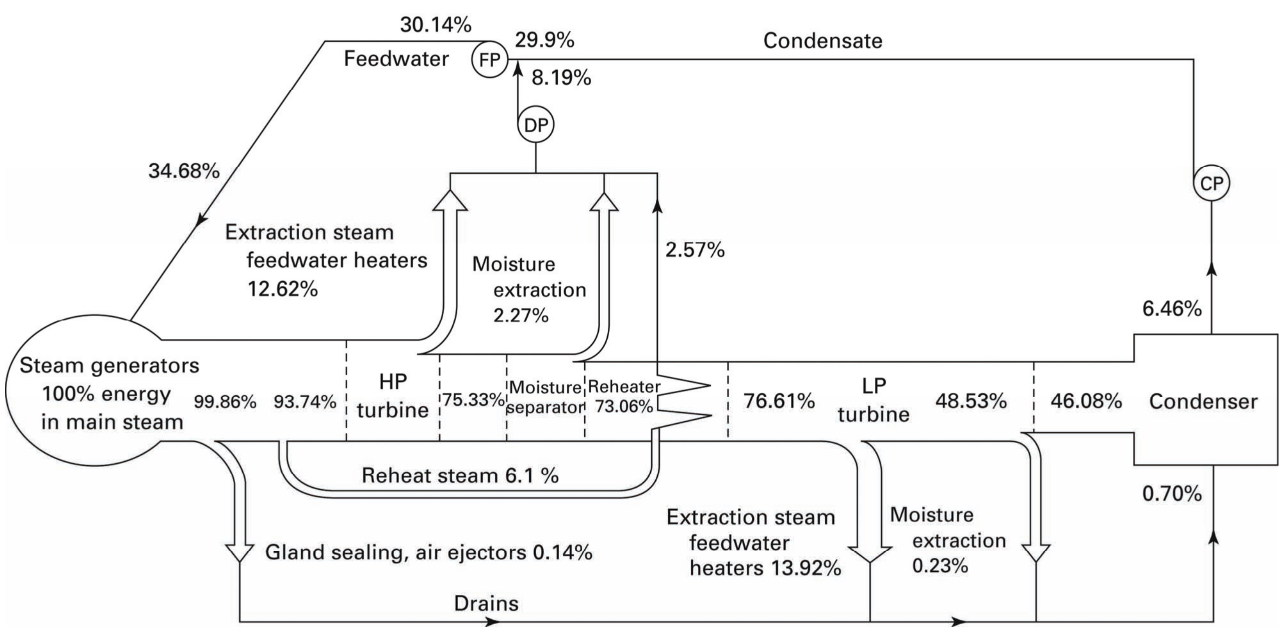


Figure 1-2.6-6 shows the constituents of the steam generator losses that account for 11% of total heat input to the cycle for a typical coal-fired unit firing a low-moisture coal such as an Eastern or Midwestern bituminous coal, and Fig. 1-2.6-7 shows the variables that make up the total losses of approximately 45% of heat input defined as cycle losses, including feedwater cycle and condenser. The breakdown of turbine-generator and station auxiliary power losses that account for 6% and 2% of total heat input, respectively, for the typical coal-fired unit is shown in Fig. 1-2.6-8.

Fig. 1-2.6-6 Typical Boiler Losses (Courtesy General Electric Co.)

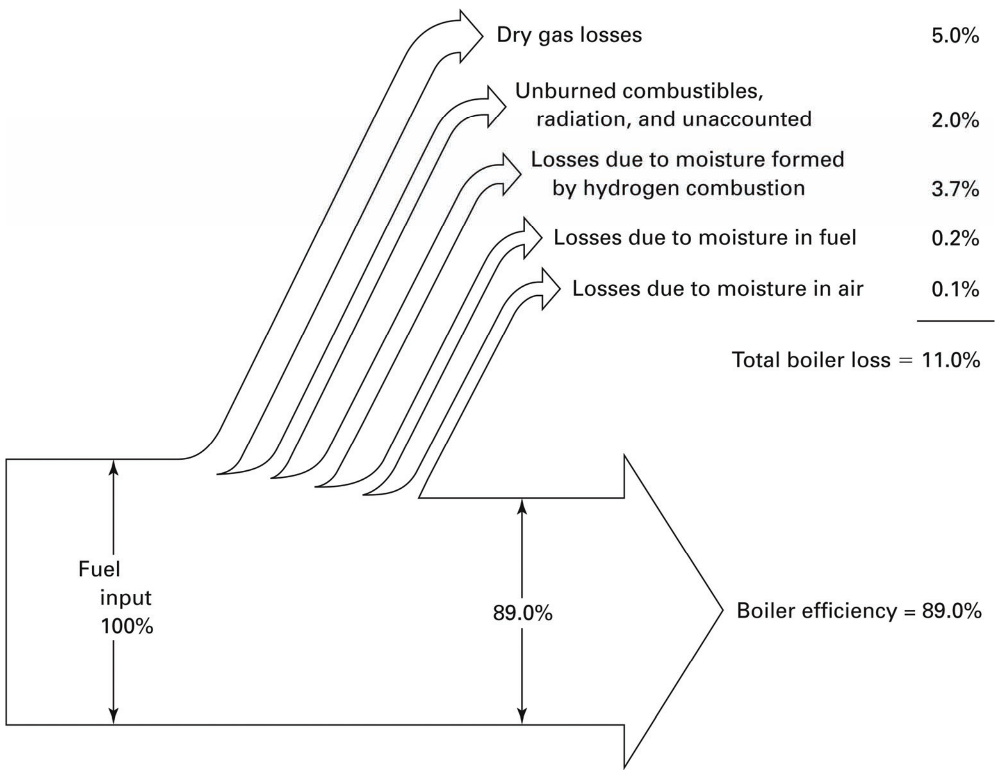


Fig. 1-2.6-7 Typical Cycle Losses (Courtesy General Electric Co.)

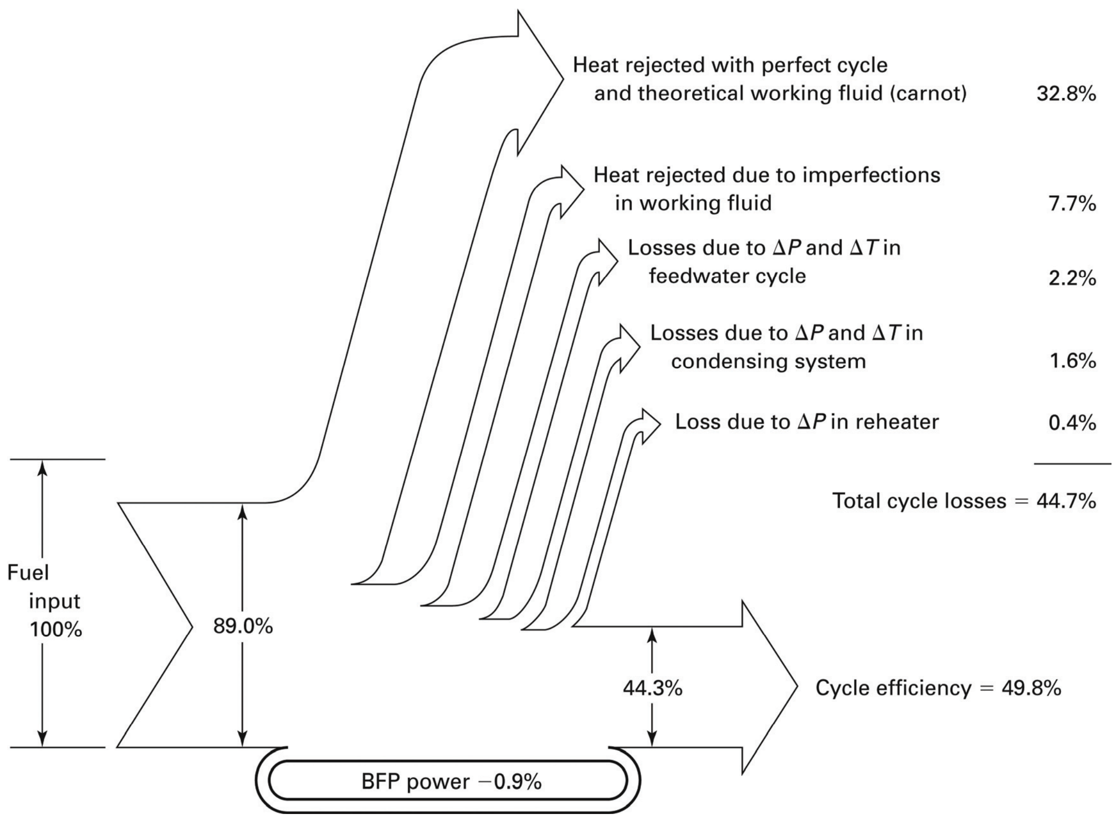


Fig. 1-2.6-8 Typical Turbine/Generator Losses (Courtesy General Electric Co.)

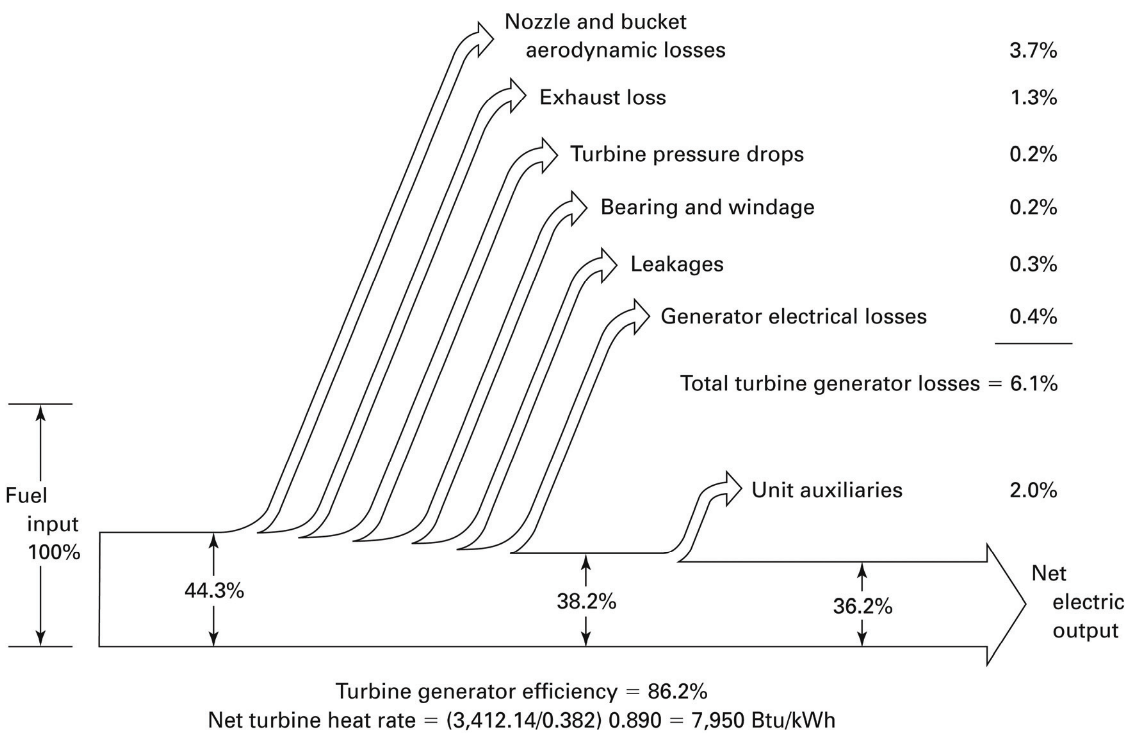


Table 1-2.6-1 Off-Design Conditions' Approximate Effect on Actual Heat Rate (Courtesy General Electric Co.)

Parameter	Change in Parameter	Change in Heat Rate
Main steam temperature	–10°F	+10 Btu/kWh
Main steam pressure	–10 psig	+3 Btu/kWh
Reheat temperature	–10°F	+10 Btu/kWh
Reheat spray	+1% (throttle flow)	+10–15 Btu/kWh
Back pressure	+0.1 in. HgA	+20 Btu/kWh
Excess O_2	+1% O_2	+30 Btu/kWh
Flue gas temperature	+10°F	+20 Btu/kWh

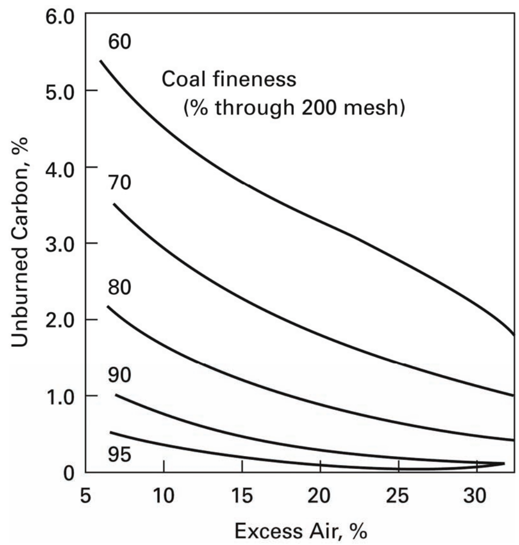
The losses shown in Tables 1-2.6-1 and 1-2.6-2 and in Figs. 1-2.6-1 through 1-2.6-8 are typical and not to be applied indiscriminately. They are affected by a number of parameters, including fuel burned, cycle design (e.g., cooling tower versus air-cooled condenser versus natural cooling source heat sink, reheat versus non-reheat, etc.), age and technology level of the unit, and ambient conditions. The losses for a particular unit need to be determined.

Table 1-2.6-2 Value of Turbine Section Efficiency Level Improvement on a Unit Heat Rate of 10,000 Btu/kWh (Courtesy General Electric Co.)

One Percentage Point On	Percent Effect On Turbine Cycle Heat Rate	Change in Heat Rate
High pressure	0.2% heat rate	–20 Btu/kWh
Intermediate pressure	0.2% heat rate	–20 Btu/kWh
Low pressure	0.5% heat rate	–50 Btu/kWh

GENERAL NOTE: Values shown in Tables 1-2.6-1 and 1-2.6-2 indicate general magnitudes of various parameters' effect on unit heat rates at VWO.

Fig. 1-2.6-9 Computed Variation of Unburned Carbon With Excess Air (Courtesy Electric Power Research Institute)



Besides knowing the order of magnitude of losses, it is essential to understand which variables are controllable from an operations point of view; which ones are controllable from an engineering perspective where equipment modifications, either by maintenance or design change, are required to effect a change in thermal performance; and which ones are controlled by nature. These performance monitoring guidelines seek to describe in detail how an effective program can be established.

There is a fourth level of loss accounting: unaccountable losses. This is the difference between the expected heat rate and actual heat rate after controllable losses, engineering change losses, and losses controlled by nature have been taken into account. Unaccountable losses are unknowns that need to be identified and addressed. Very often they are evidence of cycle isolation or instrumentation problems. Once identified, they fall into one of the other three categories.

As an example, there is a very close coupling and sensitive interaction within the steam generator cycle relating to unburned carbon, coal fineness, and excess air as shown in Figs. 1-2.6-9 and 1-2.6-10.

Figure 1-2.6-11 shows the sensitivity of heat rate to stack temperature when inlet air temperature is controlled by cycle heat (extraction steam). A good example of graphical representation of interrelationships is shown in Fig. 1-2.6-12. It shows the effect of several steam generator-related parameters. Each line can rotate

Fig. 1-2.6-10 Effect of O_2 and Coal Fineness on Unit Heat Rate (Courtesy Electric Power Research Institute)

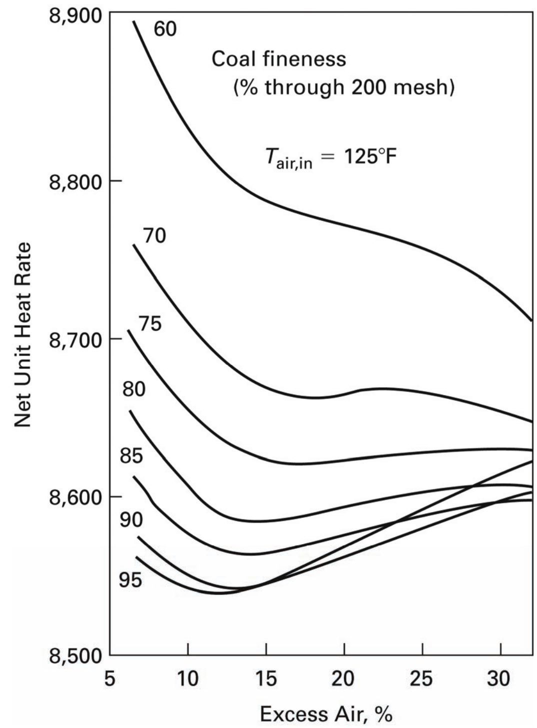


Fig. 1-2.6-11 Effect of Stack Gas Temperature on Unit Heat Rate (Courtesy Electric Power Research Institute)

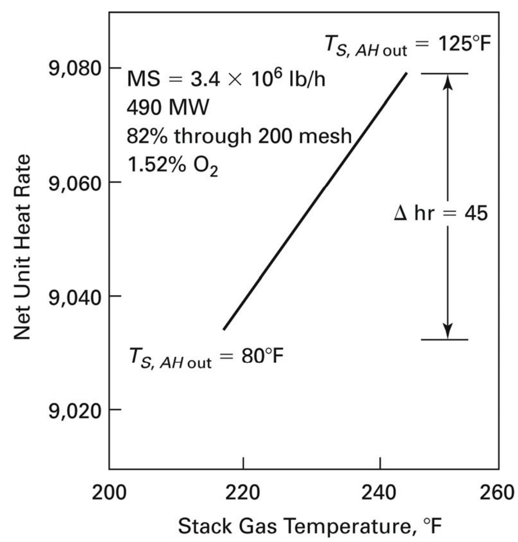


Fig. 1-2.6-12 Boiler Loss Optimization

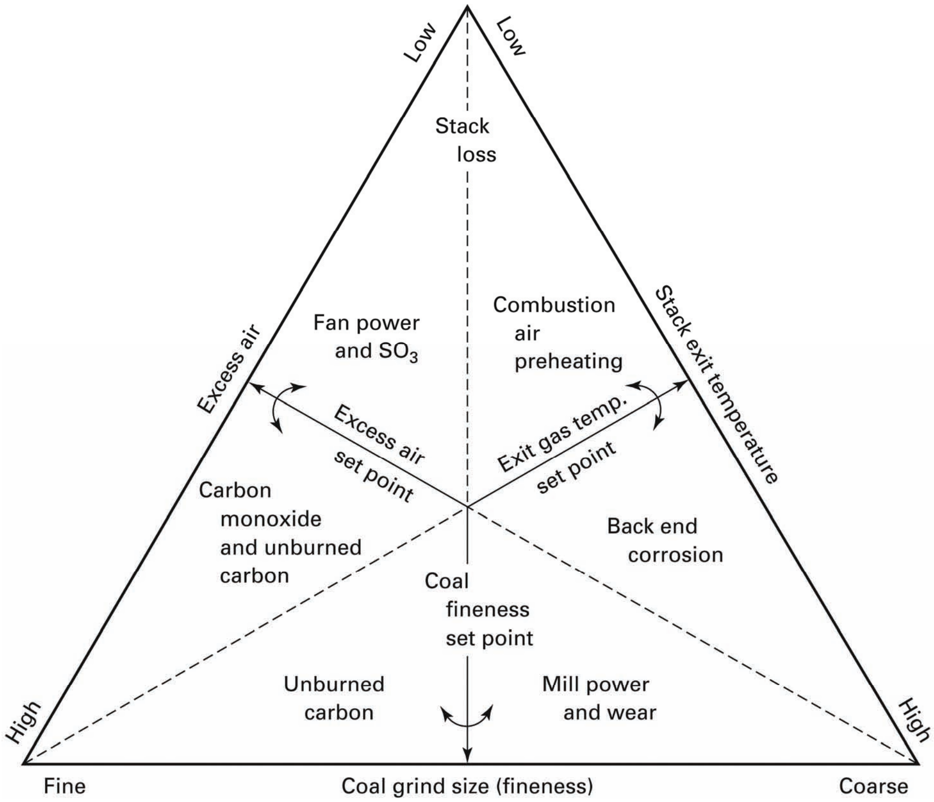


Table 1-2.6-3 Sensitivity of Heat Rate to Various Parameters for a Typical Pressurized Water Reactor Nuclear Power Plant (Courtesy Power and Energy Systems Services)

Parameter	Change in Parameter	Change in Heat Rate
Throttle pressure	+10 psig	–17 Btu/kWh
Moisture separator effectiveness	–10% points	+25 Btu/kWh
Reheat temperature	–10°F	+18 Btu/kWh
MSR cycle steam pressure drop	+10%	+6 Btu/kWh
Condenser pressure	+0.1 in. HgA	+4 Btu/kWh

about the pivot point at the center of the triangle. The arrow head can move laterally along the line it touches. The area swept by the arrow-headed line increases or decreases depending upon the direction of swing. The parameters represented by the area increase or decrease as the area changes. For example, lowering the boiler excess air below the normal excess air set point will increase the levels of carbon monoxide and unburned carbon, but will reduce fan power.

Table 1-2.6-3 shows the sensitivity of heat rate to various parameters for a typical pressurized water reactor nuclear power plant.

Given knowledge of the order of magnitude of losses and degree of controllability, priorities can and should be developed to meet overall thermal performance goals (see subsection 2-1).

1-2.7 Building Confidence in the Results

One of the most important factors in the long-term success of a performance monitoring program is that it is specified, installed, and implemented with the knowledge and care necessary to ensure confidence in the results by all parties. Extreme care should be taken to preclude loss of credibility resulting from issuance of inaccurate data or incorrect conclusions (see the diagnostic sections of subsection 2-3). Some considerations that help to ensure confidence are as follows:

(a) The most critical resource of a performance monitoring program is the personnel. Instrumentation cannot substitute for thorough analysis based on knowledge and experience. The knowledge required of both the process and the measurement instrumentation is sufficiently complex that development of expertise in the discipline requires assignments that are significantly longer in duration than those traditionally given younger engineers in corporate rotational training programs. The above, when considered together with the significant cost-benefit potential of performance monitoring, suggests the establishment of performance-oriented departments providing career path advancement opportunities and program continuity.

Direct involvement in the cost-benefit analysis of the potential savings and field implementation of projects identified by performance monitoring is a logical function of the more experienced performance personnel in an organization. This includes operational and control adjustments, revisions to maintenance scope and schedule, capital equipment additions, and unit dispatch coefficient revision.

(b) The second most critical resource is instrumentation. Selection of representative measurement locations and the appropriate specifications of pressure taps, thermowells, and flow sections should be in accordance with the PTC 19 Series and subsections 2-2 and 2-3 of these guidelines.

The performance engineer should select instruments having the necessary precision and accuracy for their intended use, combined with readout systems whose sensitivity is sufficient at the lowest loads, flows, or measurement ranges (see subsection 2-2). Instruments should be periodically calibrated at intervals that will ensure long-term accuracy. The avoidance of drift in performance monitoring instrumentation is essential in the long-term monitoring of plant performance.

The uncertainty of performance monitoring system results will vary over a wide range depending on instrument systems, economics, and expertise. It is most important that performance personnel and their management realize that the degree of uncertainty establishes an upper limit on the usefulness of the results such as the ability to establish machinery condition from machine performance.

1-2.8 Ensuring Valid Data

One measure of the validity of data is the statistical sufficiency of the data for the process conditions under which it was collected and the variation in the data as displayed by its statistical parameters (see subsection 2-3).

Another important indication of the validity of data and the calculated results is the degree of compliance with the physical laws regulating the process as discussed in subsection 2-3. This compliance as well as the analysis of a cycle and its components is most visible when monitoring is conducted over the widest range of load and flow. The shape of a given parameter's variation relative to load or flow reveals significantly more knowledge about the test data and the process than is available from a single test point.

Both of the above considerations can be implemented by utilizing a data validation program in conjunction with the performance monitoring program. Data validation routines that adjust measured data within their uncertainty bands to better conform to physical laws before use in performance calculations are becoming more widely available.

1-2.9 Making the Program Economically Justifiable

A thorough monitoring program can provide significant process performance information necessary to decisions relating to operating practice, design, and modifications. This can include capacity increases, heat rate decreases, improved reliability through decreased forced outages, decreased cost and duration of planned outages, and knowledgeable and motivated plant personnel.

In order for a monitoring program to be justified to the process managers, it must be responsive to their needs in an accurate, consistent, and dependable manner. Given that performance personnel are a critical resource, it follows that optimum use of personnel suggests use of an automated data acquisition and processing system to handle all possible labor intensive functions including

(a) instrumentation calibration

(b) data acquisition

(c) engineering units conversion

(d) data storage

(e) data averaging

(f) data sufficiency checks

(g) performance calculations

(h) steam and water properties

(i) validity checking

(j) regression analysis

(k) curve plotting

(l) graphics generation

(m) statistical analysis

(n) uncertainty analysis

(o) data retrieval

(p) data set manipulation and what-if analysis

(q) report generation

Automation of the above functions also eliminates human error and accelerates availability of recommendations to management. Systems having some or all of the above functions are available commercially or can be developed on a custom basis. Software evaluation should consider relative ease of further system expansion and modification.

1-2.10 Additional Benefits of Performance Monitoring

A thorough monitoring program can provide significant process performance information necessary for decisions relating to operating practice, design, and modifications.

(a) Examples of performance monitoring benefits include

(1) possible increased capacity if the component systems of a unit handling fuel, air, or water are not closely matched in size such that maximum unit capacity is limited by the smallest component. Optimization of the smallest component's usage can result in a capacity increase for the unit. The economic benefit to the owner of a capacity increase may exceed many fuel savings benefits and significantly enhance performance program justification (see subsection 2-3 for Cycle Interrelationships, and subsection 2-5).

(2) optimization of the procedure for starting, loading, and stopping major unit auxiliaries such as pulverizers, fans, pumps, and precipitators over the load range for power consumption. For example, if the plant design permits, the number of circulating water pumps operating may be optimized relative to the low-pressure turbine choke point to achieve savings in auxiliary power consumption.

(3) use of monitored data to calculate rates of change of temperature and/or temperature differentials in critical areas to influence operating practice toward the goal of improving equipment availability and heat rate.

(4) test instrumentation that can serve as an audit of normal plant instrumentation.

(5) clarification of the relative benefits and tradeoffs of various operating practices such as full arc versus partial arc control-valve operation and variable versus full throttle-pressure control.

(b) Typical design revision decisions that can be influenced by performance monitoring include

(1) gas turbine inlet guide vane control adjustment throughout the load curve in order to give a stable exhaust gas temperature into the heat recovery steam generator

(2) turbine control-valve operating modes, throttle-pressure set point selection, and turbine throttle steam flows and pressures for turbine retrofits

(3) establishment of the need for and benefits of boiler heat-transfer surface modifications and resurfacing requirements

(4) sootblower relocations, additions, or change in sootblower type (e.g., utilize water, steam, or air as the sootblowing medium)

(5) air preheater modifications

(6) variable speed motor drives

(7) feedwater heater replacement performance specifications

(8) condenser tube replacement specifications

(9) air preheater modifications

(10) overfiring of a gas turbine in order to generate more steam to acquire more power out of the steam turbine cycle

(11) type and size of potential combustion turbine inlet air conditioning equipment

(12) operation and setpoints of inlet air conditioning equipment when operating a combined cycle plant in a cyclic condition

(13) reduction of condenser pressure by optimizing cooling tower fan operation

1-3 DEFINITIONS AND DESCRIPTION OF TERMS

The intent of this subsection is to include terms used in these guidelines as well as additional terms of a general nature specific to performance monitoring designed to provide a basic understanding of the terminology used by the power industry.

acceptance test: a test conducted to determine if a piece of equipment meets the performance requirements of the purchase contract and is hence accepted.

accuracy: the closeness of agreement between the measured value and the true value.

air blanketing: accumulation of noncondensible gases on the steam side of heat exchanger tubes resulting in a reduction in heat transfer.

air heater: device to transfer heat from the flue gas to the air entering the boiler (recuperative or regenerative).

air heater effectiveness: the ratio of the gas side efficiency to the *X*-ratio.

air heater gas side efficiency: the ratio of the actual drop in flue gas temperature through the air heater to the maximum drop possible.

air heater leakage: leakage of air from the air side to the gas side expressed in percent of total gas flow entering air heater.

air preheater: device that controls the air temperature into the air heater so as to maintain the exit gas temperature above a minimum level.

attemperation flow: see *desuperheating flow.*

auxiliary electrical power: power used to operate the generating unit's auxiliary equipment.

auxiliary equipment: equipment needed to support the operation of the boiler, turbine, and condenser cycles.

availability: measure of a unit's ability to provide power compared to its full load capacity.

back pressure: see *turbine exhaust pressure.*

blowdown: quantity of water drained from a steam generator in a nuclear plant, the steam drum(s) in a fossil-fuel boiler, or a wet FGD system for continuous removal of impurities and sludge.

boiler air in-leakage: uncontrolled infiltration of air into the boiler through the boiler enclosure.

boiler fuel efficiency: the ratio of energy output to energy input when input is defined as the total heat of combustion available from the fuel.

boiler gross efficiency: the ratio of energy output to energy input when input is defined as the total heat of combustion available from the fuel plus heat credits.

boiling water reactor plant: type of nuclear plant that utilizes heat in the reactor directly as the source of main steam for producing power in the main steam turbines.

capacity factor: ratio of the average load on a machine for a period of time relative to the rated capacity of the machine.

cleanliness factor: ratio of the actual thermal transmittance to the transmittance at 100% clean condition.

cold leg temperature (T_{cold})*:* in a pressurized water reactor plant, the temperature of water exiting the steam generator and entering the reactor.

combustibles in ash: see *unburned carbon.*

condensate flow: flow of water from the condenser hotwell through the low pressure heaters to the boiler feed pumps.

condenser air in-leakage: leakage of air into the condenser steam side.

condenser pressure: absolute pressure on the steam side of the condenser above the tube bundles. It is sometimes referred to as *condenser vacuum* when referenced to atmospheric pressure. It may not be the same as turbine exhaust pressure.

condition-based maintenance (CBM): maintenance based on emerging failure, also known as *on-condition*, or *condition-directed*.

continuous monitoring: monitoring conducted on a uniform continuous basis, using automated data collection.

correction factors: factors to be applied to test results to correct for off-design or nonstandard conditions.

cycle isolation or alignment: the procedure used to minimize unaccounted-for flows entering, leaving, or bypassing cycle components.

data validation: process to ensure that the collected data satisfies statistical criteria and complies with the physical laws (thermodynamics, fluid dynamics, etc.) of the process.

desuperheating flow: feedwater used to control the final temperatures of the main and reheat steam flows.

economic dispatch: a method by which the loading of the units on a system is determined on a least total cost basis.

enthalpy-drop test: a test conducted to determine the turbine efficiency based on the energy removed by a turbine section.

entropy diagram: a diagram expressing entropy values corresponding to various locations in a heat balance diagram.

excess air: the amount of air in excess of the stoichiometric requirements.

excess oxygen: the percentage of oxygen present in the products of combustion. This is often confused with the term *excess air.* The terms represent different quantities and their values are not equal but are related.

exhaust loss: those losses associated with the steam exiting the low pressure turbine as a result of kinetic energy changes and pressure drops. They are usually characterized in the thermal kit provided by the turbine manufacturer.

expansion line: the locus of points on a Mollier diagram that depicts the thermodynamic states of the steam as it expands through the turbine.

feedwater flow: flow of water from the boiler feed pumps through the high pressure heaters to the boiler.

feedwater heater drain cooler approach (DCA): the difference between the shell side drain outlet and the tube side inlet temperatures.

flue gas analysis: flue gas constituents as measured on a wet or dry volumetric basis (O_2, CO_2, CO, etc.).

gross generation: total electrical output from the generator terminals.

heat balance diagram: a diagram expressing temperature, pressure, enthalpy, and flow values throughout the cycle for a given set of conditions.

heat credits: the net sum of heat transferred to the system by flow streams entering the envelope (excluding fuel combustion energy) plus exothermic chemical reactions and motive power energy of auxiliary equipment within the steam generator envelope.

heat loss method: calculation method to determine steam generator efficiency expressed in percent based on accountable losses from the boiler.

heat rate, *gross:* the ratio of the total energy input to the unit to the gross electrical generation.

heat rate, gross turbine: the ratio of the energy input to the turbine cycle to the gross electrical generation.

heat rate, incremental: the energy input change required to produce the next increment of load on the unit.

heat rate, net: the ratio of the total energy input to the unit to the net electrical generation.

higher heating value: the total energy released by the complete combustion of the fuel. This includes the heat of vaporization of all moisture.

HP-IP turbine shaft leakage: the steam leakage from the HP turbine to the IP turbine through the shaft seals of a combined HP-IP element, sometimes called *N2* or *dummy gland leakage*.

hot leg temperature (T_{hot}): in a pressurized water reactor plant, the temperature of water exiting the reactor and entering the steam generator.

incremental cost: the cost associated with the generation of the next increment of load on a unit.

input–output method: calculation method to determine steam generator efficiency expressed in percent based on the ratio of heat output to heat input.

input–output test: a test conducted to quantify the unit fuel usage versus electrical output.

log mean temperature difference (LMTD): often used in heat exchanger calculations because the temperature gradient is not constant along the length of the exchanger. Let the temperature difference of the two fluids on the A side of a heat exchanger be represented by dTA, and let dTB represent the B side. The LMTD is (dTA – dTB)/ln(dTA/dTB).

loss due to unburned carbon: heat loss expressed in Btu/lb of as-fired fuel due to unburned carbon in the ash.

loss of ignition (LOI): percent weight change when ash sample is heated to oxidize combustibles.

lower heating value: total energy released by the fuel without condensation of the water vapor in the products of combustion.

macrofouling: fouling of the cooling water flow paths caused by debris.

make-up water: water added to the cycle to replace the steam and water lost.

maximum continuous rating: the contractual maximum continuous rating (MCR) output from a steam generator.

microfouling: fouling of the condenser tube surface due to microbiological growth, deposits, or corrosion. This inhibits heat transfer through the tube walls.

moisture removal zone (MRZ): provision in turbines of nuclear plants for removal of moisture.

moisture removal zone (MRZ) effectiveness: the ratio of the moisture removed to the total moisture entering the MRZ.

moisture removal zone (MRZ) effectiveness curve: relationship of MRZ effectiveness at various MRZ steam pressures.

moisture separator effectiveness: the ratio of moisture removed to the moisture entering the moisture separator.

moisture separator reheater (MSR): device used in nuclear units to decrease the moisture content and raise the temperature of the steam going to the LP turbine.

multi-pressure condenser: condenser that is partitioned so as to operate at more than one steam side pressure.

net generation: difference between the electrical generator output and the auxiliary electrical power.

performance parameters: those variables in a cycle that can be measured or calculated that are indicative of the level of performance of a component or system.

power factor: the ratio of the true power (kW) to the apparent power (kVA).

precision: the closeness of agreement between repeated measurements.

predictive maintenance: maintenance activities that are performed based upon the prediction of failure sometime in the future. This is usually based upon past maintenance history, coupled with results from performance monitoring programs and other indicators of equipment condition. Predictive maintenance activities predict satisfactory performance until the next scheduled examination, or identify an emerging failure state.

pressurized water reactor plant: type of nuclear plant that utilizes heat generated in the reactor to indirectly generate main steam in steam generators for producing power in the main steam turbines.

preventive maintenance: maintenance activities that are performed on a scheduled basis, sometimes following manufacturer recommendations. Preventive maintenance activities are all maintenance activities performed on a scheduled basis.

output/loss method: a method by which boiler efficiency is determined by a measurement of the energy rejected in the flue gas, the combustible loss, and the boiler steam duty.

reheater pressure drop: pressure drop encountered in the reheat section of the boiler including piping.

reheater terminal difference: the difference between the saturation temperature of the heating steam and the temperature of the cycle steam exiting the reheater in a nuclear plant.

resolution: the smallest observable increment of measurement.

sequential valve (partial arc control): the operational mode to change turbine loading by which the steam flow into a turbine is governed by opening one or more control valves sequentially.

single valve (full arc control): the operational mode to change turbine loading by which the steam flow into a turbine is governed by opening all control valves simultaneously.

sliding pressure: see *variable pressure.*

special moisture removal zone: special provision in LP turbines of nuclear plants to remove moisture.

station electrical power: total electrical power used at the station. This includes auxiliary equipment electrical power and power used by support facilities (e.g., office, lighting, tank farms, etc.).

steam path audit: an audit of the turbine steam path that is used to quantify associated performance losses for each nonstandard condition. These performance losses are determined by taking detailed physical measurements of the steam path during a turbine outage.

subcooling: the temperature reduction of the fluid below its saturation temperature.

surface area ratio: the ratio of boiler heating surface areas such as superheater to reheater.

terminal temperature difference (TTD): the difference between the saturation temperature of the heating fluid at shell inlet pressure and the outlet temperature of the heated fluid.

thermal kit: a compendium of performance information, generally provided by the turbine-generator manufacturer. These include heat balances of the turbine cycle and correction curves to heat rate and load for deviations from rated values of selected performance parameters. The thermal kit is strictly intended for verification of turbine-generator contractual performance guarantees and contains several assumptions regarding components outside the scope of the turbine-generator contract. However, it does yield useful information that very often serves as the basis for designing the other components in the turbine cycle.

throttle flow: steam flow at the turbine inlet.

turbine choke point: the operating condition at which further reductions in pressure at the LP turbine exhaust flange result in no increase in turbine output for a given set of upstream conditions. This condition is typically caused by attaining sonic (choked) flow conditions somewhere within the LP turbine.

turbine efficiency: the ratio of the actual enthalpy change in the turbine to the isentropic enthalpy change (see *enthalpy-drop test).*

turbine exhaust pressure: the LP turbine exit pressure measured at the exhaust flange. This is sometimes referred to as *back pressure*. It may not be the same as the condenser pressure.

unburned carbon: carbon in the fuel that has not changed to CO or CO_2 during the combustion process.

uncertainty: the estimated error limit of a measurement, comprised of both the random and bias (fixed) components.

unit thermal efficiency: the ratio of the net generator output to the total heat input to the boiler.

valve point: the valve position just before the succeeding valve starts to open.

valve point loading: the technique of loading a unit at its valve points to maximize its efficiency.

valves wide open(VWO): the valve setting that corresponds to all turbine control valves fully open.

variable pressure operation: an operating method in which the load is changed by varying throttle pressure in lieu of changing valve position (multiple combinations of valve position may be utilized).

X-ratio: the ratio of the heat capacity of the air passing through the air heater to the heat capacity of the gas passing through it.

Section 2
Program Implementation

2-1 PROGRAM PLANNING

2-1.1 Introduction

Successful implementation of a performance monitoring project requires the development and execution of a well-defined program plan. The plan must identify operational objectives, constraints, scope and depth of coverage to be attempted, and the general technical approach. It must consider data acquisition, instrumentation, and equipment issues. It must identify resource needs and ensure the proper assignment of those resources, both financial and human. It must define and communicate roles, functions, and responsibilities. It must establish reasonable and realistic goals and schedules. It must also be flexible and able to accommodate changes in direction or priorities and unforeseen circumstances without adversely affecting progress toward the primary objectives.

The purpose of this Section is to present items and activities that should be considered during the development of the program plan. The level of detail to which each item is to be implemented is specific to the individual project. Existing organizational and corporate policy and guidelines may dictate the initiation and overall structure of the plan. Basic elements in planning a performance monitoring program should include the following:

(a) objective

(b) organization

(c) available information

(d) review of unit historical data

(e) construction of performance cause-and-effect logic trees

(f) monitoring requirements

(g) data acquisition

(h) general instrument considerations

(i) uncertainty analysis

(j) data archival and retrieval

(k) results reporting

(l) budget allocation

(m) cost-benefit analysis

(n) personnel and equipment safety

The plan needs to be carefully thought out in advance at both the general and detailed levels. The program plan must be geared towards accomplishing the identified objectives. The ultimate choice of approach that will best serve the user is a function of the objectives, the user's time frame and available resources, and other relevant factors. Information contained in this subsection is intended to help the user define the most appropriate program plan for the circumstances. One needs to recognize that the plan needs to be flexible and adaptive and that it will need reevaluation in the course of execution to maximize its effectiveness.

2-1.2 Objective

The first step of program planning is to establish a goal-oriented objective. Goals should be related to specific performance parameters. Performance parameters are those measured or calculated plant parameters that have a direct or indirect impact on performance and generating capacity. The goals should establish or enhance one or more of the following activities:

(a) efficient unit operation and high unit availability
(b) evaluation of component/cycle equipment for baseline, trending, and statistical record purposes
(c) performance optimization
(d) development of input/output dispatch curves
(e) performance problem solving
(f) timely maintenance planning
(g) performance evaluation following maintenance activities
(h) minimization of emissions

Goals should be established for each performance parameter selected for monitoring. In most cases the goal can be quantified to be a specific value or percent improvement. This is needed because the efforts must be designed to meet those objectives. For example, do the objectives involve net optimization of total production cost? Do they target a single unit, the units at one or more locations, or all the units in the system? Do they involve operational or mechanical optimization, or both? Are they aimed at efficiency or availability or both performance areas? Do they really seek improvement as opposed to optimization? Are they geared toward achievement of specific performance levels and/or cost levels? Answers to these questions will help formulate realistic goals.

Safe operation of equipment needs to be a foremost concern at all times.

2-1.3 Organization

A dedicated staff is required to carry out the objectives of the performance monitoring program. Staff personnel need to be assigned specific responsibilities and provided with a means of reporting results to management. Staffing for performance monitoring can begin at either the plant level or the corporate level [1]. This depends on whether a centralized engineering staff is in charge of overall plant activities. Two types of organizational formats are suggested: individual plant teams, or an integrated corporate-wide program. Either type should match existing plant organizational structures. It may also be appropriate to use a combined approach. A performance-monitoring team should be established to carry out the program plan, and staff positions should be defined with respect to areas of responsibility.

A considerable amount of information analysis, field and office investigative work, corrective action planning and follow-up, and other functions are necessary to make the program effective. Sufficient time must be allocated for the assigned people to properly cover these areas. If this time is not provided, and performance-monitoring functions are treated simply as auxiliary duties to other large responsibility areas, then it may be expected that the efforts may have reduced effectiveness and that program objectives may not be met.

Staffing to support the monitoring program may involve substantial cost to an organization. The specific staffing needs will vary from case to case, and should be carefully and objectively analyzed to determine appropriate assignments of people. The expected cost effectiveness of the entire monitoring program should recognize these staffing requirements and evaluate them accordingly. In any case, appropriate staffing is a vital aspect of program success, and should be given full consideration in any serious monitoring endeavor.

There is a natural tendency in performance monitoring work to concentrate on mechanical matters of equipment and units. However, there are extremely important people-related issues that affect operation and performance, and in fact, may even determine the ultimate outcome of the entire monitoring program. An in-depth coverage of the fundamentals of motivation, industrial psychology, training requirements, transition management, and the numerous other human aspects that are party to large-scope technical undertakings is not attempted herein. However, certain key human elements that should be taken into consideration in the planning, conducting, and managing of the program are included below.

(a) Upper Management Commitment. Management support must be clearly established, demonstrated, and maintained if any lasting results are to be achieved.

(b) Employee Involvement. Involving many groups, including operators, engineers, maintenance crews, and managers, in all aspects of the program will not only produce better technical and economic results than an individual or single group effort, it will also help to establish a unified team approach working toward common and mutually understood objectives.

(c) Operator Knowledge and Experience. The plant operators are of paramount importance to the monitoring program. They will strengthen the program, increase the benefits attained, and help in avoiding pitfalls and traps that may not be recognized through only engineering evaluations and management assessments.

(d) Communications. Keeping all groups with either direct or indirect connection to the monitoring program informed, from the earliest conceptualization stages through and into ongoing operation, will greatly assist understanding of and support for the undertaking.

2-1.4 Available Information

The determination of what performance information is currently available should be accomplished early in the planning stage. All available historical information relative to performance needs to be collected and centrally located. Typical sources of information include the following:

(a) records review

(b) design and as-built performance information

(c) equipment modifications that affect performance

(d) differences between design criteria and current parameters such as fuel analysis, ambient conditions, etc.

(e) results of performance tests

(f) observations of knowledgeable personnel

(g) in-service tests

(h) feed pump tests

(i) circulating water pump tests

(j) cooling tower capability

(k) feedwater heater level

(l) feedwater flow validation

(m) secondary valve leakage

(n) MSR TTD test

(o) heat exchanger tube plugging

(p) steam generator

(q) feedwater heaters

(r) condensers

(s) component inspections

(t) MSR

(u) turbine steam path audit

(v) flow nozzle

(w) vendor manuals

(x) published papers

(y) plant procedures

(z) power calculation

(aa) secondary plant ops

(bb) relating to alternate feed flow measurement
(cc) pump curves
(dd) drain pump
(ee) circulating water pump
(ff) condensate pumps (booster pumps)
(gg) feed pumps
(hh) plant calculations
(ii) power calculations
(jj) uncertainty calculations
(kk) pressure drop calculations
(ll) heat balance calculations (from A&E)
(mm) program reports
(nn) steam generator
(oo) erosion corrosion
(pp) AOV/MOV
(qq) system health reports
(rr) user manuals
(ss) plant modeling software
(tt) online monitoring software
(uu) special equipment manuals
(vv) ultrasonic flow meter
(ww) acoustic monitoring equipment
(xx) portable temperature indicators (RTD, infrared, TC)
(yy) industry contacts

Sources of information should include plant personnel interviews, design documents supplied by equipment vendors and architect-engineers, turbine thermal kit, equipment data sheets, acceptance test reports, annual test reports, routine performance testing, and industry-wide utility experience.

2-1.5 Review of Unit Historical Data

A comprehensive review of historical performance data should be conducted. The data gathered from this review should be used to establish as-built performance levels attained by the unit and associated equipment after initial startup. Determining the level of as-built performance may consist of reviewing acceptance test data, simplified baseline test data, operational startup data, and design heat balance data. Normally, more accurate baseline test data will be established following the startup period and be more representative of current performance and supersede the earlier data. Data of reduced accuracy and validity should not be used. Trending of historical data, if available, may serve as an aid in identifying problem areas. Changes in modes of operation should be noted and given sufficient consideration when sources of performance deviations are being identified. Modes of operation to be noted should include sequential valve or single valve admission, variable pressure, control valve position loading, startup practices, etc.

2-1.6 Construction of Performance Cause-and-Effect Logic Trees

Parameters that contribute to performance deviations can be identified with the aid of performance cause-and-effect logic tree diagrams [2, 3]. A logic tree is intended to be a diagnostic tool for identifying the root cause of plant performance degradation (see subsection 2-3).

The logic tree is structured to guide its user through a predetermined decision process to determine the cause of a problem by successively narrowing the problem scope based on available information. For example, a heat-rate logic tree begins with a description of the overall problem being investigated, in this case, heat rate loss. The next level identifies major areas in the plant cycle (systems, major equipment, etc.) that are potential contributors to the overall problem of heat rate loss. Typical examples are the boiler, turbine, circulating water system, auxiliary steam system, and cycle isolation. Each successive level of the tree provides more detail as to the source of the heat rate loss and is more specific than the preceding level. The tree continues until the root cause of the heat rate problem is identified. There may be more than one cause for a given symptom. Associated with each potential cause or problem of the logic tree are decision criteria. These are conditions that must be evaluated to determine if the potential cause is the actual cause of the immediate problem. In some cases, decision criteria may be based on the value of a single parameter (e.g., throttle temperature <1,000°F) or the values of multiple parameters. In other cases, the trend of one or more parameters may be appropriate decision criteria. Sometimes, more complex decision criteria are needed. These may be equations or calculations, tables or graphs of parameter values versus plant conditions, checklists of the status of various equipment, or references to tests that can be used to verify postulated problem causes. Current levels of performance for those identified contributors should be obtained from all available sources, including plant operating data, maintenance records, and outage reports. Contributors indicating deviations from expected levels should be determined using the expected levels of performance established above. The user should be aware of the source of the logic tree used, to be certain that the diagram reflects the plant's actual as-built conditions.

2-1.7 Monitoring Requirements

During the records review, information will be collected that identifies specific areas within the plant that are causing the largest degradation to unit performance. This will include availability, reliability, capacity factor, capacity, and heat rate. Deviations attributable to the following major equipment or systems should be developed, recorded, and evaluated:

(a) boiler or combustion turbine or nuclear steam supply system

(b) cycle heat rejection

(c) steam turbine

(d) moisture separator(s)/reheater(s) (for nuclear)

(e) feedwater system or heat recovery steam generator

(f) auxiliary electric power

(g) cycle alignment/isolation

(h) other balance-of-plant equipment

The results of this review of performance information will prioritize the equipment or systems to be monitored. Within each major system, subsystems may be identified to further pinpoint the areas where the initial monitoring effort should be concentrated.

2-1.7.1 Cycle Interrelationships. There are dozens of operational interactions in effect at all times on operating units resulting in significant influence on the operation and performance of those units. The list of conceivable interrelationships, operational and mechanical, obvious and subtle, is very large. It is therefore unrealistic to expect to include all possible cycle interrelationships in any performance monitoring program, even the most sophisticated. Performance monitoring personnel are thus faced with the decision of which ones to accommodate in their ongoing programs.

A monitoring envelope concept (see para. 2-3.14) is used to visualize the existence of cycle interrelationships. The envelope involves imaginary but defined boundary lines that surround the equipment component, system, or unit being monitored. Cycle interrelationships may be viewed as those interactions that cross the boundaries of the monitoring envelope. If the monitoring envelope is such that external factors are influencing performance (or indicated performance) within the envelope, then it is necessary to identify and quantitatively consider those factors and their effects in the performance monitoring processes. Applying this concept in the design and conduct of the performance monitoring program will assist the user in better meeting the monitoring objectives. It will also improve the accuracy, repeatability, and reliability of the monitoring results, by ensuring that appropriate measurement of and compensation for any important interactive effects are included in the monitoring approach.

2-1.7.2 Diagnostics. During the planning stage one must look ahead and consider how best to analyze all the data that will be collected. The goal of diagnostics is to discover the root causes for performance degradation, and to provide indications if a worsening or catastrophic performance problem is imminent. With the end in mind, the program planner can envision the inputs, instruments, software and hardware, and data acquisition equipment that are required.

2-1.7.3 Program Optimization. Optimization of resources for any endeavor requires planning. This is why the program planner must evaluate a performance monitoring program from its being up to implementation during the planning stage. In addition to optimizing the overall program, it is equally important to consider performance optimization during planning. The reader is encouraged to become familiar with subsection 2-5 during the planning stage.

2-1.7.4 Planning-Stage Questions. Responses to the following group of questions will lead the program planner to informed conclusions on which factors to include in the performance monitoring program. There is no single set of correct answers, since they will vary greatly depending upon individual needs, objectives, and circumstances. Developing valid answers to these planning-stage questions requires a reasonable understanding of performance monitoring precepts and of cycle interrelational concepts. It also requires a functional understanding of the design, operation, and general conditions of the specific equipment to be included in the monitoring.

(a) What equipment components are included within the boundaries of each monitoring envelope?

(b) For each monitoring envelope, what interrelational factors may conceivably cross its boundaries to influence the operation and performance of the equipment being monitored?

(c) For each of the interrelational factors identified, what degree of impact might it conceivably introduce into results?

(d) What is the most technically practical and cost-effective way of quantifying and incorporating each interrelational factor that could potentially have a significant impact on results?

These questions pertain to the ongoing program and to the interpretation of monitoring results, as well as to the planning stages of new programs. In either case, they provide a logical process to the consideration of cycle interrelationships, and to the sorting out of those factors that need to be incorporated into the overall monitoring requirements.

2-1.8 Data Acquisition

An important part of program planning includes determining the method(s) of data acquisition that will be used. The data acquisition method(s) chosen should allow for upgrading as new equipment and techniques become available and should also be flexible enough to accommodate additions to the number of parameters acquired should increased detail become desirable.

Data acquisition can be manual or electronic, and on-line or periodic. The objectives of the performance monitoring program may dictate the type of system. For example, information for the operators on controllable parameters should be updated frequently and will probably require an electronic system continually updating a display in the control center. However, information for the results person may be needed only periodically and to a higher level of precision, so it can be obtained by installing instrumentation for each test. Generally, use of as much electronic data acquisition as possible is recommended so that enough data is acquired over time to indicate trends. Data acquisition requirements for each performance monitoring system should be developed to meet the program objectives, and a cost-benefit analysis should be conducted to determine the number of points to be measured and the method of data acquisition.

An automated plant historian will result in gains in both plant efficiency and program effectiveness. There are many commercial systems currently available. They lend themselves to performance monitoring, problem identification, and problem resolution. The potential worth of a historian cannot be understated.

2-1.8.1 Using the Control System. The control system typically has a wealth of information useful for monitoring performance of a unit. Therefore, it may be cost-effective and convenient to extract certain performance signals from existing control system(s) for use in a performance monitoring system's data functions. Modern control systems and standardized data communication protocols have made extracting control system data more convenient and cost-effective than in the past. However, some types of control systems or plant computer systems may require upgrades to system hardware and/or software in order to provide for a computer interface that may no longer be supported by the manufacturer.

If a control system upgrade is conducted simultaneously with the implementation of a performance monitoring program, coordination of the two projects will be advantageous. When new performance instruments and plant controls are simultaneously retrofitted, proper planning will allow some transmitters, data acquisition systems, and computers to satisfy both functions in an optimum manner. The most critical consideration is that accuracy requirements be met at all points along the data acquisition chain.

2-1.8.2 Electronic Data Acquisition Components. Typical components of a data acquisition system include a sensor, signal conditioner, A/D converter, and data processor. An example illustrating a data acquisition system is temperature measurement with a thermocouple. The sensor is a thermocouple that produces a low voltage signal. The low voltage signal is picked up by the signal conditioner. The selection of the thermocouple type and other sensors is considered in Section 5.

The signal conditioner serves as an electronic link between the sensor and the rest of the system. Signal conditioners have three stages: input, processing, and output. The input stage can include amplification, measurement error compensation, noise reduction, and sensor excitation. Signal conditioners can be used to linearize the signal generated by an inherently nonlinear sensing device. Signal conditioners can also have filter circuits to reduce electronic noise. Finally, the signal conditioner produces an output signal.

For a more detailed description of data acquisition methods, the reader is referred to PTC 19.22 [4].

2-1.8.3 Installation Considerations. Consideration should be given to intermediate termination racks for input cabling to allow for future changeout of A/D hardware and computer systems without disturbing field terminations. This also affords an opportunity for test jacks, disconnect switches for calibration and maintenance, and a location for some passive signal conditioning, RTD, or other miscellaneous power supplies and thermocouple cold reference junctions. In selecting A/D hardware, consideration should be given to remote systems located in the plant near several sensors. This A/D hardware can be "smart" (engineering units conversion in the remote) or direct (where conversion register count values only are transmitted to a central CPU for further processing). The smart versions can be processing nodes on a distributed system highway or subsystems of a more traditional central CPU mainframe system.

All forms of remote systems offer the advantage of reduced cabling cost, reduced exposure to noise and interference, and unloading of the central CPU work load. PTC 19.22 discusses five types of converters and associated signal conditioning, filtering, and low level amplification [4].

Other factors to consider in selecting A/D hardware include

(a) number of inputs per A/D (one per input, one per relay card of 4 or 8, or one per input scanner system). This decision impacts system speed and calibration complexity.

(b) scan frequency (number of points/sec).

(c) variable amplifier gains (affects resolution and compatibility with various voltage levels and ranges).

(d) ease of use of on-line standards for voltage (standard cell) and resistance (precision resistor) for detecting drift or failure.

(e) conversion register bit size for determining system resolution capability [5]. System resolution should be established at the lowest operating value of a parameter.

Pressure and flow differential pressure transmitter accuracy, temperature drift, ease of calibration, and physical protection are significantly enhanced by location of transmitters in an environmentally controlled room. The added tubing expense is partially offset by reduced cabling expense. Cabling can be further reduced if transmitter power supply and A/D hardware are located in the same area. Cabling, grounding, and shielding practice should be in accordance with PTC-19.22 [4].

2-1.8.4 Master Time Base. Factors affecting data usage include the availability of a master time base in the system such that all stored data can be time tagged. This becomes more important as scan frequency is increased and transient operations are monitored. Accurate timing is necessary to cut off pulse accumulation or integration of analog rates over a fixed time period and for correct display of relationships in plots of data collected in sets such as pressure/flow relationships.

2-1.9 General Instrument Considerations

The amount of plant data readily available and its format need to be determined [6]. The goal is to accumulate a list of plant data points currently available from the data acquisition system or plant information computer. An in-plant instrument survey should be conducted to confirm the accuracy and repeatability of the measurement system.

The purpose of the instrument survey is to create and verify a current list of usable plant instruments. This list will be the basis as the instruments are checked for calibration, accuracy, and location. The survey also includes checking the scaling conversions and data signal conditioning programmed into the existing system. Primary flow pressure and temperature compensation methods should also be checked to verify that the primary readings are being compensated and that the correct flow meter coefficients are being used in the calculation procedure.

Program planning in the area of instrumentation should also include development of an adequate calibration plan and/or the utilization of a data validation program. These should identify the extent and frequency of calibrations for all instruments used in the program, so as to maintain adequate data quality. The calibration plan will vary not only between parameters, but also between instruments. Resources should be allocated in the form of technician and engineering support, and adequate calibration equipment. In addition, documentation of the calibrations should be maintained and periodically reviewed for recurring instrument problems.

Instrument installations need to be checked for correct installation practices. Instrument maintenance and calibration practices should be reviewed. These issues are discussed in subsection 2-2.

2-1.10 Uncertainty Analysis

An uncertainty analysis can be performed to determine the overall uncertainty of calculating performance parameters using the existing instruments [7, 8, 9, 10, 11, 12, 13, 14]. Uncertainty analysis is a method for calculating the propagation of instrument error and data acquisition error into a calculated result to determine the quality of the result. Conducting an uncertainty analysis consists of determining the influence or sensitivity coefficient and the accuracy of each instrument. The influence coefficient is a measure of the sensitivity of an algorithm to the error associated with a particular measurement. For example, the influence coefficient of generator output on net plant heat rate will be slightly greater than one, depending on the relative magnitudes of gross generator output and auxiliary power. For each percent error in measuring generator output there is a corresponding error in calculating net plant heat rate.

Influence coefficients are determined by performing heat balance calculations with a heat balance program. Each input parameter (temperature, pressure, flow, power, etc.) is indexed by a given amount one at a time and the heat balance is rerun. The influence coefficient is the ratio of the change in the input parameter to the change in the calculated result. If a heat balance program is not available, one can use established influence coefficients from turbine thermal kits or other publications.

Assigning accuracy values to each measurement is based on the instrument manufacturer's published data, current calibration records, and ASME Performance Test Codes and Standards such as PTC 4, PTC 19.1, and PTC 6 and its associated Report, PTC 6R. Equating the error associated with a particular measurement to instrument accuracy may be too much of a simplification. The real concern is the error in the measurement that results from installation practices, location, sampling rate, and human error.

After the influence coefficients and accuracies have been determined, the product of these two numbers, the effect, is calculated, squared, and summed with the other effects associated with the algorithm. The square root of the sum of the squares is the overall uncertainty.

The most significant outcome of an uncertainty analysis is a ranking of the instruments according to their effect on uncertainty. Listing the instruments in order of descending effect shows which instruments have the greatest effect on overall uncertainty. A rule of thumb for determining how many instruments truly affect uncertainty is the 20% rule [15]. Those instruments having an effect of at least 20% of the highest instrument should be considered for improvement.

2-1.11 Data Archival and Retrieval

With an on-line system, enormous amounts of measured and calculated data can be stored electronically. Historical data is used to troubleshoot problems and make decisions about plant improvements. Steps must be taken during program planning to ensure that historical data is being stored at an appropriate frequency and is made readily accessible.

Most commercially available data historian software packages have automated archival procedures and do not require user interaction. All measured and calculated data should be saved at a user-defined frequency and time tagged. The data historian system should provide retrieval of discrete data points for plotting versus time or any other variable. The system should also provide a means for off-loading stored data from the system to backup media.

Data compression techniques can be utilized to reduce memory storage requirements. These techniques often include assigning a deadband to variables and only storing a value when the bandwidth is exceeded. However, data resolution will become somewhat diluted depending on the size of the deadband. Therefore the deadband should be configured carefully to suit needs of the performance monitoring program. Care should also be taken when interpreting data from a discrete time in between times when data was actually stored (i.e., interpolated versus previously stored value).

2-1.12 Results Reporting

Performance monitoring typically generates a tremendous amount of quantitative and qualitative information. The effective monitoring program must reduce this information to a quantity and form suitable for decision making. The form and type of information needed will vary widely between performance monitoring objectives, and also will vary among different people within a given performance monitoring program.

Results from the data and recommendations stemming from it should be placed in the hands of the people who can act on it. Operations needs on-line data relative to operator controllable parameters. Engineers need raw data and calculated performance parameters, primarily historical information. Maintenance needs information to assist in setting priorities on maintenance activities, scheduling, and ordering parts. Management needs reports with regard to the plant and its major system summaries, trends, estimated percent improvement, efficiency recommendations, and their associated cost savings.

A team with representation from all of the above groups can be useful to communicate needs and priorities and successfully resolve problems. For example, a given problem may require Operations support to initially detect and characterize a problem, Operations and/or Engineering support to diagnose a problem to determine its root cause and solution, Maintenance support to plan and implement a solution, and Management support to coordinate and allocate resources for all of the above.

2-1.12.1 Assembling Data Results. Results reporting needs to be focused on providing the information necessary to effectively quantify those performance parameters identified in the objective. Methods of quantifying performance parameters may include absolute values, target values, and deviation from target values.

The program plan should determine whether reports are to be produced on demand, automatically, or both. Data reduction methods are discussed in subsection 2-3; however, during the planning phase, attention needs to be directed to how the output data will be assimilated and distributed. Content and format of various reports must be considered and determined.

Reporting is simply the providing of information in various forms to those groups with technical and management functions. Some reporting is needed in real time, directly to those with hands-on roles in the work. This is the case for Operations, Maintenance, and Controls personnel and at times for Engineers in the operational optimization process. Others need integrated summary information over periods of time. In general, lower levels of exact detail but higher levels of qualitative summary analysis are needed for upper management. Reduced data and results should accompany recommendations and cost-benefit analyses for operational and hardware changes. Ultimately this information may be reduced to fundamental assessments of whether the program objectives are being met, how effectively progress toward them is being made, whether the defined approaches are working or need to be modified, what performance changes are occurring, what the values of those changes are, how much they are costing to achieve, and how many people are being committed to the process.

2-1.12.2 Feedback and Follow-Up. Management controls are also a fundamental element in successful programs, providing important feedback to those directly engaged in the work; injecting resources where needed in the forms of personnel, money, and equipment; demonstrating emphasis and priority assignment to the work as warranted; and making corrections of personnel or organizational problems as necessary. Performance monitoring programs even in the best of situations are not completely automatic, self-sustaining, perpetually effective endeavors. They require and benefit from management overview of their activities, their needs, and their results. Such management controls, similarly to the need for upper management commitment, are essential elements in the successful monitoring program.

Feedback and follow-up are control functions within the monitoring process that are essential to its success. They are both technical and management responsibilities, and must be understood and executed as such by all involved parties if optimization is to be achieved.

Feedback involves examining the results of actions taken to determine if the predicted outcomes were actually achieved. Without such feedback on results, the process is very much open-ended, and the possibility exists that movement will actually be away from, rather than toward, the identified optimization objectives.

Follow-up involves taking corrective actions to keep the entire process on course. Follow-up entails a wide range of levels of action. It may be initiated directly by those with hands-on roles in either the operational or the mechanical optimization levels, or it may be directed by those with upper management responsibilities to the process, or by anyone in between. The important point concerning follow-up is that it is everyone's responsibility, and that without it, the entire monitoring program can become ineffective.

2-1.13 Budget Allocation

A budget should be prepared during the beginning planning stage to ensure that funding meets the desired program requirements. These resources include the labor and equipment needed to perform the tasks dictated by the objective. Equipment requirements include data acquisition systems, performance monitoring hardware, additional test equipment, and plant instrumentation. Another category of items to be included in the budget are software (purchased or developed in-house), heat balance programs, computerized steam tables, and other analytical software tools. In budget allocation, continued upkeep and maintenance should also be included.

It is necessary to face the issue of program cost at the same time as anticipating savings in the form of a cost-benefit analysis.

It is recommended that program costs be highlighted for management consideration in the early stages of planning. This will prevent such costs from arising unexpectedly and will allow for them to be considered, planned for, and allocated in advance of the actual need.

2-1.14 Cost-Benefit Analysis

A cost-benefit analysis should be performed to provide justification for the project. This is the most important phase of program planning. It can be a challenge because prior estimates of performance improvements are not always quantifiable.

Establish as-built benchmark performance parameters such as heat rate and capacity. This information will be used as a base for comparing off-design plant values and from that, determining avoidable cost estimates. The difference between current and best achievable performance is a measure of potential improvement.

Improvements in performance can be accomplished by operational changes, equipment changes, or both. Equipment changes can be further categorized as refurbishment to as-built condition or redesigned to new specifications.

Capital costs associated with equipment changes are then compared to economic gains attributable to performance improvements. There is some uncertainty associated with being able to credit potential gains in performance to new equipment. It may be necessary to selectively perform a sensitivity analysis along with the cost-benefit analysis. Cost savings based on operational changes are harder to quantify. Usually, performance improvements resulting from operational changes have to be estimated and the sensitivity of the parameter on overall operating costs has to be considered [16].

The final required element is the continued cost justification of the program itself. This is chiefly a management function involving the continual reassessment of how both human and financial resources are being allocated, weighed against the actual needs for and values from the allocation being done. Periodically, it is warranted to globally examine performance levels of the organization, to consider the general state and value of those performance levels, and to assess the total cost of the program itself.

It is advisable to separate the cost-benefit analysis for the justification of the performance monitoring program from those for the individual projects involving equipment upgrade of improvement. For the planning stage, the performance monitoring program will be justified on the expected benefits resulting from the program and from the possible changes to operational procedures. Subsequent to the implementation of the program, cost-benefit analysis will be performed to justify the individual improvement projects as they arise. Any additional requirements of monitoring and maintenance to sustain the improvement projects will be justified together with the projects.

Provided that these reviews indicate the program costs to be less than the total net value of the actions the program is initiating, then there is clear economic justification for the program's continuation. If program cost, however, appears to exceed net value, then the program itself needs to be closely examined. Many different options may be appropriate, ranging from strengthening and reinforcing the program to make it more effective, to minor adjustments in approach or structure to adapt it better to the current needs, to significant curtailment of the monitoring and optimization efforts. The latter should be considered an extreme measure, only appropriate under conditions of very low marginal gains that are hopefully tied to very high levels of performance already being achieved.

2-1.15 References

[1] EPRI Heat Rate Improvement Guidelines for Existing Fossil Plants, CS 4554, RP 1403-3, May 1986.
[2] EPRI Research Project 1711-2.
[3] EPRI Research Project CS-1832, May 1981.
[4] ASME PTC 19.22-1986, Digital System Techniques, Section 5, Sensor Signal Conversion, ASME, New York, NY.
[5] Ibid., Section 9, Data Management.
[6] Performance Monitoring With Plant Instrumentation, POWID 32nd Annual Power Instrumentation Symposium, May 1989.
[7] Gerhart, P. M., and Jorgensen, R., "Uncertainty Analysis: What Place in Performance Test Codes," ASME paper 84-JPGC/PTC-9, 1984.
[8] ASME PTC 19.1-1998, Test Uncertainty, ASME, New York, NY.
[9] ASME PTC 6 Report 1985, Guidance for Evaluation of Measurement Uncertainty in Performance Tests of Steam Turbines, ASME, New York, NY.
[10] Rousseau, W. H., and Milgram, E. L., "Estimating Precision in Heat Rate Testing," Journal of Engineering for Power, (73- WA-PTC-2), July 1974.
[11] Thrasher, L. W., and Binder, R. C., "A Practical Application of Uncertainty Calculations to Measured Data," Transactions of ASME (75- WA-PTC-l), February 1957.
[12] Sigurdson, S., and Kimball, D. E., "Practical Method for Estimating Number of Test Readings Required," Transactions of ASME, July 1976.
[13] Wyler, J. S., "Estimating the Uncertainty of Spatial and Time Average Measurements," Journal of Engineering for Power, (74-WA-PTC- l), October 1975.
[14] Kinney, W. F., Strandberg, W. A., and Kuchan, N. R., "The Effect of Data Errors on Performance Computing," Proceedings of the American Power Conference, 1963.
[15] Davidson, P. C., Gerhart, P., and Sotelo, E., "Uncertainty Analysis and Steam Generator Testing," JPCC, (86-WA-PTC-l), October 1986.
[16] Harmon, J. M., Napoli, J., and Snyder, C., "Real-Time Diagnostics Improve Power Plant Operation," Power Engineering, November 1992.

2-2 INSTRUMENTATION

2-2.1 General

A performance monitoring program requires instrumentation of appropriate repeatability and accuracy to provide test measurements necessary to determine total plant performance, component efficiencies, and other performance indices. The benefits afforded by a performance monitoring program are not obtained without careful selection, installation, and diligent maintenance of instrumentation.

This subsection provides guidelines for the selection, use, and calibration of instrument equipment for performance monitoring. This subsection references Performance Test Code documents for readers who would like greater detail about certain instruments. Because of the wide scope of this document, it is not possible to examine or describe all possible measuring devices or systems; however, the following guideline focuses the user on the more important aspects, allowing the user to prioritize any instrumentation upgrades that might be needed.

Performance is rarely measured directly. Temperature, pressure, and flow differential pressure are measured, and the required result is calculated as a function of the variables. Errors in measurements and data acquisition are propagated into the uncertainty of the resulting answer. This is why instrument accuracy and repeatability are critical to a performance monitoring program.

2-2.1.1 Accuracy. The degree of accuracy appropriate for performance monitoring should be assessed on a case-by-case basis, but the selection of instruments should be influenced by

(a) cost consequences of failure to detect performance deterioration

(b) cost to restore or improve unit performance

(c) sensitivity of results to small absolute change in measured parameter

(d) instrument cost relative to value of information

2-2.1.2 Repeatability. Repeatability is the ability of the measurement system to produce results of the same output/input relationship over extended time periods. Repeatability is essential in periodic monitoring so that variations in performance indicated by test results are true indications of the deterioration of the equipment tested.

Uncertainty for various measurement systems, such as those described in para. 2-1.10, is generally expressed as a single value including both the random repeatability (scatter) and bias (fixed) uncertainty components. For long-term trending, the random component or repeatability is of interest. Bias errors will affect overall uncertainty but have little effect on repeatability. Increasing the calibration accuracy of an instrument will reduce the bias component and instrument uncertainty but may not improve its repeatability. Caution should be used in applying any statistical techniques without sufficient knowledge of the relative importance of the bias and random error components. A thorough description of uncertainty analysis is beyond the scope of this document, and the reader is encouraged to consult the references listed for subsection 2-1 and for this subsection.

2-2.1.3 Data Validation. In addition to regular periodic calibration of all instruments, particularly those providing numerical values for key parameters, a process should be in place to ensure valid data. Several methods exist for data validation, but the automated systems are preferred over the manual or batch methodology. Vendor-based software packages can be installed to periodically or continuously monitor the reasonability of the data. The actions of these calibration monitors include identifying the instruments supplying erroneous or inconsistent data up to and including substitution of temporary "more valid" data.

2-2.2 Measurement of Electrical Output

Electrical output level can be determined by measuring the power generated using wattmeters or electrical energy generated using watthour meters. The most common uses for this measurement are in determining turbine cycle heat rate, unit heat rate, incremental heat rate, steam rate, and unit capacity verification. The gross electrical output and auxiliary power consumption are usually measured separately. The net output is then determined by difference.

Electrical power measurement involves much more than selection and installation of meters or transducers. The power measurement involves a system that includes the generator, instrument transformers, connecting wires, all other devices in the same circuit, and the wattmeter. All of these elements can affect the indicated power, thus greatly complicating power measurement. In addition, there are two different methods of connections for a three-phase generation system that can generally be described as three-phase, three-wire connector with no neutral, and three-phase, four-wire connector with neutral (see Fig. 4.1, ASME PTC 6 Report). Unfortunately, these different connection methods require different measurement approaches. Typically, for economic reasons, instrument transformers are omitted from one phase. The power measurement is theoretically correct if the phase voltages are balanced.

2-2.2.1 Unit Watthour Meters. Three-element watthour meters used for unit generation accounting generally have a ±0.50% to ±1.0% uncertainty (see Table 4.4, PTC 6R). This type of device is intended to sum large quantities of energy over long periods of time, so its indications change very slowly. Seemingly small integrator reading errors (for total energy) or small timing mistakes (for rate) can cause very large errors in power measurement. Great care must be exercised if data is manually collected. One very effective manual method is to photograph the time and integrator together at the beginning and end of each test. Often this type of meter can be fitted with a photoelectric pickup so that data can be collected with an automated system. Resolution with this approach can be as little as one-sixth of a disk rotation.

2-2.2.2 Power Transducers. High accuracy two-phase and three-phase transducers that indicate kilowatts or integrate kilowatt-hours are available with uncertainties of ±0.2%. These devices use the same signals as the typical panel watthour meters. In fact, many are designed to replace the electromechanical measuring devices and will fit within the existing case. These transducers can have both analog and pulse output, which allows easy automation. This device generally adds an insignificant burden to the circuits.

2-2.2.3 Indicated Electrical Output. This type of electrical output measurement is usually the least accurate (approximately ±5%). Indicated readings are tempting to use because of their convenience; however, most are intended to allow plant operators to approximate the load on the unit. These indicators may be used in performance monitoring only when more accurate measurements are not available or accessible in a convenient manner.

2-2.2.4 Power Factor. Power factor is needed along with hydrogen pressure and purity to calculate electrical losses in the generator. Power factor has a measurable effect on heat rate if it deviates far from design.

2-2.2.5 Other Considerations. Electric power is not measured in the primary circuit because of high current and voltage levels. Instrument transformers are typically used to step down maximum currents to 5 A and maximum voltages to 120 Vac. Note that typical secondary voltages are $120/\sqrt{3}$ Vac. The use of instrument transformers unfortunately introduces additional unknowns into the power measurement. The complex impedance of the secondary circuit and resulting phase angle shift affect the output of the transformers. PTC 6 requires calibration of the instrument transformers for high accuracy. This is not necessary for performance monitoring if the transformers have been installed properly. However, knowledge of the circuit is required. If the complex impedance has changed on the secondary circuit, the apparent reading can be affected.

2-2.2.6 Calibration. All power measuring devices should be bench checked. Some electronic transducers are standards, which means that they may be better than standard bench instruments. For high-accuracy power measurements, all three phases should be measured independently with dedicated calibrated instrument transformers. The secondary circuit burden should be measured so that full correction can be applied (see PTC 19.6 or IEEE 120).

The following potential transformer (PT) and current transformer (CT) correction should be applied to the power output for each phase as measured on the secondary side of the transformer:

(a) PT secondary voltage drop from the transformer to the metering instrument

(b) PT ratio correction factor as obtained from the Farber plot using PT secondary burden power factor (PF) and volt amps (VA)

(c) PT phase angle correction factor, calculated using the phase angle *w* (in minutes) obtained from the Farber plot with the above values of PF and VA

(d) CT radio correction factor (FCF) as obtained from the manufacturer-supplied CT ratio correction curve

(e) CT phase angle correction factor as obtained using the CT phase angle correction curve

Since the measurement of CT secondary burdens is risky (potential unit trip) and CT, RCF, and phase angle curves are not always available, it is recommended that a value of 1.0000 be used for the CT correction factors. The error introduced using this value is negligible.

After the above corrections have been applied to each phase, the generator outputs are summed and the total test generation (primary side) is computed by multiplying the PT and CT turns ratio times the total secondary side generation.

2-2.3 Measurement of Steam and Water Flow

Flow calculation should be based on the average of the square roots of the differential pressures. There are additional calculations in flow measurements that are not linear. It is common error to combine flow element corrections to create a constant that, when multiplied by the square root of the differential pressure, produces a flow rate. Area corrections, viscosity, compressibility, and the discharge coefficient (which is a function of the Reynolds number) should be calculated individually for the temperature and pressure at the time of each differential pressure measurement. The uncertainty in the resultant flow calculation should be considered if these corrections are not performed.

Unfortunately, most installed plant flow metering devices are not sufficiently accurate to serve as the primary flow measurements. Accurate and precise flow measurement requires the flow element to be installed in a location where the flow is free of swirl and other disturbances. To ensure necessary flow measurement repeatability, the element must be installed in a straight section of undisturbed pipe. Long radius elements are preferred due to their tendency to be less affected by fouling. In addition, a flow straightener is also recommended at the inlet of the flow section to minimize nonuniform flow profiles arising from nonideal upstream pipe configurations. Also, high accuracy flow elements generally produce large differential pressures due to low beta ratios.

To install a new primary flow meter into an existing header, several factors must be considered to ensure the accuracy of measurements and not interfere with plant operation. The requirement of a straight section of pipe with flow straighteners, and a low beta ratio element, means that significant net head loss may result. Use of recently developed low loss flow straighteners and recovery cones may minimize head loss. If a new element is built to replace an existing plant element, its design should include multiple tap sets to accommodate all required services. ASME PTC 19.5 contains an extension discussion of flow measurement techniques and calculation methods.

Differential pressure measurements shall be made with one instrument. Utilizing two separate pressure gages doubles the uncertainty and usually entirely masks the expected value, while often causing negative (physically impossible) results. The water leg correction for differential pressure measurements is only applied when the process connection taps are located at different elevations. The water leg correction applied in this case is equal to the difference in elevation between the taps.

2-2.3.1 Primary Condensate or Feedwater Flow Measurement. The Performance Test Code PTC 6 requires very accurate determination of primary flow to the turbine. The surface finish of the nozzle, flow straightener, thirty diameter length, inspection requirements, and consistency of the discharge coefficient with code limits make this type of element very expensive. This expense is easy to justify for acceptance testing or for an effective performance testing program. Remember that a 1% error in flow causes a 1% error in calculated heat rate and/or NSSS thermal power.

Most of the strict requirements of Performance Test Codes are to ensure precision and accuracy. Unfortunately, when costs are cut on the primary flow measurement installation, the accuracy and precision are lowered. For performance monitoring, the straight pipe section and flow straightener are the most important factors that will ensure repeatability of the flow indication. The effect of insufficient straight lengths can be estimated from PTC 6R.

Location can have a significant effect on the cost of the flow section and the additional instrumentation required to calculate heat rate and/or NSSS thermal power. If the primary flow element (see Fig. 2-2.3.1-1) is located at the boiler inlet (final feed), generally only a few instruments around the last feedwater heater are required to calculate feedwater flow. However, at final feedwater temperature and pressure, the element must be made with heavy walled, welded pipe section. This makes inspection impossible unless the nozzle is built with an inspection port, as described in Fig. 2-2.3.1-2.

Locating the primary flow element in the condensate line permits the use of flanges, allowing easy removal for inspection. However, it is also necessary to collect additional heat balance data around every feedwater heater downstream of the flow element, to calculate throttle flow. For a detailed discussion of the advantages of condensate flow versus final feedwater flow, see ASME PTC 6S.

The location of the primary flow measurement should be carefully chosen. The best location requires the least number of additional flow measurements. Any other flows that are used with the primary flow should be additions to this flow (such as reheat or superheat sprays) and not redundant measurements. For example, if

Fig. 2-2.3.1-1 Primary Flow Section for Welded Assembly

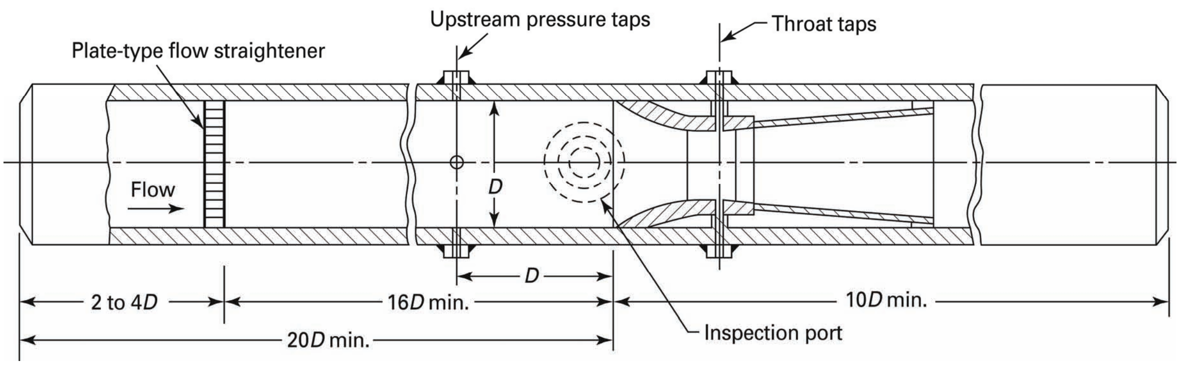
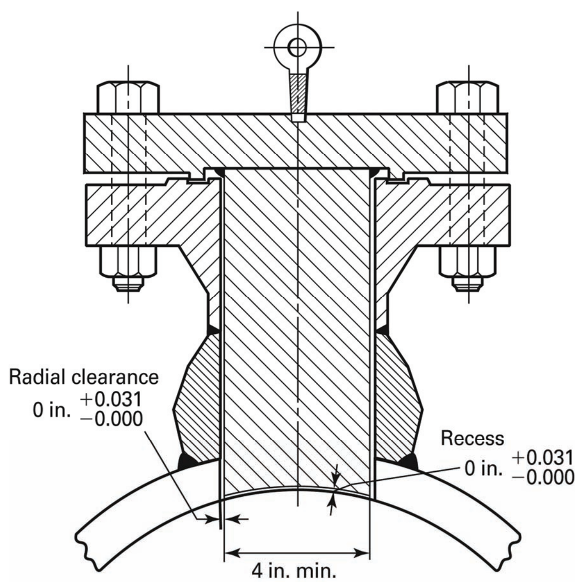


Fig. 2-2.3.1-2 Inspection Port



GENERAL NOTE: The orientation of the nozzle on the pipe is determined by the designer.

two boiler feed pumps have their own flow elements, these measurements would not be expected to be as accurate as the primary flow measurement. It is doubtful that the sum of the flows measured at the feed pumps will equal the primary flow measurement. These indications could be used carefully to bias the percent of the primary flow through each pump, but not as an actual pump flow.

Nuclear plants typically rely upon differential pressure devices to measure feedwater flow. The most common is a venturi meter chosen because of relatively low head loss as the fluid passes through the device. A major disadvantage, however, is the susceptibility of this device to fouling of the throat area causing the meter to register a higher differential pressure and, consequently, a higher flow rate than actually expected. This in turn leads to calibrating the nuclear instrumentation high, and while this is conservative with respect to reactor safety, it results in lower electrical output and loss of capacity.

To address the fouling issues as well as reduce the flow measurement uncertainties, significant efforts have been made in recent years by the power industry to find alternative means of flow measurement. One device that has been approved by the Nuclear Regulatory Commission (NRC) is an ultrasonic flow measurement device consisting of an electronic transducer that is controlled by a computer and is not susceptible to fouling due to the lack of differential pressure elements. With ultrasonic flow measurement devices, the uncertainty in feedwater flow measurement is approximately 0.5%, but the actual value is plant-specific.

2-2.3.2 Secondary Steam-Water Flow Measurements. PTC 19.5 contains discussion of proper installation of secondary flow elements. A straight pipe section and a low beta ratio are still important. In cases where the secondary flow has a significant impact on calculated heat rate (i.e., desuperheating superheat spray or steam heating coils), flow element calibration should be considered.

2-2.3.3 Valve and Packing Leak-Off Measurements. Forward reverse tubes and orifice plates are sometimes included in turbine piping by vendors. The possible influence on the calculated result should be considered before these measurements are pursued. Note that most small diameter lines have low choked flow limits; therefore, the maximum worst leak scenario will most likely have a very small effect on heat rate.

2-2.3.4 Indicated Flow Measurements. This type of flow indication is the least accurate. These devices are installed to allow plant operators to know approximate flow rate.

2-2.3.5 Circulating Water Flow Measurements. Pitot tubes can be used to accurately measure circulating water flow, provided there is a straight accessible section of circulating water pipe (typically found in power plants using cooling towers) that has been fitted with proper traverse valves. The valves should be installed so that two multipoint traverses can be performed. A multipoint traverse is essential to obtain a good flow measurement. Dye dilution and tracer techniques can also provide good flow measurements. A circulating water flow measurement can provide a good independent check of unit energy streams. However, it requires skilled labor, is time consuming, and is not reasonably automated.

Another method is flow calculated with the differential pressure drop measured from the condenser tube sheet outlet to the circulating water pipe. This measurement can be calibrated in situ with pitot tube or tracer method, and can then be used as an on-line relative water flow measurement. This flow can be used to determine total heat rejection from the turbine cycle combined with circulating water temperature rise.

For power plants with once-through cooling, circulating water flow rates may not be measured accurately due to lack of instrumentation. The flows may be determined by calculating the condenser heat rejection from an energy balance in the turbine cycle in conjunction with the circulating water temperature rise.

2-2.3.6 Other Technologies. Relatively new technologies offer other flow measurement options. Ultrasonic flow meters offer a nonintrusive and portable flow measurement method. Laser-based flow measurements offer another method.

Coriolis type meters represent a relatively new technology compared with positive displacement and differential pressure liquid flow measurement equipment. They operate on the principle of torsional strain being induced on a vibrating tube (or tubes) as liquid flows through it. The degree of torsional movement is directly proportional to the mass flow rate of liquid passing through the tubes. Electronic sensors measure the torsional displacement continually, and generate proportional signals that are then converted to indication of instantaneous mass flow. These indications may be applied directly as instantaneous, real-time flow measurements and may be integrated to determine total mass flow over periods of time.

2-2.3.7 Additional Considerations. Because of the inherent pressure pulsation and normal flow variation, a greater frequency of differential pressure measurements is required. For instance, PTC 6 requires differential pressure readings every minute of a 2-hr test period, while all other temperature and pressure readings are required only at 5-min intervals. Remember that most flow calculations are based on pounds per square inch differential (psid), not inches of water. Great care must be exercised when using water manometers. The pounds per square inch value of an inch of water is a function of that water temperature. With transmitters, use the manufacturer's conversion in inches of water to pounds per square inch differential.

While properly instrumented heat balances around feedwater heaters provide excellent calculated mass flow determination, it should not be assumed that the same is true for balances around boiler steam attemperator spray stations. It is recommended that superheat and reheat spray flows be measured with calibrated flow elements. While heat balances around attemperators can work well when the steam is superheated, the slightest spray water impingement upon the outlet thermo-well can a cause a large error in the calculation.

Some flow measurements are often not available as a measured value and must either be estimated, neglected, or isolated. These include steam drum blowdown, sootblowing extraction steam flow, auxiliary steam extraction, and others.

2-2.3.8 Calibration. Unfortunately, nozzles and venturi discharge coefficients are very sensitive to small variations in diameter, surface finish, and pressure tap construction. Because of this, it is very important that all major flow meters be calibrated before installation. Generally, it is not possible to calibrate large flow elements at high Reynolds numbers. Because of this, it is necessary to extrapolate the calibration curve to obtain the discharge coefficient for the range of use. The extrapolation must be based on the expected nature of the device and industry experience. PTC 19.5 and PTC 6 describe methods of extrapolation. For a detailed discussion of the uncertainty of uncalibrated flow elements, refer to PTC 6R, para. 4.15.

In some nuclear power plants, ultrasonic flow meters are used for primary feedwater flow measurements and to calculate NSSS thermal power. These meters are also used to validate the indications from feedwater flow nozzles. Ultrasonic flow meters therefore can be used to continuously measure feedwater flow.

2-2.4 Measurement of Pressure

Pressure is customarily selected as one of the properties measured to determine thermodynamic state of a simple fluid. Pressure is also used as an indirect means of measuring velocity and flow rate. PTC 19.2-1987 defines pressure as a force per unit area exerted by a fluid on a containing wall with respect to a reference. The pressure reference used may be the same fluid as in a differential pressure measurement, ambient pressure as in gage pressure, or zero pressure as in absolute pressure. Figure 2-2.4-1 graphically displays different pressure reference methods.

Selection of the proper pressure measurement instrumentation is highly dependent on the type of measurement and accuracy desired. Figure 2-2.4-2 displays the general accuracy of commonly used instruments. The following instruments can be used to measure pressure in performance monitoring.

2-2.4.1 Transducers. A pressure transducer converts pressure to an electrical signal, making it useful for continuous performance monitoring applications. A high-accuracy transducer is ideal for minimizing manpower requirements in performance monitoring programs. These instruments are available in various types: strain gage, variable inductance, force balance, diaphragm, Bourdon tube, and optical sensors. They can be constructed to measure differential, gages, or absolute pressure. All are easy to use, but their accuracy may be affected by the prevailing ambient conditions (see PTC 6R, Table 4.16, for uncertainty discussion).

2-2.4.2 Manometers. Manometers are typically used on low to subatmospheric pressure measurements and differential measurements across flow elements. They have an inherent high accuracy and are considered primary standards. As with the deadweight gage, the manometer requires the data taker to take readings directly and is not easily automated.

Fig. 2-2.4-1 Basic Pressure Terms From ASME PTC 19.2

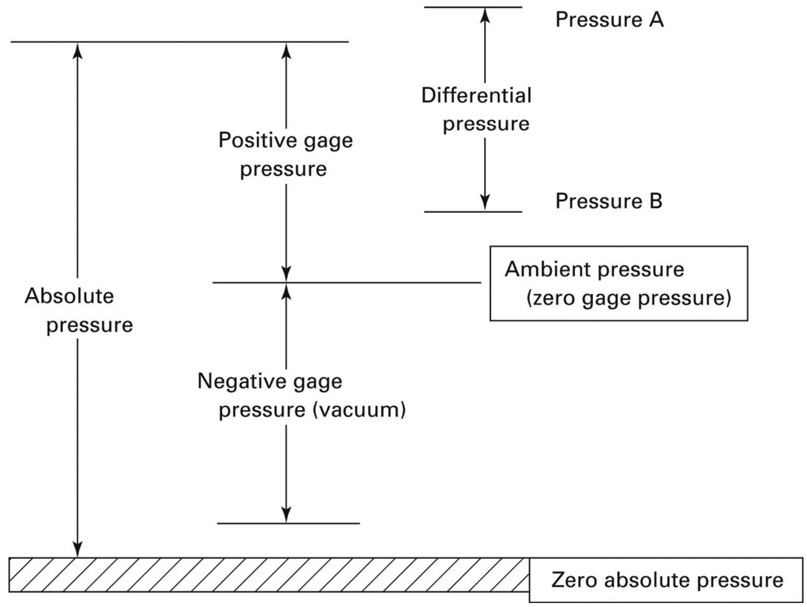
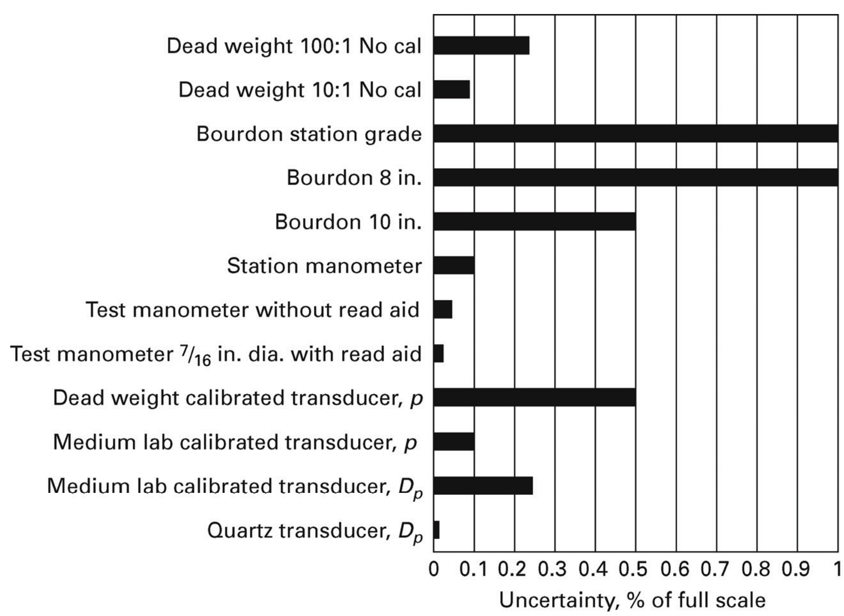


Fig. 2-2.4-2 General Uncertainties of Pressure-Measuring Devices From PTC 6 Report



2-2.4.3 Pressure Gages. Direct reading pressure gages are commonly used by plant operators to monitor power plant equipment. These gages may be of the Bourdon tube, diaphragm, or bellows type, with the Bourdon tube being the most common. These instruments are also commonly used in performance measurements where the ±1.0% accuracy readings are acceptable; however, they must be calibrated in place to obtain this uncertainty.

2-2.4.4 Deadweight Gages. The deadweight gage is an accurate pressure measurement device that is good for steady state measurements. This is an excellent calibration or check device. It is not practical for use on measurements where pressure fluctuates appreciably or in a dirty environment that could cause the piston to stick. Its main advantage over other instruments is its accuracy of up to ±0.1% uncertainty. A disadvantage is that the data taker must adjust the instrument to take the desired reading, thus requiring more manpower and a longer test period.

2-2.4.5 Considerations in Pressure Measurements. Adding new pressure measurements to an existing system is expensive when penetration of high pressure and temperature piping is necessary. Often, existing pressure taps with new tubing run to new transmitters can serve dual needs with proper precautions. Root valves at the pressure taps plus separate shutoff and blowdown valves for each branch in the line are required.

Regardless of the pressure measurement instrument selected, attention must be given to how the instrument is installed and operated. Particular attention must be paid to elevation differences in the source of the pressure and the instrument. Except for low pressure measurements, the instrument should be located below the centerline of the process. The line connecting the pressure sensing instrument is usually filled with fluid, causing the instrument to read high or low depending on the relative location of the instrument to the source. When measuring steam pressure, a column of liquid (water) will form in the sensor tubing outside the process, above the instrument. This water leg must be accounted for when determining absolute pressure used for the calculations.

To minimize the possible errors contributed by water legs

(a) minimize the length of sensor tubing between the process and the instrument.

(b) ensure the water leg is periodically blown down to ensure that no air or bubbles exist in the sensor tubing.

(c) measure the difference in elevation between the process connection and the instrument.

(d) consistently and correctly apply the water leg correction to all steam pressure measurements (i.e., subtract the pressure equivalent of the water leg from the measured pressure). (In practice, since all instruments should be placed at or below the process, the water leg is applying additional pressure to the instrument causing a falsely high reading.)

If the measured pressure is below atmospheric, as is the case for some LP turbine extractions and exhaust, ensure that the sensor tubing is sloped toward the process and periodically vented to remove trapped moisture. No water leg correction is applied to the measurement of pressure below atmospheric.

In summary, liquid-filled lines should be routed from the source to the instrument in such a manner that the line continuously slopes downward, and a low point drain should be available for purging the line. The opposite is true for gas-filled lines. Gas-filled lines should be routed from the source sloping upward, with a vent located near the top of the line, just before the instrument.

Pressure transducers are generally quite sensitive to ambient temperature changes. Because of this it has been necessary to either control transducer temperature through careful location or environmental control. However, with new technology, transducers are now available with accurate onboard corrections for temperature and static pressures. Pressure measurement on a condensing turbine exhaust must be carefully placed and use basket tips. The steam velocities in a condensing turbine exhaust annulus can affect the static pressure reading. To prevent this, cages or baskets are put on the tip of the pressure line to prevent velocity-caused static pressure errors. Unpredictable water legs may collect in exhaust pressure instrument-sensing lines that necessitate frequent manual or automated purging.

Consideration of the goals of the monitoring program will yield useful pressure values. Often pressure data alone provides little useful information when working with incompressible flowing fluids. A good example of this is multiple pumps operating in parallel between common inlet and outlet headers. Instrumenting the inlet and outlet of each pump will provide no more information than instrumenting the headers, unless pump speed, flow, and shaft horsepower measurements are also taken (and with high energy pumps, temperature rise). The largest pressure rise is not necessarily associated with the best performing pump. A lower flow rate can yield a larger rise in pressure.

Pressure accuracy requirements should be analyzed on a case-by-case basis. The influence of pressure errors and bias on turbine isentropic efficiency is not constant. To demonstrate this point, a theoretical bias and error plot of a high pressure (HP) turbine with a base efficiency of 83% at VWO is shown in Fig. 2-2.4.5-1. The bias lines are examples of efficiency change due to a 10 psi shift in both throttle and cold reheat pressures. The error lines are for shifts in the cold reheat pressure only. Note that a 10 psi bias causes about a ±0.8% shift while a ±10 psi error causes a ±1.2% change.

In Fig. 2-2.4.5-2 there is a similar analysis of an intermediate pressure (IP) turbine. Notice that for the same bias amount, ±10 psi is now worth ±3%, while the ±10 psi error in exhaust pressure is now worth ±4.6%. This indicates that identical pressure measurement errors and bias can have triple the influence on IP turbine isentropic efficiencies as they do on an HP turbine.

2-2.4.6 Calibration. It is critical that all the pressure transmitters are calibrated and that they are also maintained on a routine basis. The ideal calibration is done in a calibration lab. However, this introduces environmental effects and would not include the field signal cable or electronic measuring element in the calibration. In addition, removal and reinstallation of the transducer can subject the device to undesired stresses. Therefore, in place, source to indication is the recommended calibration method.

Fig. 2-2.4.5-1 Effect of Pressure and Bias Errors on HP Turbine Efficiency

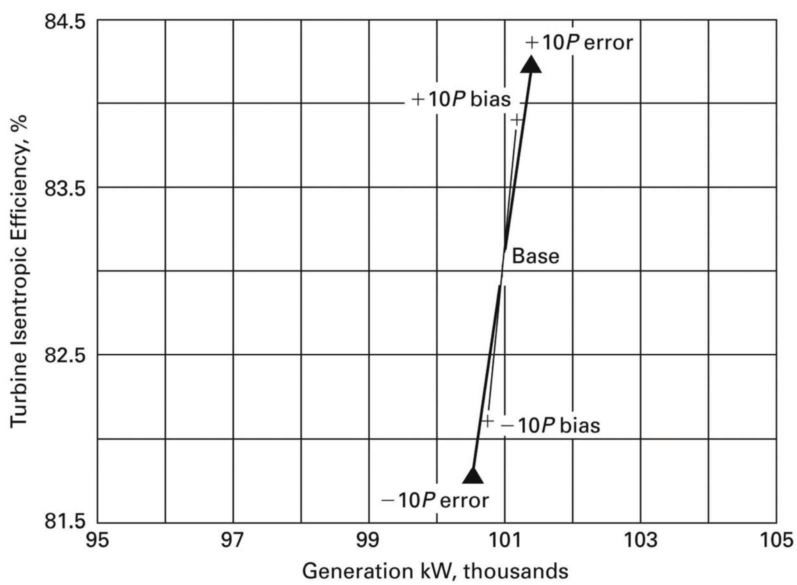
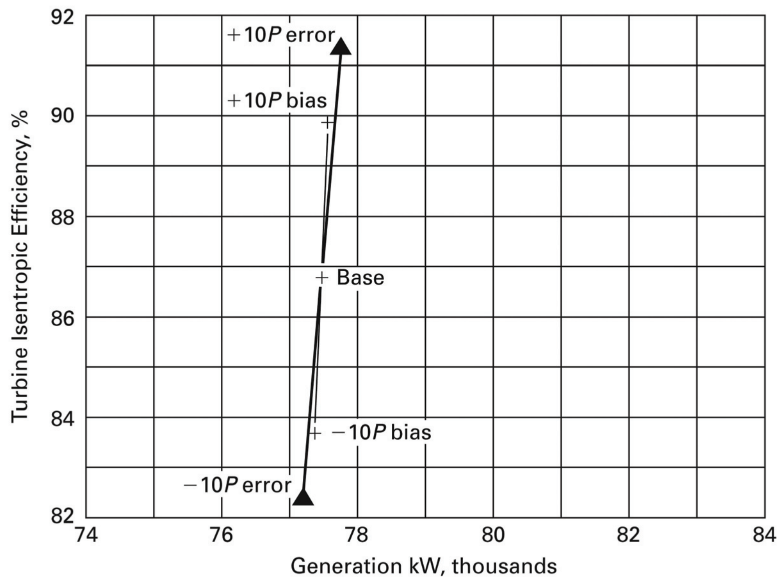


Fig. 2-2.4.5-2 Effect of Pressure and Bias Errors on IP Turbine Efficiency



Pressure transducers typically have zero and span adjustments. Traditional calibrations employ manual adjustments that actually change the output signal of the transducer. An alternative approach may be considered instead. As-found or correction curves should be developed for each transducer. Correction curves can be as simple as a linear offset to as complicated as a high order polynomial curve-fit (see PTC 19.22). This approach places correction responsibility on the software instead of in the unquantifiable adjustment of the signal transducer. This approach allows close numerical evaluation of drift and necessary calibration cycle times.

2-2.5 Measurement of Temperature

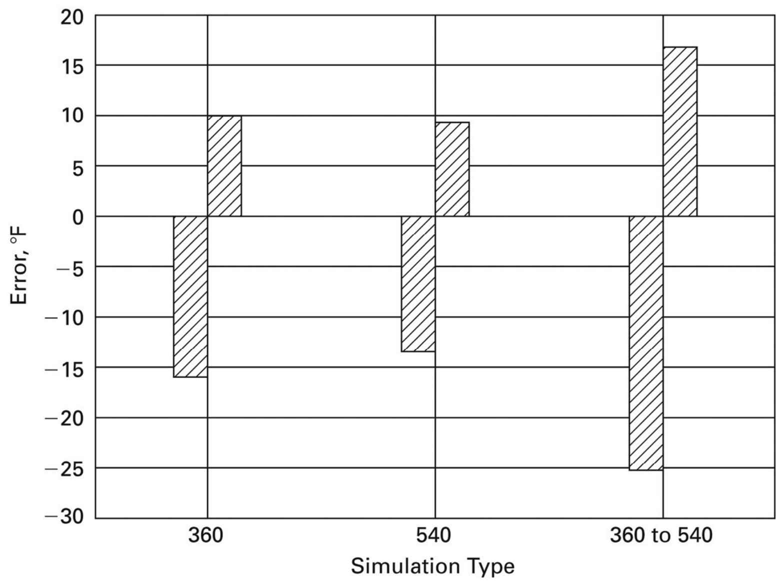
The selection of the appropriate temperature measurement instruments for performance monitoring is not difficult if the requirements for the measurement are known. All types of temperature instrumentation have inherent advantages. The summary of instruments below covers most of the temperature instrumentation commonly used in power plants for performance monitoring.

2-2.5.1 Thermocouples. The thermocouple is a temperature sensing element consisting of dissimilar electrical conductors electrically insulated from each other except where joined to form junctions. There are two junctions at the extremities of a thermocouple. The measuring junction is that which is subjected to the temperature to be measured, and the reference junction is that which is at a known temperature, usually an ice point or ambient temperature. Factors affecting thermocouples are the reference junction, the thermocouple extension lead wires, the method of manufacture, and the long-term stability of the thermocouple.

The accuracy of a thermocouple depends on the type, the temperature at which it will be used, and its calibration. Continuous lead thermocouples offer the most accurate readings because they are not subject to additional junctions. For some applications, extension leads offer a less costly alternative without influencing accuracy. Multiple readout systems where thermocouple signals are jumped or paralleled can be done with care but are not recommended. Some types of thermocouples corrode in certain environments, and some thermocouples have very low or no signal at low temperatures. Type E or Type K are preferred because of their higher signals in the temperature ranges typically experienced in power plants.

Thermocouples can be used over a very wide range of temperatures and can withstand more abuse than resistance temperature detectors. They can even be spotwelded to boiler tubes or a turbine case and still function correctly. Thermocouples are excellent for flue gas temperature measurement grids. Spools of wire

Fig. 2-2.5.1-1 TC Drift Study of Six Thermocouples Cycled 210 days to 300 days (Courtesy ISA Services, Inc.)



can be obtained with leads inside a heavy stainless steel sheath that can withstand high gas temperatures. Installed thermocouples drift as they age, necessitating recalibration or replacement. The rate of drift is related to the range and variation of temperature to which the couple is exposed (see Fig. 2-2.5.1-1).

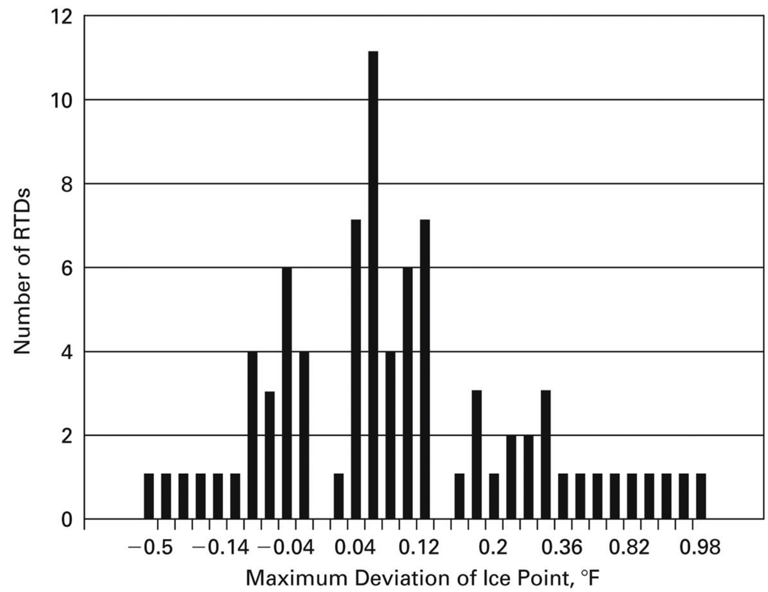
2-2.5.2 Resistance Temperature Devices. A resistance temperature detector (RTD) is a conductor of a known resistance that should be subjected to a constant current. Since the resistance of most metals varies with temperature, the resistance of an RTD can be directly correlated to a temperature with a mathematical equation. Unlike a thermocouple, an RTD does not require a reference junction. RTDs generally cost more than thermocouples and are more subject to failure, but are more accurate and linear, and operate at a higher signal level than thermocouples. RTDs are available in three types of material: copper, nickel, and platinum, with the latter being the most precise. They are also made with two, three, and four lead wires. The four-wire method minimizes the effects of extension leads. Figure 2-2.5.2-1 displays results of a cycling drift study.

A thermistor is also a temperature sensitive resistor. Thermistors are extremely nonlinear. The temperature resistivity curve for a thermistor is exponential in shape and has a negative temperature coefficient. PTC 19.3-1974 indicates that a 20% resistance variation is possible from device to device. However, thermistors are now available that usually match manufacturers' calibration curves with operating ranges up to 212°F. Thermistors have a limited range of applications; however, for temperatures less than 212ºF, thermistors are very sensitive and typically have temperature coefficients of –2 Ω/ºF.

2-2.5.3 Considerations in Temperature Measurements. The cost of additional thermowells can be avoided by the use of dual or triple element sensors, such that existing indicators or controls can have separate signals. Devices can be built with different types of thermocouples or RTDs, or an RTD-and-thermocouple combination can be combined in a single sensor. Thermocouples can be joined in series to form what is generally called a thermopile. Thermopiles can produce larger signals than a single thermocouple at the same temperature.

Many plant flow streams exhibit stratification in temperature, velocity, or chemical composition. Examples include condenser cooling water, boiler gas outlet, and air heater outlet. Unfortunately this stratification is not constant, may be significant, and varies with changes in unit operating conditions. Ideally, flow should be measured at each temperature point and used to weight the temperature; however, this is not

Fig. 2-2.5.2-1 Drift of Ice Point Resistance of 102 RTDs Cycled 810 days (Courtesy ISA Services, Inc.)



practical for continuous systems. The most cost-effective measurement solution is to use a grid of sensors located to provide a representative average of the cross section. Where possible, selection of a mixed point downstream would require fewer sensors to measure a representative value.

High accuracy temperature measurements in major inlets and outlets of turbines are very important. Temperatures should be measured at both exhaust ends of a double flow turbine, if the steam is superheated. Sometimes a significant change in turbine performance will be shown by a change of only a few degrees over time that can be obscured when temperatures are combined.

The influence of temperature errors and bias on turbine isentropic efficiency is not constant. To demonstrate this point, a theoretical bias and error plot of an HP turbine is shown in Fig. 2-2.5.3-1 with a base efficiency of 83% at VWO. The bias lines are examples of efficiency change due to a 10°F shift in both throttle and cold reheat temperatures. The error lines are for shifts in the cold reheat temperatures only. Note that a 10°F bias causes about a ±0.5% shift while a ±10°F error causes a ±3% change.

Figure 2-2.5.3-2 displays similar analysis of an IP turbine. Notice that the same bias of about ±10°F remains worth ±0.5%, while the ±10ºF error in exhaust temperature is now worth ±4%. This indicates that identical temperature measurement errors increase about 1% from high pressure to intermediate pressure. Also, in this example, a consistent shift in inlet and outlet temperatures (bias) does not appear to be very harmful to accuracy while device-to-device differences (errors) can have a significant effect.

Temperature measurements of compressed water (feedwater) or superheated steam (throttle) are very useful; however, wet steam or mixed phase data can be very misleading. For instance, condensing turbine exhaust and lowest pressure feedwater heater extraction temperatures are of little or no use.

The method of signal measurement is also very important. The ability of the power supply to provide constant current or voltage for an RTD is critical. Likewise, stable reference junction temperature is critical for accurate temperature measurement with thermocouples (see ASME PTC 19.22).

2-2.5.4 Calibrations. All temperature measuring devices should be checked. If high accuracy is desired, detailed calibration should be performed. Calibration cycles should be based on experience and the need for accuracy.

Fig. 2-2.5.3-1 Effect of Temperature Bias and Error on HP Turbine Efficiency

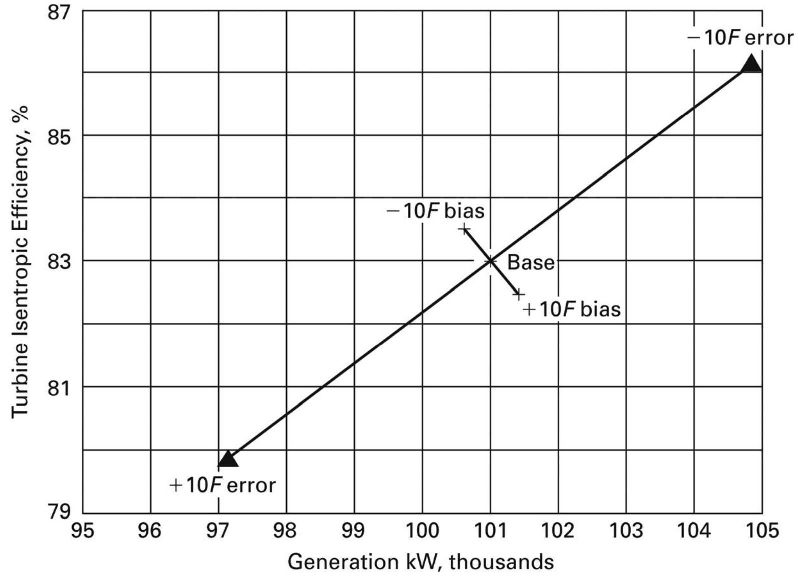
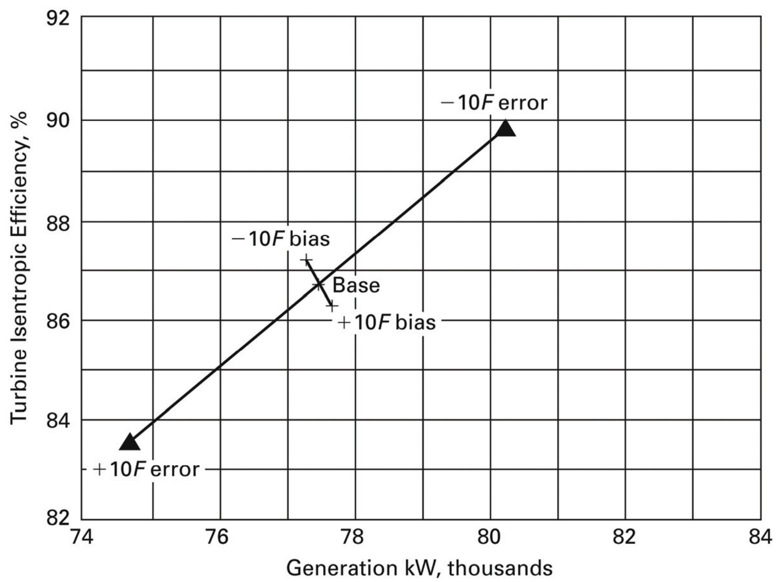


Fig. 2-2.5.3-2 Effect of Temperature Bias and Error on IP Turbine Efficiency



2-2.5.4.1 Thermocouple Calibration. It is recommended that sample thermocouples from each lot be checked against reference tables (see PTC 19.3). The average error or deviation should then be applied to the whole lot. When the highest accuracy is desired, every thermocouple should have its own correction table developed.

2-2.5.4.2 Resistance Temperature Device Calibration. All RTDs and thermistors should be calibrated. If the application does not require high accuracy, a simple check of the ice point resistance can determine if the device has drifted from its expected value. For high accuracy applications, a multipoint calibration should be run on each RTD. For some applications, a linear fit is acceptable. For ranges within the calibration range, a third order polynomial works very well. For the highest accuracy, the RTD should be calibrated with melting point standards. The data from the calibration run should be used to develop the calendar coefficients with the International Temperature Scale of 1968 (IPTS-68) corrections. For more details on the calendar equation see PTC 19.3. For temperature measurements that are read through the control system, it is good practice to compare the reading in the control system or data acquisition system with the local reading at the instrument by disconnecting the thermocouple from the DAS and connecting it to a portable reading device, then comparing the two readings at steady load. Errors and offsets may be found that should be corrected.

2-2.6 Measurement of Air and Flue Gas Flow

Air or flue gas flow rates are needed for stack loss, air mover acceptance tests, air heater tests, pulverizer tuning, and routine operations. In each case, the flow is measured by a different method. Some accurate methods involve no actual measurement of gas flow but depend on calculations that are functions of other flow measurements. The most accurate methods require multiple-point, three-dimensional traverses of ductwork as outlined in PTC 11. Some plant controls utilize gas flow elements; however, these are generally not concerned with actual gas flow rates but with relative or percent of total design. Regardless of the method, accurate gas flow is very difficult to obtain quickly.

Gas or air flow rates would allow continuous fan power analysis and boiler efficiency calculations without fuel flow or heating value measurements.

2-2.6.1 Permanent Direct Measurements. Some air and flue gas measurements are made directly with a flow element. The elements can be as simple as a fixed pitot tube or a full duct air foil or venturi. Sometimes these devices have been calibrated, or a simple duct modification will allow an in-service calibration with a pitot traverse (para. 2-2.6.2).

2-2.6.2 Direct Manual Measurements. Manual flow measurements are required for ASME PTC 11 fan acceptance testing. This method provides the most accurate gas flow measurement possible. A five-hole probe can resolve the total velocity vector.

2-2.6.3 Inferred Measurements. ASME boiler heat loss method as described in PTC 4 can be used to make an inferred gas flow calculation. The method uses lab analysis of fuel burned and flue gas analysis applied to stoichiometric combustion calculations. This method depends on an accurate fuel chemical analysis and determination of heat absorption. PTC 34 describes an additional method to measure boiler performance essentially using the boiler as a calorimeter.

2-2.6.4 Measurement of Gas Turbine Exhaust/HRSG Flue Gas Flow Rate. Gas turbine exhaust gas flow is required to support gas turbine and heat recovery steam generator diagnostics. Several methods are available to measure or calculate this flow. These include

(a) use of the calibrated gas turbine inlet scroll as a flow element for air flow to the machine. Flows calculated on this basis are often reported by the gas turbine control system. Exhaust flow then is the sum of this air flow, gas turbine fuel flow, and any steam or water injection flows.

(b) energy balance around the gas turbine as described in PTC 4.4, Gas Turbine Heat Recovery Steam Generators.

(c) energy balance around the heat recovery steam generator.

Very often, the uncertainty of this measurement can be reduced by utilizing more than one of the above mentioned methods (e.g., use of both gas turbine and HRSG energy balances).

2-2.7 Measurement of Fuel Flow

Accurate fuel flow measurement is important, but not always achievable. Whether the fuel is gaseous, liquid, or solid, the measurement of each type of fuel is vastly different.

2-2.7.1 Liquid Fuels Flow Measurement. Positive displacement meters are often used in the measurement of totalized liquid fuel flow. Their principle is quite simple. Usually, this device consists of a meter body with a rotating vane mechanism inside. This mechanism inside the body accepts and discharges a predetermined volume of liquid for every incremental turn of the rotating shaft, resulting in a positive displacement of fuel. A totalizer connected to the rotating shaft counts the total number of units of liquid that pass through the meter.

Manufacturer-stated accuracy is in the range of 0.5%. Calibrations are usually performed using either a volume tank or a prover meter. Unless the installed metering run has those features built into it, calibrations are conducted at a calibration facility, typically using a fluid other than the one being metered. Besides calibrating the basic meter, calibrations are also performed on any additional features such as temperature and specific gravity correction mechanisms. Final calibration accuracy is dependent on all of these variables. Periodic rebuilding of these meters may be necessary as wear on mechanical parts affects their accuracy.

Coriolis type fuel oil meters have advantages over the other technologies in terms of essentially no moving parts, no need for the primary fluid to be taken into pressure transducers, and virtually total electronic determination of mass flow. They are a viable alternative for new or retrofit installations where operational, accounting, or performance purpose fuel measurement is required.

Some boiler operators have identified a link between the viscosity of fuel oil that is both temperature- and oil-specific and the stack opacity. Instruments exist to continuously monitor fuel oil viscosity. For those oil boilers with the viscosity/opacity link, continuous monitoring is recommended.

2-2.7.2 Gaseous Fuels. Gaseous fuels add an additional problem to the flow measurement. First the gas is compressible. At very low pressures perfect gas equations will work well; however, at pressures above 300 psia, supercompressibility becomes a significant factor that must be corrected. With steam flow calculations this supercompressibility factor can be calculated with the steam tables; however, with compressed hydrocarbons the required information is not as easy to determine. The best reference for this calculation is the AGA method of calculation of supercompressibility of hydrocarbons. This method requires knowledge of the specific gravity of the flowing fluid.

There are well documented means of determining compressibility and densities. Some online control systems perform approximations of the standard methods. Most online performance monitoring systems offer sophisticated flow measurement formulations that can increase the accuracy of all the flow measurements. The instrumentation needs to be programmed to send the raw data (in terms of pressure, differential pressure, temperature, or flow) as well as the calculated flow such as returned by a transducer in order for the monitoring system to do the independent calculations.

2-2.7.3 Coal Flow Measurements. There are four predominant types of scales used to determine coal flow measurement in utility boilers: static scales (e.g., truck scales), conveyor belt scales, batch scales, and coal feeders (gravimetric and volumetric). Each type of measurement system is discussed in this subsection.

Devices are available that provide relative pulverized coal flow in the transport pipes. These devices rely on microwave, acoustics, or electrostatics. As the coal flow is neither steady nor evenly distributed within the transport pipes, the measurements fluctuate and contain significant uncertainty. Typically these devices require an absolute value determined via manual extractive methods to establish the range for relative flow rate readings.

2-2.7.3.1 Static Scales. Static scales, both truck and rail, are high-accuracy devices to weigh received coal in power plants. Static truck and rail scales should be calibrated using a standard railroad test car or test truck with a weight traceable to the National Institute of Standards and Technology (NIST). The test weight shall be at least 25% of the scale's rating, but it is recommended that the test weight be close to the scale's normal operating range. Rail test cars are available from the railroad company serving the plant, and test trucks are available from scale service companies and government agencies.

Prior to calibration, the scale should be inspected for wear, approach deterioration, inadequate scale pit drainage, physical damage, proper rail alignment with approach (rail scales), severe rusting and corrosion, load cell foundation deterioration, and any devices or obstructions that inhibit scale motion. If any of the above conditions are noted, they should be corrected prior to calibration. Scale zero should be checked and adjusted prior to calibration. Scale zero should also be checked between calibration runs and after the calibration is complete. If the scale zero has drifted ±1 scale division, the scale zero should be adjusted and calibration rechecked. A scale division is defined as the smallest weight the scale will indicate. Acceptable tolerances of scale error are identified in NIST Handbook 44, Section 2.20.T.N.3.1.

The scale should be checked and calibrated using three test procedures: the increasing load test, the decreasing load test, and the shift test. These tests should be performed with test weights and procedures in accordance with NIST Handbook 44. Two test runs should be made to check tolerances. A zero check should also be run prior to making adjustments to the scale.

Once adjustments are made, three repeatability runs should be made. The results should agree within the value of the scale tolerance. Specific calibration and adjustment instructions for each individual scale manufacturer and model should be obtained from the manufacturer's instruction handbook for the scale.

The requirements listed in this subsection for the calibration testing of static scales is general in nature and will be suitable for a scale whose purpose is inventory control. Should the static scale be used as a basis-of-payment scale, the scale design, calibration test procedures, tolerances, and user requirements shall be in accordance with NIST Handbook 44, Section 2.20.

2-2.7.3.2 Conveyor Belt Scales. Conveyor belt scales are used to measure both received and consumed coal for utility boilers, and are often the primary source for determining the official utility heat rate and book value of fuel inventory. The types of scales used in these two services are generally of the highest accuracy offered by a particular scale manufacturer. The best reported uncertainty for a conveyor belt scale is ±0.125% where the scales most commonly used have uncertainties from ±0.50%. Conveyor belt scales systems fall into one of two categories of service: certified for basis-of-payment or noncertified.

Weighing accuracies for belt scales in both categories of service may be the same; however, the installation and calibration of a certified scale and its associated conveyor must be in accordance with the requirements of NIST Handbook 44. All conveyor belt scales should be calibrated using a weighed material load test whenever possible. However, this method of calibration is required by NIST Handbook 44 for certified conveyor belt scale systems (NIST Handbook 44, Section 2.21. N.ll). A material test consists of pre-weighing or post-weighing a quantity of material to be passed across the belt scale on a certified static reference scale (NIST Handbook 44, Sections 2.21. N.2 and N.3.2 through N.3.2.1). This weight is then compared to the integrator reading to give the scale error in percent. Percent error = [(integrator weight – actual weight)/actual weight] × 100.

An adjustment is then made to the scale electronics or the mechanical linkages to correct for any error found. On initial calibration of the system, at least three individual tests should be conducted. On subsequent verifications, at least two individual tests should be conducted (NIST Handbook 44, Section 2.21. N.3.2f). The results of all tests should be within tolerances.

After completion of the material tests, a simulated load test (described below) should be conducted to establish the factor that will relate the results of the simulated load test to the results of the material test (NIST Handbook 44, Section 2.21. N.3.3b). The belt scale should be warmed up for a minimum of 30 min prior to calibration. This time should be extended, as conditions require, if the temperature is below 41ºF (NIST Handbook 44, Section 2.21. N.3.1). Before any scale calibration, the scale zero should be checked with the conveyor running empty. The variation between the beginning and ending indication of the totalizer should not be more than ±1 scale division after the required test duration. At no time during the zero load test should the totalizer change more than three scale divisions from its initial indication (NIST Handbook 44, Section 2.21. N.3.1).

If the integrator does change more than the allowable number of scale divisions during the test duration, the cause for the zero drift error should be investigated and the zero recalibrated. A scale division is defined as the smallest weight unit the scale will register.

At any time between material tests, or in the case of noncertified scale systems where a material test is not feasible, simulated load tests should be conducted. Simulated load tests can be conducted with a test chain, with static weights, or electronically, depending on the scale manufacturer's recommendations and user preference (NIST Handbook 44, Section 2.21. N.3.3).

For all simulated load tests, a known weight (lbm/ft^3), either physical or electronic, is applied to the scale for a specific number of whole belt revolutions (ft). At the end of this time the totalizer should register a specific, known, total weight. Percent error, if any, is calculated as shown above. In the case of noncertified scales, a calibration adjustment is generally made after the first simulated load test, if required, and then repeatability tests are run.

Three consecutive simulated tests should be conducted to check repeatability. The results of the simulated load tests that repeat within 0.1% (the difference between the greatest and least percent scale error) for these three tests should be less than 0.1%. One of the three repeatable checks may be the as-left check of the scale after making any adjustment to the scale. Errors of 0.01%, 0.02%, and -0.01% are acceptable, whereas errors of 0.1%, 0.2%, and 0.22% are not acceptable. In the case of certified scale systems, simulated test results should be interpreted and acted upon as follows (NIST Handbook 44, Section 2.21. UR3.2b):

(a) error less than 0.25%: no adjustment

(b) error between 0.25% and 0.6%: adjustment can be made if the certifying authority is notified

(c) error between 0.6% and 0.75%: adjustment can be made by a competent service person if the certifying authority is notified. If the results of a subsequent test require adjustment in the same direction, a material test is required.

(d) error greater than 0.75%: a material test is required.

Plotting as-found calibration errors as a function of time will give the user an idea as to the scale system's condition. Calibration records can indicate scale system problems. Since the conveyor is a major component of a belt scale system, the following items can be checked in the effort to identify a problem: idler alignment, belt take-up mechanism, worn scale mechanism, worn test chain, changes in belt length due to splicing, and test chain/weight certification.

2-2.7.3.3 Coal Feeders. Two types of feeders are used to measure fuel flow to the boiler: gravimetric and volumetric. Coal feeders are an integral part of a boiler's combustion control system, and in some cases, can be used for coal inventory determination.

Gravimetric feeders use conveyor belt scales to determine coal fuel flow rate. These belt scales should be calibrated at the same frequency and with the same methods as other belt scales whenever possible. Accuracies of gravimetric feeders vary with age and maintenance practices, but can achieve values of ±0.5% under optimal conditions. More reasonable assumptions for feeder accuracies may be closer to 5%. The user may need to run tests to determine as-found accuracy levels.

Volumetric feeders, sometimes referred to as table feeders, do not provide a direct measurement of mass flow rate to the boiler. Instead, the output is a volumetric flow rate that must be converted to mass flow rates based on periodic calibration. Obviously, coal bulk density has a major impact on the conversion; consequently, the mass flow rate can vary. Volumetric feeders should only be used for combustion control, and not for fuel inventory determination.

2-2.8 Measurement of Flue Gas

Flue gas analysis is used in boiler performance, air heater performance, and stack gaseous emissions determination. The three gases traditionally tested for are CO, O_2, and CO_2. In the past, Orsat analyzers were the main method available to determine flue gas composition. Orsats generally absorb (remove) CO_2 from the sample first because it would interfere with later reagents. However, with the development of accurate single gas electronic analyzers, CO_2 measurements are not required. CO_2 is not as useful as CO or O_2 for boiler tuning or control because CO_2 concentration can indicate either an excess or a deficiency of combustion air.

Flue gas flows are prone to very high degrees of both temperature and constituent stratification. It is very rare that a single sample of a gas stream will indicate the true average value. A multipoint grid must be used to obtain good flue gas measurements. Maintaining constant flow through all of the sampling ports of such grids on a continuous basis is very difficult. Instead, it is recommended that such sampling be conducted on both a periodic schedule, and also on an event-driven schedule (for example, a known or suspected change in fuel composition). The reading of the installed single point sensor can then be compared to and correlated with the average composition or temperature of the grid.

2-2.8.1 Electronic Gas Analyzers. There are a variety of electronic instruments available for measuring flue gas constituents. Most of these instruments cannot measure all three components (O_2, CO_2, and CO). The most common only measure oxygen, while some others measure oxygen and combustibles (CO, H_2). Oxygen concentration is generally measured with a paramagnetic sensor, wet electrochemical cell, or zirconium oxide cell. The zirconium oxide cell must operate at a high temperature that makes it a natural choice of in situ measuring systems. The paramagnetic sensor and wet electrochemical cell require a cooled and dried sample (see para. 2-2.8.3). The wet electrochemical cell is lightweight and is rugged, which means that it is a good choice for a portable analyzer also. Electrochemical cells have a limited life and must be replaced on a routine basis. The paramagnetic sensor is a very delicate device that makes it best suited for fixed installations.

Wet electrochemical, catalytic element, and nondispersive infrared absorption are three common methods used to measure CO concentrations. The wet electrochemical method for CO has the same advantages and disadvantages as the oxygen electrochemical cell. The catalytic sensor is temperature sensitive and is usually combined with an in situ method. This method also detects hydrogen which means that it is more appropriate for the monitoring of total combustibles. Infrared absorption analyzers must have dried and cooled samples.

2-2.8.2 Orsat Analyzer. Another instrument for flue gas analysis is the Orsat analyzer. The Orsat consists of a measuring burette and three reagent pipettes that are used to successively absorb carbon dioxide, oxygen, and carbon monoxide from the mixture. Since the combustion products are contained over water in the burette, they remain saturated with water vapor, and the volumetric proportions of the combustion gases are obtained on a dry basis. Some limitations of Orsat analysis are that it is a slow process, the reagents become exhausted and need replacing, and accurate results depend on the skill of the operator. The accurate results may be checked by plotting the readings on a dry flue gas volumetric combustion chart. Orsat analyzers are older instruments that are not in common use. Improved measurement accuracy is available using more recent analyzer technology.

2-2.8.3 Other Considerations. If the gas sample is passed through an external sampling system with a dryer, then the analysis will be on a dry basis. The concentrations measured by an in situ system would be wet and would need to be converted for comparison with dry values. See para. 2-2.9.8 for a wet-to-dry calculation method.

The desired range of measurement and highest resolution is an important decision that should be made early in the selection process. Infrared analyzers are constructed for the specific gas and concentration range required. Be aware that some gases interfere with the desired measurement (e.g., CO_2 is read as CO). Also consider that expanding the range of a CO analyzer to include upset conditions (>1,000 ppm) will diminish the accuracy of the analyzer in lower useful range (50 ppm to 400 ppm) where the high resolution is needed.

Some instruments calculate constituents instead of measuring them. Careful review of actual measurements is recommended.

2-2.8.4 Calibration. Electronic gas analyzers require calibration with zero and span gases. The necessary recalibration cycle should be based on experience with a given instrument. If Orsats are used, careful tracking of age of the reagents is required due to short useful life.

2-2.9 Fuel Characteristics

Normally, fuel characteristics are determined by an independent laboratory using standard test methods. However, these characteristics can be verified by the customer if sufficient facilities exist.

The characteristics of coals may be reported on an as-received basis that includes all of the constituents of the coal, including the moisture and ash. It may also be reported on an ash-free basis, or on a moisture-and-ash-free basis. The sum of each of these measurements should always equal 100%.

2-2.9.1 Heating Value. The heating value of fuel is one of the basic factors in determining boiler or gas turbine performance and overall plant performance. Since fuel is measured by unit of volume or weight, it is essential to know what the Btu content is per unit of volume or weight. Standard test methods for determining heat values can be found in ASTM D 5865 for coal, ASTM D 3588 for gaseous fuels, and ASTM D 3523 for liquids and solids.

The higher heating value (HHV) of a fuel includes the total energy released by the complete combustion of the fuel. This includes the heat of vaporization of all moisture. The lower heating value (LHV) includes the total energy released by the fuel without condensation of the water vapor in the products of combustion.

2-2.9.2 Fineness. Fineness of pulverized coal is a measure of the performance of the pulverizer and the primary air (transport) system. The units are normally the percent of a sample that will pass through a specific mesh screen (i.e., 70% through a 200 mesh screen). Fineness testing is discussed in more detail in para. 2-3.8.4.1 of this document, in ASME PTC 4.2, and in ASTM D 197.

Results of fineness tests can be compared to historical data and the pulverizer manufacturer's specification to determine the level of pulverizer performance. A change in fineness may be the result of deteriorated pulverizer performance, or may indicate a change in the grindability of the coal. Pulverizer capacity is inversely proportional to fineness, with about 1.5% capacity reduction per 1.0% increase in fineness. It is recommended that periodic fineness testing be performed to determine if the coal is being ground finer than necessary at the expense of pulverizer capacity. Excessive amounts of coarse particles are thought to be responsible for excessive convective section fouling, especially when firing poorer grades of coal such as PRB coals.

2-2.9.3 Grindability. The grindability of coal is a relative measure of how well the coal responds to pulverization. A coal grindability index has been developed to measure the ease of pulverization. Grindability should not be confused with the hardness of a coal. A prepared sample of a specified weight is put into a miniature pulverizer called a Hardgrove grindability machine. After running the machine for the specified amount of time, the sample is sieved and weighed. The Hardgrove grindability index (HGI) is determined by plotting these results on the calibration chart for the test machine. This test procedure is detailed in ASTM Standard Test Method D 409. The proximate and ultimate analysis of a coal does not indicate the grindability of a coal. The grindability test is a mechanical measurement that cannot be predicted using the chemical composition of the coal.

The Hardgrove index is one factor used to determine the mechanical grinding capacity of a pulverizer. A higher Hardgrove value means that the coal is easier to pulverize. Pulverizer mechanical capacity is somewhat proportional to the grindability index. However, pulverizer capacity is also a function of the required fineness of the pulverized coal, as well as the moisture content of the entering coal. Pulverizer thermal capacity is based on having sufficient heated air to dry the coal to the desired pulverizer outlet temperature. Consequently, a higher raw coal moisture level requires more drying, resulting in a reduction of pulverizer capacity. Refer to para. 2-3.8.4.1 for a discussion of pulverizer testing and capacity.

2-2.9.4 Ash Content and Analysis. Ash analysis can provide information on the effects that ash can have on the boiler (fouling) as well as giving insight to the efficiency of the combustion process.

The percentage of ash, along with the composition of the ash, in fuels has a major impact on boiler performance and emissions. Lower ash content will generally allow the boiler to remain cleaner and operate more efficiently. Higher ash content may cause increased slagging and fouling, reduced boiler efficiency, and increased wear on boiler components and pulverizers. This behavior is very dependent on the chemical composition of the ash in the fuel, and the design of the boiler and heat transfer surfaces. The ash content of coal is determined as part of the proximate analysis using ASTM D 3174.

Ash composition can also be an environmental concern. The percentage of sulfur, various metals, and percent unburned carbon can determine the disposal method of the ash or its suitability for resale. Standard test method ASTM D 3174 specifies the test for ash composition. Elemental analysis permits a better understanding of how to collect and handle the ash. Ash resistivity is directly related to ash collection in electrostatic precipitators. The knowledge of particle size distribution is important in designing and maintaining an optimal flyash collection system.

2-2.9.5 Fuel Composition and Analysis. Fuel composition is a primary factor in the design of a boiler, as well as a concern for environmental compliance. Sulfur, mercury, chlorine, and metals content are often specified in air permits and monitored by the EPA. ASTM D 3176 contains standard test methods for determining the more common elements of coal. There are other test methods for specific elements and fuels.

2-2.9.6 Loss on Ignition. Loss on ignition (LOI) is an approximate measure of the percentage of unburned carbon in the ash leaving a boiler, and is expressed in percent by weight. It is an indication of how completely the fuel has been burned in the boiler. Typical percent unburned carbon values for coal-fired boilers range from as low as 0.1% for PRB and lignite coals, to as much as 30% for eastern bituminous coals in a short furnace, poorly operating, or highly staged boiler. For most ash disposal applications, there is a maximum specification for LOI. LOI represents unburned combustibles, therefore lower numbers are desirable since they represent improved boiler efficiency.

LOI results can vary widely depending on the constituents in the ash. For example, an ash sample with a measured 2% unburned carbon may have an LOI number ranging from 2% to 9%. The LOI analysis is an empirical method that employs the weighing of a sample before and after an ignition temperature of approximately 1,400°F. In this process, several weight changes occur: hydrates are driven off, reduced iron oxide is brought to its highest oxidation state (Fe_2O_3), decomposition reactions as well as combining reactions can occur, and any residual carbon is burned. Although the LOI analysis can provide some approximation of carbon remaining in an ash of a known mineral matrix, it is generally not a very reliable figure for an unknown ash. The only accurate determination for unburned carbon is a specific analysis for carbon, where the carbon is combusted in an atmosphere of oxygen and the resulting CO_2 gas is measured very precisely by a thermal detector.

2-2.9.7 Fusibility. Ash fusibility of coal ash is a temperature measurement at which ash characteristics change. There are four different temperatures specified as the ash goes through its initial deformation stage to fluid stage. The test method for ash fusibility is specified in ASTM D 1857. Ash fusibility is a function of the coal and varies with the coal origin. Ash fusion temperature is a commonly used term in the utility industry; however, the deformation stage should also be specified, as well as whether the temperature was measured in a reducing or an oxidizing atmosphere.

2-2.9.8 Conversion of Wet to Dry O_2 Measurements

Wet O_2 readings on a percent by volume basis are obtained from in situ oxygen probes, where the weight of the water in the flue gas is included in the overall O_2 percentage calculation. Extractive oxygen analyzer systems give oxygen concentrations by volume on a dry basis. The following example illustrates how to calculate O_2 on both a wet and dry basis by developing a relationship between wet and dry O_2 readings.

(a) Obtain the ultimate analysis of fuel being fired, percent by weight.

Carbon	72.0
Hydrogen	4.4
Sulfur	1.6
Nitrogen	1.4
Oxygen	3.6
Water H_2O	8.0
Ash	9.0
Total	100.0

(b) Assuming 100 lb of fuel, calculate the number of moles of each constituent of fuel by dividing each percentage by each molecular weight.

Carbon	72.0/12 = 6.00 moles
Hydrogen	4.4/2 = 2.20 moles
Sulfur	1.6/32 = 0.05 moles
Nitrogen	1.4/28 = 0.05 moles
Oxygen	3.6/32 = 0.11 moles
Water H_2O	8.0/18 = 0.44 moles
Ash (not used)	

(c) Calculate the moles of O_2 required to oxidize the moles of each constituent of the fuel.

Carbon	$C + O_2 = CO_2$	$6.00 \cdot 1.0$	=	6.00 moles
Hydrogen	$H_2 + 1/2O_2 = H_2O$	$2.20 \cdot 0.5$	=	1.10 moles
Sulfur	$S + O_2 = SO_2$	$0.05 \cdot 1.0$	=	0.05 moles
Oxygen	$O_2 = O_2$	$0.11 \cdot (-1)$	=	–0.11 moles

(d) Calculate the theoretical O_2 requirement, which equals the total moles of oxygen required for stoichiometric combustion of the fuel (zero excess air).

$$6.00 + 1.10 + 0.05 - 0.11 = 7.04 \text{ moles } O_2 \text{ required}$$

(e) Calculate the moles of N_2 in the air used for combustion.

$$N_2 = 7.04 \cdot 3.76 = 26.47$$

where

3.76 = moles of N_2 per mole of O_2 in air

(f) Calculate the total moles of dry air required for stoichiometric combustion.

$$\text{Total} = 7.04 + 26.47 = 33.51$$

(g) Calculate a factor for the moisture in the air used for combustion.

$$Z = \frac{\text{moles } H_2O}{\text{mole dry air}}$$

$$Z = \frac{\dfrac{RH}{100} \bullet P_{saturation}}{P_{barometric} - \left(\dfrac{RH}{100} \bullet P_{saturation}\right)}$$

where

RH = relative humidity; %

$P_{saturation}$ = saturation pressure of H_2O in air (at T = ambient temperature); psia, in. Hg, or in. H_2O

$P_{barometric}$ = barometric pressure; psia, in. Hg, or in.

H_2O (consistent units required) or use:

RH = 60%

$P_{barometric}$ = 14.70 psia

Ambient

temperature = 80°F

$P_{saturation}$ = 0.5073 psia, using steam tables

$$Z = \frac{\frac{60}{100} \bullet 0.5073}{14.70 - \left(\frac{60}{100} \bullet 0.5073\right)} = 0.0211$$

(h) Calculate the moles of H_2O in the theoretical air.

$$\text{Moles } H_2O = 33.51 \cdot 0.0211 = 0.71$$

(i) Calculate the total moles of flue gas, with zero excess air.

Reaction:

$6.00\ C + 2.20\ H_2 + 0.05\ N_2 + 0.05\ S + 0.11\ O_2 + 0.44\ H_2O + 7.04\ O_2 + 26.47\ N_2 + 0.71\ H_2O$

$-\ 6.00\ CO_2 + 0.05\ N_2$ (fuel) $+ 0.05\ SO_2$

$+\ 2.20\ H_2O + 0.44\ H_2O + 0.71\ H_2O$ (air) $+ 26.47\ N_2$ (air)

Total moles = $CO_2 + N_2$ (fuel) + SO_2 + H_2O (from H_2 in fuel)
+ H_2O (from H_2O in fuel) + H_2O (air) + N_2 (air)
= 6.00 + 0.05 + 0.05 + 2.20 + 0.44 + 0.71 + 26.47
= 35.92

(j) Calculate wet and dry O_2 with 10% excess air.

Theoretical O_2 requirement = 1.10·7.04 = 7.74

Moles of N_2 = 7.74·3.76 = 29.12

Total moles of dry air = Moles of O_2 + Moles of N_2
= 7.74 + 29.12 = 36.86

Moisture in air = 0.0211·36.86 = 0.78

Total moles flue gas = $CO_2 + N_2$ (fuel) + SO_2 + H_2O (from H_2 in fuel) + H_2O (from H_2O in fuel)
+ H_2O (air) +N_2 (air) + O_2 (excess)

= 6.00 + 0.05 + 0.05 + 2.20 + 0.44 + 0.78 + 29.12 + (7.74 – 7.04)

= 39.34

$$\%O_2 \text{ by volume, wet} = \frac{\text{moles } O_2}{\text{total moles of wet flue gas}} \times 100$$
$$= \frac{(7.74 - 7.04) \bullet 100}{39.34}$$
$$= 1.78\%$$

$$\%O_2 \text{ by volume, dry} = \frac{\text{moles } O_2}{\text{total moles of dry flue gas}} \times 100$$
$$= \frac{(7.74 - 7.04) \bullet 100}{39.34 - (2.20 + 0.44 + 0.78)}$$
$$= 1.95\%$$

(k) Repeat the calculation of wet and dry O_2 using 20% excess air.

Theoretical O_2 requirement = 1.20*7.04 = 8.45

Moles of N_2 = 8.45*3.76 = 31.77

Total moles of dry air = Moles of O_2 + Moles of N_2
= 8.45 + 31.77 = 40.22

Moisture in air = 40.22*0.0211 = 0.85

Total moles flue gas = CO_2 + N_2 (fuel) + SO_2 + H_2O (from H_2 in fuel)
+ H_2O (from H_2O in fuel) + H_2O (air) + N_2 (air) + O_2 (excess)
= 6.00 + 0.05 + 0.05 + 2.20 + 0.44 + 0.85 + 31.77 + (8.45 – 7.04)
= 42.77

$$\%O_2 \text{ by volume, wet} = \frac{\text{moles } O_2}{\text{total moles of wet flue gas}} \times 100$$
$$= \frac{(8.45 - 7.04) \bullet 100}{42.77}$$
$$= 3.30\%$$

$$\%O_2 \text{ by volume, dry} = \frac{\text{moles } O_2}{\text{total moles of dry flue gas}} \times 100$$
$$= \frac{(8.45 - 7.04) \bullet 100}{42.77 - (2.20 + 0.44 + 0.85)}$$
$$= 3.59\%$$

(l) Repeat these calculations using a number of different excess air levels, and produce a graph of wet O_2% versus dry O_2%. This graph may then be used to convert back and forth between wet and dry O_2%. A new graph may need to be developed based on fuel analysis changes.

2-2.10 For Further Reference

Anderson, R. L., Lyons, J. D., Kollie, T. G., Christie, W. H., and Eby, R., "Decalibration of Sheathed Thermocouples," *Temperature, Its Measurement and Control in Science and Industry*, Vol. 5, American Institute of Physics, New York, NY, 1982, p. 977.

ASME PTC 6-1976, Steam Turbines.

ASME PTC 6 Report-1985, "Guidance for Evaluation of Measurement Uncertainty in Performance Tests of Steam Turbines."

ASME PTC 6A-1982, Appendix A to Test Code for Steam Turbines.

ASME PTC 19.2-1987, Temperature Measurement.

ASME PTC 19.3-1974, Temperature Measurement.

ASME PTC 19.6-1955, Electric Measurements in Power Circuits

Benedict, Robert P., *Fundamentals of Temperature, Pressure, and Flow Measurements*, Second Edition, 1977.

Crim, H. G., et al., "Output/Loss: A New Method for Measuring Unit Heat Rate," ASME 87-JPCC-PWR-39, 1987.

Dahl, A. J., "The Stability of Base-metal Thermo-couples in Air from 800 to 2,200ºF," *Temperature, Its Measurement and Control in Science and Industry*, Vol. 1, Reinhold Publishing Corp., New York, NY, 1941, p. 1238.

EPRI Report 1004743, Electric Power Institute Coal Flow Loop, System Description and Commissioning, March 2004.

EPRI Report 1004744, Electric Power Institute Coal Flow Loop, Extractive Methods, March 2005.

Flue Gas Measurement, Ametek, Thermox Instruments Division, 1988.

Griebenow, Caudill, Hill, Hoffman. "Precision Data Validation Boosts Process Optimization Benefits" Power Engineering, November 1999.

Kerlin, *Intech*, ISA, August 1990.

Madison, E. C., *Temperature Measurement in Industry*, 1990.

Manual of Instrument Transformers — Operation Principles and Application Information, General Electric. GET-97D.

March, P. A., and Almquist, C. W., "Techniques for Monitoring Flowrate and Hydraulic Fouling of Main Steam Condensers," TVA, Engineering Laboratory.

Miller, R. W., *Flow Measurement Engineering Handbook*, 1983.

Potts, J. F., Jr., and McElroy, D. L., "The Effects of Cold Working Heat Treatment and Oxidation on the Thermal emf of Nickel-base Thermoelements," *Temperature, Its Measurement and Control in Science and Industry*, Vol. 3, part 2, Reinhold Publishing Corp., New York, NY, 1962, p. 243.

Practical Temperature Measurements, Hewlett-Packard, Application Note 290, May 1983.

2-3 PERFORMANCE MONITORING IMPLEMENTATION AND DIAGNOSTICS

2-3.1 General

Subsection 1-2 describes the needs and benefits of a well defined and well executed performance monitoring and diagnostic program. This subsection addresses the processes necessary to obtain the desired answers about plant condition. These processes are

(a) instrumentation to provide specific information at important locations around the power generation cycle (see subsection 2-2)

(b) use of data recording, computational, and archiving equipment to display and calculate important derived operating parameters, and to archive for future recovery and redisplay the information obtained from the basic measurements and calculated parameters

(c) correction of measured data to standardized conditions or comparison with expected current values

(d) trending of parameters that provide significant information about the operation of the plant

(e) periodic testing to validate the accuracy of the continuously recorded information and to provide additional information about specific problems

(f) performance calculations and diagnostics to extract the meaning and significance of the monitored data

(g) performance calculations and diagnostics for major portions of the power generation facility, including

(1) turbine cycle

(2) steam generator equipment

(3) balance of plant (condensers, cooling towers, heaters, and pumps)

(4) combined cycle plants

2-3.2 Data Recording, Computational, and Archiving Equipment

In addition to those of direct interest, parameters such as turbine cycle and/or plant heat rate must be calculated from directly measured values. Standard practice is to use computational equipment to display the current measured value, calculate derived parameters, and archive both directly measured and derived parameters for future recall, trending, and diagnostic activities. When a parameter of interest is not incorporated into the data management system it may be necessary to keep a manual record until the system can be updated.

2-3.3 Correction of Measured Data to Standardized Conditions or Comparison With Expected Current Values

A significant problem in the analysis of measured data is to know what the value should be. This is difficult because power generation facilities rarely run at constant load, and when they do, externally imposed parameters affecting performance (air temperature, condenser cooling water temperature, etc.) change frequently. For example, the temperature values of cooling water flowing to the condenser may be the same when measured twice consecutively, but condenser internal pressure readings may differ. Does a problem exist? Condenser loading or duty is another variable difficult to measure directly. If there is a difference when the two sets of readings are taken, then the pressure reading should be expected to differ as well.

The constantly varying nature of power plant operation requires that data be corrected to a predetermined standard, usually the conditions specified for the design of the plant.

Correction of test conditions to specified design conditions is a problem that arises in all turbine acceptance tests and is dealt with in Performance Test Code PTC 6-1996. Group 1 corrections to heat rate and generation apply to variables that primarily affect the regenerative feedwater cycle, and are discussed in detail in PTC 6. Although a typical set of Group 1 correction curves is contained in PTC 6, it is recommended that, where possible, curves specific to the cycle under evaluation be established.

Group 2 corrections to heat rate and generation apply to variables affecting turbine performance and are supplied by the turbine manufacturer in the thermal kit. The thermal kit is provided by the turbine manufacturer as part of the turbine-generator contract. It contains several assumptions regarding the equipment and components in the turbine cycle outside the scope of the turbine-generator contract. When a turbine acceptance test is conducted, Group 1 and Group 2 corrections are made to the data to verify that the turbine-generator has met its contractual obligations. Since the thermal kit data relating to the regenerative feedwater cycle contains several assumptions, it is important to develop plant-specific Group 1 corrections based on plant design data.

The thermal kit is a compendium of performance information, including heat balances of the turbine cycle and correction curves to heat rate and load for deviations from rated values of selected performance parameters. Thermal kits are generally developed for new equipment contract guarantee purposes. Inaccuracies may exist if the interactions between the boiler and turbine are incorrect. For example, piping pressure drops from the turbine to the boiler may be based upon an assumed value in the thermal kit instead of their true value. Values of zero superheater and reheater spray flows are generally assumed for the turbine, while the boiler may require spray flows for temperature control. The thermal kit values should be carefully compared to the measured turbine performance. The steam tables used in the thermal kit should be specified. Depending on the date of calculations in the thermal kit, the steam functions may be based upon printed steam tables, 1967 IFC steam tables, or 1997 IFC steam tables. The calculation of turbine efficiency must be done using the same steam properties used in establishing the thermal kit. Section 6 of ASME Performance Test Code PTC 6S Report-1988 provides a thorough discussion of the equations useful for correcting measured parameters to standard conditions.

An alternate approach to correcting test results to standard conditions is to have a computerized model of the plant predict how a new-and-clean installation would respond to current operating conditions. The values of certain key measurements that define the conditions under which the plant is currently being operated, such as feedwater flow, turbine throttle and reheat conditions, condenser cooling water temperature, air temperature, etc., are entered into a computer with an installed model of the plant. The model then predicts the expected values for all parameters being measured, and reports the difference between the measured and expected values. These differences can immediately indicate unexpected conditions indicative of possible operating problems. This is a relatively new concept that is available on some of the performance monitoring equipment currently being offered to the industry.

2-3.4 Trending

Trending consists of plotting a parameter of interest against time and presenting the result as a curve to convey important diagnostic information. As an example, measurements of HP turbine efficiency made 2 mo apart may show a drop of 7%. If this change occurred quickly, it could be an indication of a blade failure or another type of mechanical deterioration that has severely damaged the steam path. If the change occurred slowly over the 2 mo period, it would be a strong indication of deposits, possibly copper, that accumulate slowly but continuously.

To maximize the ease of use and minimize the possibility of drawing incorrect conclusions, it is highly recommended that continuously trended parameters include software algorithms necessary to perform the required normalizations.

The selection of parameters to trend is left to the user’s discretion. Later sections of this document are devoted to specific portions of the power plant cycle and contain suggested lists of parameters for trending.

Load and heat rate are important basic parameters. Due to seasonal fluctuations in heat rate and generator output, plant engineers may find that an accumulative year-to-date weighted average of these indices is sometimes more meaningful than the instantaneous value.

Operators at consoles should be able to trend parameters such as calculated heat rate, turbine backpressure, and cooling water inlet, outlet, and delta temperature. Plants are complex and use many trend points (sensor inputs to data logging systems) to display data. Many plants have operations monitoring screens and protocols that allow operators to set user-defined warning alarm limits for arbitrarily trended points in the DCS or Data Historian. These points should be established based on logical analysis of the sensitivities and change probabilities that will make the trended information useful. Insensitive points that have low trend value should be avoided. Historically sensitive points should be used to trigger limit alarms to warn of undesirable conditions.

2-3.4.1 Data Validation. Data processes should ensure that measurements provide an adequate engineering basis for their intended use. To obtain accurate data, use instruments with a design and quality level that have demonstrated an ability to accurately track the parameter being measured. Next in importance is to properly calibrate the specific instrument being used. The applicable equipment Performance Test Code (PTC) specifies equipment calibration requirements to verify contractual commitments on power, efficiency, and heat rate. While the Performance Monitoring Guideline does not require the same level of calibration requirements as the PTC, by intent, good engineering practice should prevail. Where calibration is not feasible or required following installation based upon the sensitivities of the measured parameters, calibration should not be performed. An explanation should provide the basis for calibration. For example, the calibration of thermocouples used to measure boiler tube metal temperatures is not normally required. But for thermocouples used for the purpose of heat rate calculations for nuclear steam generators, the lack of calibration prevents absolute comparison with the calculated results. Sometimes equipment is provided with measurements that provide insight into performance, but do not lend themselves to calibration. Relative trend information may still be valuable. The current levels of large induction motors are an example.

If trend data is available from an acceptance test, it can be used to help identify observed data trends that appear suspiciously high or low. Automated calibration verification is now commercially available, and can be used for data validation. These software systems monitor the output of all primary instruments and compare the indications on a high frequency. The software “learns” how the values are correlated and when one strays too far from expected, specific actions can be taken. Measurement data can be noted as suspect or even removed from service. The expected value can be substituted for the suspect value until checked and either restored or manually overridden.

2-3.5 Periodic Testing

Performance testing of plant cycles and component equipment should be performed to determine how accurately plant instrumentation is identifying plant degradation, and to obtain more detailed information about an identified serious degradation problem. Testing may range from contractual acceptance/baseline testing for indexing expected levels of performance (new equipment, rebuild, or overhaul) to routine testing under normal operating conditions.

Testing should be performed at regularly scheduled intervals. Good engineering judgment should be used when determining frequency, bearing in mind the need to increase frequency when sudden or abrupt performance deterioration is suspected. Tests conducted prior to and immediately following outages provide insight on the effect of repairs or modifications done during that outage.

ASME Performance Test Code PTC 6S Report-1988 provides an excellent description of the appropriate procedures for routine performance testing of an operating power plant. The procedures recommended in that publication are recommended for use in any performance monitoring program.

2-3.6 Performance Calculations and Diagnostics to Extract the Meaning and Significance of the Monitored Data

2-3.6.1 Introduction. The purpose of this subsection is to provide guidance on converting performance results into an understanding of equipment physical condition. This information is intended to be used by operations and maintenance personnel in performance optimization.

(a) The key to an effective performance monitoring program is to obtain results that are comprehensive and adequate to make specific recommendations on

(1) operational changes

(2) maintenance actions

(3) system or equipment modifications

(4) new or replacement equipment

(b) Further benefits of a proactive, rather than reactive, detection and diagnostic process include

(1) correcting problems sooner, in order to reduce fuel costs, maintain capacity, and sometimes to reduce the cost and effort of the corrective action itself

(2) correcting problems that would have otherwise gone undetected, but would incur real performance costs

2-3.6.2 Diagnostic Methodologies. The energy conversion process is complex and involves many individual components, all of which operate as part of a system, meaning they are all affected to some degree by the performance of other components in the system. To simplify the diagnostic process, a number of logical procedures have been developed. Several of them are discussed in this subsection.

The diagnostic techniques presented in this subsection use the results of a performance monitoring program to seek out the root causes for changes in performance. The emphasis in diagnostic testing is on identification of the cause of performance changes. Often the cause can be determined from relative changes in various component parameters.

Therefore, it is desirable to monitor the various component parameters in a way that differentiates problems from normal changes that do not indicate problems. Further, the parameters monitored and diagnostic methods employed should help localize the source of problem changes, to more rapidly diagnose root cause and determine corrective action.

For example, when monitoring the heat transfer performance of the condenser, one should monitor changes in measured condenser shell pressure. In addition, it is desirable to monitor the deviation from the expected backpressure that should be achievable at current operating conditions, i.e. taking into account the varying cold water temperature and flow rate into the condenser and the heat duty being imposed on the condenser from the turbine exhaust and the rest of the cycle. This would help differentiate a problem with the performance of that component from a problem in another component, or from a normal variation.

2-3.6.2.1 Generic Performance Curves or Diagrams. Performance curves provide a means of developing target values of performance over an operating range. Target and as-operating curves may then be plotted and compared to identify operation outside of typical design or expected conditions. Off-target conditions may be caused by changes in the operating mode, equipment malfunctions, or equipment failures. Figure 2-3.6.2.1-1 is an example of a generic performance curve. Performance curves are usually qualitative in nature and describe the fundamental principles that can be customized to unit-specific conditions. This provides a simple and cost-effective means of characterizing equipment performance through a deductive process.

2-3.6.2.2 Performance Diagnostic Tables. As compared with performance curves, diagnostic tables provide a more direct way of identifying the root cause of problems. The problems, their causes, and resulting effects are usually clearly delineated, thus requiring less interpretation. Tables 2-3.6.2.2-1 and 2-3.6.2.2-2 [10] are examples of diagnostic tables for different types of equipment with different levels of information provided. Diagnostic tables usually provide sufficient guidance so that the observed symptoms can be matched with those listed in the tables, thus leading to the identification of the problem and its cause. In some cases, special actions, such as suggested testing, may be required to isolate the root cause.

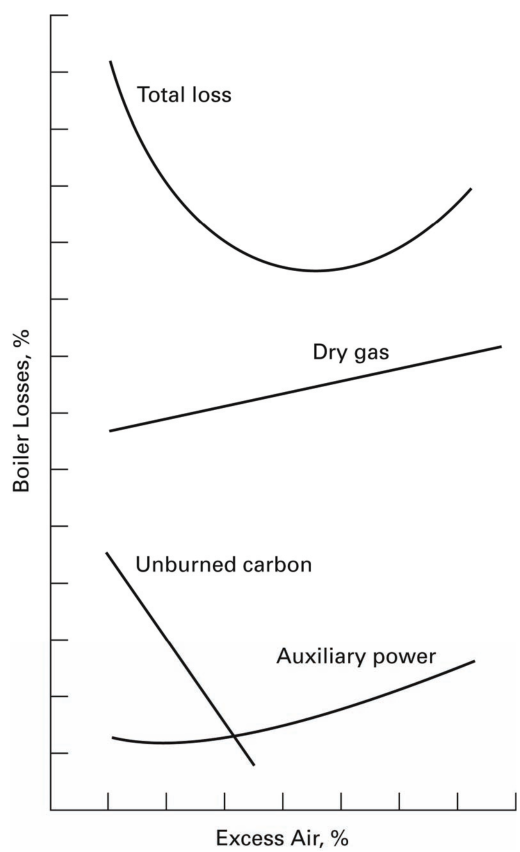
Table 2-3.6.2.2-1 Diagnostic Chart of Turbine Loss Characteristics (Courtesy Electric Power Research Institute)

Rubbing Damage on Spillstrips and Packing	
Mode of appearance:	Happens suddenly — more likely on a first startup
Local effects:	Increases flow capacity (this effect highest in HP section) Decreases section efficiency (worst on low volume flow stages) May cause IP enthalpy-drop efficiency to appear higher (opposed-flow units only)
Side effects:	Worsens flow temperature segregation Normally has little effect on thrust
Shape effects:	Ratio of % Δ efficiency/% Δ flow usually greater than 1 (absolute values)
Special dangers:	…
Solid Particle Erosion	
Mode of appearance:	Usually appears gradually
Local effects:	Increases flow capacity Decreases efficiency Worst effects usually at turbine inlets; at first stage, erosion magnitude may be worst at the inlet fed by the first value
Side effects:	Changed thrust; changed $\sqrt{p/v}$ distribution; changed flow distribution
Shape effects:	$\sqrt{p/v}$ effects may be greatest at light load Efficiency loss compared to guarantee may be greatest at light load; thrust increase may be in the same direction as flow
Special dangers:	Overloaded buckets; weakened tenons
Deposits	
Mode of appearance:	Usually gradual; may reach a self-limiting magnitude, then not increase further; may appear to decrease following a shutdown or major temperature swing
Local effects:	Decreased efficiency; decreased flow capacity
Side effects:	Changed thrust; changed $\sqrt{p/v}$ distribution
Shape effects:	Section efficiency may decrease 3–4 times as much as flow capacity Thrust changes may be opposite to the direction of flow
Special dangers:	Excessive thrust
Internal Damage	
Mode of appearance:	Usually abrupt — may have subsequent symptoms
Local effects:	Decreased efficiency; decreased flow capacity
Side effects:	Increased vibration; changed $\sqrt{p/v}$ distributions; changed thrust
Shape effects:	No consistent pattern
Special dangers:	Weakened or loosened mechanical structures

Table 2-3.6.2.2-2 Steam Surface Condenser Diagnostics (Courtesy Electric Power Research Institute)

Cause/Measured Parameter	Condenser Pressure	Cooling Water Temperature Rise or Pressure Drop	Terminal Temperature Difference
Reduced Cooling Water Flow *(a)* Debris on tubesheet/tubes *(b)* Deficient siphon loop vacuum pump *(c)* Throttled CW valves	Increase dependent on severity	Increase proportional to CW flow reduction	Increase dependent on severity
Reduced Surface Area *(a)* Tube plugging *(b)* Debris on tubesheet *(c)* Low waterbox level	Increase dependent on severity	Increase proportional to CW flow reduction	Increase dependent on severity
Tube Fouling *(a)* Deposits/growth on tubes *(b)* Air in-leakage/inadequate air removal	Increase dependent on severity	Slight increase due to increased condition duty	Increase dependent on severity

Fig. 2-3.6.2.1-1 Performance Curves to Characterize Boiler Losses — Example for a Coal-Fired Unit



2-3.6.2.3 Performance Logic Trees. Logic trees present successive levels of information with increasing detail as the tree expands [35]. Figure 2-3.6.2.3-1 is an example of a typical heat rate logic diagram. Beginning with the initiating events of the logic tree and progressing through to the other events, a set of performance parameters is identified at each branch that acts as a roadmap for the user. The set of performance parameters selected should be measured to significant accuracy in order to ensure unambiguous interpretation based on the logic tree.

In close association with logic trees are decision trees. Whereas the logic tree gives multiple choices at branch points, the decision tree requires information to make decisions on which way to proceed [35, 36]. The structure of decision trees enhances the efficiency of identification of the potential cause of performance deviations. Their design should maximize the use of past history of a unit and allow future experience to be factored into the decision process. Figure 2-3.6.2.3-2 is an example of the decision tree concept for investigating deviations of main steam temperatures from target values.

A search of the Internet will identify commercial sources of logic tree and/or decision tree software that may be helpful to the station performance engineer.

Fig. 2-3.6.2.3-1 Heat Rate Logic Tree — Main Diagram (Courtesy Electric Power Research Institute)

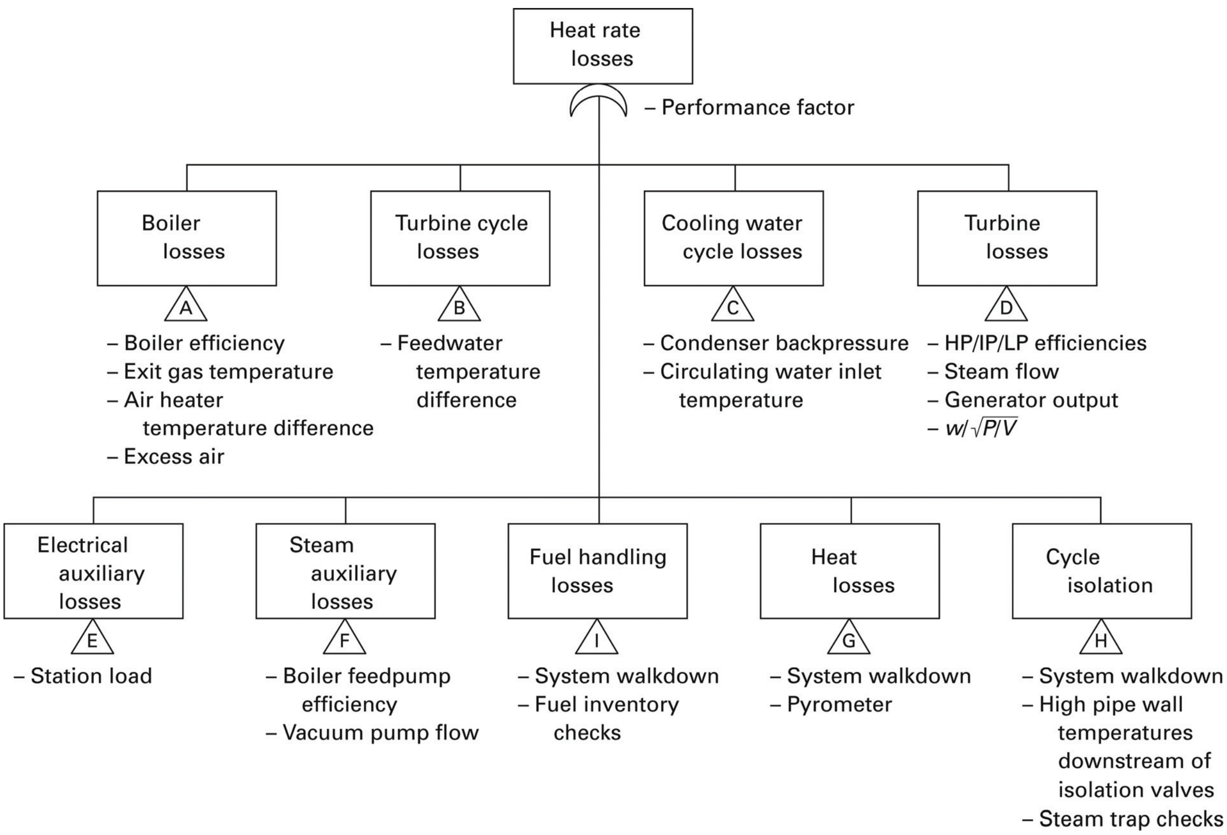
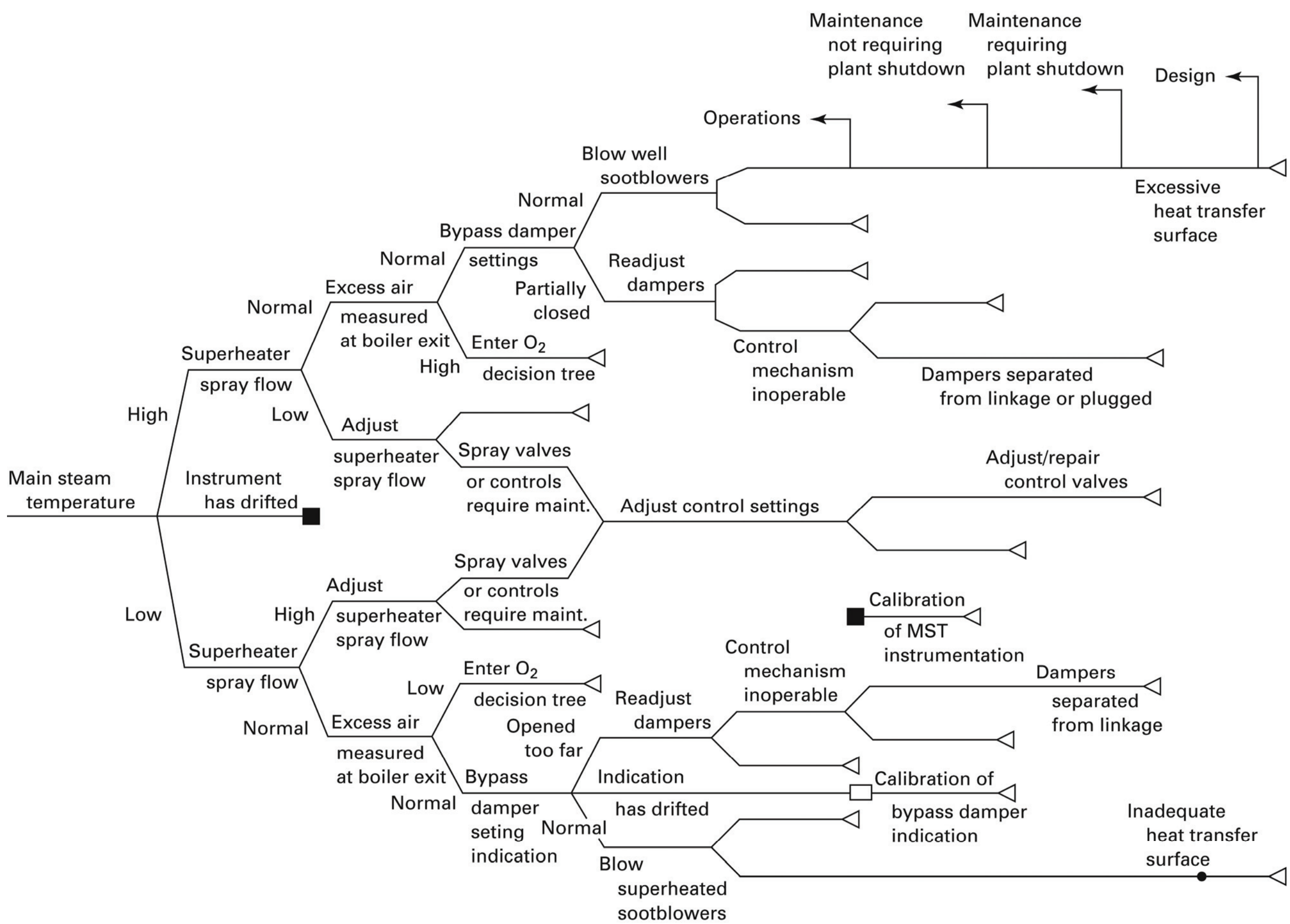


Fig. 2-3.6.2.3-2 Illustration of Decision Tree Concept for Investigating Performance Parameter Deviations (Courtesy Electric Power Research Institute)



2-3.6.3 Diagnostic Process. The diagnostic methodologies discussed in para. 2-3.6.2 should be incorporated into a formal process for tracing the root cause of an equipment or system performance problem. This paragraph describes such a generic process. The diagnostic process presented here is a simple yet formal procedure for isolating the root cause of a problem. The process presented entails the following criteria necessary for the development of an efficient diagnostic process [23]. The process should

(a) be deductive, calling for a step-by-step approach

(b) encourage the diagnostician to focus on the observed symptoms

(c) be flexible in the type of symptoms it may address

(d) provide for establishing testing and analysis programs to facilitate efficient diagnostics

(e) provide for making periodic judgments as to the cost-effective pursuit of the root cause

The root cause analysis should be carried out either to a level of detail that permits defining the corrective action necessary to prevent further occurrences of the failure, or to a level of detail at which it is judged that further analysis would not be cost effective. The following suggested diagnostic process, consisting of seven steps, represents a gradually narrowing scope of the problem [23]:

Step 1: Identify the components that are the source of the problem. In some cases this will be fairly obvious (e.g., feedwater heater out of service, condenser tube leak). In other cases, the responsible component may not be easy to identify (e.g., boiler losses can be caused by many different factors).

Step 2: Identify symptoms: for example, steam temperature or O_2 level cannot be maintained. This could be done by listing the functions of the components and determining which of these functions was impaired.

Step 3: As an extension of Step 2, describe the symptoms in as much detail as possible. For example, O_2 level is high or steam temperature is low, including under certain conditions.

Step 4: Postulate the deterioration or failure mechanisms. A deterioration or failure mechanism is defined as the physical process (electrical, mechanical, chemical, or metallurgical) or operating process that results in the occurrence of the specified problem and its symptoms.

Step 5: Define the features or characteristics of the problem that distinguish it from what it is not.

Step 6: Define scenarios that would result in the observed symptoms and postulate the root cause. The proposed scenarios could involve deviations from normal operating conditions in the plant. The most efficient way to isolate the scenario that produced the exact set of observed symptoms would be to analyze the features and characteristics defined in Step 5.

Step 7: Verify the conclusion reached in Step 6.

2-3.6.4 Plant Diagnostics. Diagnostics should proceed from a macroscopic to a microscopic view and in an orderly and logical fashion. At each step, the observed control volume becomes smaller, and key performance parameters that describe the control volume are evaluated. The following paragraphs follow this procedure, first looking at the plant as a whole, then proceeding to evaluate cycles, and then to the component level.

2-3.6.4.1 Unit Level Diagnostics. Unit level diagnostics should focus on variations in unit heat rate and maximum generator output. Investigations into the causes of an increase in heat rate are often initiated by a query from plant management simply stating that the routinely reported unit heat rate numbers have shown either an increasing trend or a sudden jump. The following discussion begins with suggested ways to logically approach the problem with no more than this minimal information.

Initial investigations of an unexplained increase in unit heat rate should focus on the behavior of the input (numerator of the equation defining heat rate) and output (denominator of the heat rate equation), keeping in mind that both of these parameters may be derived from calculation procedures incorporating many inputs and that the problem being investigated may ultimately start with one of these subsidiary inputs.

Many plants average heat rate data over periods ranging from several days to a month. This long-term averaging is useful from an accounting or business operation standpoint, but it tends to dilute equipment-specific performance information. Shorter time periods are also recommended for averaging, such as over a few hours or even minutes.

*(a) Unit Generation***.** A review of the data should look for both changes in the level and pattern of generation. Heat rate is typically higher when operating at part load. At a minimum, the instantaneous data should be examined to determine whether there have been changes in the loading of the unit that may have contributed to the observed rise in the heat rate.

Note also that it is important to determine if the maximum load capability of the cycle has changed. During periods when the system will purchase all the power the unit can produce, maximum generating capability is an important determinant of income to the plant.

*(b) Auxiliary Electric Power***.** Minimization of auxiliary electric power requirements means securing equipment when not in use, and optimizing the operating combination of equipment so as to minimize the auxiliary electric power requirements. This includes determining which pair of feed pumps to run if only two of three pumps are required for the existing load condition. Likewise, monitoring the percent auxiliary electric power (auxiliary electric power/generated load) may indicate undesirable changes in the operational configuration of some of the auxiliary equipment.

*(c) Fuel Characteristics***.** Examination of fuel characteristics and usage during the period in question is also in order. If the fuel usage reporting requires periodic estimations, changes in the fraction of the fuel usage being estimated should be noted. The magnitude of any required periodic fuel adjustments will provide insight into the accuracy of the fuel usage accounting techniques.

Inherent in the calculation of the energy supplied by the fuel is the determination of the heating value of as-fired fuel. The difficulties in determining the real-time heating value are many. On-line techniques for determining heating value fall into one of the following categories:

(*1*) lab-type techniques that can provide quasi-real-time measurement of the heating value of small samples of fuel. The resulting problem is the implementation of a sampling technique that would ensure representative samples of the fuel as it enters the boiler.

(*2*) on-line elemental or constituent analysis techniques that are capable of analyzing larger samples, but in which the heating value of the fuel is calculated, rather than measured.

Two problems dominate the consideration of as-fired fuel heating value. The first is concerned with determining a representative heating value for the period over which the heat rate is averaged. If the fuel usage has not been uniform over the period, the calculated heating value that is representative of that period should be mass weighted with regard to the fuel usage rate. A second problem is that often the fuel heating value is determined prior to the storage of the fuel in a holding facility (coal bunker or oil storage tank). To accurately determine the as-burned heating value as a function of time, an estimate must then be made of when the fuel actually enters the boiler.

Sampling errors, measurement errors, or part of both could sometimes appear as a very high or very low HHV when reported by a lab. It is desirable to verify that the reported HHV is within 1.5% or 2% of corresponding HHV correlation (Dulong's for moderate oxygen fuels, Vondracek's for higher oxygen fuels).

Consideration of varying moisture content is important. Heating value may not be varying on a dry, ash-free basis, but a higher ratio of fuel moisture for the same heating value results in lower boiler efficiency. Also, in the case of CFB Boilers, varying sulfur content will impact boiler efficiency over a measurement period.

2-3.6.4.2 Unit Cycle Diagnostics. The energy conversion cycle for a fossil plant typically consists of a boiler, turbine, condenser, feedwater heaters, pumps, and miscellaneous small heat transfer devices. Of these, it is relatively easy to analyze the boiler separately because it has only two or three principal interfaces with the rest of the cycle. These are the economizer and evaporator, the reheater, and second reheater, if it exists. It is important to recognize that the boiler may extract small quantities of hot water and/or steam from the main cycle for air preheating and supplying turbine-driven combustion air fans.

A common industry practice when preparing unit heat balances is to consider only the steam and water side of the boiler. These diagrams do include the energy used by the boiler for such things as air preheating and steam extractions, but would not include the fuel, air, and gas flows entering and exiting the boiler. These items are often available as heat balances originally prepared by the turbine manufacturer or the designers of the plant. They may also be referred to as “turbine cycle heat balances” or “steam/feedwater cycle heat balances.”

Overall unit performance diagnostics is done by comparing measured unit heat rate (meaning a heat rate calculated from measured parameters) to expected values. Once an increase in the heat rate is determined to represent a legitimate degradation in the unit performance, the evaluation should turn its focus to analyzing the boiler and the steam/feedwater cycle. The influence of these items on unit heat rate is examined through the use of the boiler efficiency and the turbine cycle heat rate.

(a) For a reheat unit, the primary transfers of energy from the boiler to the feedwater/steam cycle are

(1) the energy supplied to feedwater

(2) the energy supplied as superheat attemperation

(3) the energy supplied to cold reheat steam

(4) the energy supplied to reheat attemperation

(5) the energy supplied for auxiliary requirements

(b) The energy rejected by the turbine cycle to the condenser is composed of

(1) LP turbine energy rejected to the condenser

(2) energy rejected from all the drains dumping to the condenser

(c) Some of the more commonly monitored controllable parameters for unit level diagnostics are

(1) throttle steam temperature

(2) throttle steam pressure

(3) reheat steam temperature

(4) reheat pressure drop

(5) final feedwater flow

(6) generator output

(7) condenser pressure

(8) station electrical power

(9) excess air

(10) exit gas temperature

(11) superheat spray flow

(12) reheat spray flow

(13) steam and water loss from cycle

The difference between the as-operated heat rate and the target heat rate under the same conditions is known as the heat rate deviation. A portion of this deviation is the sum of all accountable heat rate deviations, and the remainder is known as the unaccountable heat rate deviation. Note that if the unaccountable losses are larger than the sum of the accountable losses, the performance monitoring program is deficient in scope and instrumentation. Through continued probing, the size of the unaccountable losses can be reduced by continually identifying new accountable losses. It is important to recognize that being truly effective in a heat rate enhancement effort requires the attainment of a good understanding of one’s plant and its operation.

Note that with regard to all components in the cycle, operating practices, as compared to degradation in equipment performance, can and routinely do influence cycle efficiency.

2-3.7 Steam Turbine Monitoring and Diagnostics

2-3.7.1 Turbine Measured Parameters

(a) HP/IP turbine steam temperatures and pressures (inlet, exhaust, and extractions)
(b) control valve (governor valve) position
(c) turbine exhaust pressure (see condenser section below for more information)
(d) reheater pressure drop
(e) gland seal steam leakage flows
(f) desuperheating flows
(g) feedwater and condensate flows
(h) throttle and reheat steam flows
(i) extraction flows to auxiliary equipment (BFP turbine, FD fan turbine, etc.)
(j) makeup flow to hotwell
(k) gross generator output
(l) auxiliary power
(m) turbine shaft bearing vibration
(n) turbine bearing oil drain temperatures
(o) thrust bearing metal temperatures
(p) oil cooler temperatures (inlet/outlet)
(q) pressure of generator cooling fluid (i.e., hydrogen or water)
(r) power factor

2-3.7.2 Turbine Calculated Parameters

(a) turbine cycle heat rate
(b) HP turbine efficiency
(c) IP turbine efficiency
(d) LP turbine efficiency
(e) turbine stage flow factors
(f) generator power factor
(g) corrected throttle steam flow
(h) corrected throttle steam pressures
(i) corrected stage pressures
(j) turbine section pressure ratios
(k) reduction gear box efficiency, if required

2-3.7.3 Performance Degradation Identification. Periodic monitoring of superheated turbine sections should be accomplished through enthalpy-drop testing. Data should be thoroughly evaluated for indications of degraded turbine performance. Condition of HP to IP packing leakage glands in combined HP/IP machines should be determined or estimated using HP to IP leakage procedures. Turbine condition should be quantified by conducting a steam path audit during outages. Results of the audit should be used to dictate type of repairs to be made. Upgrade of turbine design should be considered where justified.

All steam-water cycle leakages into and out of the cycle and leakages/bypasses within the cycle should be eliminated as much as possible. Sources of leaks to be investigated and corrected should include the following:

(a) feedwater heater drain system

(1) HP feedwater heater alternate drains, including emergency dump drains

(2) LP feedwater heater alternate drains

(3) deaerator auxiliary overflow to condenser

(b) extraction steam line drain system

(1) steam traps

(2) bypass orifice and drain valves

(c) steam line and turbine drain system

(d) boiler feed pump minimum flow system

(e) high pressure steam to BFP turbine system

(f) gland sealing steam system

(g) turbine water induction protection (TWIP) drain orifices

(h) safety valves

(i) steam sootblower system

(j) auxiliary steam system

(k) boiler blowdown system and sampling lines

(l) manual valves seat leakage and valve flange/packing leakage

(m) cross connect lines from adjacent units

(n) steam generator start-up pump

(o) boiler drain valves

(p) house heating steam supply from main steam or auxiliary steam

(q) main steam dump valves to condenser (nuclear and fossil)

(r) heater drain tank alternate drains to condenser (nuclear and fossil)

(s) reheater drain tank alternate drains to condenser (nuclear)

(t) steam generator pump recirculation drains to condenser (nuclear)

(u) MSR steam scavenging vent chamber drains to condenser (nuclear)

(v) long-path recirculation valves or warm-up valves (nuclear)

(w) steam supply to condenser spargers

2-3.7.4 Turbine Cycle Heat Rate. Turbine cycle heat rate testing is performed to determine the efficiency of the turbine cycle. The turbine cycle scope may range from the PTC 6 test requiring approximately 200 instruments (temperature, pressure, differential pressure, etc.) to the PTC 6.1 test requiring only 50 instruments. A combination of the two test methods may also be used. A review of PTC 6 should be conducted for a thorough understanding of the number and type of instruments required, measurement accuracy, isolation requirements, and pertinent calculations. The objective of a turbine cycle heat rate evaluation should be clearly identified initially so the appropriate level of instrumentation can be selected to meet the goal.

The PTC 6 full scale test method provides the most diagnostic information about the turbine cycle. Based on a primary flow measurement of the condensate to the deaerator, turbine cycle heat rate; HP, IP, and LP turbine efficiencies; the performance of all feedwater heaters; and generator output capacity are determined.

The alternative test method (described in PTC 6) produces a smaller set of results, ensures minimal uncertainty by employing an inspection port on the final feedwater nozzle for primary flow, and determines turbine cycle heat rate, HP turbine efficiency, top high pressure heater performance, and generator output capacity.

Another approach, using either a condensate flow nozzle or feedwater nozzle as the primary flow, determines turbine cycle heat rate, HP and IP turbine efficiencies, generator output capacity, and a combination of parameters in PTC 6 determined by the user, depending on the scope of the test. Required measurements for each method are contained in each respective performance test code.

Steam turbine performance should be benchmarked after every major overhaul or modification. Parameters of interest include maximum steam flow, section efficiencies, a pressure profile consisting of all extraction pressures, maximum capacity, and heat rate.

2-3.7.5 Turbine Enthalpy-Drop Efficiency. Enthalpy-drop efficiency testing of the superheated steam turbine sections of noncondensing turbines is conducted to detect performance deterioration in these sections of the main turbine. Required measurements include pressure and temperature data at the inlet and outlet of the HP and IP sections, the first stage pressure, and control valve stem travel for determination of valve points. Testing at valves-wide-open and at lower load valve point positions should be conducted to locate areas of degradation in the HP turbine section. For combined HP-IP turbine sections, estimates of true IP turbine efficiency and HP to IP leakage flow should be determined [1]. For double flow IP turbine sections, it is also recommended that state point conditions at both IP exhaust locations be collected in addition to the required LP crossover state point. This will ensure detection of performance problems relating to IP turbine imbalance.

The PTC 6S document should be consulted for recommended procedures when conducting this test.

2-3.7.6 Isolation Assessment. Assessment of the tightness of a turbine cycle is determined by applying the previously developed isolation criteria to the cycle and measuring the subsequent change in hotwell and deaerator storage tank level (given a constant drum level). After converting the level changes to mass flow rates, the net change is then divided by the throttle steam flow to obtain a percent leakage. Per PTC 6 requirements, the unaccounted-for leakage should be less than 0.1% of full load throttle steam flow for an acceptance test and 0.5% for a routine heat rate test. It is recommended that the 0.5% criterion be applied for performance monitoring purposes.

For nuclear-steam cycles, the same procedure should be applied to water storage in heater drain tanks, steam generators, hotwell, and condensate storage tanks.

Periodic water loss tests should be conducted to identify the magnitude of losses, and to take appropriate action to reduce their effect on unit performance.

2-3.7.7 Turbine Cycle Test Data Validation

2-3.7.7.1 Expected Relationships

(a) Extraction pressures are linear as a function of flow to the following stage (for a constant temperature) and must be zero for a zero flow.

(b) Section efficiency and pressure ratio are both constant as a function of flow except for the first and last stages.

(c) Flow factor is constant as a function of flow to the following stage.

(d) Negative temperature and pressure drops are inconsistent with basic laws of thermodynamics.

(e) Changes in enthalpy when none are expected, such as differences between enthalpy at a turbine extraction and heater inlet, should be suspect.

2-3.7.7.2 Power. Turbine shaft work as calculated by flow multiplied by tested enthalpy drop should equal measured shaft generation given proper accounting for the losses of the respective components.

On the HP shafts of cross-compound units where low pressure sections are not included, this comparison is direct, and any difference is indicative of data error. On turbine shafts including LP sections, an energy equation can be written to solve for LP section performance. The degree to which LP section efficiency falls outside expected bounds is indicative of data error.

2-3.7.7.3 Enthalpy–Drop Concerns. Turbine section expansion lines on enthalpy–entropy (h-s) diagrams must indicate less than isentropic expansion. The accuracy of extraction line pressure and temperature measurement can be displayed, in part, by variations introduced in expansion lines drawn between inlet and exit of major turbine sections. The determination of actual internal turbine enthalpy is compounded by the physical geometry of the turbine and the fact that the measurements are made remotely in the extraction piping downstream of the turbine proper. Relative to pressure, it is necessary to calculate a pressure drop from the turbine to the pressure tap in the extraction line. Relative to temperature, it must be noted that extraction pockets are located in the shell adjacent to the radial tip spill strip packing discharge point, such that the steam extracted and measured is higher in enthalpy than that flowing to the following stage in proportion to the spill strip leakage flow.

Given the above corrections, remaining irregularities in the expected shape of the expansion line are indicative of the errors in the total measurement correction process.

2-3.7.7.4 Reheater Pressure Drop. Reheater pressure drop as a percent of cold reheat pressure should be constant across the load range, and dismissing any reheater section modification, should be constant versus time. Otherwise, a change in this parameter is a result of an erroneous cold and/or hot reheat pressure or a significant change in N2 packing leakage or IP dummy flow on combined HP/IP turbine rotors (“N2 packing” is a term for the second endpacking counting from the front standard; “IP dummy” is a term for large diameter packing, which is also used to balance rotor thrust). The total reheater pressure drop from the HP turbine exhaust to the IP turbine inlet should be measured and not assumed. Turbines were often designed with a total reheater pressure drop typically ranging from 6% to 10% of the cold RH pressure. The true reheater pressure drop may be lower than these assumed design values.

2-3.7.7.5 Turbine Cycle Performance Calculations. Turbine cycle heat rate is the measure of efficiency of a steam turbine cycle. Defined as heat supplied to the cycle minus heat returned to the cycle divided by gross generator output, it is the standard by which performance is measured. Test turbine cycle heat rate is calculated using the enthalpy at the throttle steam, reheat steam, cold reheat, and final feedwater locations; measurement of boiler and reheat steam flows to the turbine; and measurement of electrical generator output. PTC 6A contains an example of how test turbine cycle heat rate is calculated.

HP turbine efficiency is calculated by dividing the used energy of the turbine section by the available energy. An example of this computation is found in PTC 6S. A plot of this efficiency versus throttle flow should be constructed to help determine areas of degradation. First stage pressure and control valve pressure drop are also recommended for use in evaluating degradation of this turbine section.

IP turbine efficiency is also calculated by dividing used energy by available energy. Although constant over the load range, the measured IP turbine efficiency for combined HP/IP turbine elements is often falsely higher than expected due to excessive N2 (IP dummy) packing gland leakage. Mixing high pressure leakage with the reheat bowl steam flow results in a higher pressure, but lower enthalpy steam condition, and therefore a falsely higher calculated value of IP efficiency. It is recommended that an estimate of the true efficiency and N2 packing leakage be calculated using published procedures [4].

The computation of LP turbine efficiency is performed only after the used energy end point has been determined from a total turbine cycle mass and energy balance, and the expansion line end point is determined by accounting for exhaust losses. PTC 6A provides an example for this calculation procedure. For turbine cycle heat rate tests in which the total cycle is not instrumented, an estimation of LP turbine efficiency is determined using the procedure found in ASME Paper No. 82-JPGC-PTC-6 [7]. Since the accuracy of LP turbine efficiency is dependent on the measurement of primary flow, it is recommended that a PTC 6 calibrated flow nozzle be used when LP turbine efficiency is to be determined.

Turbine shaft gland leakoff flows are measured and/or calculated. If measured, high pressure gland leakoffs (HP turbine inner glands) are determined using flanged union orifice plates, and low pressure gland leakoffs (HP turbine outer glands, IP turbine glands, etc.) are determined using forward-reverse pitot tubes. For calculated leakoff flows, the turbine vendor’s thermal kit should be consulted.

Measurement of these flows (versus calculation) will give an indication of seal wear and will result in a more accurate computation of heat rate and LP turbine efficiency.

2-3.7.8 Turbine Cycle Effect on Unit Performance

2-3.7.8.1 HP Turbine Efficiency. A change in HP turbine efficiency results in a change in reheater duty (for constant reheat temperature) and generator output, resulting in a change in turbine cycle heat rate. The change in generator output is counteracted by the change in reheater duty such that the heat rate is poorer with lower HP efficiency but improves from the lower reheater duty. The net effect on heat rate and generation is determined according to the equation contained in PTC 6S.

A good estimate of performance change is –0.16% change in heat rate and +0.25% change in generation for a +1% change in HP turbine efficiency at full load.

2-3.7.8.2 IP Turbine Efficiency. Although a change in IP turbine efficiency directly affects a change in generator output, the resultant energy change of the steam exiting this turbine section inversely affects the performance of the LP turbine.

A good estimate of performance change is –0.16% change in heat rate and +0.16% change in generation for a +1% change in IP turbine efficiency at full load.

2-3.7.8.3 LP Turbine Efficiency. ASME Paper No. 60-WA-139 [6] contains a method to calculate the effect on performance for a change in reheat turbine efficiency (IP and LP turbine section). Using this method, the contribution by the LP turbine efficiency is found by subtracting the IP turbine effect from the reheat turbine effect.

A good estimate of the effect on performance is –0.50% change in heat rate and +0.50% change in generation for a +1% change in LP efficiency.

2-3.7.8.4 Steam Seal Packing Flows. Excessive packing gland clearances result in an increase in leakoff flows, thus robbing the turbine steam path of motive steam to produce electrical energy. The effect on heat rate and generation can be significant, depending upon their location in the turbine cycle.

Measurement of these leakages should be performed using orifice plates and forward-reverse pitot tubes, the application of which is dependent on the location of the leakoff. For example, the use of a forward-reverse tube may not be possible if the leakoff flow path is entirely within the turbine casing, such as the leakage between the turbine first stage shell and the IP inlet on a combined HP/IP turbine in a single outer casing. Determination of excessive or less than expected flow effects on heat rate and generation should be performed using appropriate mass and energy balances per accepted engineering practice. Computer model sensitivity runs are recommended over hand calculations due to their expediency of execution and low risk of error.

Increased seal clearances in the N2 packing (or IP dummy) gland on a combined HP/IP turbine element result in a loss of steam flow through the HP turbine (downstream of the first stage blading). The net effect of the lower HP turbine output coupled with a lower reheater duty from the lower reheat flow is a lower generator output and thus a higher heat rate.

The loss in generator capacity is equal to the excess leakage flow multiplied by the difference in enthalpy of the first stage and cold reheat state points. The effect on heat rate is determined by accounting for the change in reheater duty and dividing the resultant boiler duty by the new generation.

For noncombined HP/IP turbine elements, the change in this leakage flow results in a change in HP turbine output only, since it mixes with cold reheat flow rather than hot reheat bowl flow.

2-3.7.8.5 Throttle Steam Temperature. A change in throttle steam temperature at constant throttle pressure and control valve position results in a corresponding change in throttle flow. As throttle temperature increases, the specific volume of the steam increases resulting in a lower mass flow rate. The subsequent effect on performance is a better turbine cycle heat rate and a decrease in gross generator output.

Similarly, a decrease in throttle steam temperature results in a poorer heat rate and an increase in generator output. Quantification of these effects is found in the turbine vendor's thermal kit.

2-3.7.8.6 Throttle Steam Pressure. A change in throttle steam pressure at a constant control valve setting results in a directly proportional change in throttle flow. An increase in throttle pressure results in a corresponding increase in mass flow rate from the decrease in specific volume of the steam and vice versa. The effect on performance is an increase in generator output and a decrease in heat rate. The turbine vendor's thermal kit should be consulted for determination of this contribution to performance.

Since nuclear power plants operate at a constant (licensed) reactor power, an increase in throttle pressure for a single stage reheat nuclear unit will result in a slight reduction in throttle flow, an increase in reheater heating steam flow, an increase in generator output, and decrease in heat rate. For example, a 5% increase in throttle pressure for a nominal 1,000 MWe pressurized water reactor nuclear power plant will result in an increase of about 6.3% in reheater heating steam flow, an increase of approximately 0.7% in generator output, and a like reduction in heat rate.

2-3.7.8.7 Hot Reheat Steam Temperature. An increase in hot reheat steam temperature at constant throttle steam conditions results in a corresponding increase in reheat enthalpy. The result on performance is an increase in generator output and a better heat rate. Refer to the cycle thermal kit for quantifying the effect on performance.

For a nuclear power plant with a single stage of reheat, an increase in reheat temperature will result in an increase in reheater heating steam flow, a reduction in throttle steam flow, an increase in generator output, and a decrease in heat rate. For example, an increase in reheat steam temperature of 5.0ºF for a nominal 1,000 MWe pressurized water reactor nuclear power plant will result in an increase of about 4.7% in reheater heating steam flow, an increase of approximately 0.12% in generator output, and a like reduction in heat rate.

2-3.7.8.8 Reheater Pressure Drop. The reheater pressure drop (expressed as a percent of HP exhaust pressure), defined as the percent pressure drop from cold reheat at the HP turbine exhaust to hot reheat at the IP turbine inlet, is usually stated in the turbine OEM thermal kit as 6% to 10%. However, most fossil units operate below this level, depending on the reheater design. Modifications to reheater surface area may change the reheater pressure drop and thus impact both the heat rate and generator output. A decrease in pressure drop will result in a better heat rate and a corresponding increase in generator output.

The reheater pressure drop is also used as a data validation technique for detecting measurement errors. Although this parameter should be constant over the load range, changes in operating conditions such as excess N2 packing (IP dummy) flows can change the percent reheater pressure drop by changing the flow through the reheater at constant steam conditions. Also, the addition of temporary fine mesh screens following an outage where work has been performed on the reheater will likely increase the reheater pressure drop until the screens are removed.

2-3.7.8.9 Auxiliary Turbine Steam Flow. A change in auxiliary turbine steam flow of 1% of main unit throttle flow has a corresponding 0.5% to 0.75% effect on turbine cycle heat rate and load. The exact value of the contribution is dependent on the source of the extraction steam (cold reheat or extraction after the reheater), the pressure at the source, and the percent of VWO throttle flow.

2-3.7.8.10 Superheater Desuperheating Water Flow. Superheater desuperheating water flow is used to control outlet steam temperature. Normally, this source of water is from the boiler feed pump and has only 0.01% to 0.02% effect on heat rate for a spray flow of 1% of throttle flow, depending on the throttle pressure of the turbine. The effect on generation is somewhat more at 0.07%. If the desuperheater flow originates downstream of the highest pressure feedwater heater (final feedwater), there is no effect on performance (heat rate or generation).

2-3.7.8.11 Reheater Desuperheating Water Flow. Reheater desuperheating water flow is used to control reheater outlet steam temperature. Originating from the boiler feed pump intermediate stage, the effect on performance of a spray flow of 1% of throttle flow is a 0.6% to 0.7% change in heat rate and generation, depending on cycle pressures and number of reheat stages. It is recommended that this flow be accounted for or isolated, if possible, during performance monitoring tests, since it has a direct effect on pressures at the reheat turbine and downstream, making turbine diagnostics more difficult.

2-3.7.8.12 Cycle Isolation. The following performance effects of cycle isolation are intended to aid the performance engineer's awareness of the impact that poor cycle isolation can have on the operation of a unit. It is recommended that a computer model of the specific unit be established for simulating different modes and degrees of isolation. Results of these effects should be used as feedback into the design of retrofit projects and new plants, to economically justify improvements in drain system and control logic design and in the specifications for valves, heater drain controls, etc.

2-3.7.8.12.1 HP Feedwater Heaters Emergency Drains to Condenser. When alternate high-pressure feedwater heater and deaerator drain systems are activated, or leakages occur through these systems, a loss in cycle efficiency occurs. The subsequent effect is an increase in condensate flow to the deaerator, resulting in increased extraction flows to the low pressure heaters as well as the heater immediately upstream (next lowest pressure heater) of the affected heater. The consequential decrease in steam flow through the downstream turbine stages results in less generator power output and thus a poorer heat rate.

For the top heater of a typical 300 MW fossil unit with reheat, a heat rate increase of 0.5% and a generator output decrease of 0.5% are realized for a 50% diversion of normal drain flow to the condenser. This same performance effect also holds true for the second highest pressure heater diverting 50% of its normal drain flow to the condenser. Diversion of 5% of these same drain flows results in a performance penalty of 0.1% in heat rate and generator output for the same type unit.

2-3.7.8.12.2 HP Steam to Auxiliary Turbines. On most steam-driven auxiliary turbines such as the boiler feed pump (BFP), an auxiliary source of high pressure steam is supplied for start-up purposes and low load operation. This steam source is most likely from main steam, but CRH or IP outlet can also be a source. However, HP stop valve leakage and the misconception that the normal low pressure steam supply is not adequate at higher loads often leads to the frequent supplemental use of this high pressure steam. The result is poorer cycle performance and sometimes sacrificed long term reliability of the auxiliary turbine (from simultaneous HP/LP steam admission).

For a typical 300 MW fossil unit with reheat, a 1% intentional use of main steam flow or equivalent leakage will result in a heat rate increase of 0.3% with a corresponding decrease in generator output of 0.3%.

2-3.7.8.12.3 Extraction Steam Leakage to Condenser. Leakage of high energy steam to the condenser usually occurs via cold reheat and extraction line piping drain lines used in the turbine water induction protection system.

Most drain lines are routed to a steam trap or bypass orifice/drain valve arrangement, and then to the condenser. Improperly operating steam traps and wide-open bypass drain valves (air regulator problems, etc.) allow passage of high temperature extraction steam to the condenser, thus robbing the turbine steam path of high energy motive steam and causing a decrease in power output.

Typical performance effects are a 0.2% change in heat rate and power output for a 5% extraction steam leakage of the second highest pressure feedwater heater, and a 0.3% change in heat rate and power output for a 5% highest pressure heater, deaerator, and BFP turbine extraction steam leakage. The performance effect for the highest pressure low pressure feedwater heater extraction is 0.1% (5% extraction leakage) with insignificant performance effects for the remaining low pressure feedwater heaters.

2-3.7.8.12.4 Out-of-Service Feedwater Heaters. A feedwater heater with tube leaks can be taken out of service using the bypass arrangement designed for such modes of operation. Though cost-effective in keeping the unit on-line and sometimes at a higher generator output, a performance penalty is incurred due to the lower final temperature of the feedwater to the boiler.

The performance effect for a 7-heater reheat cycle ranges from a 1% increase in heat rate for removal of the highest pressure feedwater heater to a +0.4% effect for removal of the lowest pressure heater. In a

5-heater reheat cycle arrangement, these effects increase to +1.3% (highest pressure heater) and +0.7% (lowest pressure heater).

In addition to poorer thermal performance, higher blade path flow will cause additional stress in certain parts of the turbine, such that removal of more than one heater from service is discouraged by the turbine manufacturer without a load reduction. The applicable turbine vendor instruction book should be consulted before proceeding with this mode of operation.

Most fossil-fueled utility boilers are not designed to generate rated steam flow with one or more HP heaters out of service. During this operating condition, the total amount of heat required for steam generation and hence, the rate of fuel firing, would increase significantly. Such overfiring could result in higher spray flow in the SH attemperators, higher metal temperatures especially in the primary SH, and higher loading on the air and gas fans. Unless a boiler is specifically designed for rated steam generation with HP feedwater heater(s) out of service, this operating condition can be detrimental to the boiler equipment, and represents an off-design operating condition for the boiler. The boiler vendor should be consulted about possible boiler operating restrictions under these conditions.

2-3.7.8.12.5 Manual Valves Seat Leakage and Valve Flange/Packing Leakage. The effect on a unit turbine cycle of leaking manual valve seats and flange/packing leakage will vary, but all will result in an increase in condenser makeup flow and thus a poorer heat rate and reduced power output. For example, heater bypass valve leakage for the final feedwater heater in a nuclear power plant will result in a decreased final feedwater temperature and an increase in core thermal power, thus resulting in a decrease in mass flow and electric power to maintain constant core power.

Any combination of these and/or other sources of leakage out of the cycle can be quantified using a net hotwell/deaerator storage tank level change test. With condenser makeup and all wasted boiler outputs (blowdown and sootblower steam) shut, a level drop in the condenser hotwell or deaerator will occur over a predetermined time period (usually 2 hr), given a constant drum level (for subcritical fossil units). The loss of water is calculated using the dimensions of the deaerator and hotwell storage tanks and converting the level changes to mass flow rates. Expressed as a percentage of throttle steam flow and assumed to have occurred in the steam generator, a 1% loss of water out of a typical fossil turbine cycle will result in a +0.2% change in heat rate and a –0.2% change in power output.

The throttle steam flow for a test of this type can be approximated using the turbine first stage pressure/flow curve, and the cycle loss sensitivity effect can be obtained using makeup correction curves.

In a nuclear power plant, leakage though the top heater bypass valve will lower the final feedwater temperature. To maintain reactor core power at the licensed limit, the result is a decrease in feedwater flow and reduced output.

2-3.7.9 Turbine Cycle Diagnostics

The overall performance of the turbine cycle can be characterized by comparing the unit's turbine cycle heat rate to expected values. The following diagnostic techniques are discussed in more detail in reference [6].

2-3.7.9.1 Flow Capacity (VWO) Check. A comparison to target of the turbine valves-wide-open flow capacity can be used to diagnose several problems. The throttle flow, W_t, to the turbine should be corrected to reference (design) conditions (see PTC 6S Report-1988) before comparing to a target condition, $W_{t,t}$.

The following are possible problems associated with the different findings:

(a) $W_t > W_{t,t}$

(1) first stage nozzle erosion

(2) low first stage pressure due to excessive packing clearance or enlarged second stage

(3) excess leakage and/or bypass in bell seals (piston rings), start-up drain valves, etc.

(4) valves for high-pressure steam to auxiliary turbines and steam seal systems are leaking or open

(5) flow measurement is in error

(b) $W_t < W_{t,t}$

(1) first stage nozzle area is reduced

(2) restriction in high pressure turbine reaction stages, indicated by a high first stage pressure relative to downstream pressures

(3) flow measurement is in error

It is prudent to check the flow measurement transmitters first.

2-3.7.9.2 HP Turbine Efficiency Across Load Range. HP efficiency should be plotted against percent valve position, throttle flow, or pressure ratio. If the difference between design and test is greater at minimum load than at VWO, it would indicate that the first stage performance has deteriorated more than the latter stages. Conversely, if the difference is less at part load, it usually indicates the latter stages are more affected than the first stage. Note that poor section efficiency and low first stage shell pressure may be the result of excessive leakage through the internal packing, provided that leakage is returned to the main flow prior to where the HP turbine exhaust temperature is measured. Main steam piston ring leakage is an additional factor that can affect the apparent HP section efficiency depending on which rings leak and what valves are open.

In general, the HP section performance will deteriorate much more than other sections. The most common causes of the HP deterioration are rubbed seals, excessive leakage, and solid particle erosion.

2-3.7.9.3 Turbine Flow Function Across Load Range. The flow constant, K, is an indicator of stage nozzle area changes when plotted against throttle flow.

$$K = W/(P/v)^{1/2}$$

where

K = flow constant

P = pressure (absolute) to the following stage

v = specific volume to the following stage

W = flow to following stage

Information pertinent to the internal conditions of the turbine is provided using the flow factor equation. Although this relationship is not accurate if changes in local areas occur, it can be used to recognize that a change has occurred and to help estimate the magnitude of that change. The stage flow coefficient is an indicator of test consistency, serving as a check on testing errors. The absence of scatter in the plot of flow coefficient versus throttle flow (test data lying on the same curve as design data) signifies accurate pressure and temperature measurements.

In addition, it can also be used to calculate shaft-packing and valve stem leakoff flows. The constant is calculated from acceptance or design data, and the calculated flow may be compared against the actual measured values to determine the location of the steam seal damage.

2-3.7.9.4 First Stage Pressure Versus VWO Flow. Turbine first stage shell pressure, although not providing an accurate determination of throttle flow, is a good indicator of short-term changes in throttle flow when used in conjunction with other turbine performance monitoring measurements. A comparison of design versus first stage pressure plotted against throttle flow is used to determine whether the first stage pressure is high, low, or at design. After correcting first stage pressure to reference steam conditions, the following can be considered as possible causes for a deviation from the design value:

(a) test first stage pressure is high
   (1) throttle flow measurement is in error on the low side
   (2) second stage nozzle area is restricted
(b) test first stage pressure is low
   (1) throttle flow measurement is in error on the high side
   (2) second stage nozzle area is enlarged
   (3) adjacent packing may be badly rubbed
   (4) other leakages or bypasses may exist

2-3.7.9.5 IP Turbine Efficiency Across the Load Range. For units consisting of combined HP/IP turbine elements, cooler steam leaking from the HP to the IP turbine section cools the steam entering the IP turbine. As a result, the measurement of the IP turbine efficiency will be calculated erroneously high if not properly compensated. When this cooler steam mixes with hot reheat steam, the amount of IP efficiency error will vary approximately as the difference in enthalpy between the leakage steam and the hot reheat steam. This error will thus decrease as the initial (main steam) temperature is raised and/or reheat temperature is lowered. Conversely, this error will increase if initial temperature is lowered and/or reheat temperature is raised. Utilization of this phenomenon has resulted in a procedure described in a paper by Booth and Kautzmann [4] that estimates the HP to IP leakage flow and the true IP turbine efficiency.

High IP section efficiency and low first stage pressure are usually indicative of high leakage rates.

Note that in addition to leakage in the packing between the HP and IP inlets, additional leakage flows from the HP inlet to the IP inlet may also be attributed to leakage from the HP inner cylinder joint or the piston ring seals.

2-3.8 Boiler Monitoring and Diagnostics

2-3.8.1 Measured Boiler Performance Parameters

2-3.8.1.1 Boiler Measured Performance Parameters

(a) steam and water temperatures, pressures, and flows at
   (1) economizer inlet
   (2) economizer outlet
   (3) waterwall inlet and outlet (supercritical unit)
   (4) intermediate SH and RH sections
   (5) SH and RH desuperheater inlet and outlet
   (6) main steam (SH outlet)
   (7) cold reheat (RH inlet)
   (8) hot reheat (RH outlet)
(b) auxiliary, extraction, and sootblowing steam flows
(c) SH and RH tube metal temperatures
(d) drum pressure
(e) drum level
(f) boiler circulation pumps amps, differential pressure
(g) blowdown flow from the steam drum(s)
(h) sample conditioning water
(i) fuel analysis (HHV with ultimate and proximate analysis)

2-3.8.1.2 Pulverizers and Burner Equipment Measured Performance Parameters

(a) number and location of mills in service
(b) fuel flow
(c) mill outlet temperatures
(d) mill motor amps
(e) primary and tempering air flow
(f) burner register settings
(g) fuel nozzle tilt position
(h) number and location of gas and oil burners
(i) auxiliary fuel flow
(j) atomizing steam flow
(k) windbox/furnace differential pressure
(l) windbox air damper positions
(m) overfire, underfire, and arch air damper positions and nozzle tilt
(n) mill fineness
(o) mill differential pressure

2-3.8.1.3 Air Heaters Measured Performance Parameters

(a) gas inlet/outlet temperature, pressure
(b) primary air inlet/outlet temperature, pressure
(c) secondary air inlet/outlet temperature, pressure
(d) air preheating coils inlet/outlet temperatures
(e) flue gas inlet/outlet analysis, wet or dry, O_2 as a minimum
(f) primary and secondary air outlet flows, if available
(g) air and gas pressure differentials
(h) unburned carbon in ash
(i) furnace gas pressures and differential pressures
(j) boiler exit gas temperature

2-3.8.1.4 Fans Measured Performance Parameters. Measured performance parameters for FD/PA/ID/GR fans are inlet and outlet pressures; temperatures; flows, if available; damper positions; blade angles (variable pitch fans); motor amps; ambient air temperature; barometric pressure; and humidity.

2-3.8.2 Steam Generator Calculated Parameters

(a) excess air
(b) boiler efficiency
(c) flue gas flow
(d) air flows
(e) air heater performance parameters (leakage, gas side efficiency, air-side and gas-side differential pressures, *X*-ratio)
(f) air heater exit gas temperature corrected to the zero leakage condition
(g) SH and RH spray flows
(h) boiler surface cleanliness

2-3.8.3 Steam Generator Equipment Performance Degradation

2-3.8.3.1 Pulverizers and Burner Equipment. Combustible unburned carbon data (preferred to loss-on-ignition data) from fly ash, loss-on-ignition (LOI) data should be collected, and causes of excess carbon carryover identified. If high moisture-containing fuels, such as powder river basin (PRB) and lignite are burned, testing in addition to LOI testing may be required. Air flows at mill inlets and outlets should be initially checked to ensure proper flows and distributions, and to eliminate pulverizer spillage. Mill performance should be monitored using mill fineness, mill amps, measured air and coal flows, coal spillage, coal grindability, raw fuel size, and moisture in coal. Secondary air flow distribution should be checked where practical to ensure proper air flow distribution, and to maintain the proper stoichiometry in the combustion zone.

2-3.8.3.2 Boiler. Oxygen (O_2) analyzer systems should be investigated for proper location and accuracy. Boiler casing leaks should be identified by safe available methods. All ductwork and expansion joints should be inspected for possible leakage. Excessive boiler air in-leakage can also result in changes in air heater performance, reduced boiler efficiency, and higher ID fan power.

2-3.8.3.3 Air Heaters. Levels of air heater leakage should be determined. Excessive leakage will result in increased fan power consumption, greater potential for cold end corrosion, and potential load curtailments due to insufficient combustion air or induced draft capacity. Minimization of air heater leakage should be implemented through regular seal maintenance or repair and review of available sealing system improvements or upgrades. Tubular and heat pipe air heaters should also be checked for leaks due to corrosion, erosion, or mechanical damage.

2-3.8.3.4 Fans. The actual physical position of damper blades should be checked against control room or actuator readings. A partially closed damper will reduce fan capacity, potentially affecting unit output, and will waste energy.

2-3.8.4 Steam Generator Equipment Testing

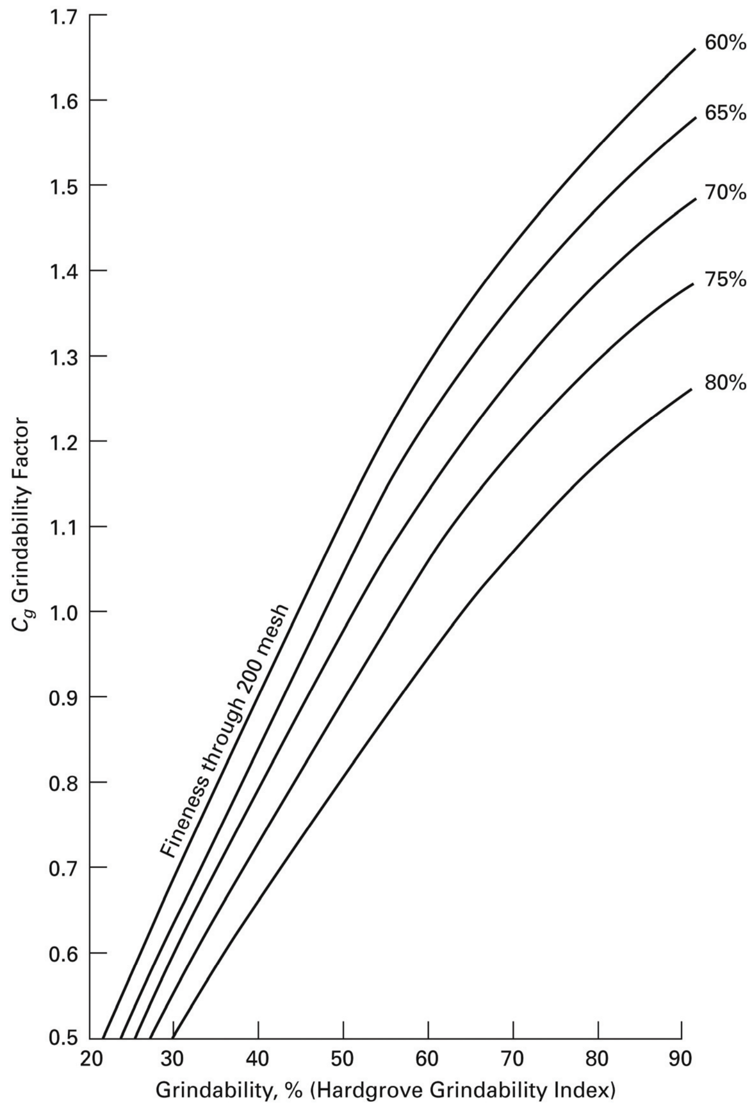
2-3.8.4.1 Pulverizer Testing. Pulverizers are selected to provide a required grinding capacity based upon a set of design criteria that include the required coal particle fineness and coal characteristics. The required pulverizer capacity is determined by the boiler heat input requirements, with some additional allowance added to account for wear of the mechanical parts. The required fineness is a function of specific coal properties such as the volatiles in coal, raw fuel feed size, and the rank of the coal. It is also a function of the type of burners, firing system, and furnace size. Coal characteristics that determine mill performance are the grindability of the coal and the moisture in the coal.

Pulverizer capacity will vary according to the current operating conditions. Figure 2-3.8.4.1-1 shows typical relationships between mill mechanical capacity, coal grindability, and coal particle fineness. As an example, assume that the coal Hardgrove grindability increases (which means that the coal is easier to grind). This figure would show that the mill would either be able to grind an increased amount of coal to the same fineness, or it could grind the same amount of coal to a greater fineness.

Pulverizer thermal capacity is affected by such factors as the moisture in the coal and the available primary air inlet temperature and flow. To a lesser extent, ambient air temperature and raw coal temperature will also affect mill thermal performance.

A higher-than-design moisture content of a coal has a negative effect on both pulverizer and boiler performance. The additional moisture, either removed from the coal in the mill and transported in the primary air, or remaining with the coal particles entering the furnace, carries heat away from the combustion process. This becomes part of the moisture heat loss that impacts boiler efficiency. The added moisture also requires additional drying and grinding time in the mill, utilizing more mill motor power for each ton processed. On average, the heat rate is impacted by 0.10% for each 1% increase of coal moisture in a typical Eastern Bituminous coal, and 0.17% for each 1% increase of coal moisture in a typical high moisture Western Sub-bituminous coal.

Fig. 2-3.8.4.1-1 Pulverizer Capacity Curve



An increase in coal moisture content can be attributed to any of the following:

(a) change in source at the mine.

(b) seasonal or isolated moisture addition from rain or snow during transport and storage. Note that a raw coal containing a high percentage of fines (greater surface area) will retain more surface moisture.

(c) inadequate coal pile drainage.

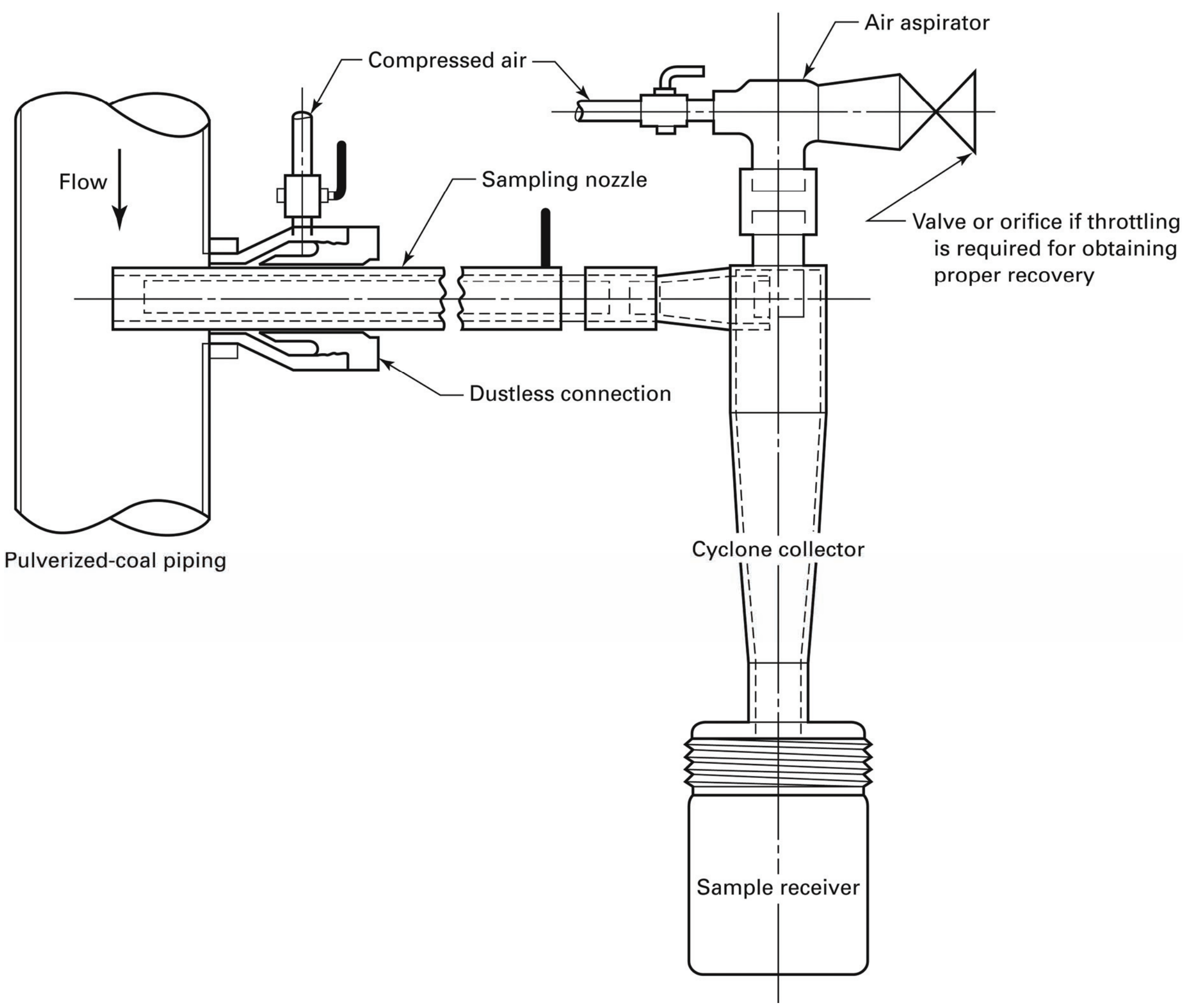
(d) change in coal blending practices.

Clean air curves are a plot of mill differential versus stationary pitot tube differential with no coal flow. They represent the pressure drop through the mill due to its internal components. As such, they give an indication of wear by comparing clean air curves over time. These curves are typically used to establish the mill-airflow calibration and to verify proper primary airflow balance to the burners. The number of hours of operation of a pulverizer, tons of coal processed, or increased pulverizer spillage are generally used to determine mill maintenance requirements.

Stationary pitot tube calibration is done to obtain the K factor of the pitot tube for use in pulverizer calibration. Individual burner lines are traversed at several different flow rates, such as minimum mill load, 50% mill load, and 100% mill load, while the corresponding pitot tube differential is measured. Based on the mass flow rate of air through the mill, a flow rate versus pressure drop relationship is determined that enables the calculation of a pitot tube calibration factor. This information is then used to establish the mill loading curve and the relationship between fuel and air mass flow rates as a function of pitot tube differential pressure.

Pulverized coal fineness is determined from pulverized coal samples taken every 30 min from mills that are tested within the guidelines of PTC 4.2. There are two methods of collecting the pulverized coal sample: the ASME PTC 4.2 method and the International Organization for Standardization (ISO) method. Figure 2-3.8.4.1-2 shows the PTC 4.2 arrangement for sampling pulverized coal in a direct-fired system using a dustless sampling connection with an aspirator and a cyclone collector. In collecting the sample, the compressed air is turned on to the dustless connection and adjusted to give a balanced pressure at the connection. The sampling tip is inserted in the dustless connection, and the compressed air is again adjusted to maintain a balanced pressure. Then the fuel transport line is traversed holding the sampling tip facing the coal-air stream at predetermined positions for equal periods of time. Samples should be obtained from two taps in the same plane, located at 90 deg to each other. Each fuel transport line leaving the mill should be tested.

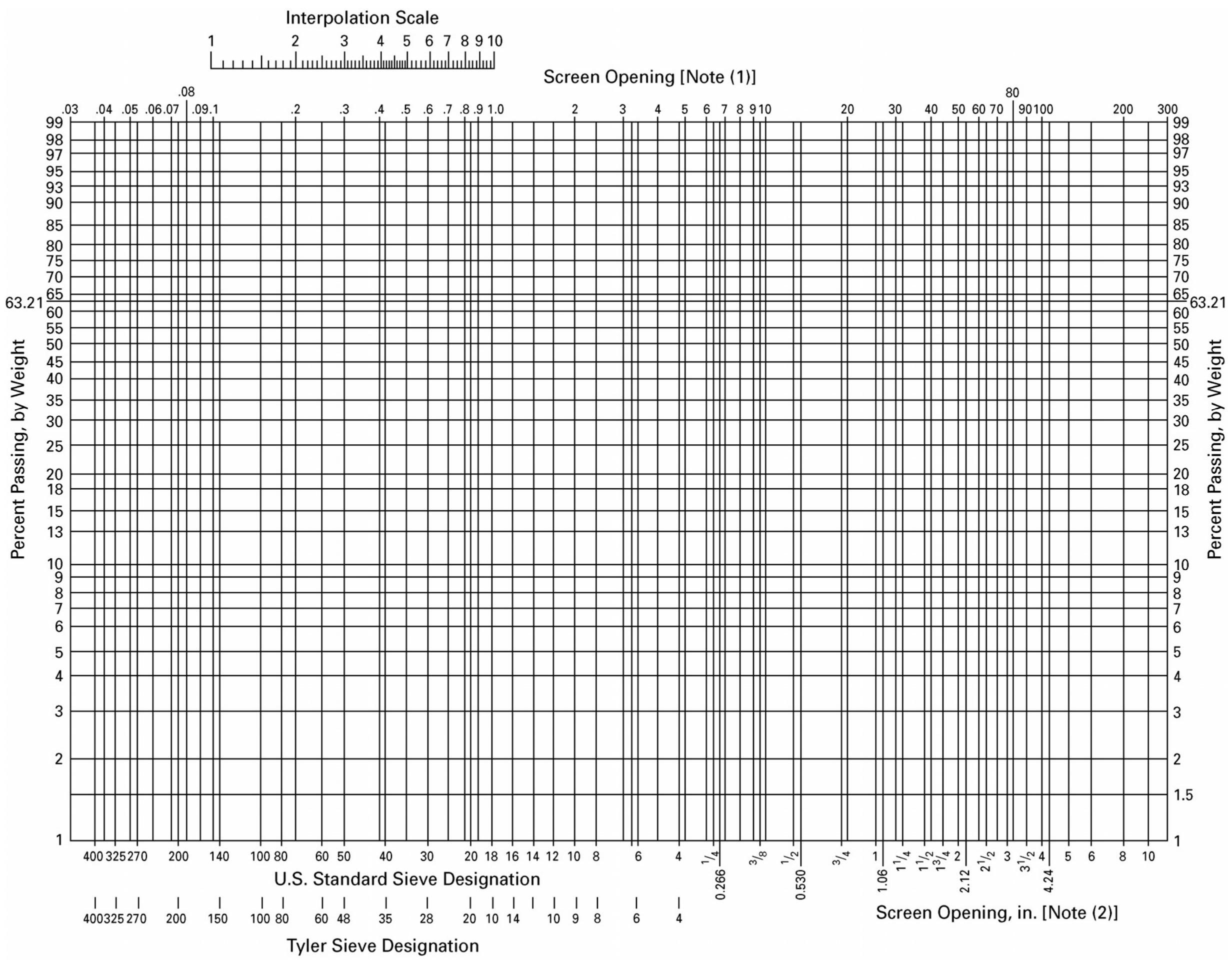
Fig. 2-3.8.4.1-2 Arrangement for Sampling Pulverized Coal



The second method of obtaining a pulverized coal sample, which is being used more frequently, is the ISO method. This method uses a transport pipe tap for probe insertion. The probe has a rotating head that allows for more sampling points than are available with the PTC 4.2 method. Additional sampling points can be an advantage when there is roping of the coal in the pipe [43, 44].

With collection complete, the pulverized coal samples from each mill are first dried then thoroughly mixed. Fifty grams of the sample is then placed in the top sieve of a nested stack of 50-mesh, 100-mesh, and 200-mesh sieves. The nest is then shaken either by hand or machine until the procedure has separated the coal particles by size. The results of the percentages of coal passing through the different mesh sizes should plot as a reasonably straight line on a Rosin and Rammler Probability Chart, Fig. 2-3.8.4.1-3. If the percentages do not fall on a straight line, either a computational error or improper sampling has occurred, or this may indicate mechanical problems within the mill. Investigation should continue until the results fall on a reasonably straight line.

Fig. 2-3.8.4.1-3 Graphical Form for Representing Distribution of Sizes of Broken Coal



GENERAL NOTE:
From: Landers, W. S., and Reid, W. T., A Graphical Form for Applying the Rosin and Rammler Equation to the Size Distribution of Broken Coal: Bureau of Mines Inf. Circular 7346; 1946.

NOTES:
(1) Any scale. If in millimeters, coincides with lower scale.
(2) Square hole if used as continuation of fine series.

Average particle size (intersection of size distribution line with 63.21% passing line)	$\bar{X}$	
Slope of size distribution line (tangent of angle)	N	

2-3.8.4.2 Steam Generator Testing. Boiler efficiency testing should be done periodically, preferably before and after unit outages, to monitor performance relative to design conditions. Also, a comparison can be made between expected performance and original design if significant modifications to the boiler, either from an equipment or operations point of view, have been made.

For rigorous performance test work, PTC 4 should be consulted for the proper procedures and calculations for the input–output and heat loss boiler efficiency tests. The heat loss method is preferred for performance monitoring.

Typically, for boiler performance testing, the control volume is from the air heater air inlet to air heater gas outlet. Air heater performance testing and boiler efficiency tests should be run concurrently to effectively use available resources (e.g., manpower and test equipment).

As noted in the instrumentation section, flue gas analysis can be obtained from either portable O_2 analyzers, CO_2 analyzers, or Orsat analysis. Again, for performance monitoring, the O_2 entering the air heater can be obtained from in situ oxygen analyzers, provided they have been calibrated and a sufficient number of them have been properly located to give a representative O_2 reading in the economizer outlet ductwork. More rigorous performance testing requires O_2 traverses at the economizer outlet due to stratification through the boiler. The air heater gas outlet ductwork needs to be traversed to obtain a flue gas analysis because many units do not routinely have O_2 analyzers located downstream of the air heaters. In the interest of time, a gas averaging apparatus can be used, where multiple gas samples can be simultaneously fed into the apparatus, so that only one point needs to be evaluated. Based on the results of this type of performance monitoring, more rigorous testing (multipoint duct traverses) may be warranted, at which time the appropriate PTC should be consulted.

During the test, steady state conditions should be maintained at rated steam conditions. Air heater bypass dampers should be closed for the duration of the test and gas flows held constant. Typically, steam generator output should be stabilized for 30 min to 60 min prior to taking data. The data necessary to evaluate performance of the boiler and heater can then be taken during the next 30 min, at 5-min intervals, provided conditions are reasonably constant. The duration and frequency of data collection is dependent on the goal of the test and the accuracy of results required. See ASME PTC 4-1998 for figures illustrating the boiler testing procedures.

When calculating air heater leakage, it is important that the performance engineer use results from the flue gas analysis consistently on the air heater inlet to outlet, either both on a dry basis or wet basis. If O_2 analyzers are used at the inlet, and CO_2 is measured at the outlet either by CO_2 analyzer or Orsat analysis, it will be necessary to convert CO_2 to O_2 on a volumetric combustion chart.

Boiler efficiency, defined generically as the ratio of the heat absorbed by the working fluid or fluids to the heat input, can be determined by either the input–output method or the heat loss method. A third method, which will be described briefly, can also be used to obtain boiler efficiency. Each test requires obtaining specific data. A review of PTC 4 should be conducted for a thorough understanding of the procedure, measurement accuracy, and efficiency calculations for the input–output and heat loss methods.

Although PTC 4 was written to provide a rigorous analysis of efficiency, alternative methods of analysis can be employed for performance monitoring that is less costly and time consuming without sacrificing a great deal of accuracy. For the purpose of performance monitoring as opposed to a guarantee/acceptance type test prescribed in PTC 4, the list of measurement and calculation parameters can be reduced to include only the major losses. Referring to PTC 4 Figs. 1.4-1 through 1.4-7, a review of steam generator boundaries, heat inputs, and heat outputs can be made to establish specific monitoring goals. For example, it may be that the immediate need is only to monitor air heater performance, or another specific portion of the boiler circuit.

The input–output method requires accurate measurements for fuel flow, fuel higher heating value, feedwater flow, steam flow, and reheater flow. This method may be best applied to boilers using gas or liquid fuel that have calibrated fuel flow meters. The steam/water flow measurements can be obtained from station flow meters or calculated from a PTC 6 Steam Turbine Test. Since this method only calculates boiler efficiency without giving a breakdown of the boiler efficiency components and is primarily used for gas and liquid fuels, it is not recommended for monitoring purposes.

The heat loss method determines the losses from the boiler. The calculated boiler efficiency resulting from the heat loss method is not as sensitive to measurement errors as the input–output method. Refer to PTC 4 for the effect of measurement errors on efficiency. In addition, the losses are quantified under loss categories that will aid the operations and performance personnel in locating and reducing the losses. For routine testing, the major losses are the dry gas loss, the hydrogen and moisture in the fuel loss, unburned combustible loss, moisture in the air loss, radiation loss, and unaccountable loss. The unaccountable loss is a constant amount to account for losses that are too difficult or too small to measure. The heat loss method is best suited for pulverized coal units when coal flow cannot be accurately measured and the higher heating value varies.

The third method for determining boiler efficiency is called the output loss method. Boiler efficiency calculated by the output loss method is a function of boiler heat duty (energy transferred to the steam in the boiler), mass flow rate of gas leaving the stack, and fan and pulverizer power. It is independent of the higher heating value of fuel and fuel mass flow rate. This method has potential for continuous on-line performance monitoring of steam generators, but it has not been accepted as a standard in guarantee testing because it yields less accurate results, compared to the heat loss method.

PTC 4 calculation sheets may be used to facilitate the calculation of results. In an effort to reduce the cost of the monitoring program, the user may wish to eliminate some of the measurements that account for the minor heat losses and credits. The values of the unmeasured losses and credits can be estimated from historical data. It is also important to note that the value of as-tested efficiency is not as important a diagnostic tool as is the trend of losses and credits corrected to reference conditions.

The major heat losses are generally heat in dry gas, moisture from fuel, and moisture from the combustion of H_2. The major losses and many of the smaller losses are all dependent on the measurement of flue gas temperature and composition at the air heater gas inlet and outlet. It is essential that a representative measurement of flue gas O_2 and temperature be obtained from a multipoint grid entering the air heater where stratification is likely to occur. It is also essential that an accurate determination of air heater exit gas temperature, excluding leakage, be made.

(a) Losses to accurately determine on a regular basis are as follows:

(1) dry gas loss

(2) water from combustion of H_2 in fuel loss

(3) water from H_2O in fuel loss

(4) unburned carbon in ash

(b) Other losses to calculate or estimate are as follows:

(1) moisture in air loss — use typical percent moisture in air or actual values if available

(2) sensible heat of refuse — use typical bottom/economizer/air heater ash splits and estimate

(3) radiation loss from hot-side precipitators — estimate from historical data

(4) radiation loss — estimate radiation losses as determined in ASME PTC 4

2-3.8.4.3 Air Heater Testing. Performance testing of an air heater is conducted to determine gas side effectiveness, air leakage, and air/gas side pressure differentials. Other related operating characteristics of significant importance are *X*-ratio and air and gas temperatures.

During unit outages, air heaters should be inspected. On regenerative air heaters, a check for proper seal clearances and tight shut-off of all bypass and/or recirculation dampers should be conducted. If necessary, all heat-transfer surfaces should be washed in accordance with the manufacturer's recommendation to eliminate any pluggage. Care should be taken to give a thorough washdown so that wet ash does not lay out between layers of baskets (hot, cold, and intermediate), which will accelerate corrosion and significantly reduce the life of the baskets.

Temperature, pressure, and oxygen sampling instruments should be located in the air and gas ducts upstream and downstream of the air heaters. The number of points to sample during a traverse will be dependent on the location of the test connections and the severity of stratification at that location.

Entering air temperature affects air heater performance. Changes in entering air temperature will result in a change in temperature head, which directly affects outlet gas temperature. A 10°F increase in air-heater inlet-gas temperature will increase the exit gas temperature by 10°F times (1.0 – gas side efficiency).

Higher-than-design gas mass flow rates will increase exit gas temperature. A common error made by performance engineers is to assume the air heater's poor performance is based on temperature measurements alone; however, the ratio of air to gas flow and its effect on performance must be known beforehand. The manufacturer should be consulted for the appropriate correction curves. For trisector air heaters, deviation from design in mass flow rates of primary air, secondary air, and gas must also be accounted for in air heater performance analysis. Excessive boiler air in-leakage can also result in changes in air heater performance and reduced boiler efficiency. Air and gas flow pressure drop, trended over time, will give the engineer and operator an idea of the degree of air heater pluggage. PTC 4.3 should be consulted for recommended procedures in conducting an air heater test.

Some applications of performance monitoring systems will require that minimal instrumentation be added to implement an air heater leakage program. As noted previously, O_2 analyzers typically need to be added to the ductwork downstream of air heaters in order to monitor air heater leakage on a continual basis. Due to the same data needs as a boiler efficiency test, air heater and boiler efficiency tests are usually conducted simultaneously.

Regenerative air heaters with seals in good condition should have leakage rates between 6% and 8% on pulverized coal fired units, and 10% to 12% for cyclone fired units. The reason for the greater leakage rate on cyclone fired boilers compared to pulverized coal fired boilers is that cyclone fired boilers have greater air-to-gas side differential pressures. The leakage rate will depend on the differential pressure between the air and gas sides of the air heater, the degree of air heater pluggage, and the seal condition.

2-3.8.4.4 Fan Performance Testing. PTC 11 provides methods for rigorous analysis of fan performance but may not be practical for a performance monitoring program. For routine performance monitoring, simplified assumptions and test methods may be employed.

PTC 11 should be consulted for the recommended procedures to conduct rigorous fan tests. An approximation of fan efficiency can be made by simply measuring fan power and static pressure rise across the fan and recording the inlet vane position. A combustion calculation using fuel flow rate, typical ultimate analysis of fuel, and back-end O_2 will yield flow rate that can be plotted on a manufacturer's curve to determine if there is a problem that needs to be investigated further through more accurate testing. Note that measured pressure and horsepower should be adjusted by the following density ratio before plotting: $\rho_{\text{curve}} / \rho_{\text{measured}}$.

For the measurement of fan flow, a grid of probes designed to measure the fluid total pressure and static pressure are required, preferably removed from obstructions. The probes can be designed such that they are sealed at the end inside the flue, pressure sensing holes are drilled along the length, and they are inserted so that the pressure sensing holes are aligned in the direction of oncoming flow.

The pressure sensing holes should be equally spaced along the length of the probe and should be sized so that the cross-sectional area of the pipe is at least 8 times the total area of the sensing holes. PTC 11 calls for elemental areas for sensing pressure not to exceed 2 ft^2, but for the purposes of performance monitoring, this requirement may be relaxed to every 3 ft^2 to 4 ft^2. When measuring the flow of particulate laden gas, sensing hole pluggage is a concern, and periodic purging of the probes may be required.

Static pressure measurement at the location of the grid is also required. The velocity pressure is the total pressure measured by the grid, less the static pressure. Fan flow, from the flow grid, is determined by the equation

$$\text{Flow} = K_1(P_v/\rho)^{0\ 5}$$

where

K_1 = calibration factor of flow grid from testing (including unit conversions)
P_v = velocity pressure from the grid
ρ = density of the flowing fluid

The calibration factor K_1 should be developed by concurrently measuring the flow using the grid, and obtaining the actual flow rate using the methods prescribed in PTC 11. The calibration should be done over the practical operating range of the fan.

Static pressures are required at the fan inlet and outlet and at the location where flow rate is measured. Static pressure can be measured with an appropriately designed probe or with pressure taps installed on the walls of the flue or duct [reference Air Movement and Control Association (AMCA) publication 803].

Air/gas temperature at the fan inlet is required to calculate density. A thermocouple, or thermocouples attached to the pressure measuring probes, can be used to measure the temperature.

Fan performance is determined by measuring fan flow rate, fan static or total pressure, and fan power at a stated speed and fluid density. Other measured values, pressures on the inlet and outlet sides of dampers, and damper position can be used to measure system pressure and fan pressure capability.

Fan flow rate is rigorously determined by measuring the velocity pressure profile in a traverse plane of a long straight duct, sufficiently removed from flow disturbances such as dampers, elbows, flow measuring devices, or the like. Acceptable distribution profiles exist when 75% of all velocity pressure readings are greater than one-tenth the maximum reading of velocity pressures. Converting the root mean square average of the readings to velocity and multiplying by the area of the traverse plane will determine the volumetric flow rate. The static pressure rise across the fan and the inlet velocity pressure should be determined with the dampers completely open in order to compare to the fan performance curve, unless manufacturer's performance curves are available for partially open inlet vanes. Gas density should be calculated as accurately as possible at the point of a volume traverse.

The following precautionary items should be noted regarding testing:

(a) all readings, regardless of algebraic sign, should be recorded

(b) all tubing or instruments must be free of moisture accumulation

(c) measuring instruments should be clean and all ports open during the test

(d) the traverse area should be representative of the area seen by the gas flow

(e) gas characteristics should be examined to determine a possible need for special instruments

2-3.8.5 Steam Generator Equipment Data Validation

2-3.8.5.1 Pulverizer. The pulverizer clean air curve is useful in assessing control air line conditions, mill condition, and pitot tube condition. The plotted clean air curve should pass through the origin. If the curve intersects the mill differential at zero primary air (PA) differential, there may be a leak in the high side of the PA differential impulse line or in the low side of the pulverizer differential impulse line. Conversely, if the curve intersects the PA differential axis, there is either a leak in the low side of the PA differential impulse line or high side of the mill differential impulse line. Comparing clean air curves over time, an increase in slope indicates the throat may be plugged with debris. If the slope decreases, the throat may be worn. Clean-air flow tests are also performed in coal pipes to compare air flows in each pipe, to ensure that there is an equal distribution of primary air to all coal nozzles.

Stationary pitot tube calibration is an important step to enable operators to fire the boiler evenly. Pitot tube calibration factors for each mill should be within 5% of one another; therefore, equal PA differentials mean that heat input is the same by each mill, assuming the same loading curve is used for all pulverizers. If the calibration factors are not within 5% of one another, they can be electronically biased if the control system has the capability for this adjustment; otherwise, the pitot tube can be twisted in the duct or a dam installed upstream of the pitot tube to change the calibration factor.

In sampling coal from a stream of coal and air, it is essential that the velocity into the sampling tip be nearly the same as the velocity in the pipe. Furthermore, the rate of movement of the sampling tip through the pipe must be uniform, and the tip should traverse the entire pipe diameter. Sampling must be taken in both directions for the same period of time. To determine if a good sample has been taken, the fineness should be determined and plotted on the Rosin and Rammler Probability Chart, Fig. 2-3.8.4.1-3

2-3.8.5.2 Steam Generators. Expected values or curves can be developed over the load range using either manufacturer's design data or historical test data for comparison with measured data. If the measured value and expected value differ by more than a prescribed amount, the data item can be flagged as suspect.

Selected data items can be backed up with redundant instrumentation or checked against an instrument making a similar measurement. For example, economizer inlet and feedwater temperature leaving last heater should match. A heat balance around attemperators using temperature data can be used to validate spray flow measurement.

Performance engineers are encouraged to become familiar with Nonmandatory Appendices A and B of ASME Performance Test Code PTC 4-2008, and to develop computer-based monitoring programs based on the material in these Appendices.

2-3.8.5.3 Air Heaters. A heat balance around an air heater can be used to calculate air outlet temperature. If measured and calculated air outlet temperatures agree, then leakage, temperature, and air/gas flow data can be assumed to be relatively accurate.

2-3.8.6 Steam Generator Equipment Performance Calculations

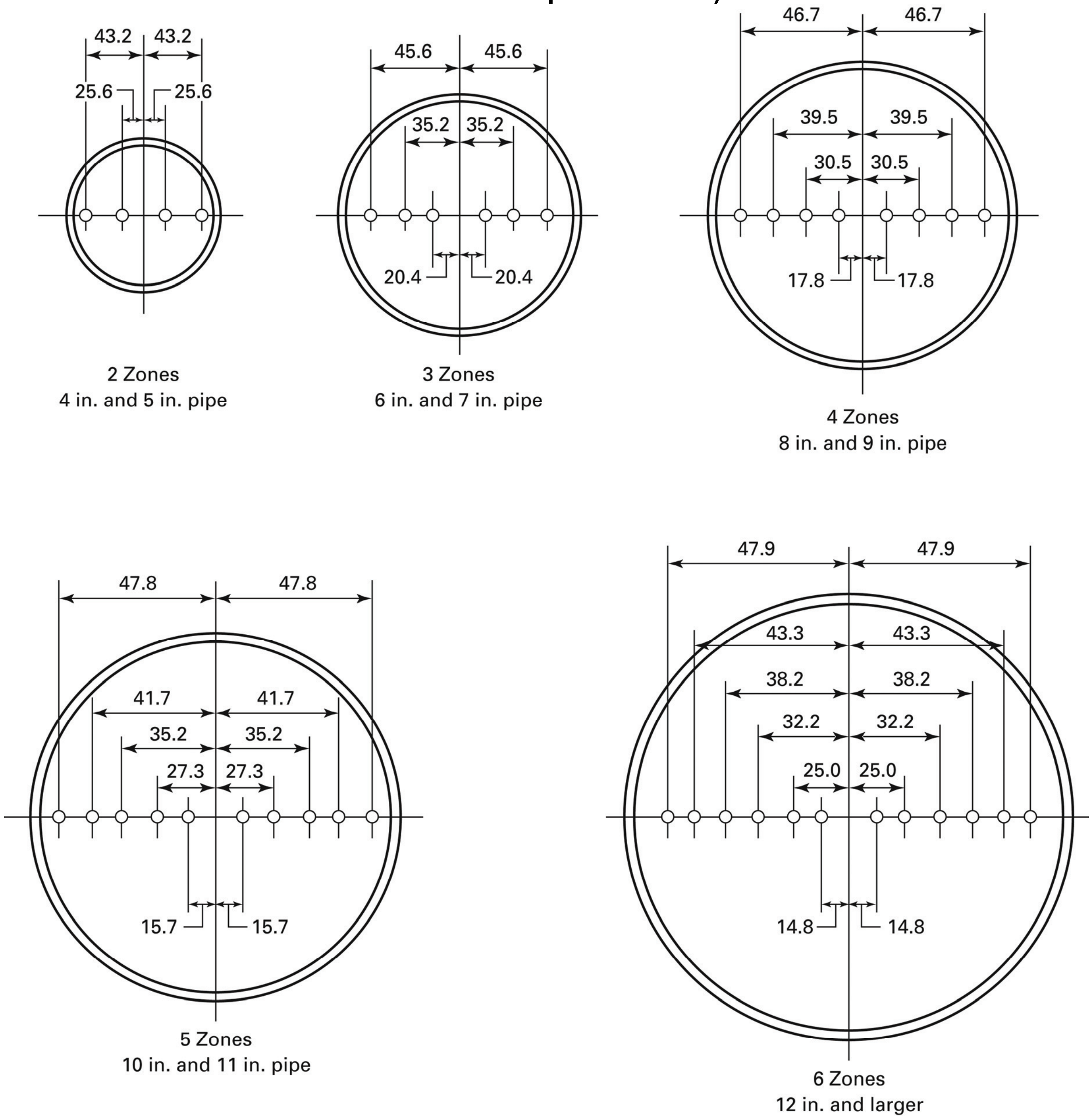
2-3.8.6.1 Pulverizer Calculations. The clean air curve requires no calculations. Simply plot mill differential (dependent variable) as a function of stationary pitot tube differential (independent variable).

The stationary pitot tube calibration requires calculations to determine mass flow rate of air through individual burner lines based on equal area pitot tube traverses. Using the simple relationship of flow rate being proportional to the square root of differential pressure, the proportionality constant, or stationary pitot tube K factor, is calibrated.

Fineness testing requires a calculation to determine that isokinetic sampling has been accomplished. The sample rate should be equal to the coal flow rate passing through a fuel transport line multiplied by the ratio of the sampling tip area to the coal pipe area. The actual sample weight should be between 90% and 110% of the theoretical weight to be considered satisfactory. In order for isokinetic sampling to be accomplished, the sample should be taken by carefully traversing at least two diameters 90 deg apart in sampling zones indicated in Fig 2-3.8.6.1-1. The sampling time at each point should be approximately 5 sec, and the sampling location should be in a vertical pipe 7 to 10 diameters from preceding bends, changes in cross section, or valves.

2-3.8.6.2 Steam Generator Efficiency (Heat Loss Method). Calculation of boiler efficiency by the heat loss method is determined using the calculation forms in PTC 4. Using these forms, the losses and credits are easily quantified such that the user is allowed to identify and reduce losses where possible. The following parameters are included in these forms:

Fig. 2-3.8.6.1-1 Sampling Direct-Fired Pulverized Coal-Sampling Stations (Dimensions Are "Percent of Pipe Diameter")



(a) dry gas loss
(b) moisture in fuel
(c) moisture in hydrogen
(d) moisture in combustion air
(e) unburned combustibles
(f) proximate analysis
(g) ultimate analysis

PTC 4 contains an example of a boiler efficiency calculation using nomographs and graphs to determine the breakdown of losses. Refer to PTC 4 for the detailed equations to calculate boiler efficiency.

2-3.8.6.3 Air Heater Calculations. Air heater gas side efficiency is defined as the ratio of gas temperature drop across the air heater, corrected for no leakage, to the temperature head. Gas side efficiency is computed as follows:

$$\text{AH gas side efficiency} = \frac{\text{Gas Drop}}{\text{Temperature head}} \times 100$$

$$= \frac{T_{g1} - T_{g2}}{T_{g1} - T_{a1}}$$

where

T_{a1} = air temperature entering the air heater

T_{g1} = gas temperature entering air heater

T_{g2} = gas temperature leaving the air heater corrected to the no-leakage condition

The gas temperature leaving the air heater, corrected to the no-leakage condition, is also referred to as the uncorrected gas temperature. This is the temperature at which the gas would leave the air heater if there were no leakage in the air heater. This temperature cannot be measured directly, but must be calculated based upon the amount of air heater leakage. The gas temperature leaving the air heater with leakage is referred to as the corrected gas temperature. This is the measured exit gas temperature and includes the dilution effect of leakage through the air heater seals and entrained leakage in the air heater baskets for a rotary regenerative type of air heater. These are further described in reference [18].

The gas side efficiency is an important diagnostic factor when considered together with other measurements and the manufacturer's correction curves. Air heater gas side efficiency may be used to correct the air heater exit gas temperature for off-design changes in the entering air or gas temperatures. This is discussed in para. 2-3.8.4.3. A change in air heater gas side efficiency is most often a result of the following three events, listed in order of their relative effect upon gas side efficiency:

(a) a change in the air or gas flow rate through the air heater

(b) a change in the cleanliness of the air heater

(c) long-term corrosion of the air heater heat transfer surface.

Air heater *X*-ratio is defined as the ratio of the heat capacity of the air passing through the air heater to the heat capacity of the gas passing through the air heater. *X*-ratio is computed as follows:

$$X\text{-ratio} = \frac{T_{g1} - T_{g2}}{T_{a2} - T_{a1}} = \frac{W_a C_{pa}}{W_g C_{pg}}$$

where

T_{g1}, T_{g2}, and T_{a1} are as defined above

T_{a2} = air temperature leaving air heater

C_p = mean specific heat of air or gas

W = mass flow rate of air or gas

X-ratio will vary based on a number of parameters, including moisture in coal, air infiltration, air and gas mass flow rates, air heater leakage, air infiltration through the penthouse or through the convective and rear-pass casing, and specific heat of air and gas at constant pressure. A change in *X*-ratio signifies a change to one or more of these parameters.

X-ratio does not provide a measure of the thermal performance of the air heater but is a measure of operating conditions. A low *X*-ratio indicates either excessive gas weight through the air heater or that air flow is bypassing the air heater. A lower-than-design *X*-ratio leads to a higher-than-design gas outlet temperature and can be used as an indication of excessive tempering air to the pulverizers or excessive boiler setting infiltration. For balanced-draft boilers, the convective pass is below atmospheric pressure. As the unit ages, air infiltration through the penthouse or through the convective and rear-pass casing may increase, thereby increasing the gas flow and lowering the *X*-ratio. Refer to subsection 3-5 for an example of the effect of high air infiltration on air heater performance.

Air and gas side pressure drops should be monitored and trended to identify degradation in the physical condition of the air heater, changes in air and gas flows, or pluggage of the heat transfer surfaces over time.

The tested air heater gas side efficiency should be corrected to the design conditions using the manufacturer's correction curves, and an expected exit gas temperature should be computed at test conditions. Comparison of the expected and tested exit gas temperatures and the deviation from design of the air heater leakage should then be used to calculate the change in dry gas loss and fan power consumption from a change in gas side efficiency.

Detailed calculation procedures for air heater leakage are included in PTC 4.3; however, some relatively simple equations for air heater leakage can be used in a performance monitoring program, depending on the level of accuracy desired and the specific data collected.

If O_2 analyzers are used, air heater leakage is approximated by the following equation:

$$\%\text{ leakage} = \frac{(\%O_2\text{ out} - \%O_2\text{ in})\ 0.9 * 100}{(21 - \%O_2\text{ out})}$$

It is imperative that the O_2 basis be the same at the air heater inlet and outlet, wet or dry. Nonmandatory Appendix C shows how to convert between wet and dry O_2.

If a combination of CO_2 and O_2 readings are taken, draw a radial line on the volumetric combination chart as defined in Nonmandatory Appendix A. Convert the CO_2 readings to O_2 using the combustion chart, making sure to be consistent in using wet or dry O_2. Do not use both. Air heater leakage is then calculated in the same way as the previous equation.

If an Orsat is used for flue gas analysis, then CO_2 and O_2 are measured on a dry volumetric basis. Air heater leakage is then calculated on a total weight basis by the following equation:

$$\%\text{ leakage} = \frac{(\%CO_2\text{ in} - \%CO_2\text{ out})\ 0.9 * 100}{\%CO_2\text{ out}}$$

Experience has shown that the use of this factor 90 will result in percentage leakage figures that are very close (±1%) to leakage determined on a weight basis.

2-3.8.6.4 Fan Calculations. The efficiency of a fan is used as a measure of fan performance. Fan efficiency is calculated using the equation

$$\text{Eff} = \frac{\text{ACFM} \times \text{static pressure rise (in. } H_2O) \times \text{compressibility factor}}{6356 \times \text{fan brake horsepower}}$$

The above equation shows that fan brake horsepower is directly proportional to flow rate. For fans with a pressure rise over 20 in. H_2O, a compressibility factor should be added to the numerator of the above equation. This factor will range around 0.95 to 0.98, but will be even lower for high-pressure fans. Consult ASME PTC 11 for calculation methods. Fan brake horsepower is typically determined from fan motor input power and an assumed fan motor efficiency, normally in the 90% to 95% range depending on motor size.

2-3.8.7 Steam Generator Effects on Performance. In analyzing the heat losses mentioned previously, the losses due to unburned combustible material and flue gases are those that are controllable.

Incomplete burning of all the carbon within the fuel will result in carbon being present in the ash leaving the furnace. This results in an unburned combustible loss. Under best conditions, some unburned carbon will exit in the ash, resulting in a slight loss in boiler efficiency (0.1% or less). Unburned carbon may appear in both the fly ash and bottom ash.

Combustible losses depend upon the type of coal used, the method of burning the coal (stoker, pulverized, or cyclone), the maintaining of the proper air-fuel ratio, and the proper maintenance of fuel preparation equipment. Boilers may also have coupled overfire air and separated overfire air for NOx emissions control. These systems can also impact combustible losses.

Wide variations of carbon loss can occur during load changes on the boiler where the air-fuel ratio may become temporarily unbalanced. Proper adjustments in air and fuel controls will minimize this loss.

Carbon loss can occur in normal operation due to an unbalance of either air or fuel to the several burners of a pulverized coal fired furnace. The unbalance results in some burners operating with too much air while other burners have insufficient air.

Coarse coal being fed to a pulverized furnace may result in an increase of unburned carbon carrying over to the boiler exit, depending on the particular coal fired. Coarse coal may also cause furnace slagging. If slagging conditions occur while burning a coal that does not normally cause slagging, carbon carryover and pulverizer fineness should be checked.

The other controllable loss due to dry flue gases is often one of the larger losses and fortunately, the easiest to control. The equation for this loss is

$$LG = (W_G)(C_{pg})(T_e - T_r)$$

where

C_{pg} = mean specific heat of dry gas
T_e = flue gas exit temperature, ^{o}F
T_r = inlet air temperature, ^{o}F
W_G = [lb of dry gas]/[lb of fuel (as fired)]

As can be seen, there are two approaches for decreasing this loss. The exit gas temperature can be lowered by either changing the heat absorbed in the steam generator by maintaining cleaner surfaces, or by changing heat exchanger surface in the gas stream. The weight of the dry gas leaving, $W_{G,}$ can be decreased by reducing the amount of excess air used for combustion. However, the operator must be informed of how low excess air can be carried. Reducing excess air below allowable limits can result in

(a) incomplete combustion resulting in CO formation and higher levels of unburned combustible

(b) excessive smoking

(c) fireside corrosion damage to furnace walls, large slag deposits, and localized overheating of the furnace

The heat loss due to moisture from hydrogen in the fuel is uncontrollable due to the presence of hydrogen in the as-fired fuel.

The heat loss due to moisture in the fuel is uncontrollable due to the moisture being in the fuel as it enters the boiler. This loss may be reduced to some extent through selective removal of coal from storage during unusually wet periods.

The heat loss due to the moisture in the air depends upon the relative humidity of the air, and is thus uncontrollable.

Radiation loss is to a great extent an inherent loss involving the size, type, and construction of the boiler unit involved. It is controllable to a small extent in that proper application and maintenance of insulation is essential to obtain best results.

The unaccounted-for loss is an approximate figure that is arbitrarily applied to account for small losses not elsewhere measured or accounted for, such as sensible heat in ash discharged from the unit. The breakup of unaccounted losses is typically provided by the boiler vendor. If this is not available, it can be approximated as 0.5%. This should not be confused with a manufacturer's margin or tolerance that may be applied by the manufacturer for contractual purposes. A typical value of 1.0% is often specified, resulting in an unaccounted-for loss plus manufacturer's margin total of approximately 1.5%.

2-3.8.8 Boiler Monitoring and Diagnostics. The overall performance of the boiler can be characterized by comparing the unit's boiler efficiency, SH/RH steam temperatures and flows, exit gas temperatures, and emissions to expected values. Major components in the boiler are the combustion region, heat transfer surfaces, the forced draft and induced draft (if a balanced draft boiler) fans, air heaters, the fuel transfer and preparation equipment, and the emission control and waste handling equipment. As discussed earlier, there are currently three methods commonly associated with the determination of boiler efficiency. These are energy-balance method, output–loss method, and input–output method.

The first and third methods are defined in PTC 4 Fired Steam Generators [13], while the second method is described in [7].

From a diagnostics standpoint, either the energy-balance or output–loss method is preferred over the input–output method for determining the boiler efficiency. The reason is that the input–output method offers no guidance on where boiler inefficiencies may exist. On the other hand, the losses and credits used in the other two methods directly identify the parameters influencing the efficiency. In fact, for diagnostic purposes, trending of the credits and losses can provide useful diagnostic information, even in the absence of determining the fuel heating value and calculating the boiler efficiency. The energy-balance (heat loss) method is also more convenient to implement with an on-line performance monitor.

Typical values of the various losses for utility boilers are given below [10, 18, 19]. The values for coal are for a typical coal-fired unit firing a low moisture coal, such as an Eastern or Midwest bituminous coal. A boiler firing higher moisture coals, such as Western or lignites, would have greater losses.

	% Loss for		
	Coal	**Gas**	**Oil**
LG (dry flue gas)	4	4	4
Lmf (moisture in fuel)	2	0	0
LH (moisture from burning of hydrogen	3	10	6
LUC (unburned carbon in ash)	2	0	0

Boiler diagnostics require the ability to separate the effects of the boiler controls, which operate on a macroscopic level, from the more localized effects. Both the combustion of the fuel and the heat transfer are localized in nature, being affected by local conditions. The boiler, however, is operated from a macroscopic perspective, as the controls deal with main and reheat outlet temperatures and fuel and air flow to the boiler.

Trending of the boiler losses is an effective way to track boiler performance from both an operations and equipment standpoint.

2-3.8.8.1 Changes in Fuel Analysis. Changes in fuel characteristics, if present, will enter almost any boiler diagnostic evaluation. Therefore, a determination of whether or not changes occurred and, if so, they are pertinent to the current evaluation, should be a standard step in a boiler diagnostic procedure.

The characteristics of coal greatly influence plant performance, emissions, capacity (increased or derated), and overall generation costs. Proper evaluation of the effects of coal quality variations can be very complex and time-consuming. Several software tools have been developed for conducting these evaluations [31, 32, 33, 34, 37, 38]. These software tools provide means for rapid evaluation of performance and economic factors relating to coal quality, and typically include detailed predictive performance models for all equipment affected by coal quality. The program predictions of system performance are based on equipment configuration and component type information provided by the user. They make use of state-of-the-art equipment models to calculate coal quality impacts. Derates are analyzed using a Monte Carlo simulation. Specific component models, such as the boiler and electrostatic precipitator, can be calibrated by matching the model predictions with user-supplied performance test data for a calibration coal. Maintenance and availability costs are determined by a detailed component failure model that is sensitive to coal quality effects on performance and failure rates.

The bottom line of this type of evaluation is to provide total fuel-related costs for alternative coals. The costs consider all cost components associated with combustion of each coal supply, including plant efficiency effects, emissions, auxiliary power requirements, steam desuperheating requirements, equipment replacement costs, maintenance costs, waste disposal costs, replacement power costs due to differential unit availability or capability, fuel costs, and fuel transportation costs. Another important output of this type of software tool is a summary of projected operating limitations on a system-by-system basis, which may reduce the risk of burning troublesome coals.

These tools can be applied in the areas of fuel supply, fuel planning, coal supply, fuel management, and plant engineering. The following applications have been demonstrated:

(a) evaluate the effect of variations in the fuel on superheat and reheat temperatures, spray flows, exit gas temperatures, emission rates, slagging, and fouling conditions

(b) evaluate potential coal supplies and assist in fuel procurement decisions

(c) establish unit-specific coal specifications and property range limits

(d) develop premiums and penalties for key coal quality parameters

(e) assess changes in maintenance and availability costs

(f) screen alternative coals prior to test burns

(g) support engineering studies to predict impacts of equipment modifications on overall unit performance and economics

(h) document and standardize the fuel procurement decision process

2-3.8.8.2 Flue Gas Measurements. Flue gas measurements can be used to diagnose several problems. Diagnostics will focus on the boiler's ability to combust all the carbon to CO_2. Measurements should include those of O_2, CO, and carbon in the fly ash at the economizer exit to characterize the combustion occurring in the furnace. The O_2 present in the flue gas is indicative of the excess air provided to the combustion process and is an input to the boiler controls. The CO and carbon in the fly ash, conversely, are indicative of the degree of incomplete combustion occurring. If a burner is operated with a deficiency of air (reducing atmosphere), the result will be the presence of unburned carbon (C) in the ash and carbon monoxide (CO) in the flue gases.

Typical ranges of excess air at full load for the various fuels are as follows [19]:

Fuel	% Excess Air	%O_2
Coal (pulverized)	15–40	3–6
Oil	3–15	1–3
Natural Gas	3–15	1–3

For gas-fired boilers, only a small amount of CO (typically between 100 ppm and 400 ppm locally and less than 250 ppm on the average) should be present in the flue gas at the optimum operating condition. On coal-fired boilers, in particular, both CO content in the flue gas as well as carbon carryover in the fly ash should be monitored. The reason for this is that the carbon carryover in the fly ash increases significantly before normal detection of a significant increase in the CO content of the flue gas.

Below 800˚F, CO combustion becomes relatively negligible. Since the flue gas temperature typically drops below this level by the time it exits the economizer, any CO present at the economizer exit will be present in the flue gas.

If the O_2 measured at the economizer exit is within acceptable limits, but CO and/or carbon in the fly ash values are high, the following combinations of events are probably occurring:

(a) localized combustion in a reducing atmosphere is occurring at one or more of the burners

(b) air infiltration into the boiler through casing leaks, cooling air for unused burners or registers, or an oxygen deficient environment exists at one or more of the burners

To identify the location of the combustion problems, operational modifications (burner air register positions, etc.) can be made, one at a time, while simultaneously looking for a reduction in the CO level in the flue gas [8].

Measurement of O_2 and CO at the exit to the air heater may be easier from the standpoint of a lower flue gas temperature at that location, but any air in-leakage into the air heater will dilute the CO concentration, thus making it harder to detect. In addition, the closer to the furnace exit the CO is monitored, the greater the ability to identify the source.

Continuous tracking of CO, O_2, and carbon in the fly ash is an effective diagnostic technique for monitoring combustion efficiency. Once O_2 is minimized for a given CO level (a suggested control range for CO is between 150 ppm and 250 ppm of CO [10]), increases in CO and/or carbon in the fly ash over a short period with no increase in excess O_2 can indicate a combustion stability problem. This may require operational adjustments or maintenance. Increased excess O_2 requirements over a long period of time to maintain combustion stability (constant CO and/or carbon in the fly ash levels) can be used as an early warning of combustion problems. Follow-up diagnostic procedures can then identify specific problems, and the appropriate maintenance can be scheduled. With regard to carbon in the fly ash, for high-moisture fuels, the foam test run by the ash purchaser is the most sensitive test.

This methodology must take into account the increased requirements for excess air as the load drops off. Reductions in fuel and air flow reduce the mixing efficiency and must be compensated for by providing a higher percentage of excess air. Minimum air flows are required for proper boiler operation during start-up and low loads, and as a safeguard against furnace explosions [18].

2-3.8.8.3 Gas Measurements at the Furnace Exit. Measurements at the furnace exit, with a high velocity temperature probe [9] are advantageous because they identify local problems before the gases undergo additional mixing in the convection pass. Variances to the flue gas analysis profiles (O_2, CO, carbon in the fly ash, and temperature) may be tied to specific burners or groups of burners by initiating intentional burner disturbances and noting the effect of the change on the measurement profile. Fuel/air ratio balancing can be adjusted by damper manipulation at the local burner or by windbox supply dampers where available. Off-line opportunities include coal pipe flow balancing (orifices and riffles) to adjust fuel delivery. O_2 and CO concentrations as well as temperatures and fly ash samples can be taken at up to 30 or more locations across the furnace exit. Relatively uniform O_2 (±20% of the average) and CO (average <100 ppm, and no one measurement >200 ppm) concentrations and temperatures (±150°F) across the furnace exit indicate good mixing within the furnace, while large variations in these parameters indicate that changes in operation may be required.

For pulverized coal-fired boilers, the unburned carbon loss may be used to detect problems with either local fuel/air balance or pulverizer grind size. A temperature of about 1,500°F is required to complete combustion of the carbon char. After leaving the furnace, the flue gas temperature drops quickly below the ignition temperature of the carbon. Complete combustion of the carbon must therefore be accomplished in the furnace. Incomplete combustion can indicate that either the particles are too large (problems with grind size) or the residence time in the furnace is too short (approximately a 3 sec residence time is generally required). Fireball location will impact residence time of the coal particles in the furnace also. On boilers capable of changing the burner angle and raising the fireball, such as tangentially fired boilers, the positioning of the burner tilts will therefore affect the residence time.

2-3.8.8.4 Boiler Surface Cleanliness. Boiler heat transfer surfaces are sootblown to enhance heat transfer and reduce gas path restrictions. The trending of boiler heat transfer surface cleanliness can identify performance changes and assist in the improvement of heat rate by allowing selective sootblowing in a manner that will result in an improvement in heat rate or boiler operation. The effect on the unit performance that cleaning a specific heat transfer surface would have can also be used in maximizing the heat rate improvement resulting from selective sootblowing.

Boiler cleanliness is a relative parameter, based on the conditions used to define the 100% clean conditions of the specific heat transfer surface under evaluation. The cleanliness of a specific component or area is expressed in terms of an associated ratio of the as-operating overall heat transfer coefficient in Btu/hr-ft^2-°F, to the target overall heat transfer coefficient in Btu/hr-ft^2-°F under clean conditions. This ratio is then expressed as a percentage.

All modes of heat transfer must be considered. The 100% clean overall heat transfer coefficient may be determined by either of two methods. The first method is analytical and consists of determining the 100% clean overall heat transfer coefficient to equal either the maximum theoretical heat transfer rate, or the maximum design heat transfer rate. This value would be calculated based upon measured or known parameters such as surface geometry; steam, water, air, and gas flows through a section; materials properties; gas composition and specific heat; and other parameters that are necessary to determine heat transfer rates. The second method is operational and consists of determining the 100% clean overall heat transfer coefficient based on a set of operating conditions or operating points that represent the normally achievable clean condition. The cleanliness value under these desired conditions would be set arbitrarily to 100% to represent the targeted clean condition.

Instrumentation that may be used for the determination of the overall surface cleanliness of a section includes thermocouples mounted on the inlet and outlet headers or links for a given boiler section, thermocouples imbedded below the surface of boiler tubes, or special thermocouples mounted on the surface of tubes or on the walls between tubes. Systems that trend the overall heat transfer coefficient based on such temperature measurements are sometimes referred to as heat flux monitors. These may be used as inputs to intelligent sootblower control systems. There are also software programs and turnkey systems available to calculate the cleanliness of tube sections for monitoring and trending, and to aid in the frequency of sootblower operation. Excessive sootblower operation may lead to excess tube erosion. Accurate monitoring of section cleanliness can help reduce sootblower erosion, reduce sootblowing steam usage, improve efficiency, and optimize cleanliness of heat transfer surfaces.

The burner tilt positioning control mechanism of tangentially fired boilers is usually directly driven by, or a function of, reheat temperature control. In a radiant reheat type of boiler, the reheater heat absorption and outlet steam temperature are an interrelated function of many factors. These include the surface cleanliness of all upper and lower furnace heat transfer surfaces. As such, the burner tilt position can be used as an indicator of the cleanliness of these surfaces. It is not, however, entirely indicative of the surface cleanliness of any one particular surface. For example, during a certain time period of operation, the burner tilt angle on a tangentially fired boiler may be observed to be steadily decreasing in order to maintain a constant final reheater outlet temperature. This trend in burner tilt angle may indicate that the furnace waterwall cleanliness has decreased, the reheater cleanliness has increased, the superheater cleanliness has increased, or a combination of these and possibly other factors.

Routine boiler tube or air heater inspections can be used to locate problem areas of ash deposits, measure specific deposit weights, and acquire samples for chemical analysis. The use of an on-line steam and water chemistry monitoring system may assist in the detection or prevention of waterside internal tube deposits and corrosion.

The flow rate to the superheater attemperator may be used as an indicator of effective boiler operation. Excessive spray flows may be the sign of improper sootblowing practices, improper sizing or ratio of steam generating surface area, air or fuel flow imbalances, excessive air flow rates, or nonoptimal tilt settings.

The effects of increased fouling and plugging of furnace waterwall surfaces, convective section surfaces, and air heaters can include any or all of the following:

(a) lower position of fuel and air nozzle tip tilts (tangentially fired furnaces only)

(b) increased SH and RH desuperheater spray water flows

(c) higher boiler economizer exit gas temperatures

(d) higher air heater exit gas temperatures

(e) increased draft losses (depending on ash fouling characteristics)

(f) pluggage of close-spaced convection section surfaces, particularly with coals having ash with low fusion temperature

2-3.9 Balance of Plant (Condensers, Cooling Towers, Heaters, and Pumps) Monitoring and Diagnostics

2-3.9.1 Balance of Plant Measured Parameters

2-3.9.1.1 Condenser

(a) condenser air in-leakage

(b) circulating water inlet/outlet temperatures

(c) condenser pressure

(d) condensate temperature

(e) tubeside pressure drop (waterbox differential pressure)

(f) vacuum exhaust air temperature

(g) vacuum exhaust air flow rate

2-3.9.1.2 Cooling Towers

(a) wet bulb temperature

(b) dry bulb temperature

(c) fan power (mechanical draft)

(d) cold water temperature

(e) hot water temperature

(f) circulating water flow rate

(g) wind direction

(h) water TDS

(i) temperature inversion

2-3.9.1.3 Feedwater Heaters

(a) heater inlet and outlet feedwater temperatures

(b) extraction temperature

(c) extraction pressure and/or heater shell pressure, or both, if available

(d) cascade drain inlet temperature

(e) drain cooler exit temperature
(f) tubeside feedwater/condensate flows
(g) feedwater heater level and control valve positions, both normal and alternate drains
(h) tubeside pressure drop
(i) shellside pressure drop
(j) drain flow
(k) level control valve position

2-3.9.1.4 Centrifugal Pumps

(a) suction/discharge temperatures and pressures
(b) flow rate
(c) seal water injection flow rate, if applicable
(d) balancing drum leakoff flow rate, if applicable

2-3.9.1.5 Moisture Separator Reheater (MSR)

(a) cycle steam outlet (hot reheat) temperature
(b) reheater heating steam inlet temperature or pressure
(c) reheater heating steam drain temperature
(d) reheater heating steam flow
(e) LP turbine inlet pressure
(f) HP turbine exhaust pressure
(g) drain flow rate
(h) drain temperature

2-3.9.2 Balance of Plant Calculated Parameters

2-3.9.2.1 Condenser Performance

(a) condenser pressure deviation
(b) condenser heat duty
(c) circulating water flow rates
(d) condenser cleanliness factor
(e) condenser log mean temperature difference
(f) condensate subcooling
(g) hotwell temperature

2-3.9.2.2 Cooling Tower

(a) deviation from design cold basin temperature
(b) approach
(c) range
(d) capability

2-3.9.2.3 Feedwater Heater

(a) terminal temperature difference (TTD)
(b) drain cooler approach (DCA) temperature difference
(c) temperature rise (TR)
(d) extraction line pressure drop.

2-3.9.3 Balance of Plant Degradation Identification

2-3.9.3.1 Condenser. The condenser is the heat sink for the power cycle. While references to condensers in these Guidelines are written mainly concerning surface condensers, a brief discussion of air-cooled condensers is at the end of this subsection. Heat from the condensation of turbine exhaust steam and incoming drains is removed by the condenser circulating water (CCW). During the process, it is necessary to remove any noncondensible gases that collect, or local pockets of these gases will result in regions in which little or no heat transfer occurs. These pockets are often referred to as regions of air blanketing.

Evaluation of the condenser may be divided into the following four areas of focus:

(a) the demand being placed on the condenser in terms of the magnitude of the total heat load

(b) the ability of the CCW system to provide adequate CCW flow rate and inlet temperature

(c) the resistance to heat transfer from the condenser shell into the condenser cooling water

(d) the adequacy of venting to minimize dissolved gases

Performance monitoring of the condenser should begin by noting the condenser pressure and its relationship to the expected value for the given load and inlet CCW temperature. In a condenser, the condensation temperature, and therefore the pressure, will rise until the condenser is removing the required heat load. If the condenser cannot remove the required heat load to maintain turbine backpressure below a set limit, the unit will have to be derated to prevent a turbine trip from excessive backpressure.

Unusually high CCW temperature rises across the condenser typically point to either less than normal CCW flow or excessive heat load on the condenser. Improperly operating drains and dumps are often the cause of excessive heat loads. Low CCW flow rates can be caused by either partially open valves in the system, closed valves in the system, condenser waterbox debris, CCW pump degradation, or severe tube fouling.

As a historical rule of thumb, 1 scfm to 2 scfm air in-leakage for each 100 MW of installed capacity is considered an acceptable level of air in-leakage. Recent advances in instrumentation have suggested that air in-leakage limits be increased to the pump capacity for well-designed condensers, at which point condenser pressure has been shown to increase [46]. Sources of air in-leakage can be from cracked condenser expansion joints, valve packing, pump seals, or any other areas subject to subatmospheric pressure. Condenser performance monitoring should also include investigation of potential areas of air in-leakage into the condenser. Noncondensible vent-gas flow rates from the condenser should be continuously monitored. Note that air in-leakage can also come from regions of the LP turbine where internal pressure is below atmospheric. In-leakage where shaft packing boxes are bolted to the casing has been reported by users. Noncondensible flow rate from the condenser should be checked. The flow rate should not exceed 25% of the capacity of the air removal equipment, or unit performance can be expected to deteriorate due to the development of excess condenser pressure. Drain lines to the condenser are normally designed with fittings (valve, etc.) within 2 ft to 5 ft of the condenser. This fitting acts as a throttling device, causing the water to flash before entering the condenser. It is not desirable to have the liquid water flashing in the condenser itself. Evidence of a water leg in these drain lines in the vicinity of the condenser indicates improper action of the throttling fitting.

The pressure drop across the tube bundle on each waterbox should be checked. Increased pressure drop indicates either macroscopic fouling of the tubes or the tube sheet, or higher than normal CCW flow. Therefore, the CCW flow rate should be continuously monitored to identify significant fouling or onset of CCW pump degradation. If real-time CCW flow rates are not available, the CCW pump performance should be routinely monitored in order to differentiate pump degradation from condenser fouling issues.

The waterbox air removal system should be checked by examining the water level in the waterboxes. This system usually removes air from the top of the exit waterboxes, but is sometimes also installed on the inlet waterboxes and inlet and outlet circulating water tunnels or pipes.

A few degrees of hotwell subcooling can typically be found in many condensers, expressed as the calculated difference between saturation temperature and the hotwell condensation temperature. This subcooling may indicate such problems as excessive circulation water flow, excessively low inlet temperature, excessive air-in leakage, impaired air-removal capability, decreased deaeration capability, excessive makeup for a given heat rejection rate, or air binding, known to be a cause for excess condenser pressure [47]. Less than normal subcooling can indicate improper condensate level or sealing problems in the division plates on the bottom of the condenser, as a precursor to a measured increase in condenser pressure. Chronic hotwell subcooling is indicative of air binding or deficiency of tube bundle venting, both of which are correctable by condenser retrofitting [48]. Hotwell subcooling is usually associated with elevated dissolved oxygen [46].

2-3.9.3.1.1 Air-Cooled Condensers. Air-cooled condensers (ACC) move ambient air over tubes carrying exhaust steam from the turbine. The steam condenses in the tubing. There is no circulating water system. Three of the main performance-monitoring points for an ACC are

(a) turbine backpressure

(b) dissolved oxygen in the condenser

(c) corrosion products in the condensate

If the backpressure is higher than expected for ambient conditions with number of fans operating, the problem could be from fan degradation or, more likely, from air in-leakage. Check the trends in dissolved oxygen.

With the steam and condensate traveling in significantly greater lengths of piping and tubing compared to a surface-condenser system, there is greater potential for corrosion products in the condensate. These products can plug strainers and cause boiler/turbine fouling. Condensate samples should be monitored for corrosion products.

2-3.9.3.2 Cooling Tower. The cooling tower is one type of heat sink for the condenser. The heat rejected from the power cycle to the condenser is absorbed by the condenser circulating water (CCW) and in turn again rejected to the atmosphere in the cooling tower.

Evaluation of the cooling tower may be divided into the following three areas of focus:

(a) the demand being placed on the cooling tower in terms of the size of the total heat load

(b) the ability of the CCW system to provide adequate CCW flow rate

(c) the resistance to the heat transfer attributable to any and all of the various contributors in the cooling tower

Cooling tower performance should be monitored by comparing actual cold basin temperature to the design value based on ambient air temperature (wet and dry bulb), heat duty, and circulating water flow.

Cooling towers should be inspected to determine water flow distribution by direct observation. On mechanical draft towers, the inspection should include an evaluation of the power consumption of the fans. Excessive power consumption can denote electrical or mechanical problems with the fan.

2-3.9.3.3 Feedwater Heaters. Feedwater heaters preheat the water fed into a steam generator using steam extracted or “bled” from various stages of a steam turbine. They improve Rankine steam cycle efficiency by utilizing the heat of the extracted steam, including the latent heat that would otherwise be rejected to the condenser heat sink.

Guidelines for testing feedwater heater performance are described in para. 2-3.9.4.3. Calculations for TTD, DCA, and TR (temperature rise) are defined in para. 2-3.9.6.3. Potential impacts to overall unit performance are described in para. 2-3.9.7.3. Diagnosis of feedwater heater problems is covered in para. 2-3.9.8.3.

Root causes of feedwater heater performance degradation may be divided into the following three areas of focus:

(a) cycle isolation problems external to the heater

(b) cycle isolation problems internal to the heater including level control, inadequate venting of noncondensible gases, and division plate leakage

(c) actual heat transfer performance

Detecting degradation usually begins by monitoring the TTD, DCA, and TR of each heater along with heater level and positions of drain valves both normal and alternate. These parameters should be monitored continuously, if possible, or otherwise checked on a routine basis. Small changes can provide advance notice of developing problems.

Tube leakage can occur suddenly or start more gradually. Typically, in either case, it should be isolated promptly for two reasons: first, to protect the heater from further damage due to high-velocity water spraying from the leak impinging on other tubes and other internal structure; and second, to reduce risk of turbine water induction from extra water in the heater traveling up through extraction piping.

In the absence of sudden tube leakage or tube rupture, changes in feedwater heater performance are relatively gradual. However, variations in heater performance will result in serious harm to cycle efficiency if not immediately corrected. As indicators of the onset of such deterioration, TTDs, DCAs, and tubeside temperature differentials should be monitored continuously or checked on a routine basis. The terminal temperature difference (TTD) is the saturation temperature at the shell pressure minus the feedwater temperature leaving the heater. The drain cooler approach (DCA) is the drain temperature minus the entering feedwater temperature. Abnormal heater levels and fluctuating drain control valve positions may be caused by increasing heat duty from a flow unbalance or upstream heater bypassed, or a tube leak. Level control valve positions should be monitored to determine deviations from their normal position. Tube pluggage should be quantified and symptoms of excessive tube leakage identified.

2-3.9.3.4 Boiler Feed Pump. BFP performance may not have a significant impact on unit heat rate, but it will impact the capability of a unit to obtain full load.

BFP performance is monitored in the same way as any type of centrifugal pump. It is important to know the basis for the manufacturer's pump performance curve with respect to extraneous flows into and out of the pump, such as seal water injection (SWI) and balancing drum leakoff (BDLO).

For normal pump performance degradation over time, a standard should be established for each pump to determine rebuild cycle, preferably based on deviation from design head. In addition to the general pump performance from a capacity standpoint, high SWI and BDLO flows indicate wear by problems in the seal area, and will help in establishing work scope for pump rebuilds.

Monitoring pump performance is also useful to help detect related problems that affect pump system performance, but are not caused by that pump's degradation. Two common problems that affect pump flow rate are recirculation loop flow and uneven flow rate through parallel pumps.

(a) Recirculation Loop. Most BFPs have a recirculation flow path that opens to protect the pump during periods of very low flow rate at start-up and very low load. If the recirculation valve is open or leaking through, then the pump has to handle extra flow rate for no benefit. This will consume extra pump power, and could limit the maximum output of the unit if pump capacity cannot handle extra flow rate.

(b) Parallel Pumps With Uneven Flow Rate. In a system with parallel pumps, performance degradation on one pump causes uneven flow rate through the parallel pumps and increases the power consumed by all pumps in the system. The better performing pump will flow more than its share of the system flow to produce same head as the poorer pump. Individual flow measurements per pump can help diagnose which pump has a problem, or else pumps may have to be taken out of service individually to determine which pump has a problem.

For turbine-driven boiler feed pumps, the boiler feed pump determines the amount of power required by the boiler feed pump turbine. In order to monitor the performance of the boiler feed pump, the extraction flow to the feed pump turbine should be quantified either through direct measurement or a first stage pressure/steam flow curve relationship. HP extraction steam leakage to the turbine should also be monitored using the first stage pressure as an indicator.

2-3.9.3.5 Moisture Separator Reheater (MSR). The moisture separator reheater (MSR) is a nuclear cycle balance of plant component located between the HP and LP turbine sections. Its purpose is to remove moisture from the HP turbine exhaust and add superheat to the cycle steam prior to entering the LP turbine. Proper operation of the MSR results in a reduction in the amount of moisture formation in latter LP turbine stages. This results in increased LP turbine section efficiency, improved reliability, and an improvement in unit heat rate and megawatt output.

The MSR consists of two major sections: the moisture separator, and the reheater sections. The moisture separator section consists of plates, screens, chevrons, or other apparatus that mechanically remove moisture from the steam. The reheater section may consist of one or two stages. In some cases, no stages of reheat are present.

The HP reheater stage consumes throttle steam, while a low pressure reheater stage, if present, consumes HP turbine extraction steam. An excess steam vent also is present to motivate drain flow and to minimize subcooling of condensate in the reheater tubes. Condensate from the moisture separator and reheater sections drains into feedwater heaters or drain tanks and is usually pumped forward into the feedwater flow.

A problem with MSR performance may first be indicated by a drop in cycle steam outlet (hot reheat) temperature and a change in HP or LP reheater heating steam flow. These parameters illustrate the primary inputs and outputs of the MSR and should be monitored on a daily basis. A decrease in cycle steam outlet temperature of 2 deg would indicate the need for further investigation.

Once a problem is evident, additional data is necessary to attempt a diagnosis of the problem. The terminal temperature difference (TTD) of each reheater stage should be calculated and compared to design (targeted value). HP reheater TTD can be calculated from available plant data as the difference between the heating steam inlet and cycle steam outlet temperatures. LP reheater TTD can be calculated using the same approach when the HP reheater stages are taken out of service. HP reheater TTD should be calculated daily while LP reheater TTD, due to the difficulty of isolating the HP reheater stage, can be calculated when a problem is indicated.

Increase in reheater stage TTDs combined with reduced heating steam flows would indicate the possibility of cycle steam bypassing the reheaters, reheater tube fouling, partially closed heating steam supply valves, inadequate purge steam flow, or poor heating steam drainage control. Increased TTD with increased heating steam flow would indicate possible increased duty of the reheater or an excessive vent steam flow. Increases in reheater heating steam flow and increased TTD may also indicate possible reheater tube leaks.

An increase in reheater duty is most likely due to poor moisture separator performance. Problems in moisture separator effectiveness result in increased moisture carryover into the adjacent reheater stage. The reheater heating steam flow increases as additional steam is required in order to evaporate this moisture. The reheater surface area available for superheating is then reduced, thus lowering the cycle steam outlet temperature.

A more direct indication of moisture separator performance can be determined by the monitoring of MSR shell drain flows. A decrease in moisture removal effectiveness will result in a reduction in shell drain flow. Decreases in HP turbine efficiency can also cause this result; however, the total cycle steam inlet moisture content would be less at the higher HP turbine expansion endpoint.

Other parameters to monitor include the cycle steam flow rate and the MSR shell pressure drop. Cycle steam flow can be monitored by the trending of LP turbine inlet pressure. The MSR shell pressure drop can be calculated by subtraction of this pressure from the HP turbine exhaust pressure. A change in MSR shell pressure drop would indicate possible erosion, steam bypass, or obstruction of the cycle steam flow path through the MSR.

A change in LP turbine inlet pressure with a proportional drop in HP turbine exhaust pressure indicates a change in cycle steam flow. This change in flow and the corresponding change in MSR duty will result in minor changes in MSR performance parameters. These parameters may be monitored as infrequently as once a month.

2-3.9.4 Balance of Plant Testing

2-3.9.4.1 Condensers. Many power plants routinely monitor condenser performance on a weekly or monthly basis. It is also possible to calculate and monitor condenser performance on a continuous basis for trending and analysis using typical station instrumentation, and to use that information to indicate when more rigorous troubleshooting analysis may be worthwhile. Both thermal and hydraulic performance of the condenser should be evaluated. Conditions that can be identified include condenser tube fouling, waterbox pluggage, low waterbox level, excessive air in-leakage, and diminished air-removal capacity.

The most widely accepted procedure for conducting a condenser thermal performance test is PTC 12.2. That Code provides guidelines for the determination of the performance of a condenser with regard to one or more of the following:

(a) the absolute pressure that the condenser will maintain at the steam inlet nozzle at a specified heat duty, circulating water flow rate, inlet water temperature, and tube cleanliness

(b) the thermal transmittance of a condenser for specified operating conditions

(c) the amount of subcooling of the condensate

(d) the amount of dissolved oxygen in the condensate

Most tests conducted on condensers are done in accordance with the first method above. The test code specifies a method to determine the condenser cleanliness factor that entails extending selected tubes through the waterbox so that the inlet and outlet conditions for the tube may be monitored external to the condenser. This process is both time-intensive and labor-intensive.

The parameters that must be measured or calculated, at a minimum, to determine an average thermal transmittance for the condenser are the total surface area of the condenser tubes, the entering and exiting circulating water temperature, the condensing steam temperature, and the heat load on the condenser.

Most condenser performance tests do not utilize extensions of tubes through the waterbox that deviate from PTC 12.2. Performance tests commonly monitor macroscopic or average values of various parameters to determine the as-operating performance of the condenser and compare it to the as-designed performance under similar operating conditions.

Historically, the most widely used technique to determine the heat load on the condenser was to utilize boiler and turbine measurements and calculate the circulating water flow rate as that necessary to give the measured inlet and outlet circulating water temperatures. This required additional measurements and introduced the potential of additional errors in the analysis. Techniques are currently available to monitor the flow rate of the condenser circulating water utilizing the pressure drop through the sudden contraction from the outlet waterbox to the outlet water pipe. This allows the determination of the heat load on the condenser directly from the inlet and outlet circulating water temperatures and the measured flow rate of the circulating water through the condenser.

2-3.9.4.2 Cooling Tower Performance. Many power plants conduct routine cooling tower performance tests during the summer or fall months when the greatest demands are placed on the towers. The frequency is dependent on the rate of performance degradation experienced by the cooling tower. Both thermal and hydraulic performance of the cooling tower should be evaluated.

Cooling tower thermal performance is normally expressed in terms of tower capability as a percent of design. This compares the condenser circulating water flow rate that the tower can actually provide at a given cold water temperature (condenser circulating water inlet temperature), ambient wet bulb temperature, and heat load to the design value of the condenser circulating water flow rate that should be provided under the same conditions.

The ASME procedure for the conduct of a cooling tower thermal performance test is PTC 23, Atmospheric Water Cooling Equipment. Makeup and blowdown to the tower should be isolated during the test. The most important phase of testing is the operating conditions during the test. Tests should only be conducted when conditions fall within the following limits:

(a) wind velocity <10 mph

(b) wet bulb temperature = 3°F above to 7°F below design

(c) cooling range = 20% of design

(d) water rate = 10% of design

(e) heat load = 20% of design

The accuracy of the test results depends on stable operating conditions. When possible, all conditions subject to control should be closely regulated. All testing should be conducted for a period of 1 hr after steady state conditions have been established.

2-3.9.4.3 Feedwater Heater Performance. Routine feedwater heater performance testing involves the collection of data necessary to determine terminal temperature differences, drain cooler approaches, and tubeside feedwater temperature rise. When TTDs and DCAs suggest abnormal heater performance, more detailed thermal performance testing of the heater should be conducted. ASME PTC 12.1 contains instrumentation requirements and procedures to be consulted for conducting closed feedwater heater tests, including formal acceptance testing. However, PTC 12.1-2000 also provides useful guidelines for routine performance testing and routine operation. In particular, two things must operate properly when testing a heater: water level control within the heater and adequate venting of noncondensible gases.

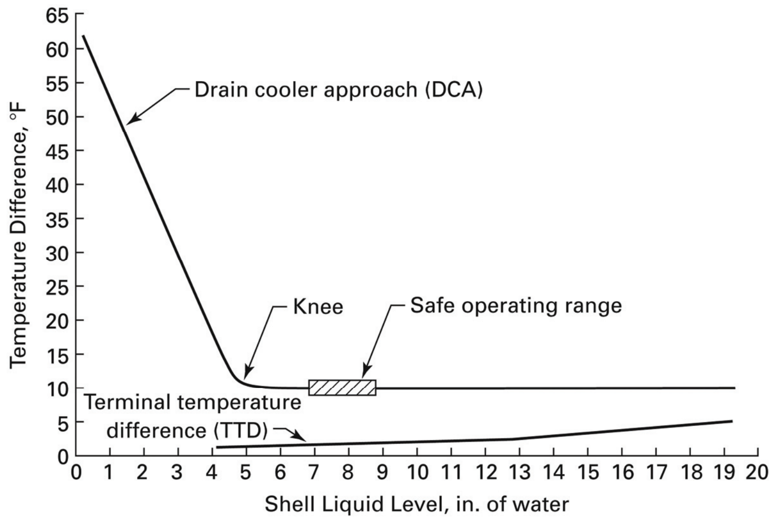
(a) Water Level. It is important that water level at the drain cooling zone inlet is maintained as close as possible to proper water level for that heater. If the water level is higher, then extra heat transfer surface area in the condensing zone will be flooded, which may reduce heat transfer capability and adversely affect TTD. If the water level is too low, then steam will enter the drain cooling zone, which will significantly increase the DCA and may gradually cause drain cooler damage. Therefore, proper water level should be verified before beginning the test, and maintained for good operation.

The overall plot of DCA versus heater level should resemble the shape of the curve depicted in Fig. 2-3.9.4.3-1. The optimum level is determined by finding the knee break of the curve and adding an appropriate safety margin (2.0 in.) of liquid to determine a safe operating level that can withstand some fluctuations while still properly sealing the drain cooler.

The following method can usually be used to determine proper water level inside horizontal and vertical channel-up heaters. (Vertical channel-down heaters should be set at the manufacturer's recommended normal liquid level.) The liquid level controller setpoint shall be adjusted in step increments of approximately 1 in. Each step increment should be held for 5 min or until drain temperature is stabilized prior to recording the drain temperature and calculating DCA. If the DCA was operating close to the design point prior to adjusting level, then the level should be lowered in each step, and the steps repeated until the DCA shows a sharp upward break with a rapid increase in DCA, as shown in Fig. 2-3.9.4.3-1 for typical DCA and TTD versus level.

Conversely, if the DCA is noticeably higher than normal prior to adjusting water level, then the opposite approach should be taken. The water level should be increased during each step, still pausing to stabilize and determine DCA after each level increase, until the DCA stops decreasing with level as shown in Fig. 2-3.9.4.3-1 for typical DCA and TTD versus level. However, certain internal problems could cause the DCA to remain high, such that increasing the water level cannot bring DCA down towards normal levels as

Fig. 2-3.9.4.3-1 Typical DCA and TTD Versus Internal Liquid Level



shown in Fig. 2-3.9.4.3-1. If this is experienced, then the water level should be restored to original, and other causes of high DCA should be investigated, such as a path for steam to enter the drain cooling section other than the normal inlet from the bottom.

(b) Venting. Noncondensible gas accumulation in either the condensing or drain cooling zones of the heater will degrade the performance of the heater by blanketing some heat transfer surface area. It may also lead to corrosion of heater internals. If a heater is not performing properly, the venting system operation should be checked. One simple check is to compare heater performance before and after purging the heater by opening a bypass valve around the vent orifice. The opened bypass should be maintained for approximately ½ hr to sufficiently purge the heater of noncondensibles. Comparison of the time-averaged feedwater outlet temperatures before and after the purge should yield close agreement. If there is a significant difference, an improperly sized or obstructed orifice should be suspected and corrective actions taken.

(c) Parallel Heaters. For heaters arranged in parallel, flow rate may not be split evenly between strings due to differences in piping, number of tubes plugged, etc. However, for routine performance monitoring, an even flow split is often assumed for in-service heaters. PTC 12.1 gives further guidance on performance of parallel feedwater heaters.

(d) Stability. Testing should be conducted at the load conditions corresponding to the rated performance conditions specified by the heater vendor. Tubeside flow should be within 2% of the rated heater flow. The turbine cycle should be operating at normal conditions with the turbine in condition such that the extraction state of the steam is as close to design (heater specifications) as possible. Adjacent feedwater heaters should not be in a bypass mode, and start-up vents, shell drains to slab, channel drains, level control drains, safety valves, and emergency drains to the condenser should be in normal mode of operation. Prior to testing, the heater level should be adjusted within safe operating limits to optimize the DCA. PTC 12.1 provides additional guidance on stability requirements when testing performance of closed feedwater heaters.

Calculations for TTD, DCA, and TR (temperature rise) are defined in para. 2-3.9.6.3. Potential impacts to overall unit performance are described in para. 2-3.9.7.3. Diagnosis of these conditions is covered in para. 2-3.9.8.3.

2-3.9.4.4 Centrifugal Pump Performance. The objective of monitoring performance of centrifugal pumps is to determine the degree of performance deterioration through internal wear. Testing before and after overhaul is done to verify the degree of improvement attained. Test data can also be used to identify when pump maintenance is required.

Design data required to evaluate a pump's performance is the manufacturer's head versus capacity curve, along with the pump's design capacity and total developed head. Operating data required is the pump suction and discharge pressure and temperature, flow, and speed (both motor and pump). When considering boiler feed pump performance, particularly multistage pumps, it is imperative to know the seal water injection, balancing drum leakoff, and attemperation flow rates (if attemperation is taken off between stages), and how the manufacturer accounts for these miscellaneous flows in their design capacity/head curves.

If the efficiency of the pumps is a desired end result of the performance monitoring, then additional data required for analysis is voltage, amperage, and power factor for motor-driven pumps.

Given that pump performance has very little impact on heat rate, pump efficiency is seldom of concern to a performance engineer. The bottom line is determining whether or not the pump is meeting its capacity/head curve within a given tolerance. Typically, deviation from design is expressed as a percent deviation in head at a given flow rate, or percent deviation in flow at a given head. For pumps with relatively flat characteristic curves in the operating range, the more significant comparison is percent deviation in head for a given capacity, because a minor difference in head between the design curve and operating curve will yield a high percentage difference in capacity, yet the pump's performance would be acceptable.

Usually, calibrated station instrumentation is acceptable for determining pump performance with sufficient accuracy for performance monitoring. In cases where special problems need to be evaluated, such as pump cavitation, more precise instrumentation may be required using PTC 8.2 as a guide.

For turbine driven boiler feed pumps, boiler feed pump turbine (BFPT) efficiency is difficult to calculate. In the rare case where the exhaust steam from the BFPT is still superheated, turbine efficiency may be determined through direct measurements as detailed in PTC 6. If, however, in the more frequent case, the exhaust steam is saturated, difficulties arise in calculating turbine efficiency. For this second case, it is necessary to perform calculations similar to those used in determining low pressure turbine efficiency. As mentioned in para. 2-3.9.6.4, the efficiency may be found only after the used energy end point has been determined from a total turbine cycle mass and energy balance, and the expansion line end point is determined by accounting for exhaust losses. An example of this calculation may be found in PTC 6A.

2-3.9.4.5 MSR Performance (Refer to PTC 12.4). In order to calculate the parameters required for MSR performance evaluation, the following data should be obtained:

Parameter	Repeatability	Recommended Frequency
Cycle steam outlet (hot reheat) temperature	±2°F	Daily
HP reheater heating steam	±2°F	Weekly
LP reheater heating steam	±2°F	Weekly
HP reheater heating steam flow	±5%	Daily
LP reheater heating steam flow	±5%	Daily
LP turbine inlet (hot reheat) pressure	±0.5%	Weekly
HP turbine exhaust pressure	±0.5%	Weekly
Heating steam drain temperature	±2°F	Weekly
MSR shell pressure drop	±2.25%	Weekly

Cycle steam outlet temperature should be measured for each MSR in order to determine relative performance of all MSRs on the unit. Heating steam temperature may be measured in a common header, although a pressure reading downstream of any control or check valves near the inlet to each reheater stage is preferred. The heating steam temperature can be calculated from measured pressure for all saturated steam units. Units with superheated throttle steam must measure HP reheater heating temperature directly.

Heating steam flows can be monitored by simple trending of indicated differential pressure readings only. However, direct measurement of heating steam flows or calculation of flow from observed differential pressures, using the James equation is preferred [49].

The measurement of heating steam drain temperature (or pressure) to obtain the drain enthalpy (saturated fluid) will enable the calculation of moisture carryover from the separator. Equating the heating steam reheater duty to the cycle steam reheater duty will result in the calculation of enthalpy out of the separator (the cycle steam flow to the LP turbine is determined using either a design flow factor or a design pressure/flow relationship). With the assumption of design cycle steam inlet (HP turbine exhaust) and moisture separator pressure drops, the separator exit pressure and the calculated separator exit enthalpy will result in the determination of quality, and thus moisture carryover.

2-3.9.5 Balance of Plant Data Validation and Normalization

2-3.9.5.1 Condenser. Tests should be conducted before and after cleaning to determine improvement in condenser cleanliness factor. If fouling or plugging of the condenser occurs, circulating water flow will decrease and condenser pressure will increase. Waterbox differentials should be checked along with waterbox level to make sure pluggage is not a problem and the tubes run full. Poor heat transfer due to tube fouling is apparent throughout the load range, although mostly noticed at full load.

2-3.9.5.2 Cooling Towers. A deviation in cold water temperature directly affects unit performance. Cooling tower capability provides a measure of cooling tower performance independent of ambient conditions that is useful for monitoring cooling tower condition over time.

2-3.9.5.3 Heaters. Based on design TTD at a given load, a comparison of actual versus expected TTD values will indicate if further analysis is required. If the TTD is higher than design, then several conditions may exist including improper water level, inadequate heater venting, abnormal bleed steam conditions, waterbox partition plate leakage, tube leaks, or dirty tubes.

Calculations for TTD, DCA, and TR (temperature rise) are defined in para. 2-3.9.6.3. Potential impacts to overall unit performance are described in para. 2-3.9.7.3. Diagnosis of these conditions is covered in para. 2-3.9.8.3.

2-3.9.5.4 Pumps. Comparing deviation from design flows at a given head can give an unrealistic condition assessment of a pump. The analysis should be made along the system resistance line to get a handle on the deviation from design head and flow simultaneously.

For turbine-driven feed pumps, comparison of baseline data such as mass (steam) flow rate, BFP speed, BFPT speed, control valve position, feedwater flow, and original design power to present working conditions under similar operating conditions will yield insight into efficiency losses. All losses should be accountable as large deviations in BFPT efficiency are not expected. Deviations greater than a few percentage points may be in error.

2-3.9.6 Balance of Plant Calculations

2-3.9.6.1 Condenser. Condenser cleanliness factor (C_f) is a term used to express the degree of tube fouling: actual as obtained from tests, or estimated for use in design in determination of condenser size. The cleanliness factor is defined mathematically by the Heat Exchange Institute (HEI) for comparison to design parameters, while PTC 12.2 defines procedures for determining the cleanliness factor through performance testing based on clean tube conditions.

The HEI condenser cleanliness factor is a comparison of the as-operating thermal performance of the condenser to the as-designed thermal performance of the condenser with 100% clean tubes for a given set of operating conditions. HEI's mathematical definition of the cleanliness factor is as follows:

$$C_f = \frac{U_{\text{actual}}}{U(100\% \text{ clean tubes})}$$

where

U is the overall heat transfer coefficient, Btu/hr-ft-°F

$$U_{\text{actual}} = \frac{Q\left(\ln[(T_s - T_{cwi})/(T_s - T_{cwo})]\right)}{A(T_{cwo} - T_{cwi})}$$

A = heat transfer surface area of the condenser, ft^2
Q = heat transferred, Btu/hr
T_{cwi} = circulating water inlet temperature, °F
T_{cwo} = circulating water outlet temperature, °F
T_S = condenser inlet steam temperature, °F
U (100% clean tube) = see manufacturer's data sheets, Btu/hr-ft-°F

The condenser duty is the total heat load on the condenser from the LP turbine(s) and all the drains and dumps that exhaust into the condenser. The duty may be calculated from the condenser circulating water side as the energy gain of the water as it flows through the condenser. The mathematical definition of the condenser duty, Q (water side), is as follows:

$$Q = W_{cw}C_p(T_{cwo} - T_{cwi}), \text{ Btu/hr}$$

where

W_{cw} = mass flow rate of the circulating water, lbm/hr
C_p = mean specific heat of circulating water, Btu/lbm-°F

The condenser heat duty can also be calculated by doing an energy balance around the steam side of the condenser, taking into account exhaust steam from LP turbine(s) and all the drains that exhaust into the condenser.

PTC 12.2 defines the cleanliness factor as a ratio of thermal transmittance of tubes in service to the thermal transmittance of new clean tubes, all under identical operating conditions of circulating water temperature and velocity and the same external steam temperature and flow. The Code outlines three methods for determining tube cleanliness.

As a word of warning, HEI uses the cleanliness factor in the design of condensers, for example, manufacturer's condenser pressure curves as a function of heat duty (Q) and circulating water temperature are based on the HEI cleanliness factor. However, the HEI cleanliness factor is not intended for performance test purposes. The HEI cleanliness factor is a unique value, since it applies to only one specific operating condition; HEI correction factors may not be applicable for every installation for the purpose of performance testing. Using HEI correction factors to account for off-design conditions such as circulating water inlet temperature may result in inaccurate performance test results.

2-3.9.6.2 Cooling Tower. Cooling tower capability is defined as the ratio of the test range to the range at test conditions of circulating water flow and relative humidity. The test range is the difference between the hot water temperature and cold water temperature.

Test range = (hot water temp) – (cold water temp)

$$\text{Capability (\%)} = \frac{\text{adjusted test water rate}}{\text{predicted test water rate}} \times 100$$

The capability of the tower is the ratio of the as-tested thermal performance of the tower to the predicted performance of the tower at the test conditions.

Cooling tower temperature range is the difference between the hot and cold water temperatures.

2-3.9.6.3 Feedwater Heater TTD and DCA. Feedwater heater terminal temperature differences (TTD) and drain cooler approaches (DCA) should be calculated.

TTD = saturation temperature of shell pressure minus feedwater outlet temperature

DCA = drain outlet temperature minus feedwater inlet temperature

TR = feedwater outlet temperature minus feedwater inlet temperature

Extraction line pressure drop = extraction pressure near turbine minus heater shell pressure (or extraction pressure near heater)

2-3.9.6.4 Centrifugal Pump. Centrifugal pump performance calculations are based on a few fundamental definitions and relationships.

The work or hydraulic horsepower of a pump depends on the mass flow of fluid being pumped and the differential pressure developed.

$$\text{Hydraulic HP} = \frac{\text{lbm of fluid per min} \times \text{total developed head (ft)}}{33{,}000}$$

The equation for pump power input for a motor driven pump (brake horsepower, bhp) is

$$\text{bhp} = \frac{\text{kW}}{0.746} \times \eta_g$$

where

kW = kilowatt input to motor

η_g = motor efficiency

Pump efficiency is then defined as the ratio of hydraulic horsepower to brake horsepower

$$\eta_p = \frac{\text{hydraulic horsepower}}{\text{brake horsepower}}$$

where

η_p = pump efficiency

The flow and head should be corrected for the temperature of fluid being pumped to yield correct results.

2-3.9.6.5 Moisture Separator Reheater. MSR TTD is defined as the difference in the heating system inlet saturation temperature (at heating system inlet pressure) and the cycle steam outlet hot reheat temperature. The shell pressure drop across the MSR is calculated as the difference in pressure from the cycle steam inlet to the cycle steam outlet (hot reheat) on the shell side of the MSR. Moisture carryover is determined from an energy balance around the tube and shell side of the reheater to obtain separator outlet (reheater inlet) enthalpy. Using this enthalpy with a proportioned separator outlet pressure (to design), the quality (and thus moisture carryover) is determined. PTC 12.4 should be consulted for a detailed calculation of these parameters.

2-3.9.7 Balance of Plant Effects on Performance

2-3.9.7.1 Condenser. Deviations in the condenser pressure, and therefore the turbine backpressure, are one of the greatest contributors to heat rate deviations in the power house. The condenser pressure establishes the temperature at which the condensation of the steam occurs in the condenser. Subcooling of the condenser occurs in the hotwell and the subcooled temperature becomes the thermodynamic low point in the steam cycle. The effect on performance of a change in turbine backpressure of 1 in. HgA is approximately a 2.5% change in heat rate and generator output, all other parameters held constant. The user must refer to specific turbine manufacturer's thermal kit for a given machine to determine the absolute value of a deviation in condenser pressure. This actual effect is dependent upon the actual inlet water temperature and is much greater at higher temperatures.

The greater the difference between the condenser saturation temperature corresponding to the condenser pressure and the condensate hotwell temperature (i.e., degrees of subcooling), the greater the duty placed on the lowest pressure feedwater heater. The subsequent increase in heater extraction flow results in an increase in heat rate and decrease in generator output (due to lower exhaust flow).

Due to the necessity of continually blowing steam impurities out of the steam generator, makeup water is supplied to the condenser hotwell at a rate proportional to the rate of blowdown. As the amount of cycle water loss (from poor isolation) increases, the amount of makeup is increased. The resultant effect is an increase in heat rate and a decrease in generator output. Typical values of these effects are +0.2% change in heat rate and –0.2% change in load at 100% throttle flow for 1% makeup flow.

The circulating water inlet temperature affects unit performance through its influence on the condenser pressure. If the circulating water temperature is higher than design conditions, then the condenser pressure will be higher than design value for that set of conditions. On a unit with a closed circulation water system, the cooling tower is responsible for providing the proper CCW inlet temperature.

The cleanliness of the condenser affects the performance of the unit through its influence on the condenser pressure. The flow of the condenser circulating water is established by the circulating water pumps, and the circulating water inlet temperature is established by either the cooling tower or inlet water supply as applicable. The outlet circulating water temperature is then established by the heat load on the condenser, which in turn is established by the unit demands. The condenser pressure will be established by the cleanliness of the condenser such that the condensation temperature of the steam is adequate to provide the necessary steamside to waterside temperature differential across the condenser tubes to reject the required heat load.

2-3.9.7.2 Cooling Tower. There is a direct negative affect on unit performance as the deviation from design cold basin temperature increases. The actual value is determined from manufacturer's performance prediction curves.

2-3.9.7.3 Feedwater Heaters. The most common parameters used for assessing feedwater heater performance are TTD and DCA temperatures. The DCA is an indicator of heater level and is used primarily as a diagnostic tool for detecting tube pluggage, leaking tubes, or a cracked subcooling baffle. The TTD is a general indication of the amount of heat transfer to the heater.

The top (highest pressure) heater TTD has the most significant impact on performance. The approximate effect on performance for 5°F increase in TTD is a 0.1% increase in heat rate, all other things being equal. However, the actual effect can be greater or less depending on the integrated effect on the specific unit. For example, the 5°F increased TTD provides colder final feedwater into a fired steam-generator, which shifts its pattern of heat absorption. It typically decreases the boiler exit gas temperature that improves unit efficiency, and also either causes steam temperatures to increase towards setpoint (improve heat rate) or else causes spray flows to increase (worsen heat rate).

The top heater TTD also has the most significant impact on unit capacity. If all components of the unit are capable of the extra duty, then a 5°F increase in TTD can provide approximately a 0.4% increase in generator output, if the unit load is limited only by turbine governing valves open 100%. Otherwise, if the unit maximum load is limited by any other component, then the maximum load will decrease by the same percentage as heat rate will increase.

The TTD calculated locally to the heater is a good indication of the heater itself, using shell pressure and feedwater outlet temperature local to that heater. However, it can also be useful to monitor a less local TTD for the highest pressure heater, which can help capture performance effects of related problems including feedwater bypass leaking around the heater or excessive pressure drop across its extraction line (valve not fully open.) That larger "external" TTD is calculated from the saturation pressure of the extraction pressure near the turbine (typically cold reheat) minus the final feedwater temperature entering the boiler (downstream of heater's bypass). However, when comparing that external TTD to a baseline target, the baseline should be calculated on the same basis.

The other non-highest pressure heater heaters also affect unit performance, but to a lesser degree than the top heater. For a typical feedwater heater train with six stages of feedwater heating, the impact of subsequent heater's TTD is summarized in the following table:

For a +5°F Increase to Heater TTD	% Increase to Cycle Heat Rate
Highest pressure heater	0.09%
Second highest pressure heater	0.07%
Third highest pressure heater	0.05%

Similar to the "external" TTD mentioned above for the highest pressure heater, bypass around the other heaters or excessive pressure drop may not affect that heater's local TTD, but does affect overall performance.

2-3.9.7.4 Pumps. As noted previously, deviation from design has an insignificant impact on unit performance relative to other major power plant equipment. The basic issue that needs to be addressed is whether the pump can handle the service requirement, not its efficiency.

2-3.9.7.5 Moisture Separator Reheater. Because the HP reheater takes throttle steam from upstream of the HP turbine control valves, any variation in heating system flow causes a change in turbine control valve position. Depending on the mode of control valve operation, the sensitivity of MSR performance on unit heat rate and load can vary. When operating in the partial arc admission mode (sequential valve control), an increase in MSR reheater TTDs or a decrease in moisture removal effectiveness causes an increase in unit heat rate and a drop in load. For a 1°F rise in TTD, the increase in heat rate is about 1.5 Btu/kWhr, and the drop in load is approximately 100 kW.

The sensitivity of these parameters when operating in this mode and with a constant thermal power was discussed by Spencer and Booth [10]. When operating in full arc admission mode (single valve control), the sensitivity is much less, and in some cases, poorer MSR performance results in decreased heat rate and increased load. This phenomenon is discussed in more detail in a paper by Campbell [11].

An increase in MSR shell pressure drop results in a higher HP turbine end point enthalpy, thus resulting in less work output by the HP turbine.

For a 1% point increase in pressure drop, the heat rate increase is approximately 12 Btu/kWhr and the load reduction is about 900 kW.

An increase in moisture carryover to the reheater results in more heating steam flow, thus decreasing the steam flow to the HP turbine. The resultant effect is less generator output and an increase in heat rate. For a 1% point increase in carryover, the increase in heat rate is approximately 25 Btu/kWhr and the reduction in generator output is about 1,700 kW.

The effect of these MSR thermal parameters on heat rate and load were based on a 750 MW, single stage reheat secondary turbine cycle with 745 psia, 0.25% moisture, 510°F inlet steam conditions at a turbine exhaust pressure of 2.5 in. HgA.

2-3.9.8 Balance of Plant Diagnostics

2-3.9.8.1 Condenser Diagnostics. The condenser is the heat rejection component for the cycle. Important associated components include the condenser circulating water pumps, the air exhausters, and the heat sink (cooling tower, river, lake, pond, etc.). Condenser diagnostics should include ambient considerations, as they affect the achievable cold water temperature and flow rate of the circulating water.

An important parameter for condenser diagnostic work is the condenser loading. This is the heat in the steam from the turbine plus the heat in all drains that flow to the condenser. While this may be determined from a full scale PTC 6 performance test, an easier and more common method of determining the condenser loading (sometimes called condenser duty) is to measure the cooling water inlet and exit temperature and its flow. Since flow meters are not normally installed in the cooling water circuit, the flow may be inferred from pressure drop across the tube bundle or from measured pressure and power and the circulating water pump curves.

Excess heat load can be due to a loss of turbine efficiency, a feedwater heater dumping directly to the condenser, an auxiliary dump open to the condenser, or heaters in the condenser neck with lagging off or extraction line connections leaking.

Low condenser circulating water flow rate can be due to excessive condenser macrofouling, excessive condenser circulating water system resistance, poor condenser circulating water pump performance, or low river/lake levels (open system). High condenser circulating water flow rate can be due to reduced condenser circulating water system resistance, or the pump operating above the pump curve.

Apparent excess levels of air in-leakage to the system can be due to the water chemistry being excessively out of limits, requiring larger than normal amounts of chemical treatment. This can generate higher than normal levels of noncondensibles, primarily ammonia.

The process of enhancing the performance of a condenser should include an initial baseline performance assessment and continuation with a performance trending program. Boundary conditions on the condenser include

(a) heat load

(b) circulating water flow rate

(c) circulating water inlet temperature

(d) air (noncondensible) in-leakage rates

(e) air (noncondensibles) removal equipment capacity.

Condenser diagnostics seeks reasons for differences in the condenser pressure from target values. For a specified circulating water flow rate, inlet temperature, and heat load on the condenser, the condenser pressure is dependent on the thermal transmittance and the size and distribution of the effective heat transfer area. Concerns to be investigated include the degree of steamside and waterside fouling of the condenser tubes, the steam flow distribution through the tube bundle, the water flow distribution through the tubes, the performance of the noncondensible gas exhausting system, the number and location of leaking or plugged tubes, and the amount of air in-leakage.

2-3.9.8.1.1 Cleanliness Factor. The condenser cleanliness factor is commonly used in diagnostics as an indicator of thermal fouling of the heat exchange surface. The HEI guidelines [26, 27] provide an empirically based method to determine a design value for the cleanliness factor. These have historically been used as a design tool for the industry. However, when using these guidelines for evaluating cleanliness of as-operating condenser performance, users must realize they are meant for overall performance of the condenser rather than cleanliness, specifically. The HEI methodology assumes that the heat exchange area is constant, though in realistic operation, the effective surface area can be reduced by air binding, low waterbox level, or high hotwell level. Any operating influence that reduces the effective heat exchange area will be reflected in a reduced cleanliness factor. The methodology also incorporates an empirically determined correction curve for inlet water temperatures other than 70°F. The accuracy of this curve diminishes with temperatures lower than 55˚F to 60˚F. A correlation of predicted heat transfer as a function of water velocity through the tubes is also provided. Condensers are typically designed for a relatively narrow range of velocities, usually between 6 ft/sec and 8 ft/sec. Outside of this region, the accuracy of the provided heat transfer-velocity correlation is suspect. Therefore, an empirically derived correction may need to be created by operating the condenser with different water velocity at otherwise similar operating conditions, such as varying number of circulating water pumps in service or adjusting valves in circulating water piping.

The target cleanliness factor should also vary as a function of load. It has been suggested that as the load decreases, the flow regime in the water film on the tubes changes from turbulent to laminar. This effect combined with an increased LMTD at reduced load is the reason for the observed variation in performance factor with load [39].

To more effectively utilize the cleanliness factor methodology for condenser performance evaluation and diagnostics, the cleanliness factor of the condenser should be benchmarked during clean conditions, such as after an acid or mechanical cleaning of the condenser tubes or a retubing of the condenser. Likewise, the waterside pressure drop across the tube bundle can be benchmarked after an acid or mechanical cleaning of the condenser tubes, or a retubing of the condenser, for comparison of the hydraulic performance of the condenser.

Routine condenser tube inspections can also be used to measure specific deposit weights and perform deposit analyses. The potential for waterside deposits and corrosion problems can be detected through the use of on-line steam/water chemistry monitors.

2-3.9.8.1.2 Condensate Subcooling. Increased levels of dissolved oxygen or severe corrosion of the bottom tubes of the tube bundle may indicate excessive subcooling of the condensate. The condensate temperature should be periodically compared to the saturation temperature for the operating condenser pressure. This is especially the case at part load operation and situations with low condenser circulating water flow rate.

Condensers are designed to avoid hotwell-condensate subcooling, at least at design conditions. The energy removed in subcooling the condensate has to be replaced by heat added in the boiler. This is a net energy loss to the cycle, increasing the heat rate. Actual condenser designs reduce subcooling by bypassing part of the exhaust steam to the condenser hotwell to reheat the condensate to its saturation temperature. This has the added benefit of vaporizing dissolved gases, such as oxygen, in the condensate. These gases accumulate in the air cooler section of the condenser and are removed by the air ejection system. If the condensate remains subcooled, elevated levels of dissolved oxygen can occur. To maintain dissolved oxygen at reasonable levels, extra chemicals must be added to the condensate. One of the breakdown products of the chemicals is ammonia, which can become concentrated in the bottom section of the condenser tube bundle. This can lead to severe corrosion problems in that area of the tube bundle depending on the tube material.

2-3.9.8.1.3 Air Binding. Decreased values of the cleanliness factor may be the result of air binding. Noncondensible gases will be present in any operating condenser, accumulating in pockets in which mass and heat transfer are so inhibited that little heat transfer takes place. In well designed condensers operating at low air in-leakage rates and at their design condenser pressure, these pockets are confined to the air removal section of the tube bundle. If the air removal rate of the air removal equipment is within its capability, this should be the case.

If an air removal flow rate measurement exists (typically on the air removal equipment), this should be periodically monitored to detect an increase in air in-leakage flow rate. Also, a simple diagnostic test can help detect or rule out either excessive air in-leakage or inadequate air removal capacity; however, it may not indicate which problem is the case. With the unit operating at near steady conditions, temporarily activate extra air removal equipment (either redundant back-up devices or start-up equipment) and observe if condenser performance appears to improve. If the unit is operating with as-designed low air in-leakage rates and the air removal equipment was adequately removing the air and preventing air-binding, then the activation of additional air-removal equipment should have no effect on condenser performance (and should be turned back off to eliminate extra parasitic consumption). Alternatively, if the condenser performance noticeably improves (backpressure decreases) with extra air removal equipment, then either the air in-leakage flow rate is excessive and beyond the normal capacity of the air removal equipment, or else the previously active air removal equipment was inadequate to remove the normal air flow rate. Switching between identical alternate standby air removal equipment may help indicate if one is underperforming relative to the other.

Note also that air in-leakage can occur anywhere in the cycle that operates below atmospheric pressure, not only the condenser; however, that air will flow to the condenser and its air removal equipment, since that is the lowest pressure in the system. Common examples other than the condenser itself include lower pressure feedwater heaters, shaft-seals on condenser hotwell pumps (especially redundant and out-of-service pumps), and a turbine gland sealing system. To determine specific location of air in-leakage, several leak detection technologies exist including helium tracers and sonic or acoustic methods.

2-3.9.8.1.4 Macrofouling. Macrofouling is the blockage of condenser tubes, usually caused by the accumulation of debris or a large air pocket in the inlet waterbox. The former is usually the failure of the inlet screens to function properly, insufficient waterbox velocities, or pluggage by mechanical cleaning devices. It can also be caused by biological growth such as freshwater snails within the condenser tubes. Large air pockets can be the result of air entraining vortices in the pump inlet, a malfunction of the waterbox air removal system, or air entrainment in the system piping. Since these tubes are physically blocked, there is a reduction in surface area for heat transfer. This will result in a reduced value calculated for the cleanliness of the condenser and a higher than predicted condenser pressure. The water velocity through the remaining tubes is increased, resulting in an increased system hydraulic resistance. This may reduce the water flow through the condenser.

2-3.9.8.1.5 Microfouling. Microfouling is the fouling of the heat exchange surface by biological or chemical deposition. Most closed cycle systems and some natural waters are supersaturated with respect to calcium carbonate, $CaCO_3$. Since the solubility of $CaCO_3$ decreases with increasing temperature, the stability of the water is decreased by the increasing water temperature within the condenser.

Chemical deposition occurs by precipitation of $CaCO_3$ on the tube surface usually beginning at the cooling water outlet. The chemical composition of the cooling water should be monitored to establish whether such deposition is likely to occur. In closed cycle systems, chemical dispersants are sometimes added to prevent the attachment of the precipitate to the tube walls.

Chemical deposition can also occur from a low flow rate of the water through a tube or tubes, resulting in boil-off of the water and thus leaving chemical deposition. Biological fouling is caused by microorganisms that attach themselves to the tube walls. In addition to decreasing the heat transfer rate, some of these microorganisms can induce corrosion around the weld joints. Disinfection of the cooling water, usually by chlorination, is necessary to prevent biological fouling. The effectiveness of the disinfection can be monitored by biological assay of the cooling water at the discharge of the condenser.

Symptoms of microfouling may be a reduced heat transfer rate and increased turbine exhaust pressure, and a resulting decrease in the calculated value for the cleanliness of the condenser. Since all other problems discussed in this paragraph will also cause a decrease in the apparent cleanliness, and microfouling has no other symptoms, its diagnosis will be accomplished by eliminating the other possible causes of the decreased performance (air binding, low waterbox level).

Corrective action for microfouling depends on the nature of the deposit, but can include various technologies including mechanical cleaning as well as chemical surfactants and biocides. Mechanical cleaning methods exist in both batch processes that typically require waterboxes to be isolated and out of service, as well as in continuous processes that require installation of specialized equipment.

2-3.9.8.1.6 Low Waterbox Level. The condenser's inlet and outlet waterboxes should be full of circulating water at least to the top of the tubes to ensure all tubes are full of water. However, air or other dissolved gases from the circulating water can collect at tops of the waterboxes and so prevent normal water flow through upper tubes, reducing the effective heat-transfer surface area of the condenser.

There are two causes for this. First, cold circulating water into the condenser is typically saturated with dissolved gases (air). As the circulating water is heated in the condenser tubes, it cannot contain as many dissolved gases, so those gases come out of solution. Second, depending on system pressure drops and elevation changes between the circulating water pumps and condenser and siphoning effect of outlet flow, the circulating water can actually be slightly below atmospheric pressure. Therefore, those waterboxes operating below atmospheric pressure can be subject to air in-leakage.

For either cause, the waterbox priming and/or venting system is designed to remove those gases to ensure the water level in the waterboxes stays above the tubes. Therefore, the water level should be periodically monitored during normal operation, and if tubes are not flooded, then troubleshoot the priming and/or venting system. Also, circulating water valves at condenser outlet can be partially closed to help increase water level above tubes; however, that will also reduce total water flow rate, which will provide worse performance than would normal water flow rate through full tubes.

2-3.9.8.2 Feedwater Heater Diagnostics. Feedwater heater problems generally fall into one of the three following categories:

(a) deposits, both tube-side and shell-side

(b) internal problems in each heater, including both tube leaks and leaks in baffles, seals, and level-control valves

(c) external problems related to each heater, including leaks through bypass valves and level control valves, as well as excessive pressure drop through partially closed valves in extraction piping

The focus of this paragraph will be on diagnostics associated with identification of problems falling into the first three categories. Recommended parameters to trend for diagnostic purposes include the terminal temperature difference (TTD), temperature rise across the heater, and drain cooler approach (DCA) temperature difference (see para. 2-3.9.2.3). Also, external leaks (to atmosphere or open drains) are normally identified during equipment walkdowns, other than expansion joints for extraction piping located in the condenser neck, or any entire heaters located inside the condenser neck.

Deposits result in increased TTD and decreased temperature rise across the heater. Both TTD and temperature rise are functions of load; however, for diagnostic trending purposes, target curves should be established across the operating load ranges. A significant tube-side buildup of deposits can also result in increased tube bundle pressure drop. Routine tube inspections can also be used to measure specific deposit weights and deposit analyses. The potential for waterside deposits and corrosion problems can be detected through the use of on-line steam/water chemistry monitors.

Tube leaks can have a number of effects on the efficient operation of a feedwater heater. Large tube leaks will be characterized by an increased demand on the feedwater pump (high-pressure heaters) and opening of emergency drain valves to maintain the heater shell-side level. This will result in increased condenser duty. Increased demand on the normal cascade drain and emergency drain can reduce the pressure in the drain cooler section causing flashing and an increase in the drain cooler approach (DCA) temperature difference, as well as possible damage to the feedwater heater. Particularly on HP heaters, leaks can be detected during equipment walkdowns, sounding very much like a jackhammer operating inside the heater.

Leaks in baffles, seals, and valves constitute the remaining area requiring diagnostic procedures. Leaks in HP heater emergency drain valves can often be detected by listening through a solid wooden rod placed against the valve. Internal leaks through seals and baffles cause short-circuiting of the steam path and result in varying degrees of increased TTD, reduced temperature rise, and increased DCA, depending on where the leak is. A methodology [10, 11] for detecting the location of internal baffle or seal leaks involves varying the shell-side liquid level, allowing the feedwater heater parameters to stabilize and recording the DCA and TTD as a function of level. Deviations from a normal or target curve generated under operation with no leaks present will indicate the level of the leak. Reference [22] discusses the use of the Wilson Plot to differentiate between tube fouling and leakage within the waterbox division plate, also known as the pass partition plate. The technique involves measurement of feedwater flow rate and the temperature of the inlet and outlet feedwater and the drain temperature. A heat transfer factor is plotted as a function of a velocity factor over several loads. A reference line of zero leakage and 100% cleanliness is plotted. Fouling will map as a line parallel but apart from the reference line. Leakage will increase the slope of the test line from that of the reference line.

2-3.9.8.3 Boiler Feed Pump Diagnostics. Pump diagnostics must first separate the system effects from those due to variations in pump performance. For this reason, several operating points should be evaluated. Diagnostic procedures can include both hydraulic and vibration analyses. If possible, multiple speeds can be run to establish the system curve, and various flow rates can be examined to establish the pump curve. Reference [25] contains several tables of cause-and-effect relationships that can be used for boiler feed pump diagnostics.

2-3.10 Combined Cycle Plants

2-3.10.1 Combustion Turbines

2-3.10.1.1 Combustion Turbine Cycle Diagnostics. The overall performance of the combustion turbine can be characterized by comparing the unit's base load power output and simple cycle gross heat rate (or, alternatively, simple cycle efficiency) with expected values. Because of the significant impact of ambient conditions on gas turbine performance, actual performance measurements and expected performance must be at a common set of conditions. For diagnosing problems at current operating conditions, it is common to adjust parameters at design or reference conditions to what they would be at test conditions. These are then the expected values at current conditions, and a direct comparison can be made. For evaluating and diagnosing temporal changes in performance, the measured parameters at each time period are adjusted from actual values to what they would be at reference conditions; this is referred to as "corrected performance."

Reference conditions may be ISO conditions of 59°F, 14.696 psia, and 60% relative humidity, or they may be reference conditions specific to the site. Adjustments between test and reference conditions can be made in one of the following three ways:

(a) use of curves provided by the combustion turbine manufacturer.

(b) use of a thermodynamic model developed by the combustion turbine manufacturer or others.

(c) use of dimensionless or quasi-dimensionless parameter groups. Because of complex turbine cooling arrangements, control algorithms that address pollutant formation as well as thermodynamic behavior and other factors provide at best only an estimate of the relative performance between test and reference conditions.

2-3.10.1.2 Combustion Turbine Diagnostics. Changes in combustion turbine performance parameters are affected by how the combustion turbine is controlled, as well as the thermodynamics of the cycle itself. The objective of the combustion turbine control algorithm is to limit the temperature of the working fluid in the cycle to levels compatible with the turbine materials and cooling technology. At base load, this generally means limiting the temperature of the hot combustion products entering the combustion turbine expander section, referred to as the firing temperature or turbine inlet temperature. Because this temperature is generally too high to measure reliably, the control algorithm is based on thermodynamic relationships between this temperature and other cycle parameters. For heavy duty industrial units typical of large power generation applications, the control algorithm is generally based on measured turbine exhaust temperature and compressor discharge pressure or pressure ratio. For aeroderivative machines, the controlled temperature may be the compressor discharge conditions, gas generator exhaust temperature, and/or the power turbine inlet temperature.

Combustion turbine diagnostics are generally performed for base load operation. If changes in part load performance are to be diagnosed, the effects of modulated inlet guide vanes or variable stator vanes on compressor performance, exhaust flow, and exhaust temperature must be taken into consideration.

2-3.10.1.2.1 Combustion Turbine Inlet Pressure Drop and Exhaust Pressure. Before addressing performance issues of the core machine (compressor, combustion section, and expander), the impact of the pressure drop across the inlet conditioning equipment (filters, evaporative coolers, etc.) and the exhaust pressure drop should be addressed. Inlet pressure drop will be influenced by cleanliness of the filters and high-humidity conditions. Exhaust pressure will be influenced by the condition of the exhaust diffuser and stack, and path through the HRSG for combined cycle applications. Increases in inlet pressure drop and/or exhaust pressure will result in reduced base load output and increased base load heat rate.

2-3.10.1.2.2 Compressor Section Efficiency. Calculated compressor section isentropic efficiency provides a direct indication of the condition of this section of the machine. Efficiency can be reduced due to compressor fouling, or due to compressor blade wear, damage, or erosion. Contaminants contributing to compressor fouling include dry, hard contaminants such as dust, sand, and dirt, and soft, sticky contaminants such as pollen, oily vapors, or airborne insects. If evaporative coolers are used, waterborne contaminants may also be introduced. Dry, hard contaminants can also contribute to compressor blade erosion.

Compressor section performance degradation, especially due to fouling, occurs gradually over time. A sudden change in compressor section performance may be an indication of foreign object damage due to compressor icing or upstream hardware breaking away and being ingested into the compressor.

The main impact of reduced compressor section efficiency in heavy-duty, or frame, units is to reduce the pressure-producing capability of the compressor and mass flow through the machine. A low calculated efficiency should therefore be corroborated by a lower than expected pressure ratio and exhaust flow. Turbine exhaust temperature may also be increased, unless the limit has been reached, at which point generation will be reduced to maintain the exhaust temperature at the lower compressor efficiency. For a typical heavy duty industrial combustion turbine, a 1% decrease in compressor section efficiency will have the following approximate impacts:

(a) 3.5% decrease in base load power output

(b) 1.5% increase in base load heat rate

(c) 2% decrease in compressor pressure ratio

(d) 2% decrease in turbine exhaust flow

(e) 6°F increase in turbine exhaust temperature

A portion of compressor section performance deterioration due to fouling may be recoverable with an on-line or off-line compressor water wash.

2-3.10.1.2.3 Exhaust Temperature Spread. The combustion turbine exhaust temperature spread provides an indication of the health of the combustion section of the turbine. Although the combustion system is not likely to be the direct cause of performance deterioration, with time [40], degradation in this section of the turbine can result in increased emissions of NOx and other pollutants, and can have an impact on the long-term reliability of the machine.

The spread in exhaust temperatures is the result of nonuniform combustion conditions among the combustors. This may be due to fuel nozzle plugging or wear, fuel distribution problems, or cracking or other thermal distress of combustion section components. The resulting uneven temperature profile at the entrance to the expander section can lead to deformation of downstream components from thermal stress and consequent deterioration in their performance. Also, although the average turbine inlet temperature may be within machine design limitations, locally high temperatures can lead to thermal damage or failure of hot gas path components.

The following observations should be made relative to the exhaust temperature spread:

(a) maximum spread (difference between the maximum and minimum temperature in the exhaust temperature profile).

(b) deviation of maximum and minimum exhaust temperatures from the mean exhaust temperature.

(c) location of the maximum and minimum temperatures in the exhaust profile. From the physical location of a temperature in the exhaust profile, the combustor contributing to this portion of the combustion turbine exhaust can be identified by applying appropriate corrections for exhaust swirl.

2-3.10.1.2.4 Expander Section Performance. The performance of the expander section is generally inferred from the overall machine performance and the performance of other sections. Although it would be desirable to calculate a section efficiency such as is done for the steam turbine and the combustion turbine compressor section, for example, this is not feasible for two reasons. First, as noted previously, the temperature of the gases entering the expander section is not measured; thus, the thermodynamic state at the beginning of the expansion is not known. Second, as a result of the cooling of the nozzles and rotating blades, the expansion process is not adiabatic; thus, calculation of isentropic efficiency is not an appropriate performance measure. Definitions have been proposed for cooled turbine efficiency [41, 42], but there is no generally accepted definition. Also, the proposed definitions suffer the shortcoming that they require measurements of cooling air flows not normally available for an operating unit.

Expander section performance can deteriorate over time as a result of high temperature oxidation, increased clearances resulting from vibration or blade tip rubs, blade erosion or corrosion, thermal damage due to combustion section problems, or problems with turbine cooling air. Step changes in performance may be the result of foreign object damage. Because the turbine is generally fired to control exhaust temperature, deterioration of the expander section will result in underfiring the combustion turbine. For a typical heavy duty industrial combustion turbine, a 1% decrease in expander section efficiency will have the following approximate impacts:

(a) 2.6% decrease in base load power output

(b) 1.8% increase in base load heat rate

(c) 0.2% decrease in compressor pressure ratio

(d) 0.2% increase in turbine exhaust flow

(e) no change in turbine exhaust temperature

(f) 13°F decrease in firing temperature (not measured)

A comparison of these changes with those that occur for compressor section performance deterioration shows that the magnitude of the output change relative to the heat rate change is a good discriminator between problems in the two sections.

2-3.11 Heat Recovery Steam Generators (HRSG)

2-3.11.1 Heat Recovery Steam Generator (HRSG) Cycle Diagnostics. The overall performance of the HRSG can be characterized by comparing the HRSG's efficiency with expected values. Because the majority of HRSGs used in power generation applications generate steam at multiple pressure levels, it is also necessary to compare the capacity of the HRSG with expected values at each pressure level. If the high-pressure section of the HRSG is deficient in steam production, more energy will be available in the exhaust gas stream for steam production at lower pressure levels. Thus, it is possible to have an HRSG that is at or near design efficiency, but has a deficit in high-pressure steam production, coupled with a surfeit of steam production at lower pressure levels. Because the high-pressure steam has more available energy to do work in the steam turbine, this results in reduced steam turbine output.

In evaluating HRSG performance, interrelations with other parts of the combined cycle must be recognized and accounted for. Boundary conditions for the HRSG are the combustion turbine exhaust energy, energy from any supplemental fuel firing, outlet from the HRSG high pressure section to the steam turbine cycle, and cold reheat from or hot reheat to the steam turbine cycle. For an unfired HRSG, the entire energy input is from the combustion turbine exhaust. This is the primary determinant of the steam-generating capacity of the HRSG. However, the pressure at which the steam is produced also influences how much steam is produced. For a given exhaust energy, steam production at a given HRSG pressure level varies inversely with the pressure. Since the pressure at the steam turbine throttle varies directly with flow to the throttle for the sliding pressure mode of operation typical of combined cycle units, equilibrium is established when the pressure-flow characteristics of the steam turbine match the pressure-steam production characteristics of the HRSG. Changes in the pressure-flow characteristics of the steam turbine (e.g., due to erosion or deposits) will thus influence the performance of the steam generator.

2-3.11.2 Cycle Isolation. Cycle isolation is a significant issue in HRSG testing. To verify appropriate cycle isolation, mass balances around the HRSG overall and at individual pressure levels should be verified. Assuming that instrument calibration issues have been addressed and calculated flow uncertainties have been taken into account, significant mismatches indicate cycle isolation issues. A specific example is comparison of the high-pressure steam flow with the high-pressure feedwater flow. If the HP steam flow is significantly less than the HP feedwater flow, the problem may be related to

(a) HRSG boiler tube leak.

(b) excessive HP drum to IP drum cascading blowdown. If this situation exists and the IP drum continuous blowdown is normal, then IP steam generation will be greater than IP feedwater flow.

(c) leakage through the main steam to cold reheat line bypass valve.

Other potential cycle isolation problems include leakage through the hot reheat emergency bypass to the condenser and through the low-pressure steam bypass to the condenser. Suspected changes in steam flows due to cycle isolation problems should be corroborated by changes in steam turbine pressures.

The effect of cycle isolation for HRSGs may be significant during testing. For periodic testing, it is recommended to verify isolation. For online monitoring, poor performance results may indicate problems in normal operating valve alignment affecting cycle isolation.

2-3.11.3 HRSG Diagnostics. HRSG diagnostics are designed to identify the reason(s) for decreased flow to the steam turbine from that expected for the given HRSG heat input. As noted above, this should start with verification of proper cycle isolation. HRSG efficiency will provide an indication of proper heat transfer from the combustion gases to the steam/water circuits in the HRSG. Unlike the steam generator in a conventional Rankine cycle unit, the HRSG is evaluated based on the lower heating value fuel heat input to the combustion turbine and the lower heating value of any supplementary fuel firing. Both input–output and loss methods for evaluating HRSG efficiency are presented in PTC 4.4 for heat recovery steam generators. The loss method is generally preferred due to lower uncertainty. The major losses to be considered are the stack loss and the radiation/convection loss. The radiation/convection loss is often assumed constant at its design value; however, for outdoor installations, this may understate the true loss, particularly if the testing is done in cold weather or windy conditions.

2-3.11.4 HRSG Efficiency. HRSG efficiency may be reduced due to fouling of the heat transfer surfaces. For natural gas-fired turbines, HRSG fouling from combustion products is normally not an issue. If the combustion turbine is fired with a liquid fuel, especially one with measurable ash content, fouling due to combustion products may impair performance over time. Another source of fouling exists in HRSGs with selective catalytic reduction systems from deposition of reaction products, notably ammonium salts, in the lower temperature sections of the HRSG. Corrosion products resulting from dewpoint corrosion in the feedwater heater (low temperature economizer) section can impair heat transfer in this area. HRSG efficiency may also be reduced as a result of exhaust gases bypassing portions of the heat transfer surface, or channeling due to uneven flow entering the HRSG.

2-3.11.5 Effectiveness. Effectiveness of the HRSG or HRSG section is the ratio of the heat removed from the exhaust gas to the maximum theoretically possible heat removal, as limited by the temperature of the cold-side fluid. Depending on the extent of gas side temperature measurements, it may be possible to calculate the effectiveness of individual heat transfer sections or groups of sections. This is useful in identifying any localized heat transfer problems in the HRSG.

2-3.11.6 Pinch. Evaluation of the HRSG pinch points provides a quick indication of the heat transfer occurring in sections upstream in the gas stream of the HRSG.

2-3.11.7 Approach. The evaporator approach (saturation temperature minus feedwater inlet temperature) affects HRSG performance, and in many cases may be influenced by HRSG operation. Often, there is a controlled bypass around one or more economizers that can be used to control the approach temperature, or may be designed to control stack gas temperature when burning liquid fuels in the combustion turbine. If the approach is too low, natural circulation in the evaporator section may be impaired. This could lead to reliability problems and a reduction in steam production. If the approach is too high, it will also reduce steam production at this pressure level.

2-3.12 SCR and FGD Performance

2-3.12.1 Selective Catalytic Reduction System. A selective catalytic reduction (SCR) system removes nitrogen oxides (NOx) by injecting ammonia vapor into the flue gas upstream of a catalyst bed. An SCR system is typically monitored for NOx removal or outlet NOx, reactor pressure drop, air preheater pressure drop, inlet flue gas temperature, and ammonia feed rate. High pressure drop usually indicates plugging of the catalyst bed or the air heater, and low NOx removal indicates an ammonia feed or distribution (uniform mixing with the flue gas) problem or ammonia nozzle plugging. Catalyst activity is followed over time so it should not be a sudden problem and is usually evidenced by increased ammonia consumption.

Increased ammonia consumption is an indicator of catalyst degradation, catalyst pluggage, poor mixing, poor ammonia or temperature distribution, improper flue gas temperature, or combinations of these and other possible factors. For operating or fuel conditions that deviate from the guarantee basis, the following correction curves from the SCR or catalyst supplier should be used to compare ongoing operation with the guarantees:

(a) NO_X Reduction Efficiency (%). A family of curves on one chart depicting NO_X reduction efficiency versus inlet NO_X concentration (ppmvd) at various flue gas temperatures (°F), unit loads, O_2 content, and gas flows.

(b) SO_2 Oxidation (%). A family of curves on one chart depicting at various curves the SO_2 to SO_3 oxidation versus flue gas O_2 concentration, flow rate, and SO_2 inlet concentration.

(c) Catalyst Activity (%). Catalyst activity versus catalyst life.

(d) Gas Side Pressure Drop (in. wg). Gas side pressure drop for both the catalyst alone and the full system versus gas flow (acfm).

2-3.12.2 Wet FGD Performance. A wet flue gas desulfurization (FGD) system removes SO_2, SO_3, and sulfuric acid mist from the flue gas. It is monitored for SO_2 removal efficiency, flue gas pressure drop including individual absorbers and mist eliminator pressure drop, and pH in the reaction tank. These are the key indicators of proper operation. Low SO_2 removal could mean spray nozzles are plugging or a malfunction in one or more of the slurry feed pumps. Low pH could be caused by limestone blinding or limestone feed problems. High mist eliminator pressure drop indicates plugging in the mist eliminators. High absorber pressure drop indicates plugging in the absorber. FGD system testing should be patterned after the instructions in ASME PTC 40.

2-3.12.3 Dry FGD Performance. A dry flue gas desulfurization system removes SO_2, SO_3, particulates, and sulfuric acid mist from the flue gas. These removals are accomplished by the injection of fine droplets containing lime slurry into the flue gas stream in a vessel called a spray dryer. In the spray dryer, the droplets contact the hot flue gas and, simultaneously with the evaporation of the water in the droplets, acid components in the flue gas react with the lime contained in the droplets. This process produces a dry, powder-like particle that is carried by the flue gas into a particulate removal device, usually a fabric filter. As the particles collect on the filter bags or the electrostatic precipitator plates, there is further contact with the flue gas and hence further removal of acid gas. Unlike a wet FGD system, the flue gas is not quenched down to its adiabatic saturation temperature, but rather to a temperature high enough to ensure that condensation of water vapor does not take place in either the spray dryer or the fabric filter, thereby protecting both of these vessels against corrosion and plugging.

Dry FGD (spray dryer and fabric filter) is monitored for SO_2 removal or SO_2 outlet concentration, inlet and outlet gas temperature, pressure drop, high vibration (when rotary atomizers are used), baghouse pressure drop, lime slurry feed system pressure, stack opacity, and lime feed rate. Low SO_2 removal can indicate a problem with the lime feed system or dual-fluid nozzle, or atomizer wheel plugging. Continued high-pressure drop in the fabric filter indicates plugging or blinding of the filter bags. Low outlet temperature indicates too much water in the system. For operating or fuel conditions that deviate from the guarantee basis, the following correction curves from the dry FGD supplier should be used to compare ongoing operation with the guarantees:

(a) SO_2 reduction efficiency (%)

(1) as a function of approach to adiabatic saturation temperature

(2) as a function of inlet SO_2 loading

(3) as a function of the inlet flue gas mass (lb/hr) flow rate

(4) as a function of fresh lime feed to the spray dryer

2-3.13 Results Reporting

Once the contributors to unit performance degradation have been identified and the impact on total unit efficiency determined, reporting of this information to the appropriate level(s) of management should be performed. Presented in the form of a report, the information should be clear and concise and should emphasize the deviations between actual and expected (targeted) levels of performance. The report should give reasons for lower than expected performance and should address each deficiency equitably based on significance and cost. If gains in performance have been identified by the performance engineer, then the report should contain recommended solutions. Grouping of the contributors (e.g., unit losses, system losses) in a prioritized manner according to performance impact and/or the cost of correction should also be performed. This will enable management to have the consideration of a full scope of candidate projects before subsequent budget decisions are made.

The presentation of data in these reports should be in a format that is readily understood and usable by the readers of the report. Presentation should include a summary of the major contributors, including a synopsis of the system's heat rate and generation for the period in question and reasons for deviation from expected. Other performance indices that might be included are heat rate rankings of units, efficiency factors, unit losses, system losses, and unaccounted-for losses.

Although the frequency of feedback will depend upon the performance philosophy of the users, it is recommended that management receive input on no less than a quarterly basis.

To aid in the development of corrective courses of action for each contributor, diagnostic techniques should be utilized to focus the corrective action on the correct equipment component or subcomponent. In addition, performance optimization of controllable parameters (operational and/or equipment repair) should be established for optimum operation and to eliminate cycle deficiencies otherwise hidden. Subsection 2-5 should be consulted for information on these techniques.

2-3.14 Cycle/Operational Interrelationships

Operational interrelations include those interactions that involve operating conditions and operational parameters. Typically, these parameters are temperatures, pressures, and flows for the equipment. The resulting interactions are quite variable and flexible, often under the control or influence of the unit operators. The effects on operation and performance of the overall unit or particular equipment components are frequently significant.

Since limited resources are available to diagnose and correct problems and to optimize performance, it is useful to apply those limited resources in a manner that provides the greatest benefit. To do that, it is useful to estimate potential impacts of detected problems and optimizable choices before pursuing them, so as to prioritize efforts to those problems that provide the greatest benefit.

When determining the impact of a given problem or action, it is important to consider the interrelationship of different parts of the plant cycle, since commonly expected impacts of a single parameter can be wrong (even reversed) when considering only a portion of the plant cycle without interrelationship with the rest of the cycle.

For example, increasing reheat spray flow or bypassing highest pressure FWH are both often expected to increase maximum steam turbine electrical output, which can be true if the unit's capacity is limited only by the turbine governing valve 100% open position. However, a given unit's boiler may not be capable of providing the additional steam heat duty required (e.g., fan-limited, pulverizer-limited, etc.), or the unit's heat rejection system may not be capable of rejecting the additional heat duty at an acceptably low turbine exhaust pressure. In either case, taking either of these actions would reduce maximum capacity rather than increase it, due to the worsened turbine heat rate and resulting interrelationship with the capacity-limiting component.

The following are several examples of operational interactions:

(a) increasing the reheat spray flow increases the megawatt output of the unit and, for a fixed throttle flow, increases the unit heat rate.

(b) lowering the condenser pressure increases the megawatt output until LP exhaust becomes choked. After this point, lowering the condenser pressure further decreases megawatt output because the cooler condensate requires more extraction steam for feedwater heating for a given set of upstream conditions.

(c) increasing the excess oxygen (excess air) in the boiler decreases the unburned carbon loss but increases the dry gas loss because of a greater mass weight of flue gas leaving the boiler.

(d) extracting steam from the turbine for a process will increase the pound steam per kilowatt hour of the turbine/generator (i.e., lower the amount of electric power generated per pound of steam flowing through the inlet). This is the case when an industrial steam host exists. The overall plant thermal efficiency will increase since the steam that is extracted will be used in the process and not add to the heat through the condenser.

(e) increasing the coal particle fineness from the pulverizer (i.e., increasing the percentage of coal particles passing through a given mesh) reduces the carbon loss but increases the auxiliary power requirements and increases wear on the components. The net result can be an increase in efficiency. In addition, furnace absorption and slagging/fouling can be affected, which affects the turbine cycle through changes in final steam temperatures and desuperheating flow rates.

(f) main steam pressure and temperature variations affect the feedwater outlet temperatures of the feedwater heaters by affecting the saturation temperature of the steam inside the heater.

Operational interrelationships lend themselves to optimization. The process of identifying and using the best overall combination of conditions in normal operation is described in detail in subsection 2-5. Some operating parameters cannot readily be optimized or controlled. For example, ambient air and water temperatures, coal quality, unit load, environmental restrictions, and limitations placed on equipment for safety and availability considerations are not controllable by the operator. Yet, operating parameters such as these have significant influence on unit and equipment performance. Thus, they need to be accounted for in the performance monitoring program when interpreting results.

2-3.15 Mechanical Interrelationships

Unlike the operational interactions, mechanical interactions tend to be fairly uniform and stable until the mechanical condition changes. If the mechanical condition changes, the interactions tend to change, too, but would stabilize at a new level if the mechanical condition stabilizes. The following are several examples of mechanical interactions:

(a) turbine steam path deterioration has an effect on stage pressure, section efficiencies, final feedwater heater temperature, gross electric output, and unit heat rate.

(b) replacing the finned tubes in a boiler economizer has an effect on boiler efficiency, boiler fan power consumption, gross electric output, and unit heat rate.

(c) the condenser pressure can be affected by the fouling of the condenser tubes.

(d) air heater seal degradation causes a greater leakage of air into the gas stream, thus lowering the gas temperature leaving the air heater and increasing fan power. To maintain the minimum cold end temperature, air preheating (steam coils) or air heater bypassing may become necessary.

It is important in performance monitoring to recognize changes in mechanical conditions in the early stages. For example, a change in turbine condition will be revealed through changes in turbine cycle heat rate, turbine section efficiencies, and other turbine indicators, and will also affect the performance of the feedwater heaters and boiler operation. Understanding mechanical interrelationships and incorporating that knowledge into the interpretation of the results is vital to the performance monitoring program.

2-3.16 Matrix of Cycle Interrelationships

A matrix identifying interrelationships that may occur in a typical fossil-fueled electric generating plant is shown in Table 2-3.16-1. The vertical axis lists 13 operational parameters by the plant components or systems with which they are most directly associated. The horizontal axis identifies six performance results categories. The matrix is not all-inclusive, but identifies significant interrelationships that need to be recognized and accounted for in a performance monitoring program. The user is encouraged to modify and expand the matrix to meet unit specific needs.

Table 2-3.16-1 Matrix of Cycle Interrelations

	Performance Results					
Cycle Parameters	**Boiler Operation and Performance**	**Turbine Operation and Performance**	**FW Heaters Operation and Performance**	**Cond./C.T. Operation and Performance**	**Electric Output**	**Fuel Burn Rate**
1. Boiler Fluid Outputs Main steam flow Main steam pressure Main steam temperature	XXX	XXX	XX	XXX	XXX	XXX
2. Reheater Fluid Inputs Cold RH steam flow Cold RH steam pressure Cold RH steam temperature RH spray flow, pressure, temperature	XX	XX	X	X	XX	XX
3. Reheater Fluid Outputs Hot RH steam flow Hot RH steam pressure Hot RH steam temperature	XX	XX	X	X	XX	XX
4. Turbine Sealing, Cooling Steam Inputs Gland sealing steam F., P., T. Stem cooling steam F., P., T.	...	XX	X	XX	X	...

Table 2-3.16-1 Matrix of Cycle Interrelations (Cont'd)

	Performance Results					
Cycle Parameters	**Boiler Operation and Performance**	**Turbine Operation and Performance**	**FW Heaters Operation and Performance**	**Cond./C.T. Operation and Performance**	**Electric Output**	**Fuel Burn Rate**
5. Feedwater Heater, Extraction Stream, Drainage	X	XXX	XXX	X	XX	X
FWH extraction steam flows						
FWH extraction steam pressure						
FWH steam temperature						
FWH drainage pressure						
FWH drainage temperature						
6. Other Turbine Cycle Fluid Extractions	XX	XX	XX	XX	XX	XX
(all F., P., T.)						
Aux. steam						
House heating steam						
Air preheating fluid						
Process/external system						
7. Condenser Steam Side Conditions	XX	XXX	X	XXX	XX	XX
Condenser pressure						
Hotwell condenser temperature						
Air in-leakage/removal rate						

Table 2-3.16-1 Matrix of Cycle Interrelations (Cont'd)

	Performance Results					
Cycle Parameters	Boiler Operation and Performance	Turbine Operation and Performance	FW Heaters Operation and Performance	Cond./C.T. Operation and Performance	Electric Output	Fuel Burn Rate
8. Condenser, Cooling Tower Heat Sink Conditions	XX	XXX	X	XXX	XX	XX
Circulating water flow						
Circulating water inlet temperature						
Cooling tower air flow						
Barometric pressure						
Ambient air temperature						
Ambient air relative humidity						
9. Condensate, Feedwater System Conditions	X	...	XX	...	X	X
L.P. FWH temperature rises						
BF Pumps temperature rises						
FWH pressure drops (tube side)						
H.P. FWH temperature rises						
10. Boiler Fluid Inputs	XXX	X	XXX	...	XX	XXX
Final feedwater flow						
Final feedwater pressure						
Final feedwater temperature						
Main steam DSH flow						

Table 2-3.16-1 Matrix of Cycle Interrelations (Cont'd)

	Performance Results					
Cycle Parameters	**Boiler Operation and Performance**	**Turbine Operation and Performance**	**FW Heaters Operation and Performance**	**Cond./C.T. Operation and Performance**	**Electric Output**	**Fuel Burn Rate**
11. Boiler Combustion Air Inputs	XX	…	…	X	X	XX
Primary, secondary, tertiary air flows						
Barometric pressure						
Ambient air temperature						
Relative humidity						
12. Boiler Incidental Inputs	X	…	…	…	…	X
Seal air sys. flow						
Sootblowing system						
Boiler air in-leakage						
13. Boiler Heat Inputs	XXX	X	X	X	XX	XXX
Fuel flows						
Fuel temperature						
Fuel analyses						
Atomizing steam supply						
Air preheating system						

GENERAL NOTE: Notations are as follows

XXX = high degree of influence
XX = moderate degree of influence
X = low degree of influence
… = negligible to none

2-3.16.1 Purpose. The purpose of the matrix is to assist the performance engineer in recognizing the existence of different interrelationships within the equipment, system, or unit being monitored. Furthermore, it gives a general idea of the degree of influence the interrelationship may have to other equipment, systems, or units. The matrix expresses the typical interrelationships in qualitative rather than quantitative form for several reasons, the more significant of which include

(a) interactions quantitatively tend to behave differently between different units, even between similar units.

(b) interactions frequently tend to behave differently on the same unit at different times, even when conditions appear to be reasonably similar.

(c) many interactions tend to behave very differently on the same unit as operating and mechanical conditions of the unit change

2-3.16.2 Effects of Operational Parameters. The matrix lists operational parameters in equipment or system subgroupings. The effects of the parameters, indicated in general terms in six performance results categories, are

(a) Boiler Operation and Performance. This is the results category identifying the interactions that influence the boiler operation and performance. This category includes the boiler auxiliary equipment such as fans, pumps, air heaters, and environmental controls.

(b) Turbine Operation and Performance. This is the results category identifying the interactions that influence the turbine operation and performance. This category excludes feedwater heaters and the heat rejection equipment (condenser, cooling towers), which are treated separately.

(c) Feedwater Heater Operation and Performance. This is the results category identifying the interactions that influence feedwater heater operation and performance. Only feedwater heaters are included in this category.

(d) Condenser and Cooling Tower Operation and Performance. This is the results category identifying the interactions that influence the condenser and cooling tower (heat rejection equipment) operation and performance. All auxiliary equipment such as vacuum pumps, air ejectors, circulating water pumps, and cooling fans are included.

(e) Unit Net Electric Output. This is the results category identifying the interactions that influence the net electric output.

(f) Fuel Burn Rate. This is the results category identifying the interactions that influence the fuel burn rate.

The relative degree of influence between the operational or mechanical parameter and the performance result categories are shown below.

Symbol	Degree of Influence
XXX	High
XX	Moderate
X	Low
…	Negligible to none

A few rules for using the matrix or for creating a unit specific matrix are in order. It must be recognized that any analysis of this type is dependent upon the level to which the user wishes to pursue the problem. The matrix is intended to convey a general idea. The six performance results are very broad in scope. A unit-specific application of the matrix concept would focus in more detail on the subject problem area. For example, condenser analysis would require monitoring the following operational parameters: tube cleanliness, air in-leakage, circulating water temperature, and flow rate. The corresponding condenser performance parameters would be condenser pressure and condensate temperature.

The relative impact that each general category has on the performance is given in the matrix. However, the user is encouraged to develop unit-specific impacts for each parameter under the 13 general categories.

The matrix in Table 2-3.6-1 is intended as a guide only. All the interrelations on the horizontal axis are valid only within the context of the 13 individual categories. Vertical comparison under each performance category should not be interpreted on a relative basis.

2-3.17 References

[1] ASME PTC 6S-1988, Simplified Procedure for Routine Performance Tests of Steam Turbines, ASME, New York, NY.
[2] ASME PTC 6A-1982, Appendix A to Test Code for Steam Turbines, ASME, New York, NY.
[3] ASME PTC 6-2004, Steam Turbines, ASME, New York, NY.
[4] Booth, J. A., and Kautzmann, D. E., "Estimating the Leakage from HP to IP Turbine Sections," EPRI, Power Plant Performance Monitoring Conference/Workshop, October 1984.
[5] Booth, J. A., and Schofield, P., "Testing to Detect Changes in Performance of a Steam Turbine Cycle," EPRI Heat Rate Conference Proceedings, 1988.
[6] EPRI, Power Plant Performance Monitoring and Improvement, Vol. 5: Turbine Cycle Performance Analysis, EPRI CS/EL-4415 Vol. 5, December 1987.
[7] EPRI, Power Plant Performance Monitoring and Improvement, Vol. 4: Boiler Optimization, EPRI CS/EL-4415 Vol. 4, December 1987.
[8] Biarnes, R., "Optimizing Combustion To Save Energy," Instrumentation in the Power Industry, Vol. 30, ISA, 1987.
[9] Storm, R. F., and Riley, T., "Coal Fired Performance Improvement Through Combustion Optimization," ASME paper 87-JPCC-Pwr-G.
[10] EPRI, Heat Rate Improvement Guidelines for Existing Fossil Plants, EPRI CS-4554 Project 1403-3 Final Report, May 1986.
[11] Linley, Fred H., "Feedwater Heater Operation," 1983 Annual Conference, Engineering and Operations Division, April 13–15, 1983, New Orleans, LA.
[12] Spencer, R. C., Cotton, K. C., and Cannon, C. N., "A Method for Predicting the Performance of Steam Turbine Generators 16,500 kW and Larger," General Electric Company GER-2007C, July 1974, based on ASME paper 62-WA-209.
[13] ASME PTC 4-1998, Steam Generating Units, ASME, New York, NY.
[14] Air Preheater Company, Inc., "Cold-end Temperature and Material Selection Guide for Ljung-strom Air Preheaters," Air Preheater Company, Inc., Wellsville, NY.
[15] Shriver, J., "CO/O2 Boiler Control: Setpoint is Vital," Power, October 1982.
[16] Cotton, K. C., and Schofield, P., "Analysis of Changes in the Performance Characteristics of Steam Turbines," Journal of Engineering for Power, April 1971.
[17] Carcich, J. A., and Shafer, H. S., "Development of a Rational Procedure for Monitoring Steam Turbine Performance," ASME paper 81-JPGC-Pwr-20.
[18] Combustion Fossil Power, Singer, Joseph C., ed., Fourth Edition, 1991, Published by Combustion Engineering, Inc., Windsor, CT.
[19] Steam/its generation and use, Babcock and Wilcox.
[20] Salisbury, J. K., Steam Turbines and Their Cycles, Kreiger Publishing Co., Malabar, FL, 1974.
[21] Bartlett, R. L., Mulder, J. E., and Sheldon, R. C., Steam Turbine Performance and Economics, McGraw-Hill, 1958.
[22] Hoelscher, E. C., and Stiller, M. A., "Put the Wilson Plot to Work as a Feedwater Heater Maintenance Tool," Power, February 1962.
[23] EPRI, Utility-Oriented Approach for Root-Cause Analysis of Power Plant Equipment Problems, EPRI CS-1832, May 1981.

[24] Missimer, J. R., and Brackett, C. A., “Condenser Monitoring for Plant Heat Rate Enhancement,” ASME paper 86-JPCC-Pwr-61.
[25] Karassik, I. J., Krutzsch, W. C., Fraser, W. H., and Messina, J. P., Pump Handbook, McGraw-Hill, 1986.
[26] Standards for Steam Surface Condensers, Eighth Edition, Heat Exchange Institute, 1984.
[27] Addendum 1, Standards for Steam Surface Condensers, Eighth Edition, Heat Exchange Institute, 1989.
[28] ASME PTC 19.11-1997, Water and Steam in the Power Cycle (Purity and Quality, Leak Detection and Measurement), ASME, New York, NY.
[29] Mizzell, G. E., Kleppe, I. A., and Nuspl, S. P., “Application of Acoustic Pyrometry in Boiler Testing,” Proceedings of EPRI 1991 Heat Rate Improvement Conference, 1991.
[30] Kitchen, D., and Whitten, P., “Acoustic Pyrometry for Improved Boiler Operations and Maintenance,” Presented to POWER-GEN ‘88, Orlando, FL, 1988.
[31] Stallard, C. S., Dennison, K., Calluzzo, N. C., and Mehta, A. K., “EPRI Coal Quality Impact Model,” Presented to Impact of Coal Quality on the Economics of Thermal Power Seminar, Halifax, Nova Scotia, June 15 and 16, 1987.
[32] Stallard, C. S., Mitas, D. W., Brown, R. E., and Pavlish, J. H., “Implementing EPRI’s State-of-the-Art Coal Quality Impact Model into Fuel Procurement Practices,” Presented to EPRI Conference on Effects of Coal Quality on Power Plants, Atlanta, GA, October 13–15, 1987.
[33] Stallard, C. S., Mitas; D. W., Pavlish, J. H., and Mehta, A. K., “Assessing the Impacts of Cleaning Coal: An Example of EPRI CQIM Flexibility,” ASME paper 89-JPGC/FACT-1.
[34] Frendberg, A. M., “Effects of Fuel Changes on Boiler Performance,” Presented to Pacific Coast Electrical Association, Engineering and Operating Conference, San Francisco, CA, March 18 and 19,1976.
[35] Kessler, T., Frogner, B., and Divakaruni, S., “Two Expert Systems Approaches to Heat Rate Diagnostics for Fossil and Nuclear Power Plants.”
[36] Frogner, B., Huang, S., et al, “EPRI Demonstration of Decision Tree Approach to Heat Rate Diagnostics for Fossil Power Plants.”
[37] Hargrove, M. J., Chow, O. K., and Nsakala, N. Y., “Combustion Characterization of Beneficiated Coal Based Fuels,” EPRI Coal Quality Conference, September 1990.
[38] Callagher, J. M., “Potential Benefits of an On-Line Coal Quality Monitor and Control System Scheduler,” EPRI Coal Quality Conference, September 1990.
[39] Putman, R. E., “Steam surface Condensers: Basic Principles, Performance Monitoring, and Maintenance,” ASME Press, New York, ISBN 0-7918-0151-9.
[40] Diakunchak, I. S., Performance Deterioration in Industrial Gas Turbines, J. Eng. For Gas Turbines and Power, Vol. 114, April 1992.
[41] McDonel, J.D. and Eiswerth, J.E., Effects of Film Injection on Performance of a Cooled Turbine, AGARD CP 229, 1977.
[42] Kurzke, J. Performance Modeling Methodology: Efficiency Definitions for Cooled Single and Multistage Turbines, ASME Paper 2002-GT-30497.
[43] EPRI, Coal Flow Loop: Evaluation of Extractive Methods, EPRI 1004744, March 2005.
[44] EPRI, Coal Flow Loop: Evaluation of Extractive Methods-Addendum, EPRI 1010319, October 2006.
[45] ASME PTC 12.1 Closed Feedwater Heaters, ASME, New York, NY.
[46] Harpster, J.W., “Reducing Dissolved Oxygen Under Conditions of High Air Ingress” ASME International Joint Power Generation Conference, Phoenix, AZ, June 24–26, 2002.
[47] Harpster, J.W. and Rubin, L.J., “Recovery of MWe Capacity Through Condenser Improvements” EPRI NPPI Conference, Austin, TX, June 28 and 29, 2004.
[48] Harpster, T.J. and Harpster, J.W., “Improving Condenser Performance and Reducing Dissolved Gases By Tackling Their Common Root Causes,” The 27th Annual Electric Utility Chemistry Workshop at the University of Illinois, Champaign, IL. May 15–17, 2007.
[49] James, R., “Metering of Steam-Water Two Phase Flow by Sharp-Edged Orifices,” Proc. Inst. Mech. Engineers, 1965–1966, Vol. 180, No. 93, pp. 549–566.

2-4 INCREMENTAL HEAT RATE

2-4.1 Introduction

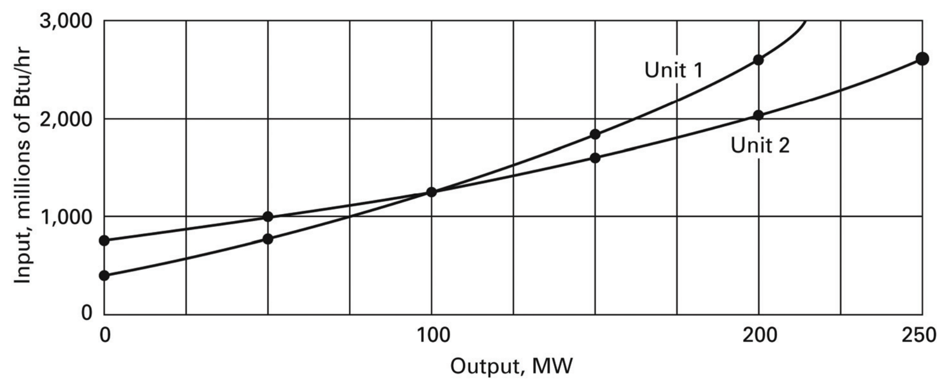
An economic choice of which generating unit should provide the next increment of power to a change in load demand should be based on incremental costs. The change in fuel input required for an incremental change in power output, and the cost of that quantity of fuel must be determined. A performance monitoring program can provide these answers as well as optimize costs for independent power producers and electrical interconnection sales by improving overall delivered heat rate.

2-4.2 Input–Output Relationships

Input–output relationships provide the basics for building incremental cost data. The relationship between input, generally expressed in terms of energy (kJ/hr or Btu/hr), and output, typically expressed in megawatts (MW) for power plants, does not necessarily vary uniformly over the entire load range. The input–output relationship may be determined from design or guarantee data supplied by equipment manufacturers or from test data. Figure 2-4.2-1 illustrates input–output relationships for two steam-cycle generating units. Figure 2-4.2-2 illustrates the same relationship for a 2 × 1 combined cycle facility.

For most fossil base loaded units, the incremental heat rate curve is relatively flat from minimum to full load. A fossil unit's incremental heat rate curve outside the startup block is close to linear in nature. For most fossil units, boiler efficiency decreases with increasing load, causing incremental heat rate to also increase as load increases, but the input–output relationship is impacted by many variables, including turbine cycle design, control valve operation, condenser pressure, ambient conditions, spray flows, auxiliary equipment operation, steam path degradation, feedwater heater operation, and boiler efficiency. There may be certain load ranges where the heat input required shifts one way or the other depending on the number of auxiliaries required to support the increased load, or if certain equipment becomes limiting, such as the cooling tower's capacity to maintain the steam turbine back pressure at optimal conditions. These effects on the incremental heat rate performance of the unit need to be understood so that the performance monitoring program can account for them in dispatch and when making recommendations for optimizations, especially in an on-line system.

Fig. 2-4.2-1 Input/Output Curves for the Two Typical Thermal Units (Based on Fig. 20, p. 40, of *Power System Operation* by R. H. Miller; McGraw-Hill, 1983; Published by Permission of McGraw-Hill)



GENERAL NOTE: Although the no-load fuel for Unit 2 is greater than that of Unit 1, at loads above 100 MW, the heat input for Unit 2 is less than that of Unit 1.

Fig. 2-4.2-2 Input/Output Relationships for a 2 × 1 Combined Cycle Facility

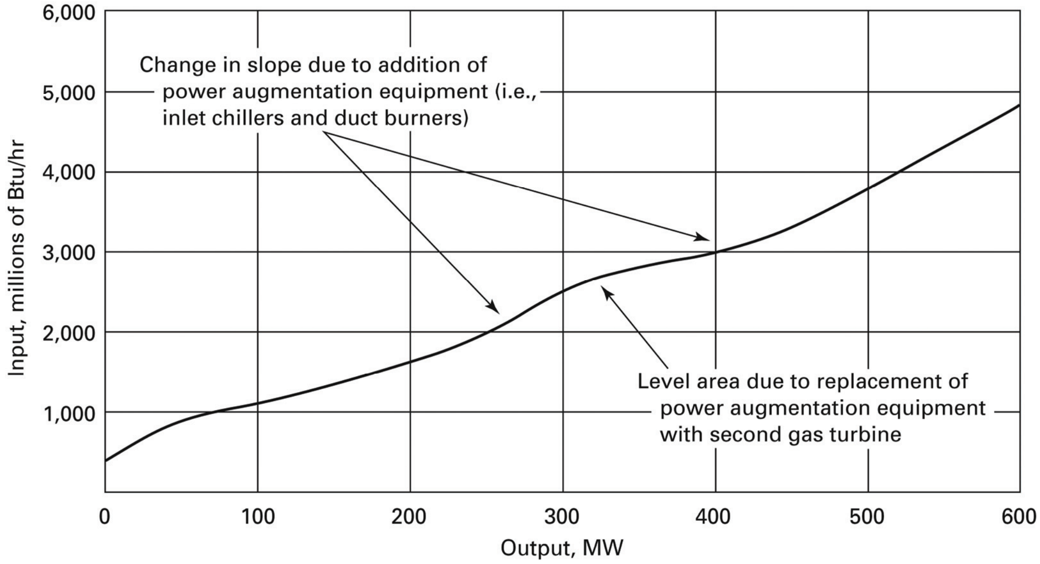
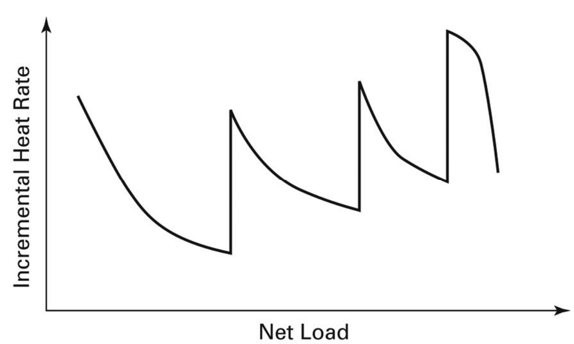


Fig. 2-4.2-3 Incremental Heat Rate for Steam Turbine With Sequential Valve Operation (Courtesy General Physics Corporation)



For partial-arc admission steam turbines, the incremental heat rate varies substantially from one megawatt to the next due to throttling losses that occur across the control valves (see Fig. 2-4.2-3). However, because most power control centers can only accept incremental cost relationships that have a continuous positive slope, the true incremental heat rate in Fig. 2-4.2-3 cannot be used. Instead, a compromise is made by fitting a smooth curve to the input–output relationship and taking its derivative to determine its slope (dQ/dMW), and thus incremental heat rate (Btu/kWh or KJ/kWh). The input–output curve may be second order polynomial or higher, with a positive slope typically required by most dispatch systems. The slope of the incremental heat rate curve varies between units and is predominantly a function of the turbine cycle. Equipment degradation and normal operating conditions can impact incremental heat rate. Once placed into dispatch, generating units with shallow slopes will reach full load more quickly than those with steeper slopes.

For gas turbine-based combined cycle units, the incremental heat rate can be more complex due to several operating modes consisting of gas turbines often being in and out of service, fired and unfired HRSG operation, and simple cycle operation. Each operating mode has its own distinct input–output and incremental cost relationships. Gas turbines are most efficient at full (base) load. As the unit is lowered in load, the heat rate increases significantly. Gas turbine heat rates at minimum load can easily be 200% of the base load heat rate. Combined cycle plants may also include power augmentation equipment, such as gas turbine inlet air conditioning systems, steam injection to the gas turbine, and/or duct burners in the HRSG. These additions to the cycle also impact the incremental heat rate. For example, output gained by increasing the gas turbine 10 MW from part load to full load will be accomplished with less fuel (Btu/hr) than a 10 MW increase accomplished by using the duct burners in the HRSG. Understanding the component fuel usage as well as the impact to the overall cycle is required to make recommendations and optimize the output and heat rate of the overall plant. In some cases, it may be desirable to model the power augmentation equipment as separate units for the sake of incremental heat rate analysis.

2-4.3 Incremental Costs

Incremental costs are calculated from the input–output relationships developed by factoring in fuel costs. Table 2-4.3-1 illustrates incremental rates for the two generating units shown in Fig. 2-4.2-1. The table illustrates that it is far less expensive to load up Unit 2 to higher levels of capacity than it is to load up Unit 1.

Incremental costs for combined cycle units not only encompass which unit to load up next, but also if it is more economical to add load by adding an additional gas turbine, or by utilizing power augmentation equipment such as inlet cooling, steam or water injection, or HRSG duct burners. The relative costs of incremental output from each piece of equipment can change based on the prices for fuel, water, electricity, and expected maintenance considerations. Table 2-4.3-2 illustrates the relative incremental costs associated with a combined cycle facility.

Table 2-4.3-1 Incremental Rates for the Two Generating Units in Fig. 2-4.2-1

Load MW	Instantaneous Heat Rate	Incremental Heat Rate, Btu/kWh	Million Btu/hr	Dollars/hr [Note (1)]	Charge for Load Increment, dollars/hr	Incremental Rate, dollars/MWh
Unit 1						
0	…	…	400	1,000	…	…
50	14,000	6,000	700	1,750	750	15.00
100	12,000	10,000	1,200	3,000	1,250	25.00
150	12,000	12,000	1,800	4,500	1,500	30.00
200	13,000	16,000	2,600	6,500	2,000	40.00
250	16,000	28,000	4,000	10,000	3,500	70.00
Unit 2						
0	…	…	800	2,000	…	…
50	20,000	4,000	1,000	2,500	500	10.00
100	12,000	4,000	1,200	3,000	500	10.00
150	10,667	8,000	1,600	4,000	1,000	20.00
200	10,000	8,000	2,000	5,000	1,000	20.00
250	10,400	12,000	2,600	6,500	1,500	30.00

NOTE:
(1) At $2.50 per MBtu/hr

Table 2-4.3-2 Relative Incremental Costs Associated With a Combined Cycle Facility

Load MW	Instantaneous Heat Rate	Incremental Heat Rate, Btu/kWh	Million Btu/hr	Dollars/hr [Note (1)]	Charge for Load Increment, dollars/hr	Incremental Rate, dollars/MWh
2 × 1 Combined Cycle						
0	…	…	400	1,000	…	…
50	18,170	10,200	910	2,275	1,275	25.50
100	10,980	3,800	1,100	2,750	475	9.50
150	9,070	5,200	1,360	3,400	650	13.00
200	8,130	5,400	1,630	4,075	675	13.50
250	7,850	6,600	1,960	4,900	825	16.50
300	8,270	10,400	2,480	6,200	1,300	26.00
350	7,940	6,000	2,780	6,950	750	15.00
400	7,390	3,600	2,960	7,400	450	9.00
450	7,260	6,200	3,270	8,175	775	15.50
500	7,490	9,600	3,750	9,375	1,200	24.00
550	7,780	10,600	4,280	10,700	1,325	26.50
600	8,030	10,800	4,820	12,050	1,350	27.00

NOTE:
(1) At $2.50 per MBtu/hr

2-4.3.1 Optimum Load Division. Once the incremental cost relationships are known for two or more generating units, the division of load among the units leading to the lowest overall fuel cost (economic dispatch) may be determined. A mathematical proof will show that the least cost for fuel is achieved when all units are loaded at equal incremental cost. This is shown graphically in Fig. 2-4.3.1-1.

Referring to Fig. 2-4.3.1-1, the total load demand is 585 MW; therefore, the sum of Unit A and Unit B outputs must equal 585 MW. While the capacity demand can be met by any arbitrary combination of A and B outputs that sums to 585 MW, the least cost operating point is achieved when both units are operated at loads corresponding to the same incremental heat rate. For the 585 MW case, operating Unit A at 347 MW and Unit B at 238 MW results in the least overall cost. This is demonstrated in the following paragraphs.

The total heat input is expressed as unit heat rate multiplied by the unit load

$$Q_{PLANT} = HR_A \times MW_A + HR_B \times MW_B$$

For two units firing the same fuel, the total fuel cost is calculated as

$$\$_{PLANT} = \$/MMBtu \times \left(HR_A \times MW_A + HR_B \times MW_B\right)$$

Table 2-4.3.1-1 shows plant economies corresponding to unit capacity combinations that meet plant demand of 585 MW.

Because Unit A is more efficient than Unit B over its load range, the tendency might be to load Unit A to full capacity and then bring Unit B on to make up the difference. This case corresponds to Scenario # 8, which has a cost of \$19.73/MWh. However, loading these units at equal incremental heat rates results in a

Fig. 2-4.3.1-1 Optimum Load Division by Equal Incremental Heat Rate (Courtesy General Physics Corporation)

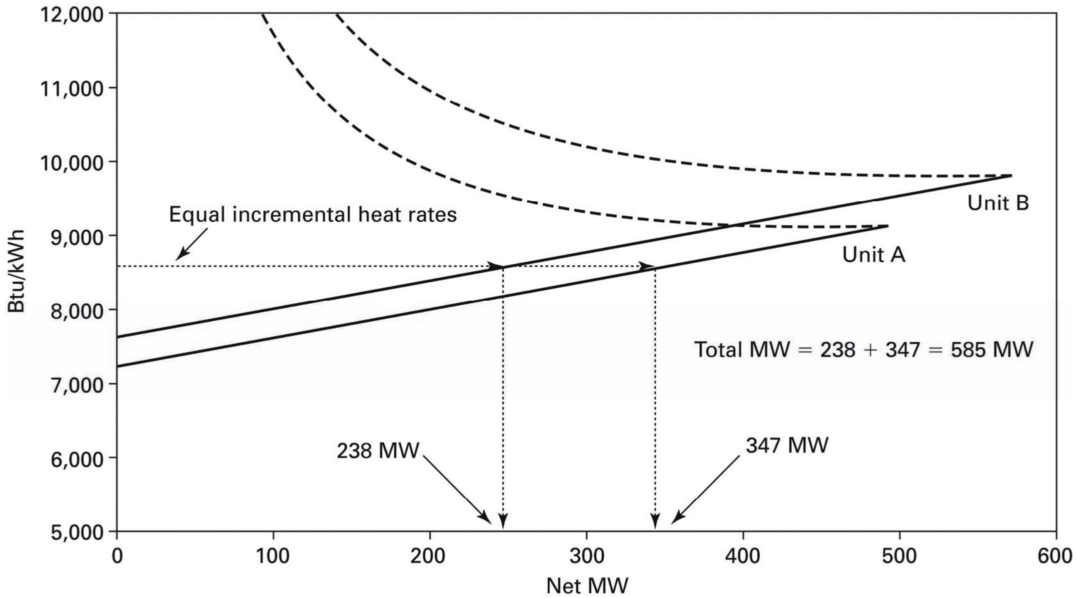


Table 2-4.3.1-1 Impact of Load Division on Plant Economy (Courtesy General Physics Corp.)

Scenario	Unit "A" Output (MW)	Unit "B" Output (MW)	Plant Output MW	Unit "A" Input (MMBtu/h)	Unit "B" Input (MMBtu/hr)	Plant Input (MMBtu/hr)	Plant Heat Rate (Btu/kWh)	Plant Fuel Cost ($/MWh)
1	100	485	585	1 205	4 756	5 962	10 191	20.38
2	150	435	585	1 591	4 287	5 878	10 048	20.10
3	200	385	585	1 987	3 827	5 814	9 938	19.88
4	250	335	585	2 392	3 376	5 768	9 860	19.72
5	300	285	585	2 807	2 934	5 741	9 813	19.63
6	**347**	**238**	**585**	**3 205**	**2 527**	**5 732**	**9 799**	**19.60**
7	400	185	585	3 664	2 078	5 743	9 817	19.63
8	450	135	585	4 108	1 664	5 772	9 867	19.73

GENERAL NOTE: Fuel cost is assumed to be $2.00/MMBtu.

lower fuel cost of $19.60/MWh. This corresponds to approximately $222,500 annually for this 1,085 MW plant. This demonstrates the order of magnitude of savings that can be achieved with economic dispatch.

2-4.3.2 System Constrained Economic Dispatch. Dispatch of generating assets must also consider operating and transmission system constraints. These include conditions internal to the plant such as minimum stable operating load and maximum output, as well as external limits such as the location of the plant on the transmission grid. The dispatch solution must be determined in compliance with these constraints.

2-4.4 Incremental Heat Rate by Test

Establishing the input–output relationship from measured data requires careful attention to data validation and instrument accuracy.

(*a*) Incremental heat rate is a measure of the heat required to generate the next increment of power above (or below) the current operating level of generation. It is the information the power dispatcher uses to economically determine which unit or units to adjust as system load varies.

(*b*) The development of incremental heat rate is a procedure that utilizes the same performance monitoring data that is collected for the various other purposes described in this document. In establishing test loads and procedures (for all uses) it is necessary to understand the unique nature of incremental heat rate characteristics.

(*1*) Incremental heat rate is the rate of change or derivative of the firing rate that determines the incremental value; therefore, the determination of shape or curvature is more important than absolute level.

(*2*) Incremental heat rates are not always monotonically increasing (positive), and may be less than overall heat rates. Examples of incremental heat rates that are not monotonically increasing include those caused by a partial arc turbine control valve loop, or the reduction in overall heat rate observed when changing from a combined cycle unit operating in a 1 × 1 configuration (one gas turbine and one steam turbine) with full inlet chilling and duct burner firing, to a 2 × 1 configuration (two gas turbines and one steam turbine) with no inlet chilling or duct burner firing required (see Fig. 2-4.4-1).

(*3*) Many dispatch systems can accommodate multiple curve segments that can be situated so as to coincide with changes in equipment operations, including control valve intercepts or valve points, and auxiliaries such as duct burners or coal pulverizers.

(*4*) The curvature of the heat input or firing rate relative to net output (or load) determines the slope (and therefore the unit loading rate) of the incremental curve.

(*c*) It should be noted that the sensitivity of incremental heat rate to test measurements is much higher than for simple heat rate only, resulting in larger influence coefficients as described earlier in this guideline. This should be taken into consideration when selecting instrumentation used to measure operating parameters when determining incremental heat rate is one of the objectives.

The above factors, in addition to those considered as good practice for collecting performance monitoring data, must be considered in establishing test protocol for incremental heat rate determination.

(*d*) Once validated test data has been acquired, the steps of incremental heat rate development are

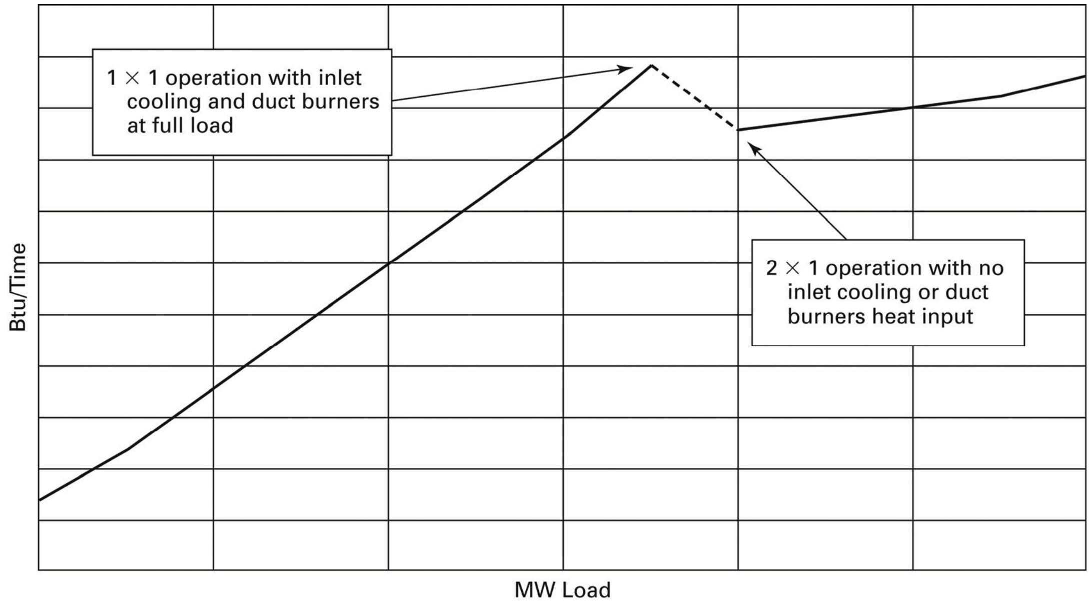
(*1*) plot the firing rate versus flow or load relationship as determined by test and illustrated in Fig. 2-4.4-2.

(2) curve fit the data to one or more polynomials (or other curve fit algorithm) acceptable to the applicable dispatch algorithm.

(*3*) take the derivative of the function relative to load to find the instantaneous incremental heat rate for any load point, or calculate the slope at the load points of interest.

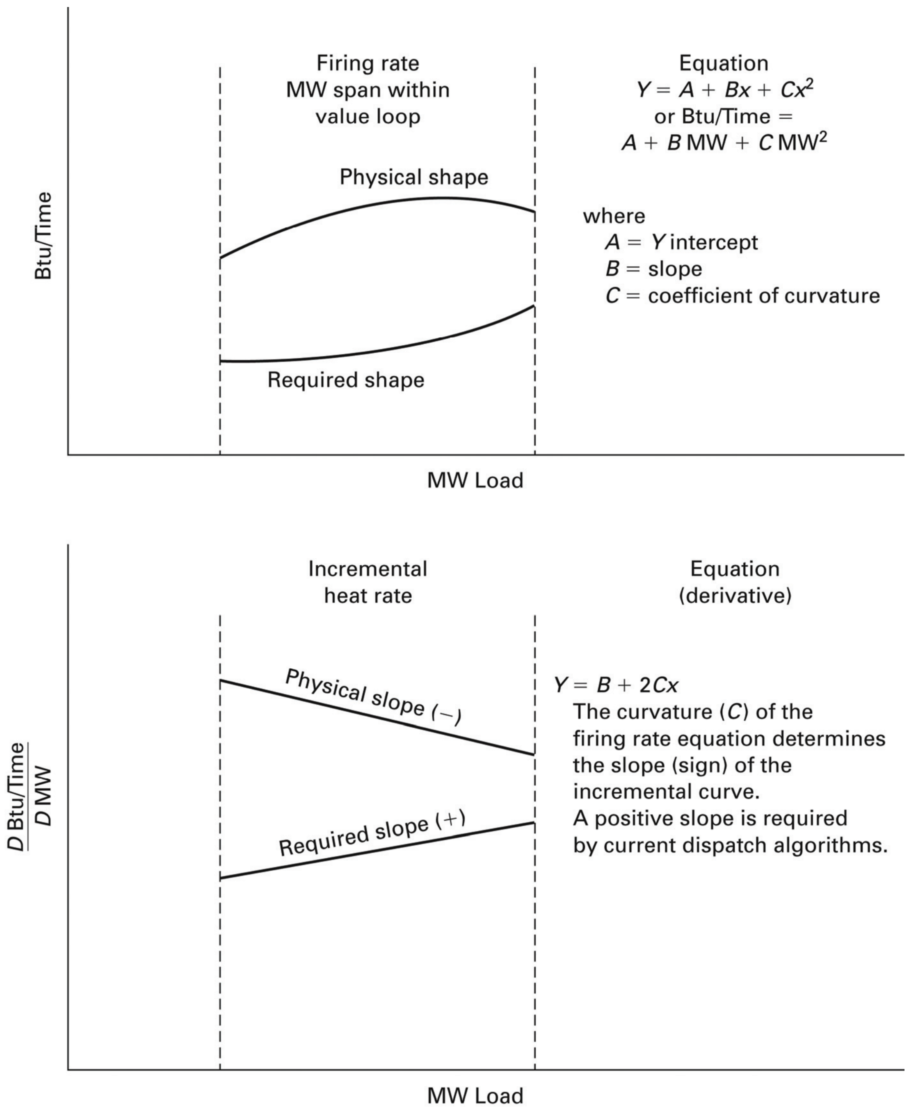
(*4*) determine the additional heat input required for a set change in load in cases where the dispatch algorithm may be set to require such information, for example, how much additional heat input is required for an increase in load of 25 MW. In this case, the heat rates at specific points along the firing rate curve will be needed for the incremental heat rate determination.

Fig. 2-4.4-1 Example of Heat Rate Not Monotonically Increasing in a 2 × 1 Configuration



Load MW	Million Btu/hr	Dollars/hr	Charge for Load Increment, dollars/hr	Incremental Rate, dollars/MWhr
290	2,370	5,925	…	…
295	2,420	6,050	125	25.00
300	2,480	6,200	150	30.00
305	2,540	6,350	150	30.00
310	2,600	6,500	150	30.00
315	2,660	6,650	150	30.00
320	2,720	6,800	150	30.00
325	2,790	6,975	175	35.00
330	2,730	6,825	(150)	(30.00)
335	2,740	6,850	25	5.00
340	2,750	6,875	25	5.00
345	2,760	6,900	25	5.00
350	2,780	6,950	50	10.00
	—			
325	2,790	6,975	175	35
330	2,730	6,825	–150	–30

Fig. 2-4.4-2 Incremental Curve Shape



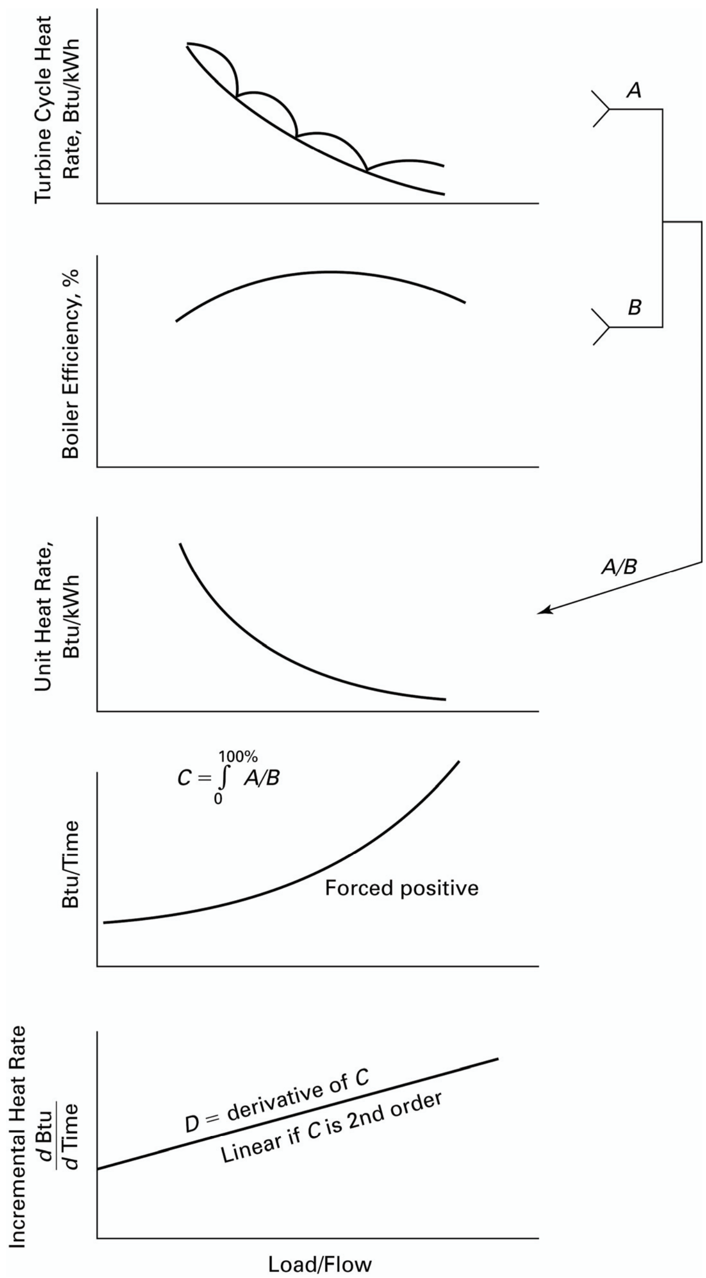
(*e*) In the absence of accurate heat input measurement capability, which is often the case for coal-fired units, the task of unit characterization is best determined by boiler and turbine cycle energy balances.

The steps required, as illustrated in Fig. 2-4.4-3, are to

(*1*) develop a net turbine cycle heat rate that subtracts from gross generation all auxiliary power consumption, including that used to support boiler operation.

(*2*) develop a boiler efficiency curve as defined previously in this guideline.

Fig. 2-4.4-3 Illustration of Development of Incremental Heat Rate Information From Basic Plant Measurements



(*3*) calculate a unit heat rate by dividing the turbine cycle heat rates by their associated boiler efficiency

(*4*) integrate firing rates from unit heat rate

(*5*) curve fit the firing rate data to one or more acceptable polynomial equations

(*6*) multiply by fuel cost data and other miscellaneous costs attributed to maintenance or environmental factors

(*7*) apply corrections to account for seasonal conditions

(*8*) apply corrections for line or transformer losses depending upon the ultimate end user of the power

(f) Impact of Measurement Uncertainty on Incremental Heat Rate Relationship. Applying uncertainty principles to incremental heat rate testing provides valuable insight into the limitations of the test procedure and instruments chosen. It can also be helpful in interpreting the validity of changes in incremental heat rate observed between tests. This is especially important since the end result of incremental heat rate testing has a direct impact on commercial operations.

The slope of the incremental heat rate curve is very sensitive to the shape of the input–output curve. For a second-order polynomial input–output relationship, the uncertainty of incremental heat rate may be as much as 3.7 times the uncertainty of the heat rates determined from the same curve. This sensitivity places a very significant demand on the quality of test data used to develop incremental cost curves.

There are three types of error that contribute to the overall uncertainty of the incremental heat rate: bias error, random error, and residual error. Bias error in the test data can be minimized by using calibrated instruments for all critical measurements, especially fuel or feedwater flow, fuel heating value, and power output. Random error can be minimized by collecting data at a high frequency with the unit in a stable operating mode. Residual error is a function of how well the input–output curve predicts the unit's heat rate from the calculated test results. This can be minimized by using a detailed thermodynamic model, in which energy and mass are conserved, to understand the nature of a particular unit's input–output relationship prior to selecting a curve order. The dispatch or power control center may also place constraints on the type and order of curve employed.

Regardless of the care taken when collecting input–output test data, bias and random errors combine to make any incremental heat rates determined from test data alone highly uncertain. The impact of these errors on the incremental heat rate of a fossil generating unit is illustrated in the following example. Figure 2-4.4-4 shows the design net unit heat rate and incremental heat rate plotted versus load. In this case, the incremental heat rate is represented as the first derivative of a second-order polynomial curve fit of the heat input and net power output. A 1% bias error in net unit heat rate results in the range of possible incremental heat rates shown in Fig. 2-4.4-5. This shows that the net unit heat rate and incremental heat rate are affected equally by bias error. As such, heat rate bias error has a minimal impact on the slope of the incremental heat rate curve.

Unlike bias error, random error in the test data has a pronounced effect on the shape of the input–output curve, and therefore the slope of the incremental heat rate. A random error of ±0.5% applied to the upper (+1%) and lower (–1%) bias error boundaries results in the range of possible incremental heat rates shown in Fig. 2-4.4-6. Thus, while the test heat rate uncertainty may be considered reasonable (±1.42%), the incremental heat rate uncertainty may be unacceptable (as much as 3.7 times higher or ±5.3% for this example). This demonstrates the importance of understanding the impact of test uncertainty when establishing incremental heat rate relationships solely from test data.

2-4.5 Incremental Heat Rate by Model

Computerized thermodynamic models, in which mass and energy are conserved, are commercially available and can provide insight into the nature of a unit's incremental heat rate. This is especially the case with multiple operating modes of combined cycle plants that can often not be tested fully due to constraints on personnel resources and commercial availability. Such models can also help minimize the large potential

Fig. 2-4.4-4 Heat Rate and Incremental Heat Rate Versus Load Fossil Unit (Courtesy General Physics Corp.)

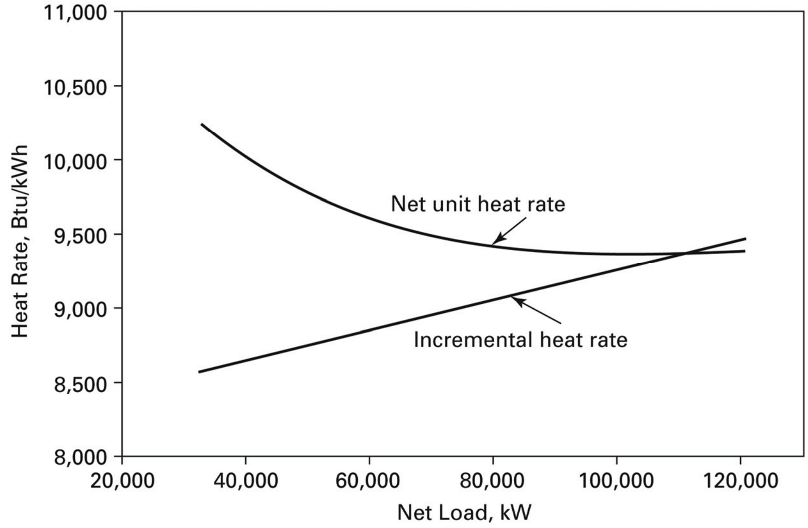


Fig. 2-4.4-5 Heat Rate and Incremental Heat Rate Versus Load Bias Error (Courtesy General Physics Corp.)

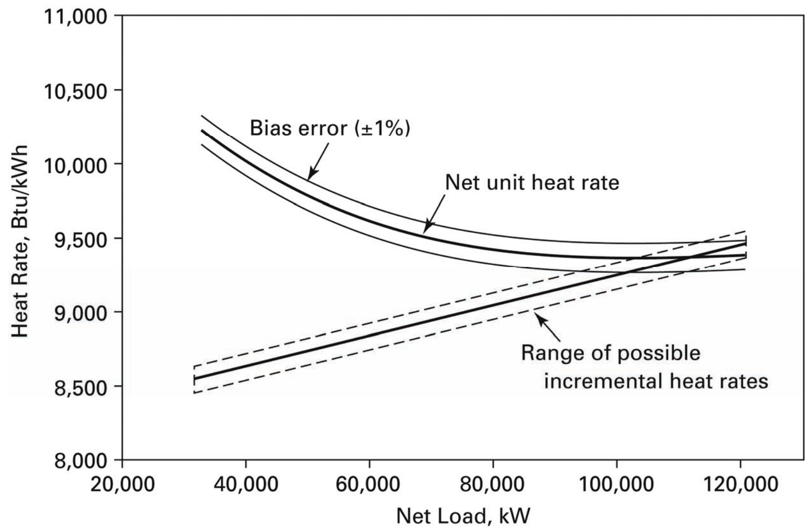
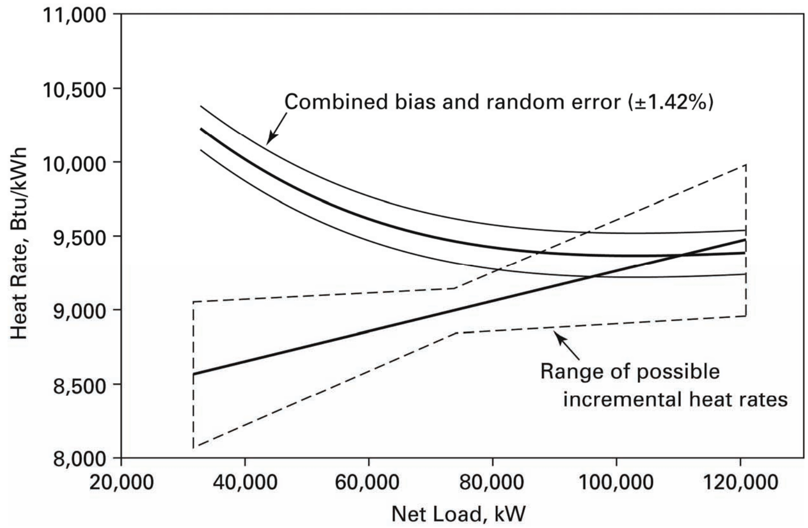


Fig. 2-4.4-6 Heat Rate and Incremental Heat Rate Versus Load Combined Bias and Random Error (Courtesy General Physics Corp.)



uncertainty of incremental heat rates calculated using test data alone. For fossil plants, a thermodynamic model can be used to validate test data collected for the purpose of developing the input–output and incremental heat rate relationships.

2-4.5.1 Fossil Plant Incremental Heat Rate. The procedure for using a thermodynamic model for determining the incremental heat rate of a fossil generating unit is as follows:

(*a*) Create a turbine cycle model that matches the turbine cycle design heat balance. Tune the design model to match observed test results across the load range (e.g., HP turbine efficiency, IP turbine efficiency, condenser pressure, and feedwater heater performance).

(*b*) Create a boiler model that matches the design of the unit. Tune the design boiler model to match observed test results across the load range, for example, fuel constituents, excess air, steam production, superheater and reheater outlet temperatures, spray flows, and air heater performance.

(*c*) Determine the auxiliary power consumption across the load range for normal operating conditions.

(*d*) Combine the above three models into an integrated plant model that can be exercised across the load range to establish a model-based input–output curve that has been tuned to actual equipment conditions. The tuned model, when bounded with actual test conditions, should produce the observed power output for the test fuel or feedwater flow.

Using a thermodynamic model tuned to actual plant conditions provides the ability to develop consistent incremental heat rate relationships with varying seasonal conditions (cooling water temperature, ambient air temperature) and operating modes (top heater out of service, fuel switching, fouled condenser, etc.). The tuned model should have the ability to accurately predict plant performance throughout the normal operating range.

2-4.5.2 Combined Cycle Plant Incremental Heat Rate. The procedure for using a thermodynamic model for determining the incremental heat rate of a combined cycle generating unit is as follows:

(*a*) Create a gas turbine model that matches the original OEM specifications. Tune the design model to match observed test results across the load range (e.g., gas turbine output, heat rate, exhaust flow, and exhaust temperature).

(*b*) Create an HRSG model that matches the design of the unit. Tune the design HRSG model to match observed test results across the load range, for example, steam production, steam temperatures, and efficiency.

(*c*) Create a steam turbine model that matches the original OEM specifications. Tune the design model to match observed test results across the load range (e.g., throttle flow, power output, and HP and IP efficiencies).

(*d*) Determine the auxiliary power consumption across the load range for normal operating conditions.

(*e*) Combine the above four models into an integrated plant model that can be exercised across the load range for each operating mode (simple cycle, unfired, fired, etc.) to establish a model-based input–output curve that has been tuned to actual equipment conditions. The tuned model, when bounded with actual test conditions, should produce the observed power output for the test fuel flow.

A thermodynamic model of a combined cycle plant, when tuned to actual conditions, allows the impact of taking gas turbines, evaporative coolers, duct burners, and other power augmentation equipment in and out of service to be readily predicted. Careful tuning to match conditions observed during routine performance monitoring is required to validate the model.

2-4.6 Variation of Heat Rate During Normal Operation

(*a*) The preceding assumes a fully off-line process. The advantage of using on-line data in an automated system is that operational, maintenance, or ambient conditions that are different from the most recent test period can be reflected in incremental costs.

(*b*) Performance factors affecting the incremental costs during normal operation include

(*1*) step changes in auxiliary power requirements

(2) steam temperatures

(3) steam pressures

(4) excess air

(5) exit gas temperature

(6) moisture in the fuel

(7) onset of desuperheating sprays

(8) condenser pressure (ambient or equipment condition)

Continuous or periodic updating of the incremental costs is desirable as equipment degradation or maintenance activities affect the performance of each unit.

2-4.6.1 Special Considerations for Combined Cycle Plants. The thermal performance of gas turbine-based power plants is especially sensitive to changes in ambient temperature, where capacity can decline dramatically as air inlet temperature increases. Heat rate also increases as air inlet temperature increases. Heat rate characteristics for a typical 1 × 1 combined cycle power plant are shown in Fig. 2-4.6.1-1. The three operating modes are clearly differentiated: simple cycle, combined cycle unfired, and combined cycle fired. For the temperature range considered (60°F to 120°F), and for the three operating modes, base load plant capacity changes by approximately 24%, 20%, and 14%, respectively. Heat rate variations are less pronounced at 8.4%, 2.7%, and 3%, respectively.

2-4.6.2 Combined Cycle Incremental Heat Rate. When determining the most economical unit commitment, an asset's incremental heat rate (and cost) is taken into account. The incremental heat rate is defined as the ratio of a change in fuel heat input to the resulting change in electrical output. Engineering units for incremental heat rate are the same as for unit heat rate (Btu/kWh, kJ/kWh), but the meaning is quite different. Incremental heat rate represents the additional heat input required for a given change in unit output. In practice, it is most often represented as the mathematical slope of the input–output relationship (see Fig. 2-4.6.2-1). For the subject 1 × 1 combined cycle plant, this slope varies substantially with operating mode and with changes in unit load (see Fig. 2-4.6.2-2). For the temperature range considered (60°F to 120°F), the incremental heat rate (and thus cost) varies by as much as 1,500 Btu/kWh in simple cycle mode, 750 Btu/kWh

Fig. 2-4.6.1-1 Combined Cycle Heat Rates Versus Ambient Temperature (Courtesy General Physics Corp.)

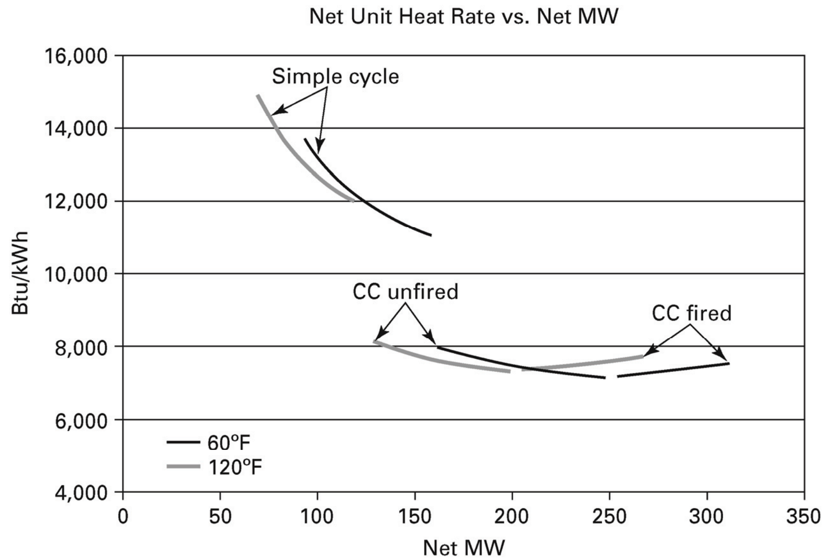
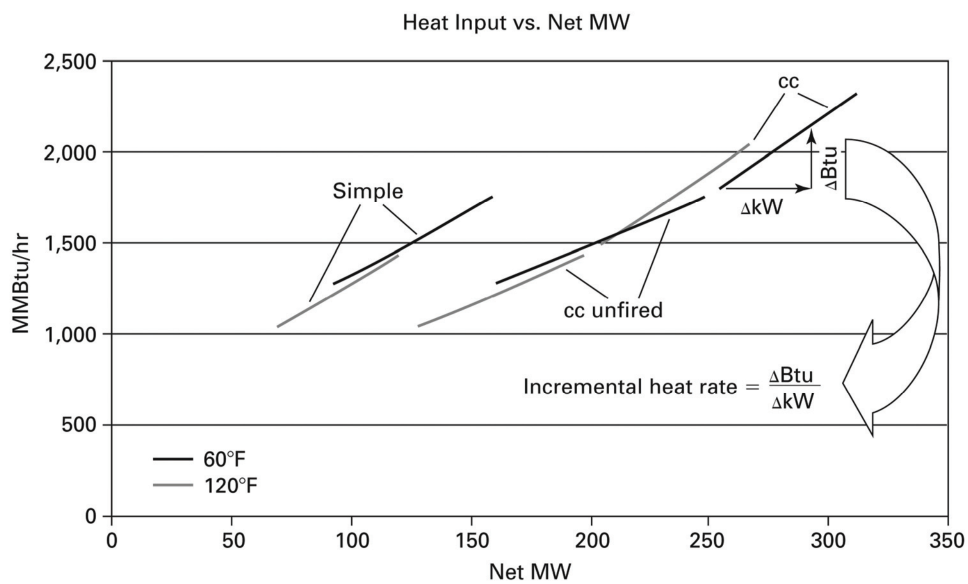
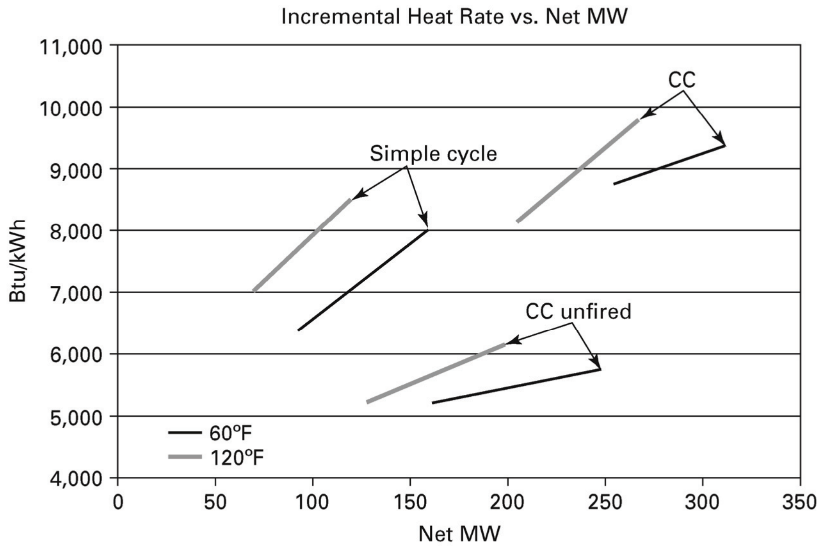


Fig. 2-4.6.2-1 Combined Cycle Input/Output Relationships (Courtesy General Physics Corp.)



in unfired combined cycle mode, and 1,000 Btu/kWh in fired combined cycle mode at the same load. With the gas turbine in base load mode, this difference is a consistent 500 Btu/kWh. Unless considered in unit commitment decision support systems, these variations can result in substantial differences in projected and realized profits.

Fig. 2-4.6.2-2 Combined Cycle Incremental Heat Rates Versus Ambient Temperature (Courtesy General Physics Corp.)



2-5 PERFORMANCE OPTIMIZATION

2-5.1 General

Optimization is one of the most important objectives of a performance monitoring program. The objective of performance optimization is to produce the most cost-effective performance possible. This subsection provides guidance on optimization concepts and techniques. Benefits gained from effective performance optimization may be

(*a*) economic, through reduction of net total production costs, effective maintenance planning, and determination of future capital improvements

(*b*) technical, through generally improved equipment conditions by way of taking corrective actions, both operational and through mechanical work

(*c*) informational, through revelation of previously unknown, little understood, or unquantified technical and economic information concerning units being monitored and optimized

Subsections 2-1, 2-2, and 2-3 are all particularly relevant to performance optimization. It is suggested that they be used as a reference by engineers who are experienced in optimization work. For the less experienced, it is suggested that these paragraphs be carefully studied before attempting optimization.

2-5.1.1 Definition and Explanation. Optimization is the process through which performance is taken to the highest levels that are technically feasible and cost effective. Optimization will generally lead to performance improvement, and may also result in capacity increase.

Performance optimization means more than a one-time or a short-term effort. Such an optimization effort may result in some cost-effective performance improvement, but there is no assurance that it will be sustained. In order to achieve the most value over the long term, it is necessary to maintain the optimization efforts at a level that will continue to produce the desired results.

2-5.1.2 Applications. Performance optimization can be approached from an economic, operational, or maintenance and modification (mechanical) aspect of power production. The three applications involve different concepts and different approaches. They are very much interrelated, however, since all three applications strive to achieve the most cost-effective performance possible. There are further interactions between the mechanical and operational applications, due to the interdependency between equipment operation and the maintenance and modification requirements of that equipment.

(a) Operational. This involves operation of equipment in such a manner as to produce the best performance possible. Optimization is accomplished through economic tradeoff of various operating modes. Operational optimization is normally geared toward efficiency improvements in the boiler, turbine, or balance-of-plant equipment, to reduce unit heat rates or emissions. Beyond these traditional considerations, operational optimization may also raise capacity, improve availability, and reduce maintenance.

Operating practices to achieve best performance must be governed at all times by certain limitations and constraints. It is necessary to operate safely, reliably, and with environmental responsibility. Maximized performance is not always consistent with these considerations. However, optimized operation recognizes them as vital aspects, and seeks to identify the best balance points in operation that are safe, reliable, and environmentally sound, and that at the same time deliver the best performance achievable.

(b) Maintenance and Modification (Mechanical). This application involves the conduct of mechanical work on equipment in such a manner as to produce the greatest ratio of value from the work to the cost of performing that work. Maintenance work refers to the correction of problems without modifying or upgrading the equipment. Modification work may involve design changes, equipment replacements, or any significant modifications, upgrades, or retrofits. For the balance of this subsection, the term "mechanical" will be used synonymously with maintenance and modification to distinguish from the operational approach to optimization. Limitations and constraints on mechanical optimization must recognize the same safety, reliability, and environmental considerations as operational optimization.

(*c*) *Economic*. Optimizing for economic considerations is market-based and normally a short-term application used to capture the maximum profits during short, high-priced market hours. During times when the power plant may be selling generation on the open market, the goal of the optimization routines is to maximize profits. A simple approach would be to maximize generation at the expense of current efficiency. This may be as simple as turning on inlet air conditioning equipment or steam injection systems for a combined cycle plant, or running in an over-pressure condition on a boiler plant. It is important to consider the long-term impact of running at these maximum capacity conditions, since the impact to major maintenance cycles (and therefore maintenance costs) will reduce and may even remove any additional profits gained by selling the additional capacity.

2-5.1.3 Additional Information and References. In addition to this subsection of the guidelines, there are many other information sources that address performance improvement and optimization. Some of these are wide in scope, and attempt to provide broad treatment of the subject. Other sources are much more specific, and focus on defined aspects of optimization. Users are referred to the list of references at the end of this subsection.

2-5.2 Operational Optimization—Empirical Techniques

2-5.2.1 General. The following information is given as guidance on empirical optimization of most equipment comprising fossil-fueled steam units. The specific operating and equipment circumstances must dictate the actual techniques to be used. User judgment will have to prevail and drive the overall empirical optimization process. Paragraph 2-5.3.2 describes four different methods for approaching operational optimization. None of the four can be considered a fully stand-alone approach. However, the first method, operational testing, is generally believed to be the most productive and the most powerful in terms of identifying the most efficient operational practices to employ for all applications.

Some of the more important performance factors that can be evaluated through operational testing are discussed below. Guidance is given according to major equipment and to the various parameters of equipment operation. In almost all cases, the evaluation of these factors may be done through the methods described in para. 2-5.3.

2-5.2.2 Nonboiler Applications. Optimization of non-boiler applications centers around the combined cycle facility that consists of one or more gas turbine generators, heat recovery steam generators, steam turbines, condensers, and heat rejection equipment (such as a wet cooling tower or air cooled condenser). Optimization of combined cycle facilities can occur at both the component and system levels.

2-5.2.2.1 Gas Turbine Optimization. Most operating groups have limited control over the operation of the gas turbine. The OEM is responsible for tuning the unit to meet its initial performance, emissions, and combustion stability requirements. Once the tuning is complete, the operator can select the load desired, and has some latitude with respect to the operation of auxiliary equipment (e.g., cooling tower fans, NOx injection water), but is at the mercy of ambient conditions and machine health and cleanliness.

(a) Inlet Air Conditioning. Gas turbines are very sensitive to the conditions of the air at the inlet to the compressor. The largest dependence is to the dry bulb temperature of the entering air. Many facilities are equipped with inlet air cooling that, in many cases, can be controlled by operations. For large frame units, the unit capacity is inversely proportional to the temperature of the inlet air.

Limitations on cooling the air are based on icing concerns. For some aero-derivative gas turbines, the compressors operate at a much higher pressure ratio, and have multiple control limits such as compressor speed, discharge pressure, and discharge temperature. At high compressor inlet temperatures, the unit may be limited to turbine inlet temperatures; at low temperatures the engine controls limit the capacity to the compressor limits, such that any further decrease in inlet air temperature will actually reduce engine capacity. Knowledge of the unit being operated and how it responds to inlet air conditions is paramount to optimizing the unit.

(*b*) *Load Control Selected*. Most gas turbines are designed to operate at the best efficiency when at maximum or base load. Some gas turbines allow the operator to select a higher firing temperature, which is known as peak firing the gas turbine. While this mode of operation may provide additional capacity, the maintenance costs for operating at higher temperatures can be significant. When optimizing for economic objectives, the maintenance costs must be considered.

(*c*) *Compressor Water Wash Schedules*. In some cases, the water wash schedule for gas turbines may be predetermined by warranty issues or long-term major maintenance agreements. Where this is not the case, it may be beneficial to determine the water wash schedule from an economic perspective. Conducting a compressor water wash can incur significant costs, such as those associated with the chemicals used, the power and water consumed, the disposal of the wash materials, and in some cases, reduction in unit availability. When these costs are compared to the gains in compressor efficiency and the resulting savings on fuel costs or increased available capacity, the water wash schedule can be optimized to meet the goals of the facility.

2-5.2.2.2 Heat Recovery Steam Generator (HRSG) Optimization. HRSGs in combined cycle operation have very few controllable parameters. The hot gases entering the HRSG are a function of the gas turbine operation. The steam pressures are often controlled by the steam turbine, unless the steam turbine is placed in inlet pressure control (IPC) mode. The following paragraphs contain information on operating the HRSG and steam turbine in sliding pressure mode and other optimization techniques common to both combined cycle operation and fossil boiler operation.

2-5.2.2.2.1 Steam Pressures. Control of the steam pressures at the HRSG is normally accomplished by changing the position of the steam turbine control/governor valves. Changing the pressure levels in the various HRSG sections can effectively move the heat transfer from the high-pressure section of the HRSG to the lower pressure sections, or vice versa. Since most work produced in the steam turbine comes from the higher pressure steam, the optimal setup will be to maximize the heat transfer in the high pressure section, thus maximizing the high-pressure steam production. This is best accomplished by operating the steam turbine in sliding pressure mode, allowing it to control the steam pressures.

2-5.2.2.2.2 Drum Level Control. Many of the combined cycle facilities are equipped with three-element control on the boiler feedwater system. The three elements are the drum level, feedwater flow, and steam flow rate. When operating in three-element control, the feedwater flow to the drum can be controlled in a way that minimizes swings in pressure in the final steam flow to the steam turbine. By minimizing pressure swings, the unit is able to operate more efficiently with less wear and tear on the mechanical and control systems.

2-5.2.2.2.3 Steam Attemperation. The use of attemperation flows should be avoided. The system was designed to operate at design steam temperatures with little or no spray flow; therefore, any additional spray flow causes a heat rate penalty.

2-5.2.2.2.4 Duct Burner Operation. Duct burners are used in fired steam plants for NOx reduction. The same type devices are used in combined cycle facilities to increase steam generation and steam turbine output. Those optimizing this process should be aware of the costs involved. The incremental cost of firing the additional fire relative to the NOx reduction or production increase should be understood.

2-5.2.2.3 Steam Turbine Optimization

(a) Inlet Pressure Control. Inlet pressure control has been discussed in the previous paragraphs and will be discussed again in the following paragraphs.

(b) Back Pressure Control. Back pressure on the steam turbine should be optimized to maximize the output and minimize the heat rate for the steam turbine. During the design phase, the cooling system and condenser selected for the LP exhaust steam is sized to provide optimal back pressure at loads just below the design load point. As loads are increased from this optimal point, the heat load to the condenser increases, and the back pressure will also increase. As loads are decreased from the optimal point, the heat load also decreases, resulting in a lower back pressure and higher exhaust velocity. The low back pressure can eventually cause a choked flow condition in the exhaust of the lowest pressure steam turbine. This overcooling or subcooling actually increases the losses from the unit. This negative effect can be avoided by adjusting the circulating water flow through the condenser, the cooling tower fan operation, and/or the number of cells in operation at the cooling tower (if applicable). A review of the steam turbine correction curve for output versus back pressure will identify the load conditions at which choked flow may occur. Additional concerns related to the condenser are discussed in para. 2-5.2.4.5.

2-5.2.3 Boiler Cycle Equipment Evaluation

2-5.2.3.1 Combustion Parameters. Almost any operating actions that can be taken to improve combustion efficiency will have a net positive effect on unit performance. Exceptions include those actions that require more energy than the energy savings they create. An example includes raising combustion air temperature to improve flame quality and efficiency, but requiring more net energy to raise air temperature than the additional energy derived from combustion improvement. Exceptions also include those types of actions that improve combustion efficiency but create greater losses elsewhere in the cycle. An example is increasing excess air in order to improve combustion and raise steam temperatures that results in net reduction in boiler efficiency and an increase in fan power.

The general approach to optimizing combustion parameters involves combustion efficiency and the placement and configuration of the flames themselves. These conditions are highly dependent on the fuel type or types being burned, the furnace design, and many other factors. Some of the combustion parameters that can be evaluated empirically for optimization are listed below.

(a) Boiler Excess Air Levels. The lowest levels achievable that create no adverse combustion, safety, or environmental conditions are generally sought. Insufficient excess air may result in incomplete combustion and an increase in the associated losses from increases in unburned carbon in the ash from coal fired plants. Excess air has a significant effect on NOx emissions from a boiler. Increased air in-leakage in the convection passes of a boiler affects the validity of the exit gas O_2 measurements, subsequently impacting the combustion process in the furnace because of erroneous excess air levels in the furnace.

(b) Variations of Primary, Secondary, and Tertiary Air. Evaluation is conducted as a means of finding the optimal combination of air flows, temperatures, and points of admission to flame area.

(c) Burner Positions. At reduced load regions, various combinations of burners in and out of service are evaluated to produce the best combustion and best steam temperature results. By design, many units are extremely responsive to the combination of which burners are in and out of service and to how they are set up.

(d) Burner Set-Ups. Set-up details such as register positions, damper settings, burner tilts, tip and sprayer plate sizes for oil burners, and other variables are evaluated for best combustion results. Some of these factors are important for burners out of service as well as for burners actually operating. An evaluation as to which burners or burner rows should be utilized under specific operating or load conditions is recommended.

(e) Fuel Conditions for Oil. Temperature, pressure, and atomizing steam flow (if used) are evaluated for optimal results under various loads, numbers of burners in service, air flows, and other firing conditions

(f) Fuel Conditions for Coal. Coal fineness is evaluated for best combustion, lowest excess air requirements, cleanest stack conditions, and minimum carryover of unburned carbon into ash. The best net efficiency will generally be achieved at the finest grind size of coal particles that can be produced. However, this is a complicated function of equipment, fuel, and operating variables and needs to be verified on a unit-specific basis to ensure that optimal grind size is being produced.

(g) Draft and Fan Conditions. Optimal settings of furnace draft, windbox-to-furnace differential, flue gas recirculation fan, and other air flow and gas flow controls are evaluated for best net results

(h) Pulverizer Performance. Mill outlet fuel-air temperature, consistent with safe operating practice as specified by the manufacturer, is evaluated for maximum safe temperature matched to the moisture and volatility of the coal being fired.

Operation at a mill outlet temperature below design in a manner that calls for reduced hot primary airflow subsequently reduces the airflow through the air preheater. This in turn increases the air heater gas outlet temperature. The effects on performance and heat rate due to elevated air heater gas outlet temperature are to decrease boiler efficiency and increase the unit heat rate.

In evaluating these and perhaps other combustion parameters, users may find it necessary to assess some of the variables in groups instead of individually. Optimal fuel oil temperatures, for instance, may be dependent on the burner tip sizes installed. Optimal coal fineness may vary with the level of excess air in use. Minimum excess air achievable at reduced loads may vary with the final arrangement of burners in and out of service, and with the register settings on all of those burners. There are many factors to evaluate that affect combustion, and these factors are highly interactive. It takes time and effort to assess them fully. However, working with these key combustion variables in a carefully planned and executed empirical process is almost certain to yield significant performance advantages to the user.

2-5.2.3.2 Main and Reheat Steam Temperatures. It is generally advantageous to maximize steam temperatures up to their design levels. However, certain factors and conditions indicate that reduced temperatures are better, or necessary, at times. High reliability may call for slightly reduced temperatures below design point if experience has shown a high tube failure rate at full temperature. Age of materials, the number of cycles the unit has experienced, or metallurgical analysis may also suggest that lower temperatures be carried. In a practical sense, many units are simply unable to carry full load design temperatures at reduced load levels for a number of reasons, assuming they are designed for full load design temperatures at reduced loads. All of these factors indicate that design steam temperatures are not always attained over the unit's load range; therefore, optimization is needed to identify best values throughout the load range.

There are a large number of operating factors that either directly or indirectly affect the temperatures produced. While these factors will vary between units, the most influential are listed below, along with brief comments on their roles in empirical optimization.

(a) Unit Load Level. Particularly in lower load regions, design temperatures may not be achievable. If load is dispatched economically, this is not a function under direct operator control, and temperature reduction, if it occurs on the unit, may be unavoidable. However, the amount of reduction may be controllable and subject to optimization through other means. If it is possible to improve upon steam temperature patterns at reduced loads, this may positively affect the incremental cost curve for the unit and increase operation as well as improve efficiency. Note that since boilers are typically designed such that steam temperatures are lower at reduced loads, the manufacturer's instructions should be referenced for curves of design steam temperatures versus load. These curves should be followed over the range of load points.

(*b*) *Direct Temperature Controls.* Desuperheating water systems, gas flow control dampers, sootblowing, and firing controls may all be used to maximize steam temperatures produced. They may also be monitored to ensure that they do not inadvertently decrease temperatures, through improper use at the wrong times. Unit heat rate is increased by the use of superheat and reheat desuperheater spray water for steam temperature control, although the heat rate penalty from SH sprays is considerably less than the effect from RH sprays.

(c) Bias Control Between Main and Reheat Steam. In some units there is ability to bias between main and reheat steam, for the purpose of balancing temperatures. This control creates an optimizing opportunity through empirical testing to identify the best combination of temperatures available at any given load point. Depending on the unit and circumstances, the optimal combination may be the highest temperatures in both main and reheat steams, or optimum may occur through favoring one steam temperature at the expense of the other.

(d) Boiler Excess Air. In many units there is a direct relationship between increases in excess air and increases in steam temperatures. The air increase by itself is less efficient from a boiler perspective, though the resultant temperature increase by itself is more efficient from a turbine cycle perspective. This relationship creates a performance tradeoff with the opportunity for identifying the optimal combinations through empirical testing. Note that NOx emissions also need to be considered here; if the excess air is higher, NOx will increase.

(e) Gas Flow Controls. Gas flow controls such as gas recirculation systems and gas flow dampers influence steam temperatures considerably, and can be optimized through empirical evaluation.

(f) Burner Positions and Register/Damper Settings. These have a large effect on steam temperatures in many units, and can be evaluated to determine which burner arrangements will produce the optimal temperature results. Particularly at reduced loads, the selection of which burners are in and out of service and the register and damper settings for burners both in and out of service may have a pronounced effect on steam temperatures produced.

(g) Changes in Fuel Type. Changes such as from coal to oil or gaseous fuels, or changes in fuel quality, have significant effects on the ability to attain design steam temperatures. Changing the coal classification has a major effect on boiler performance, such as when changing from a low moisture coal to a higher moisture coal. Ash properties affect heat transfer section cleanliness and heat absorption, which may reduce the final steam temperatures.

(h) Changes in the Boiler Boundary Conditions. Degraded turbine performance or replacement of turbine components will change the feedwater and reheater conditions entering the boiler, and boiler performance may need to change to account for these off-design conditions.

It will be found on many units that steam temperatures and combustion parameters are so closely interlinked that they cannot be separated in optimization work. Such interaction calls for empirical evaluation of the multiple parameters together. This is not necessarily a problem, but it requires close attention during data analysis to sort out the multiple cause-and-effect relationships that occur in the empirical testing. For instance, controlled changes in excess air levels may produce changes in both the main and reheat steam temperatures, and probably in other parameters as well. The evaluation will need to recognize that any net effects on total unit performance will be a result of all the changes and not simply a result of the excess air changes alone.

2-5.2.3.3 Air Heaters. Normally, optimal unit performance may be achieved in part through operating air heaters to produce maximal heat transfer while maintaining exit gas temperatures above acid dewpoint. Air heaters and their associated temperature control equipment are very influential on unit performance. As the last heat recovery component on most units, they provide the last opportunity to retain and recycle valuable energy. In addition to heat recovery, they influence combustion efficiency, help protect downstream equipment (ductwork, precipitator, stack liner) from corrosion, and affect the particulate removal efficiency of cold side electrostatic precipitators. Therefore, there are a number of reasons optimal performance of this equipment is needed. Some of the empirical considerations to its optimization are listed below.

(a) Maximizing Air Heater Efficiency. This is one of the more valuable actions that can be taken. It is done through monitoring air heater performance and making use of air heater sootblowing and washing as indicated by the performance monitoring results. In addition to temperature-based indications such as gas side efficiency or *X*-ratio, pressure differentials on the air and gas sides referenced against specific air or gas flows are also valuable indicators of air heater cleanliness. Sootblowing or washing to correct fouling and differential pressure problems will not only improve heat transfer, it will also reduce fan power consumption for a dual improving effect on unit heat rate. If performance deteriorates for reasons of seal wear, high internal leakage, element degradation, or mechanical failures of any sort, maintenance action should be planned and taken. None of these measures involve empirical evaluation in the sense of other optimization work; however, they are all done in order to maximize air heater heat transfer.

(b) Air Heater Control Systems. Some units may be equipped with control systems that allow temperatures of the gas and air streams to be regulated. At least two types of these systems exist: preheating coil arrangements that are supplied with steam or hot water from other sources, and bypass duct and damper arrangements that permit some of the air stream to bypass the air heater elements.

Preheating coils are very useful for introducing heat into cold boilers, for raising the temperature of combustion air to improve combustion, and for regulating exit gas temperatures for stack discharge reasons or for corrosion control. However, these systems use large amounts of energy from their heat sources in the process, and may result in net performance losses under some conditions. Empirical evaluation can reveal the break points at which use of the preheating systems becomes uneconomical. Air preheat may be used to control the air heater gas outlet temperature, since the amount of preheating reduces the total heat transfer in the air heater and therefore raises the air heater gas outlet temperature. This is a control parameter if the unit is burning high sulfur fuels to reduce or prevent air heater cold-end dew point corrosion.

Bypass systems are not as flexible in their use as preheating arrangements, but they have certain operational and maintenance advantages over the preheating coils. Bypass systems are simpler, involve less installed equipment, and may require less maintenance than preheating systems. However, they cannot introduce heat into a cold boiler or into a cold air stream during very low ambient temperatures. In general, bypass systems are of value in allowing air heaters to be used to maximum advantage in reclaiming heat, utilizing the bypass only at those times when the exiting gases are too low for reasons of stack discharge or corrosion control. Empirical optimization of bypass systems basically involves minimizing their use except during those times and conditions when they are needed for the reasons cited above.

Both systems of exit gas temperature control, when used to raise exit gas temperatures, are inherently inefficient. Whether either or both systems are installed on user’s equipment, they need to be carefully evaluated through modeling or empirical testing to determine optimal operating conditions.

2-5.2.3.4 Sootblowers. Sootblower operation throughout the boiler can be optimized through frequency of use and the selective use of blowers in specific boiler sections. Sootblower operation is dependent upon many factors, including boiler type and design; sootblower type, design, and energy consumption; fuel type and characteristics; ash com-position and characteristics; firing system type, design, firing rate, and system maintenance; steam temperature control requirements; current operating load and conditions; draft losses due to fouling; reduced boiler capacity due to fouling; exit gas temperature limits; reduction in boiler efficiency; increase in unit heat rate; minimization of tube erosion; and other factors.

Sootblowing optimization has several considerations, including energy used, effects on steam temperatures, environmental emissions, maintenance of equipment, and other factors, balanced against its advantages, such as improved availability and efficiency.

Sootblowing optimization may be approached through observation of its impacts on boiler efficiency; tube metal, steam, and exit gas temperatures; and reduction of fouling or pluggage in specific boiler sections. The sequence of sootblower use, frequency of use, and use based on specific measured or observed conditions will usually have sizeable impacts on sootblowing effectiveness. Where the benefits of sootblowing are measurable or observable and where their value exceeds the costs of negative factors listed above, sootblower use is being optimized. However, where the measurable or observable benefits are indiscernible, their use should be minimized at that time and under those operating conditions.

The optimization of sootblowing needs to be a dynamic process, since it is dependent on the interaction of many variables that themselves may be almost continually changing. However, since the potential benefits and certain detriments are so significant, this is an aspect of boiler operation that warrants careful and ongoing attention by plant personnel. Sootblowing optimization may be better done with software programs and turnkey systems, considering the complication and interaction, as discussed in para. 2-3.8.8.4.

2-5.2.4 Turbine Cycle Equipment Evaluation

2-5.2.4.1 Fixed Versus Sliding Steam Pressure. There are tremendous opportunities on many units for achieving efficiency and other technical benefits through use of variable steam pressures. For units operating at or near their design full load point, there is generally no choice but to carry main steam pressure at the fixed, design level. However, as operation is required at lower load regions, the option exists for many units to carry lower pressures and to potentially obtain significant benefits in the process. Note that sliding pressure is more often a consideration for subcritical, drum-type units. Supercritical, once-through units will less often have the option of operation below design pressure. The exceptions to this will be a unit originally designed with special boiler and turbine circuitry to accommodate such operation, or one where provision for this has been made through redesign and retrofit.

The concept of sliding pressure optimization is briefly explained here. Reduced main steam pressure inherently degrades turbine cycle heat rate. However, if a unit operates at reduced loads with significantly reduced steam temperatures, and its boiler feed pumps are of the variable speed type, then the savings of reduced pump power and improved steam temperatures from reduced steam pressure may be greater than the thermodynamic losses of the pressure reduction. The ideal pressure condition at reduced load is that which allows most of the turbine control valves to be wide open, creating minimum throttling loss, while at the same time being able to produce and control the desired electric output. Optimum steam pressure relative to unit load level will be unique for each unit and may vary over time and with different operating conditions.

Main steam pressures can be empirically evaluated at different levels to identify the optimal pressure at any given load region. There will be practical operating limits to how low a variable pressure should be attempted, typically involving ability to control boiler feed pumps, feedwater flow, and/or drum level. Minimum pressure may also be set by the ability to deliver sufficient steam flow to the turbine to maintain desired electric output. Additionally, there may be some reduction in ability to respond to rapid load changes if pressure is at low levels. In spite of these considerations, it may be feasible to operate in some circumstances with main steam pressure as low as 30% or less of rated pressure, with potentially very large performance advantages resulting. Therefore, there is considerable potential for empirical exploration of sliding main steam pressure. It is not limited to very small incremental adjustments. Note that the amount of pressure reduction that can or should be considered is dependent upon unit design and operating and mechanical conditions. It may or may not be feasible for the user's particular unit.

Other considerations include circulation and heat transfer patterns in certain sections of the boiler. Observation of temperature monitoring devices throughout the boiler may alert operators to any unusual or high heat conditions. However, originally installed temperature instruments may not be sufficient to give total recognition of overheating conditions that could be induced by sliding pressure.

Sliding pressure operation represents the potential for significant performance improvements, but carries with it certain operational and mechanical risks. These need to be carefully evaluated before pressure variation is attempted, and should be monitored during periods of reduced pressure operation. Consultation with boiler, turbine, and control system manufacturers is recommended for information specific to the user's equipment.

Attention should also be given to the possibility of boiling taking place in the economizer, which could potentially occur under conditions of low flow, low pressure, or during unit start-up. Economizer boiling, while not directly detectable by normal measures, can be evaluated by measuring economizer outlet temperature, comparing it to the calculated saturation temperature at the economizer pressure, and maintaining a reasonable differential of the saturation temperature above the economizer outlet temperature. Localized boiling could also occur in certain sections or assemblies of an economizer, but in such low quantities that it would not affect the measured economizer outlet temperature to a significant degree.

2-5.2.4.2 Control Valve Sequences. Taken as a partner with steam pressure considerations, certain valve sequences can produce significant efficiency improvements. These involve valve settings in sequential or partial arc versus in unison or full arc control modes. Generally, the choice of how to arrange control valve sequences is not within operator control, and at times may not be a matter of choice at all. However, there are potential opportunities for examining and modifying valve sequences, and the techniques of empirical evaluation are very useful for this work. In those circumstances where some degree of latitude exists in valve sequencing, given the appropriate attention to design and reliability considerations, there may be opportunity for optimization gains through the best setting of these sequences.

2-5.2.4.3 Valve Point Operation. Valve point operation represents potential economic value for single units, as well as for large, multi-unit systems. This is a concept involving the external dispatch considerations of units as well as the operational procedures of units themselves. Valve point operation entails operating units at or very near the points at which some or all turbine control valves are fully open, and at which no valves are in their throttling ranges. This mode of operation reduces throttling losses, but also somewhat restricts the operation of units by requiring units so dispatched to be loaded in fairly wide increments.

Relating valve point operation to control valve sequences, a full-arc unit dispatched at valve point would theoretically have only one operating point. A partial arc unit with eight valves, in which the first four operate in unison and the last four operate in sequence, would have five operating points if dispatched at valve point. The unit, if dispatched by normal means (not at valve point), could conceivably have an infinite number of operating points.

There are optimization considerations to valve point operation, but they are much broader than those considerations that are internal to the bounds of a single unit. Valve point operation involves optimization elements of multi-unit systems, with clearly increased complexity of evaluation. However, it is an aspect of operational optimization, and is included here for user consideration.

2-5.2.4.4 Feedwater Heaters. Feedwater heaters represent opportunity for significant economic gains through improved operation. Feedwater heaters are subject to optimization in a number of ways. Their controls can be set up to produce maximum heat transfer, regulate levels at the optimal points, dump high level drainage at the right times and conditions, and bypass or trip the heaters when necessary to protect equipment. All of these controls are vital to the safe and reliable operation of the unit, as well as to the net efficiency of the unit. Some feedwater heater controls can be tuned through an empirical approach within a narrow range of variation, but generally will produce the best results when set at or near design points.

Note that feedwater heater controls may be sensitive to unit load in their ability to maintain desired level setpoints. They may also be load sensitive in terms of which levels produce the optimal performance results. While the state of control technology may limit the equipment's ability to produce load-sensitive adjustments, this sensitivity should be considered when making adjustments to feedwater heater controls.

For shell and tube feedwater heaters, the optimal condensate level setpoint may be established in the following manner. With drainage controls in manual mode, and with unit load and feedwater conditions as stable as possible, record the drain cooler approach temperature and terminal temperature difference while dropping condensate level in the heater in step increments of approximately 1 in. When level reaches the point of blowing through, meaning that extraction steam is passing through the heater without condensing, the drain cooler approach temperature will dramatically climb, producing a "knee" in the curve of level versus DCA. Level should then be raised to a point at least 2 in. above that where the knee break occurred to provide an operating safety margin. For vertical feedwater heaters, a slightly larger margin may be necessary. If level is set too close to the knee break, the heater may be unstable during transients or reduced loads, and damage may be incurred to both the subject heater as well as the heater receiving its drainage. If level is set too far above the knee break, optimal heater efficiency will not be achieved. In the extreme case of level being set far too high in the heater, the damaging possibilities of heater flooding and turbine water induction may be created. Thus, there is opportunity to empirically evaluate and set the optimal condensate levels in individual feedwater heaters, but it needs to be approached with knowledge and care.

Normally the optimal performance from a unit's feedwater heaters will be achieved with all heaters in service, drainage levels established and stable in all heaters, drainage cascading downward from upper to lower heaters, and no high level or start-up drain activity. Heater vents will be open to release noncondensibles, and bypass valves around heaters or groups of heaters will be fully closed. To the extent that operators are able to maintain these conditions, operation of the feedwater heaters will be optimized.

Under some conditions, a group of feedwater heaters may be out of service while the unit continues to operate. This may restrict unit operation through limitations of the turbine, boiler, or both, and will adversely affect unit performance. If a parallel group of heaters remains in service, it may be feasible to utilize the operating heaters for full unit feedwater flow while the unit is at reduced loads, provided that neither flow nor heat transfer design limits are exceeded on those heaters. This may be an infrequent application for most units, but it does offer the potential at times for performance improvements under unfavorable feedwater heater conditions.

2-5.2.4.5 Condensers. The condenser represents the major single point of unavoidable energy loss from the cycle, as well as one of the potentially largest single areas for unit performance improvements. In general, performance improvement for the unit will result from improved heat transfer in the condenser, and from lower back pressure (higher vacuum) being achieved at any given load point.

Exceptions to these rules include cases where more energy is consumed to achieve condenser improvement than is gained as a result of that improvement. This may be empirically evaluated, as described below. Exception also exists to the generalization of "better vacuum, better performance." In many units, there is a point at very high vacuum levels beyond which further vacuum increases result in slight heat rate deterioration. This point can be seen in turbine thermal kit data, but can also be evaluated through computer models and could possibly be detected empirically, though with some difficulty. Means through which condenser operation may be optimized include the following:

(a) Tube Surface and Tube Sheet Cleanliness. Attention to condenser performance will indicate when that performance is deteriorating. Frequently, this will be a result of fouling of the waterside tube surfaces through silt accumulation, debris, or biological growth. Accumulation of debris from the circulating water system may also frequently occur on the tube sheets. Most of these materials, with the exception of biological growth, may be removed either by entering the condenser and doing normal tube and tube sheet cleaning or by backwashing the condenser on-line, provided that the circulating water system is designed with this capability. Growth inside tubes is normally controlled with biocides. For once-through units, the use of biocides must comply with environmental regulations. The expense for continuous chemical feed should be understood and evaluated. When materials accumulate throughout the tube surfaces, it is necessary at times to do a more thorough cleaning with brushing and/or flushing methods. Units with on-line condenser tube cleaning systems may also have these systems optimized.

All such maintenance of condenser cleanliness is needed to ensure maximum heat transfer, as well as to protect reliability of the condenser itself. Routine cleanliness maintenance can be optimized rather than maximized, by performing the work at the most economic times and by doing it when the performance improvement to be gained will exceed in value the costs of doing the work.

(*b*) *Minimized Air In-Leakage*. In many units, the presence of air in the condenser may be the single largest detractor from condenser performance. This conceivably could mean that air in-leakage at times could be the largest single cause of performance loss on the entire unit. In addition to reduced heat transfer through air blanketing of the steam sides of tube surfaces, air also results in corrosion throughout the cycle, requires expensive chemicals to be neutralized, and creates other adverse economic and technical effects. Air blanketing may also contribute to condensate subcooling on some units, further detracting from unit net performance.

As a general rule, the optimum approach to air in-leakage is to locate and minimize it. Measurement equipment is installed on many units to indicate the amount of air in-leakage, or more appropriately, the amount of air removal. Even without such instrumentation, it is possible to detect condenser performance degradation and to identify some or all of it as an increase in air in-leakage. When leakage has increased, or if it has existed for some time at excessive levels, a search for the leakage sources is needed. There are many methods for locating air in-leakage, including aural (manual listening), sonic detection, foam methods, tracer gases, hydrostatic testing, and others. Back-tracing of steps is helpful if leakage has taken a steep increase during a recent operating procedure. Some leaks may also be traced through examining the conditions under which they occur, such as during a crossover of certain unit equipment from positive to negative pressure. Such equipment may include low-pressure feedwater heaters, shaft seals, expansion joints, and various drain and vent connections. Some of these detection methods are used in service, while others require the unit to be out of service. Regardless of the methods used, it is advantageous to locate the sources of air in-leakage, and to correct them as soon as possible.

Of value in locating condenser air in-leakage is the development of potential leak location checklists and procedures to be used when leakage is at unacceptable levels. The location checklist eliminates the random approach and ensures that at least most of the potential areas will be checked.

Optimization of air in-leakage would imply that if leakage rates are at very low levels, it may be uneconomical to attempt to locate and correct the remaining problems. Quantifying what are acceptable levels of air in-leakage is a matter of management judgment, but as a general position it may be safe to consider that the lower the leakage, the better.

(*c*) *Optimized Air Removal*. Condenser air removal equipment is of many varieties, with various means of optimizing its use. Usually there will be two or more air removal devices per unit, permitting one or more to be removed from service with the unit in operation for maintenance, operating, or economic reasons. As a general empirical approach, in cases where air in-leakage is so low that a removal device can be shut down without any impairment of condenser performance, the optimal action is to operate without the device. On the other hand, where condenser performance is improved through operating additional removal devices, it is normally optimal to use as many as are available. The exception to this is when the economic gain of condenser improvement is less than the energy cost of operating the additional devices.

Operation of air removal equipment can be optimized in some cases to permit maximum removal capacity to be achieved. With mechanical types of vacuum pumps, such optimization is generally limited to maintaining proper operation of the sealing systems, minimizing any restrictions on the pump discharge side, and requesting maintenance assistance when pump capacity appears to diminish. Liquid ring type pumps have considerably more options for optimization, involving the pressure and temperature of seal systems, conditions of pump coolers, and flows and levels maintained by the seal systems. Air ejector systems also have options for optimization, involving conditions of the motive fluid supplied to the ejectors. In all cases, regardless of the type of equipment, maximum capacity will be aided by minimizing pressure losses or restrictions in either the suction or discharge sides of the equipment.

For units with oxygenated water treatment containing carbon steel components in contact with the steam or water, all feedwater heater vents should be kept closed to avoid flow-accelerated corrosion. These vents may be opened periodically for a short duration. For those feedwater heaters with air in-leakage problems operating near or below atmospheric pressure, when the vent is closed, the heat transfer performance deteriorates quickly.

(d) Optimized Circulating Water Flow. Circulating water flow can be optimized through the number of pumps operated at any given time. During periods of reduced load and low circulating water temperatures, there may be the potential to operate with fewer than all pumps without detracting from condenser performance. This operating mode can be evaluated empirically, as well as through the use of computer models

or through manual calculations. Other considerations of shutting down circulating water pumps, besides condenser performance, include pump motor reliability, reliability of the condenser and the unit itself, vibration characteristics of condenser tubes, and various tube corrosion and other factors. Many of these factors are of concern in arrangements where only one circulating water pump can supply water to a given waterbox. Where waterboxes can be supplied from multiple sources, the main considerations of number of pumps to operate are unit efficiency and pump motor reliability.

Another area for flow optimization involves water levels in the condenser inlet waterboxes. The waterboxes are normally equipped with priming systems connected to the main air removal equipment in order to remove air that enters either through leakage or entrainment. To the extent that these priming systems are kept in good condition and waterbox levels are full at all times, this will help to optimize overall condenser performance. Having access to sight glasses showing the top three feet of the waterbox level may prove helpful in monitoring waterbox levels.

2-5.2.4.6 Cooling Towers. Cooling towers present a limited opportunity for efficiency improvement through operational action. Nonetheless, this opportunity should still be explored. Mechanical draft towers have a greater degree of operational flexibility, with options available on selective use of fans. To the extent that additional air flow reduces circulating water temperature, it is generally a cost-effective practice to implement. A limit on this would be when the cost of auxiliary power consumed by additional fans exceeds the value of performance improvement achieved through colder water. Another limit would occur on some units at the lower ranges of condenser pressure, where back pressure improvement would cause the turbine exhaust to reach the choke point. During warmer ambient periods, maximum air flows will generally produce the best net unit performance.

Natural draft towers have very little opportunity for operational actions to improve their performance. The extent of this may be to seek optimal circulating water flows associated with specific ambient conditions, and to operate with these flows during the appropriate times.

During cold weather operation, the cooling tower system may be further optimized through bypassing the tower fill in whole or in part and routing the bypassed flow directly to the tower basin in order to reduce pump power and condensate subcooling.

In areas with extreme conditions between summer and winter operation, it may be beneficial to adjust the pitch of the cooling tower fan blades from season to season in order to maximize the operation of the fans during hot weather, and to prevent over-amperage of the fans during cold weather when the air is much more dense. If a midpoint fan pitch is selected for year-round operation, the fans may not be providing enough air flow during hot, low-density air conditions, causing an observed reduction in cooling tower capacity and resulting in a higher steam turbine back pressure than necessary. Variable speed fans may also be used to optimize the cooling provided by the tower.

2-5.2.4.7 Sealing Steam Systems. Sealing steam systems are important to unit performance in several ways. They can be empirically optimized through evaluation of different steam supply pressures and different sources of the sealing steam. Sealing steam condensers or exhausters can also be empirically optimized through the number of fans or blowers operated, and through the level of vacuum maintained in the condenser or exhauster.

Two specific areas of caution are mentioned for the user's reference with regard to sealing steam system operations. Since both areas represent potential for significant equipment damage through misoperation and/or poor maintenance, they are emphasized here for the user's consideration.

(a) Turbine Water Induction. These systems are potential water induction points to the turbine, being sources of possible low temperature steam or of condensate being carried by the steam. Temperature indicators are important to assist in early recognition of impending water or low temperature steam. Likewise, trap and drain systems within the sealing steam system need to be operated and maintained properly in order to afford full protection. Additional information on the prevention of water induction can also be found in ASME

TDP-1, Recommended Practices for Prevention of Water Damage to Steam Turbines Used for Electric Power Generation.

(b) Water Infusion to Oil Systems. These sealing systems are sources of water entrance to turbine oil systems as well as to generator lubricating, sealing, and cooling systems. Close attention needs to be given to the possible increase of water in any of these systems as operating procedures are changed through any empirical evaluations.

2-5.2.4.8 Auxiliary and Process Steam Systems. These systems represent economic opportunities wherever energy levels or steam flows delivered are minimized or optimized, and wherever condensate or higher energy drain flows can be feasibly returned to the cycle. Almost all main turbine cycles have one or more forms of extraction steam systems in addition to the steam used for regenerative feedwater heating. Auxiliary steam systems typically supply steam for house heating; fuel, oil, and other process heating; turbine gland sealing; auxiliary turbine operation; and other purposes. Some units are designed for high volume steam delivery to internal or external process uses. In many cases the steam extracted from the main turbine cycle does little or no direct work after extraction, but rather supplies energy to some use that may or may not benefit unit performance.

Optimization of auxiliary and process steam systems is limited, but possible to some degree in almost all cases. The general approach is to use extraction steam at the lowest conditions of pressure possible that will still satisfy the intended needs. Designs of some of these systems permit extraction from more than one point in the turbine cycle. Testing with steam taken from the various extraction points will reveal the optimal source under specific unit loading and process conditions. Further optimization is also possible through determining ways to minimize the amount of steam extracted. This may be an obvious approach, but it is indicated here because of potentially large energy losses through steam wastage.

2-5.2.4.9 Drain, Recirculation, and Bypass Systems. Sizeable efficiency advantages exist in these systems through minimizing or stopping their flows except when specifically needed for operation or equipment protection. Throughout a main turbine cycle there are many drains and bypasses intended for water removal, start-up, pre-warming, equipment protection, and other purposes. These range in magnitude from very small, orifice-equipped continuous drains and small tap-equipped drain lines, to the larger high level and start-up drains to the condenser from feedwater heaters, to high flow recirculating lines of pumps and condensate systems, to the very large partial or full bypass lines around the turbine or individual turbine or boiler sections in some units. These drain, recirculating, and bypass systems serve very important purposes. However, through leakage and excessive use, they may contribute to major cycle energy losses. It is conceivable that in some cases, cycle losses through drain, recirculating, and bypass systems may represent one of the largest, if not the largest, sources of controllable unit energy loss.

Optimization of these systems is best done through a three-pronged approach, consisting of effective monitoring for unintended flows, correct operating actions of the system components, and well-directed maintenance when needed to rectify problems. These are described below.

(a) Monitoring of drain, recirculating, and bypass systems for unintended flows can be done by several means. These include remotely — read thermocouples or RTDs, local temperature devices, flow sensors, open/close and position indicators on control valves in the systems, sonic detection, and manual, hands-on surveillance. Monitoring these systems for unintended flows alone does not constitute correction. The information needs to be interpreted and acted upon either through operational or maintenance attention.

(b) Correct operating actions for the systems are specific for each individual unit and are dependent upon equipment conditions and operating mode at any given time. However, it needs to be recognized that any drain, recirculating, or bypass systems will generally create an adverse effect on unit performance if used in excess of their intended purposes. Operators can minimize these losses through ensuring that the systems are open or properly controlling only when needed for equipment protection or unit controlling purposes. At other times, operators need to ensure that the systems are securely closed and absent of unintended flows through the monitoring methods available to them.

(c) Maintenance attention to system problems is needed to correct whatever conditions are beyond operator control. This may be as simple as adjustments or tuning of recirculating or feedwater heater controls, may involve repairs or replacements of system valves, or could even require system modifications in some instances. While these maintenance activities are not within the scope of operational optimization, they are included here because operators are generally the ones to detect system problems first, and therefore will need to initiate corrective actions as required.

2-5.2.5 Auxiliary Equipment Evaluation. Auxiliary equipment of a unit typically consumes a significant portion (several percent) of total unit energy. For electrical equipment such as pumps, fans, control systems, and lighting, internal usage of some of the unit's gross electric output is necessary, thereby reducing net power output. For steam-driven equipment such as turbine driven pumps and compressors, thermal energy is extracted from the cycle, indirectly reducing net power output. There are a number of opportunities for optimizing the operation of auxiliary equipment, accompanied by a number of important considerations. Some of these factors are described below.

2-5.2.5.1 Running Versus Not Running Auxiliaries. Choices for operating or not operating certain auxiliaries afford economic opportunities to minimize power consumption and to optimize equipment configurations. In many situations, operators may have options of which auxiliaries to operate, and in what quantity. For example, if a unit is equipped with three boiler feed pumps, each of 50% unit capacity rating, there are options in the lower load ranges of running one pump, and in the upper load ranges of running two pumps, with the third being an out of service spare. Similar options may exist for fans, condenser circulating water pumps, vacuum pumps, air compressors, and other major and minor auxiliary apparatus. In some cases, the use of variable speed or variable frequency drives can add additional flexibility and the ability to optimize auxiliary equipment.

The efficiency effect of which number of auxiliaries to operate is based on the shape of the pump, fan, or compressor curves. In many cases, it will be advantageous from an efficiency perspective to operate the minimum number of pumps, fans, or compressors that will be sufficient for the output needed. In other cases, it may be advantageous to operate more than are needed, having some or all operating at reduced capacity to meet the output needed. Important operational factors beyond efficiency need to be considered in determining the optimal numbers of auxiliaries, such as unit response requirements. This also needs to be recognized as a dynamic process that may vary as operating conditions or mechanical status of the equipment changes. In simple terms, there are usually both advantages and disadvantages in the selection of which auxiliaries to operate and in what quantities. Testing processes as well as calculation methods can be used to help identify the balance of advantages and disadvantages for those situations where the options exist. From the testing, and from considering other important factors, operating procedures can be developed to identify the optimal combinations of auxiliary equipment at any given time.

2-5.2.5.2 Net Economic Judgments. The economic balance in auxiliary operating choices generally lies between auxiliary component efficiency characteristics and considerations for any adverse impacts of not running those auxiliaries. The net economic value in operating or not operating auxiliaries is influenced by at least two significant factors, as described below.

(*a*) *Efficiency Curve*. The number of auxiliaries run, and which specific ones to operate, can be based on operating at or close to the most efficient range, as described by the equipment's efficiency curve. The conclusion that would be drawn here is that the minimum number of auxiliaries required to satisfy the process needs should be run. Equipment that may be optimized on this principle includes condensate and boiler feed pumps, boiler fans, ash system equipment, fuel equipment such as fuel oil pumps and coal pulverizers, as well as other equipment types. There are important considerations beyond simple economics, however. These are described in subparagraphs below.

(b) *Performance Consequences of Shutdown Auxiliaries.* These may be a determining factor, due to the economic effects of not running certain auxiliaries. The example is used of one of two circulating water pumps being shut down in an attempt to reduce auxiliary power and thereby improve performance. Under some conditions, such as low load or cool ambient conditions, condenser performance may not degrade from the partial loss of flow, while under other conditions, such as during hot, humid ambient conditions, performance would degrade significantly. The theoretical economic break point would occur where the net loss due to condenser performance degradation would just equal the net savings of removing the pump from service. Equipment that this optimization principle applies to includes circulating water pumps, vacuum pumps, cooling tower fans, and other types of equipment.

Both of these factors can be evaluated empirically through a careful process. First, a pre-evaluation of likely possibilities for successful removal of equipment from service is needed. The intent is to determine which auxiliaries offer the possibility of shutdown and under what general conditions, without regard to whether the net result will be a gain or a loss. Next, the procedures for operators to follow in shutting down any auxiliaries need to be developed, such that the necessary operating safeguards are in place and so that appropriate indicators of gains and losses will be observed. Next, the testing itself can be conducted and observations made. Finally, the information derived from multiple tests can be evaluated, and appropriate economic conclusions can be drawn.

2-5.2.5.3 Safety and Reliability Considerations. The question of operating or not operating specific auxiliary equipment should not be answered purely on the basis of economics. Safe operation of equipment needs to be a foremost concern at all times. It is possible that in some situations, operating without certain auxiliary equipment could jeopardize operating safety. This needs to be considered in advance of any empirical testing, as well as in advance of implementing any new operating procedures involving shutdown of auxiliaries. While no specific guidance is given here on evaluating the relative safety of various auxiliary combinations, it must be given proper attention by those engaged in the planning, evaluating, and operating processes.

2-5.2.5.4 Auxiliaries Cycling Considerations. The adverse consequences and risks of cycling auxiliary equipment must be fully recognized and carefully weighed in any decision to cycle such equipment.

2-5.3 Operational Optimization — General Methodology

The following information describes general approaches and methods for operational optimization. There are many different approaches, each with its own advantages and disadvantages relative to the other methods. This subsection identifies some of the fundamental requirements of operational optimization, describes the general operational approach to optimization, and explains in overview several of the different optimization methods.

2-5.3.1 Operational Approaches. A general approach to operational optimization is described here. Several fundamental issues are described below before any information is given on actual optimization techniques. This is for the purpose of building a solid foundation upon which users may build their optimization efforts. Issues for user consideration are discussed in paras. 2-5.3.1.1 through 2-5.3.1.6.

2-5.3.1.1 Operational Assessment. Before beginning any optimization work, it is wise to do a careful and objective assessment of the overall type and quality of operation that the subject equipment currently experiences. For example, is the equipment operated chiefly at a steady, full-load design point, or does it experience much transient operation? Is it cycled out of service frequently? Does it experience changes in fuel type? What levels of proficiency do its operators possess? Do they operate the equipment well and conscientiously? How old is the equipment? What state is it maintained in? What are its future prospects?

These questions are meant to be representative of the types of areas to examine in assessing the equipment's overall operation. The results of such an assessment are of much value in determining the most appropriate course for the optimization efforts to follow.

2-5.3.1.2 Operator Involvement. It will frequently be true that the operators of the subject equipment have a very solid base of knowledge, experience, and operating intuition upon which the optimization efforts may be built. It is most important that these personnel have significant involvement in the entire optimization process. When coupled with the active participation of others, including engineers, maintenance personnel, and plant management, it is likely that very effective results may be obtained from the optimization efforts. Conversely, if the active involvement of key and knowledgeable personnel is not applied in the process, it is likely that results will be limited at best.

2-5.3.1.3 Operating Standardization. The concept of standardization, or uniformly applied operating methods, is a fundamental issue of optimization. The optimization process arrives at the best types of operation to produce the desired results. The corollary is that once these operating methods have been determined, they need to be applied in normal operation with reasonable uniformity. This does not exclude operator judgment and active involvement, which is of paramount importance to the success of optimized operation. However, the reasonable uniformity or standardization of operating practices is needed to achieve the fullest benefits of optimization.

2-5.3.1.4 Optimal Operating Modes. This fundamental aspect of operating optimization involves the setting of operating values at those levels producing the best results. In optimization, this equates to the highest performance levels that can be cost-effectively and feasibly achieved, within the constraints of safety, reliability, and environmental soundness. Normally the objective is to maximize operating efficiency; however, there are also considerations of equipment preservation and reliability that may limit pursuit of maximum efficiency. An example would be the limiting of main steam temperatures to some value below design, if operation at the higher temperatures has resulted in boiler tube failures and expensive outages. Maximized efficiency would suggest operation at the higher temperatures, but the constraints of equipment safety and reliability present an offsetting factor, with a slightly reduced temperature level found to be optimal.

Optimal operating modes are expressed very specifically, typically describing load points at which these operating modes would change, and values of parameters such as excess oxygen, steam temperatures, condenser pressure, and others as functions of unit load and other variables.

2-5.3.1.5 Operator Controllable Parameters Programs. Of extremely high value in operational optimization work are computer-based programs that provide information to the operator on controllable parameter optimization. Typically these programs continually receive various unit data and perform calculations on equipment performance. The results are presented to the operators on displays that allow them to see the impact of various controllable parameters. In some cases, the program may make suggestions to the operators, such as increasing fan speeds on a cooling tower, or adjusting boiler operating pressures. The results of the program are often left as recommendations only, since the operator will be aware of external factors that may prevent them from operating the unit at the recommended values (such as equipment out of service, or market constraints). Most systems will also produce summary performance reports for operator and management use.

2-5.3.1.6 Management Information and Controls. Operational optimization typically generates a very large amount of information. The effective program must reduce this to a quantity and form suitable for assessment. This applies both to the developmental and application stages of optimization work.

2-5.3.2 Methods of Determining Optimal Operating Modes. There are several methods that can be used for determining the optimal means of operating equipment. Generally, a combination of methods will best serve the user's purposes. Four basic approaches are briefly described below.

2-5.3.2.1 Testing. This method involves the use of specialized types of operational testing of equipment. This testing is used to determine the actual effects of various operating techniques and values on performance of the overall unit. There are a number of advantages in this method. For instance, it tends to capture the net results of many interactive effects that occur with a set of specific operating conditions. It tends to reflect the real life effects of the operating techniques and values being studied. It also is geared toward operational changes that are within the operator's direct control or influence. The empirical testing method does have its drawbacks, including the complexity of some of the testing, the need for very carefully controlled conditions, the difficulty at times in interpreting results, and the costs of conducting the testing itself. Overall, the empirical testing approach may generally be the most fruitful of all methods in revealing the truly best combinations of operating techniques and values.

2-5.3.2.2 Design-Based Operation. Optimal operating modes may be based in part or in whole on design data provided by equipment manufacturers. Use of design data offers the reassurance that operation is being guided by the techniques and values that were intended by the original designers and manufacturers. Design data also provides certain safety limits, such as maximum or minimum operating values, that are intended to safeguard the equipment. It may be found that design-based operation is very sound as well as being capable of producing optimal or near-optimal results in those situations where normal operation is chiefly in the design region.

Design-based operation, however, has many limitations. Frequently there will be different design values for equipment components making up a unit. For example, a turbine manufacturer may specify a constant steam temperature pattern through the load range, but the boiler manufacturer may identify a sloping pattern as the norm. Equipment frequently may be required to operate far from its design regions, in areas where operation was never anticipated and where design values are simply not available. For instance, it is not uncommon for some units to operate at minimum loads that are 10% or less of full rated loads. It also often occurs that the operating values of certain parameters have to change over time. An illustration is the derating of steam temperatures in some cases far below their design values, due to poor operating experience with the boiler, turbine, or piping components. Another shortcoming of design-based operation is that the design values may not produce optimal operating results, which operational optimization attempts to achieve. Additionally, the fuel being burned may not be the original design fuel, which may affect performance and make some performance parameters no longer achievable.

2-5.3.2.3 Computer Modeling. Commercially available computer models and smaller scale computational programs are also useful in evaluating optimal operating modes. These tools allow the knowledgeable user to evaluate different combinations of hypothetical or actual operating conditions for their net effects on unit performance. Three examples of how such modeling can be used are given below.

(a) Excess Oxygen and Steam Temperature Modeling. Assume that the interactive effects of boiler excess oxygen and main and reheat steam temperatures have been accurately duplicated for a specific unit in a computer model. A number of iterative model runs can be conducted to predict the optimal combinations of the three conditions at various points throughout the unit's load range.

(b) Steam Pressure and Turbine Valve Point Modeling. Assume for the same unit that the interactive effects between main steam pressure and the turbine have been modeled. Iterative runs can be conducted to predict the optimal pressures at various points of unit load and turbine control valve position.

(c) Condenser Modeling. Assume again for the same unit that the condenser is fully incorporated into the model. Runs can be made to determine the effects of reduced circulating water flows on unit heat rate and electric output, for the purpose of identifying at what points a circulating water pump may be removed from service to save auxiliary power, without creating a net loss through heat rate increase and/or electric output decrease.

Such modeling techniques are very powerful. Drawbacks include their cost, the work required to duplicate the conditions of specific units or equipment, and the need for skilled and knowledgeable users to operate these models. Overall, they are of much value in operational optimization as well as in other aspects of performance analysis when used in conjunction with other methods.

2-5.3.2.4 Manual Engineering Methods. Manual calculation methods are also useful in some aspects of determining optimal operating modes. These include calculator-derived or manually derived heat balances, various energy calculations on equipment or systems, graphic analysis using nomograms and charts, and other methods of manual calculation and analysis. Graphic methods in particular can be helpful in checking data quality and in getting quick, approximate answers to analytical questions. Graphic and other manual methods also may help an engineer to visualize the process being analyzed.

Certain equipment types lend themselves readily to manual engineering methods, notably feedwater heaters, condensers, the turbine itself, and some aspects of boiler operation.

Manual calculations of complex cycle questions, however, tend to be very time consuming and difficult, and when compared against computer-based analysis, the manual methods can be prohibitive for very involved evaluations. However, manual methods are valuable as cross-checks and verifications of the computer methods, and help to establish credibility in the more sophisticated approaches. In spite of their drawbacks, manual engineering techniques have a role in any optimization efforts, and it may be of value to include them as a part of the overall methodology.

2-5.3.3 Empirical Approach to Operational Optimization. The empirical approach for this work, as stated above, is recommended as potentially being the most fruitful for most operating situations. It is not a stand-alone method, certainly needing the thorough consideration of all appropriate design information, and benefitting from some degree of manual and automated evaluations of unit conditions. However, the approach is primarily empirical, a hands-on method relying heavily on operational testing and on careful assessment of the information produced from these tests.

Certain cautions are expressed to users of this method. First, it requires a solid base of operating knowledge, experience, and judgment. This experience and judgment must be sought out and utilized, particularly if the persons guiding the process are relatively unfamiliar with operations in general or for a specific unit. Second, it is important to recognize the need for, and clearly establish, the operating limits and constraints on the process. These constraints involve such parameters as minimum boiler air flows and excess oxygen levels; minimum levels of superheat in main, reheat, and extraction steam systems; maximum allowable temperatures in boiler tube banks; and other similar conditions. These limits and constraints may be based on design data, operating experience, judgment, and common sense. They are intended to protect the equipment and people during the optimization process that may enter previously unexplored operating regions. They are thus important guides to set the general bounds of optimization testing and operation, and should be established before and reexamined throughout the process as it proceeds.

Following are the general processes of empirical optimization. These are not necessarily sequential, and are not given in sufficient detail to be considered total and absolute. Rather, they are given as general guidance to the overall empirical process.

2-5.3.3.1 Operational Testing and Analysis. The process is largely based on conducting specially designed operational tests and analyzing the information from these tests to determine optimal operating modes. In very simple and abbreviated terms, these tests involve making controlled changes in a single parameter or groups of parameters while the equipment or unit is under stable test conditions. The results of these controlled changes are then measured in order to establish cause-and-effect relationships. Iterative changes are continued for the parameter until its optimal operating levels and modes have been determined.

In concept, the process is no more complex than stated above. In practice, it is not complex, but requires time, patience, and perseverance; careful planning and execution; and reliance on operating knowledge, experience, and judgment. Guidance on conducting the process for various operating parameters is included in this subsection.

The effects of parameter changes may be gauged through a number of means. Resultant changes in unit gross or net electric output, or in unit heat rate, may provide clear indications of effects on total unit performance. There are many conditions, however, where these indicators may not be appropriate, and where substitutes may be needed. This may require relying on measures of performance or output of equipment components. Examples of this include observing condenser pressure or condenser effectiveness for test conditions affecting the condenser; tracking feedwater heater temperature rise for conditions affecting heater performance; monitoring boiler fuel flow or other fuel indicators for conditions influencing boiler efficiency; recording feedwater flow, steam flow, or turbine first stage pressure for tested parameters influencing turbine efficiency; and other similar means. These examples are by no means complete, and may not be appropriate in some situations. They are meant to illustrate the general process of measuring the effects of controlled changes in operational parameters.

It is not the measures themselves, but the cause-and-effect relationships that are important. Determining the proportional and directional effects with a sufficient degree of repeatability to establish confidence in the results is essential to the success of the approach.

Analysis of test information is of key importance in the process. Under the best of conditions, some analysis of the effects measured is needed to determine the location of the optimal operating point for the parameter being varied. Under more typical conditions, the user will encounter conflicting data, different cause-and-effect indications (at times even possibly being directionally different), and a combination of parameter changes occurring in addition to the one or the few being intentionally varied for the tests. Users may also encounter some degree of instability in the test conditions in some, many, or perhaps all of the tests run, in spite of the best efforts to regulate test conditions according to plan. All of these circumstances require careful analysis of information to draw valid conclusions as to optimal operating modes. This analytical process takes time, care, and attention to detail, but it is well worth the effort to pursue, particularly if large gains in operational performance are being sought.

2-5.3.3.2 Sequence of Equipment Evaluation. There are many operating parameters that could potentially be evaluated for optimization. These range in magnitude of impact from almost negligible (such as only a few Btu/kWh effect on heat rate) to major (conceivably hundreds of Btu/kWh effect for a single parameter). Selection of which equipment and operating parameters to optimize needs to be based on the user's objectives and the operating conditions at the start. However, the sequence in which to address the equipment and parameters in the optimization process is quite important. This is because work that is done on certain parameters may have a sizeable influence on other parameters. If a sequence is used that does not recognize these interactions, it is possible to render some optimization efforts invalid.

The criteria for determining optimization sequence is suggested to first handle those parameters that are likely to have the largest effects on other parameters. This will permit the greatest degree of control by operators over their optimization variables. It will also result in more effective optimization work, minimizing the need to repeat work already done that would almost certainly occur if highly interactive variables are addressed out of sequence.

There is not an exact, universal approach that may be used to establish best optimization sequence for all units. For any particular case, the optimization circumstances need to be examined by the user, including the types of interrelationships that may be present between the parameters in question. However, as a general guide to sequence of optimization work, the following order is suggested as being appropriate for many fossil steam units:

(*a*) combustion parameters (coal grind size, fuel oil viscosity and temperature, unburned fuel, excess oxygen, main and reheat steam temperatures, and boiler exit gas temperature)

(*b*) main steam pressure and turbine control valve operations

(*c*) condenser and cooling tower operations

(*d*) feedwater heater operations

(*e*) turbine operations

(*f*) auxiliary power levels and use of auxiliary equipment

2-5.3.3.3 Test Conditions. It is important to establish test conditions that are as stable as possible. The purpose of the testing is to create baseline conditions, cause controlled change, allow restabilization, and observe and measure the effect. If test conditions are unstable before or after the controlled change is made, it will be difficult, if not impossible, to evaluate the effects of the change. This does not imply that absolutely rigid, static test conditions have to be achieved for the process to be of value. Such conditions almost never occur in real operation. However, the more stability that can be achieved, generally the more repeatability of results and ease of interpretation can be expected.

In addition to having stable conditions during any one test, it is desirable to establish as repeatable a set of conditions as possible during subsequent repeats of that test. This is because differences in the base conditions, particularly if they are large differences, can create very conflicting appearances in the cause-and-effect data and cloud the interpretation. As with stability, this does not imply that exact repeatability of test conditions is necessary, but the closer that similar conditions can be reestablished for repeat tests, the more consistency can be expected in the test results.

2-5.3.3.4 Assurance of and Confidence in Results. The empirical testing process is used to arrive at information on operating modes and values that will be used in routine operation. It is therefore vital that a high confidence level be established in the results, with strong assurance that the results describe the true optimal state. In general, this confidence can be established best through proven repeatability of the observed cause-and-effect patterns in the testing. It is recommended that any given series of tests be conducted at least three times, at well-separated points in time. This repetition is done to verify that the effects of the controlled changes are directionally consistent, that their magnitudes are reasonably repeatable, and that the optimal operating value identified is approximately the same in each series.

2-5.3.3.5 Data Evaluation and Conversion to Standards. Results of all the testing require evaluation to assimilate them into practically useable operating modes and values. For example, assume that a large amount of testing of combustion parameters has been done in the low load regions of an oil-burning unit. Evaluation may indicate that the best net results occurred with a specific arrangement of burners in and out of service, with specific settings of air registers. The best boiler draft levels, settings of recirculating fans, and excess air amounts may have been determined. Optimal burner tip sizes, and fuel oil supply temperature and pressure may also have been identified through the testing. Only through careful evaluation of the data can these conclusions be drawn. However, once identified with a high confidence level, this information can be integrated into operating procedures for the most effective use of it.

Conversion of the empirical results to standards involves specifying the patterns to be followed in normal operation through the full load range for each parameter studied. Building on the example of combustion testing, the final set of standards would specify the optimal level of excess air at each load region from minimum to full. It would also identify the burner combinations to use and the conditions of all burner parameters (register settings, tip sizes, fuel temperature and pressure, etc.) at specific load regions, and would identify the points and conditions at which to change over to the next set of burner specifications. A complete set of standards for the unit may include more detail on the combustion parameters, and would include similar information on the other important operating parameters related or unrelated to the combustion items. All of this information, however, would be derived from the empirical testing process, careful evaluation, and consideration of other important factors such as design data and safety, reliability, and environmental aspects.

2-5.3.3.6 Use of Standards in Routine Operation. An important issue concerning use of standards involves the all-important element of operator judgment. Standards are meant to be general guides for the operation of equipment under normal conditions. They cannot be taken to be absolute operating methods and values that must be followed under any conditions. The need for operator understanding and judgment of the appropriate use of established standards at any given time should be reinforced as an important aspect of the standards concept.

Another significant matter involves the use of standards under reasonably steady-state conditions as opposed to during highly dynamic or transient operating periods. There are transient operating conditions where standards may provide a general guidance, but where it may be extremely difficult for operators to adhere to them while trying to operate through the transient. This also is a matter for operational management decision, calling for reliance on operator judgment, but it is an aspect that needs to be considered in using standards in normal operation.

Many other factors, major and minor, are involved in routine operational use of these concepts. The factors and issues cited above are specific ones to be considered in advance, but there are others as well. These are meant to indicate the general types of considerations in the use of standards, which users need to evaluate on a case-by-case basis.

2-5.4 Mechanical Optimization — General Methodology

The following information is given to acquaint the user with the general approaches to mechanical maintenance and modification optimization. The process described below assumes that a subject unit is being evaluated for overall mechanical optimization. However, aspects of this process may be applied to very specific problems or equipment components, without engaging in the total unit evaluation.

There are important interactive issues between operational and mechanical optimization. It is necessary for users to understand these issues for reasons of maximizing the effectiveness of their optimization efforts as well as for total economics. The integration of these interactive issues is described in this paragraph.

2-5.4.1 Assessment of General Mechanical Conditions. To fully describe the mechanical optimization concept, illustration is given of the overall mechanical evaluation process. As with operational optimization, it is wise to begin with an assessment of the unit's general mechanical condition. This is needed to assist in planning the general approach to mechanical optimization. The assessment is intended to identify the more significant mechanical problem areas that may benefit from repair and/or engineering attention. Assessing the unit's mechanical conditions may be approached as follows.

2-5.4.1.1 Efficiency. Regardless of the overall unit efficiency levels, it is valuable to conduct an efficiency assessment of selected equipment. This should be targeted toward the mechanical conditions of equipment, including at least the turbine, boiler, condenser, cooling tower if equipped, and feedwater heaters. Major auxiliary equipment may also be included in this evaluation if the user wishes to assess at this level of detail.

In an ongoing performance monitoring program, efficiency degradations would be revealed, creating the opportunity to analyze them and take corrective actions. Such monitoring-based information may not be available at the time of an initial overall efficiency assessment if a monitoring program has not been previously used. In this case, the initial assessment helps to establish baseline data, while the subsequent monitoring program will reveal changes that occur after the initial assessment.

2-5.4.1.2 Availability. Many times the mechanical problems of a unit affect its availability. Availability in this context refers to the unit's ability to reliably remain in service when needed, and to be able to produce full output. The assessment of availability conditions therefore seeks to identify those problems that have been or potentially could be contributing to unit outages or to reduced output.

Information sources for availability problems may be more sophisticated and better documented than those for efficiency detractors. The North American Electric Reliability Council (NERC) maintains a very extensive data base on causes and magnitudes of availability detractors. For units that are participants in this database, information is available either in-house or through NERC that details the units' specific availability problems in terms of outage causes. This information, when combined with other data obtained through in-house records and discussions, becomes valuable in identifying and prioritizing availability detractors on the unit.

2-5.4.1.3 Overall Maintenance Levels. In addition to assessing specific mechanical problems affecting efficiency and availability, it is advantageous to evaluate the overall maintenance levels of the subject unit or units. For example, is the equipment highly maintained and subject to possibly no changes or else only refinements to its maintenance practices? Or is it maintained at a very low level, with perhaps much opportunity for mechanical optimization? Other key questions deal with the drivers of maintenance practices and levels. For example, is the maintenance highly constrained or limited by finances? Is much preventive maintenance done, or are repairs driven largely by breakdowns? Is maintenance driven more by availability, efficiency, or other objectives? Is maintenance being targeted toward long-term unit service or does it anticipate a limited remaining service life?

These aspects of maintenance practices may seem general, but they are vital elements to be considered in designing the mechanical optimization approach. It is these factors that may need to be either modified or accommodated in the optimization program. However, they certainly need to be identified and acknowledged, since they represent the starting point from which maintenance and modification optimization efforts will proceed.

2-5.4.2 Optimization Cost, Performance Value, and Net Worth. All mechanical optimization is based on consideration of technical and economic factors in each specific case. At times, the technical considerations of a problem may outweigh the economics, and the appropriate action will be problem correction without direct economic justification. Other factors may also at times outweigh economically preferred actions. These factors may include safety, environmental impacts, and others, and they may dictate that a noneconomic course of action be taken.

In most cases, however, economic analysis of a mechanical problem can serve as a primary decision-making tool. Concepts of such analyses include optimization cost, performance value, and net worth. These are briefly described below, not at the level of detailed treatment of engineering economics, but sufficiently enough to introduce the general process to users.

2-5.4.2.1 Optimization Cost. For any repair or engineering solution that is being considered, it is necessary to identify costs as accurately and completely as possible. Significant factors in cost identification include all materials, labor, installation and subsequent maintenance requirements as appropriate, cash flow projections, cost of money, and expected life of the repair or engineering action being considered. It may also be appropriate to consider the costs of the current system or problem that is being corrected. All of the identified costs, both costs incurred and costs reduced, need to be converted to a consistent basis such as total present worth. Once in this form, the costs for each optimization option may be evaluated relative to the value of the expected performance change for each option.

2-5.4.2.2 Performance Value. The expected performance changes that would occur with each correction option and the probable economic value of these changes represent the other side of the optimization equation. For actions targeting the efficiency problems, expected changes may be expressed in terms of heat rate, operation or performance factor, fuel consumption, or another factor that can most readily be converted to economic value. For availability problems, expected changes may be expressed as reduced number or duration of outages, lesser number of load reductions, change in equivalent availability or forced outage rate, or another measure convertible to economic value. In addition to estimating the magnitude of performance changes, it is also necessary to project the longevity over which they are expected to be effective. This is a very important factor, since differences in longevity may often cause an option to emerge as the most favorable, which on the basis of otherwise apparent cost may not have been selected.

There are many different methods by which the economic value of performance changes can be evaluated. For multi-unit systems, production cost computer models are frequently used by the operating companies. If access to such models is available, or if special evaluation runs can be requested, they represent one of the more objective means of evaluating economic effects of performance changes. Where computer models are not available, other means must be used. Considerations for evaluating efficiency and availability changes are briefly described below.

(a) Efficiency Changes. Some of the factors that need to be considered to quantify the value of efficiency changes include average heat rate change expected in specific load ranges, average load expected to occur in each load range, number of hours expected to be spent in each load range over the time period being computed, fuel type or types being burned, and the fuel cost or costs anticipated.

Combining these factors will enable the user to estimate the direct fuel savings associated with the expected efficiency change. There are other economic effects of efficiency changes such as interconnection accounting effects and operational impacts on other units through incremental dispatching. These are complex variables that cannot be accurately recognized in a simplified value quantification. However, a simplified quantification of direct fuel savings may be supported with reasonable and realistic assumptions. It may often be approximated when more sophisticated, model-based calculations are not available.

(b) Availability Changes. The process to convert changes in availability is a quite different efficiency valuation. A simplified evaluation is shown.

Assume that a unit rated at 250 MW has a recurrent availability problem, correctable through an engineered course of action. The problem causes an average of four two-day unplanned outages through the course of a year, typically occurring during full load operating periods. Assume for estimation purposes that these outages occur once each during winter, spring, summer, and fall periods. The unit has an average operating cost of $24/MW-hr. Expected operating costs for available replacement power in the first year the improvements will be in effect average $42 in the winter, $36 in spring and fall, and $60 during summer periods.

The value of this performance change, if the correction is made, is approximated as follows:

Winter:

(250 MW)*(2 days)*(24 hr/day)*($42/MW-hr – $24/MW-hr) = $216,000

Spring:

(250 MW)*(2 days)*(24 hr/ day)*($36/MW-hr – $ 24/MW-hr) = $144,000

Summer:

(250 MW)*(2 days)*(24 hr/day)*($60/MW-hr – $ 24/MW-hr) = $432,000

Fall:

(Same values as Spring) = $144,000

Total First Year Performance Value = $936,000

As with the efficiency method, this method of estimating availability value is an approximation only. Other factors that are not represented but could affect the true value include reserve capacity penalties, start-up costs for this unit as well as for replacement units when it is out of service, and many others. It is also an extreme simplification to represent operating costs as average values and to apply these values to around-the-clock full load operation, as is done in the example. However, this method illustrates one approach, even if very simplistically, to quantifying the value of availability change.

2-5.4.2.3 Net Worth. Once the optimization costs have been determined for each correction option, along with valuation of the expected performance changes, it is relatively simple to determine the net worth and the cost-benefit ratio of each option. Optimization by definition seeks the highest performance levels that are feasible and that are cost effective to achieve and maintain. Therefore, the calculation of net worth and cost-benefit ratio is of key importance in making decisions as to which actions to take.

It should be mentioned that net worth is not exclusively a before-the-fact concept to be used in economic justifications. It definitely can be determined after corrective actions have been taken. This permits the true total costs to be evaluated, which may or may not be the same as the costs projected in advance. It also allows the actual performance changes to be measured and assessed, to be compared to the before-the-fact expected changes. Most importantly, such calculations after the actions have been taken allow the true net worth to be evaluated and to serve as data for the total performance monitoring and optimization program.

2-5.4.2.4 Economic Analysis. It is necessary to evaluate all costs and benefits associated with any performance changes, efficiency and/or availability, in accordance with the economic evaluation criteria used within the organization. This may involve calculations of revenue requirements, required internal rates of return, payback periods, net present values, or other criteria. It is important to maintain perspective and perform an economic analysis to the level of detail needed, based on the magnitude of overall costs and benefits to the organization. Because of the potentially large costs and high values associated with optimization work, the proper evaluation of these net values is warranted. For further information on performing these evaluations, financial analysts within the organization may be consulted, or textbooks on economic evaluation and engineering economics may be referenced.

2-5.4.3 Mechanical Optimization in Practice. Mechanical optimization requires a higher level of analysis and commitment to net economics than some of the more traditional approaches to maintenance and modification work. More factors need to be considered in arriving at the overall best results of performance levels and economics. Some of the practical considerations of mechanical optimization are described below.

2-5.4.3.1 Single Versus Multiple Unit Factors. A range of different performance problems with various correction options exists for the different problems. Optimization analysis for the single unit will, therefore, evaluate each of the more significant problems and their correction options. Choices can then be made as to which options to pursue and in which order. Where multiple units are involved, the optimal course of action is to pursue corrections on those units that will yield the highest net worth and the best cost-benefit ratios.

2-5.4.3.2 Outage Planning Considerations. The timing and planning of scheduled outages is an important factor in mechanical optimization. Factors such as system loading and demand, local area conditions, replacement energy costs and sources, fuel availability, and manpower considerations all have significant impact on total costs. Another consideration is the downtime required to perform corrective actions.

2-5.4.3.3 Engineering Optimization. This work involves design changes, equipment replacements, modifications, upgrades, or retrofits. By nature it is generally expensive and time consuming to accomplish. It is highly recommended that any such work that is undertaken to correct efficiency or availability problems be subjected to the closest, most objective, and most critical analytical scrutiny possible. This is because the costs can run so high, as well as the technical ramifications of this level of work. Close attention needs to be given to the expected performance changes to ascertain if they are realistic. It is also important to be sure that all reasonable alternative options have been considered and properly evaluated before the final choice of corrective action is made. Where major physical work is involved, it is advantageous to have the plans reviewed in advance by operators, maintenance personnel, engineers, specialists such as chemists or metallurgists, and others, as appropriate to the nature of the work being planned.

The engineering approach to mechanical optimization is an important and valuable option, often being the far superior action compared with repetitive, symptomatic repair efforts. However, because of the factors cited above, the higher initial cost and larger technical ramifications of this more complex approach deserve very careful optimization analysis before the work is committed to and undertaken.

2-5.5 Integrated Operational and Mechanical Optimization

In practice, it is difficult to separate completely these two areas of optimization. Operational efforts will require certain maintenance and modification support; similarly, maintenance and modification efforts must be accompanied by sound, knowledgeable, and conscientious operating practice.

Generally speaking, operational optimization can be accomplished at relatively low cost once it is under way. It is helpful to have certain prerequisites provided, particularly including good operating instrumentation, an operator-controllable parameters program, training, and a positive environment. Operational optimization can also be done under a wide range of equipment conditions, including impaired and in need of major work. Under very impaired equipment conditions, a unit will not be able to perform well even with the best of operator attention, but performance improvements can be made, nonetheless.

Operational optimization also serves as a valuable source of information for mechanical optimization efforts. Through operational work, mechanical problems that otherwise may have been too subtle to have been recognized can often be detected and brought to attention. This does not imply subtlety in terms of their effects on performance, but rather, in their degree of visibility.

Mechanical optimization may often require higher cost than operational work to implement and maintain. It requires or benefits from some analytical tools that may be fairly complex, and requires cost outlay to perform the corrective actions indicated through optimization analysis. The repair level is an ongoing function essential to reliable unit operation, but requires a commitment of human and financial resources to perform that maintenance at a reasonable level. Certainly the engineering level of mechanical optimization may be extremely complex and expensive, yet in some circumstances it can also yield the most valuable results concerning major or recurrent problems.

In the ongoing optimization effort, it is best not to attempt to separate the operational and mechanical areas from each other. Rather, they can be viewed as highly integrated aspects of unit performance that need to work collaboratively and effectively if the overall objectives are to be met. Ultimately, the efficiency and availability levels achieved for a unit in a cost-effective manner are the result of team effort and commitment. That commitment must be to both the results of performance optimization as well as the team approach needed to achieve them.

2-5.5.1 Reliability Versus Efficiency Tradeoffs. Many tradeoffs exist in the enhancement of power plant performance. With respect to modifying operational methods, short-term gains in efficiency can be made by sacrificing long-term availability. There are also methods to increase generating capacity that are accompanied by a heat rate penalty. Several methods to reduce boiler emissions result in heat rate and capacity penalties, and may reduce reliability. A few examples of these will be given and discussed. This will not be an all-inclusive list, as many others exist.

Increasing main steam temperature improves the efficiency of the steam turbine, but can lead to damage of both the high pressure turbine and the associated piping if design limits are exceeded. The turbine, boiler, and piping manufacturers provide recommended limitations for main steam and reheat steam temperatures. Those limits are usually adhered to, though for some plants that compliance comes with a cost. If combustion controls alone (including burner tilts in tangentially fired boilers) cannot maintain main steam temperatures at or below the recommended level, attemperation water is injected that causes a small heat rate penalty. Unit heat rate is increased by increasing the use of main steam and reheat attemperation, although the heat rate penalty for the main steam is considerably less than the effect from reheat spray. Allowing the temperature to increase beyond the recommended limits improves the heat rate and efficiency of the steam turbine, but will lead to eventual damage of the high pressure turbine, associated piping, and some boiler components.

A common method to reduce the formation of NOx is to stage combustion by limiting the amount of secondary air introduced at the burners and permit the completion of combustion at lower temperatures higher in the furnace by injecting over fire air (OFA). Another option is to utilize various coal nozzle designs that have longer flames and lower peak temperatures. Combustion at lower temperatures will reduce the formation of thermal NOx. If sufficient over fire air is correctly introduced, then complete combustion should occur and boiler efficiency will not suffer, as the amount of unburned fuel and dry gas losses remain unchanged. Boiler efficiency would decrease should the percentage of unburned carbon in the ash increase; this is a frequent consequence of low NOx firing systems for certain type of fuels.

In a limited number of cases, waterwall tube wastage has been seen in the reducing zone between the burners and the over fire air ports. This is a function of a number of factors, including the fuel fired, boiler design, and a preexisting corrosion problem.

As mentioned earlier in this subsection, the steam supply systems are typically designed with water attemperation as a method to control high and potentially damaging steam temperatures. The heat rate penalty in the case of the reheat attemperation spray flow is greater than that incurred on the main steam system. However, the generated output increases when introducing RH attemperation flow, since the total reheat steam flow that enters the boiler and the IP turbine is increased. While this is an atypical case, it is an example of incurring a heat rate penalty while increasing MW generation.

The optimal point of operation can be estimated on a system or component basis utilizing fuel and operating costs, known heat rate penalties, generation gains or losses, emission allowance costs, and an estimate of the reliability effect. This engineering economic evaluation contains a moderate degree of uncertainty. Several of the longer term inputs are estimated. These include the reliability effect and capacity factor. Additional contributors to the overall uncertainty include the inadvertent variations in operating parameters and interrelationships between parameters throughout the plant. Even with these uncertainties, these evaluations are key to optimizing performance and in many cases can be proved or disproved in the long term via careful testing or performance monitoring.

In some cases, the economic analysis does not support revising operational methodology to improve heat rate as the penalty for decreased reliability is too great. These cases can be the impetus for research and development. For example, decades ago, the desire for increased efficiency via higher steam temperatures led to improved materials used in steam piping and boiler heat transfer surfaces. In some cases, fuel switching can either improve or further worsen the prospects to make an operational change economically attractive.

In summary, these three examples depict typical tradeoffs power plant operators encounter. Finding the optimal point of operation is difficult because for most cases, that best point of operation is not constant day to day, or even hour to hour. Small variations in fuel or changes in unit load can move the optimal point. Also the control variables have complex relationships. The best trained operators are often unable to take all the subtle and even obvious interrelationships into account, as they would be overwhelmed with information. In these cases the use of computer-based optimization systems may be beneficial.

2-5.6 Online Optimizers

2-5.6.1 Introduction and Description. In step with the increasing complexities in the interrelationships of the controllable parameters of power plants came the increased capabilities of computer-generated solutions to complex problems. Optimizers were first developed and installed on power plant boilers in the early 1990s. These devices were focused on the complicated role of reducing NOx emissions without increasing unit heat rate. They were used in an operator advisory mode, where their outputs were suggestions to the plant operators. While the operator gained trust in the recommendations of the optimizer, as he/she manipulated the equipment controls, the optimizer “learned” from the results of the operator’s actions and the process improved upon itself. Optimizers and their users later permitted closed-loop operation, where the optimizer, once set into action, directly manipulated certain controls without operator interaction.

Optimizers are software packages that directly affect the manner of equipment operation, in some cases with minimal manual interaction. The optimizer senses the cause and effect of the adjustment of certain parameters and strives to improve performance based on predefined end goals. These packages are analytical in nature (neural networks, fuzzy logic, or mathematical based) and utilize a set of measured values with complex interrelationships. From those values, the optimizer revises control settings to move the unit’s performance towards it goal. The optimizers are tied directly to performance monitoring, but typically without continuous human interaction and the accompanying limitation.

The first applications for optimizers were to reduce NOx emissions without increasing heat rate. Reasonable success was experienced in that area, so most optimizers currently in power plants focus on the boiler and combustion. Focus has recently expanded to sootblower operation and steam temperature stabilization.

The tradeoff between reliability and efficiency is also very great in soot blowing operation. Too little sootblowing reduces boiler efficiency and increases NOx emissions. Too much sootblowing wastes the media, typically steam, water, or air, that causes another heat rate penalty and can damage the boiler tubes causing reliability problems.

New and future optimizers are attempting to optimize more than just the boiler operation; they will cross equipment, system, and unit boundaries to optimize the total plant performance. These future optimizers can be multiobjective, attempting to optimize the circulating water system, backend emissions controls (e.g., SCR, ESP, and FGD), and total plant performance.

2-5.6.2 Methodology. The brain of the optimizer is a computer-based solution technique. These techniques include neural networks, fuzzy logic, linear modeling, and mathematical solutions, some of which operate in a feed forward mode to predict, preempt and compensate for the cause and effect of certain operating modes.

The first step in installing an optimizer is to set its objectives. These can be as simple as increasing boiler efficiency or reducing emissions. Next, the control variables must be identified and methods to monitor the results should be developed.

The actual installation is involved on two planes: software and hardware. On the software side, links must be established to the control variables, data inputs, and resulting outputs. The hardware installation includes a computer where the optimizer resides; wiring, cabling, or wireless connections; the connection to data highway/DCS; and additional instrumentation as needed.

After initial installation, parametric testing using all optimizers must be conducted, where the cause and effect of the adjustment of certain parameters are recorded. This establishes the initial model and teaches that model by establishing relationships between variables. This action is conducted in-plant, either manually with operator intervention every step of the way, or, in some cases, automatically via the optimizer's connections to the control variables.

Once the initial model is created, the plant operator sets limits and constraints to ensure the optimizer does not permit operation that might compromise equipment or personnel safety. For example, if the sole purpose is to minimize NOx emissions, without any constraints, the optimizer might back down air to the level that dangerously high amounts of CO remain in the flue gas. Once the model has been installed, the plant operators conduct reasonability tests in an advisory mode. If accurate, the optimizer has learned the cause and effect relationships and proves itself, so operator confidence rises and closed-loop operation can commence.

Optimizers have a self-tuning process, where through continuous or batch learning periods, its internal model and the variable interrelationships are modified in time to better achieve its goals.

2-5.6.3 Issues. While optimizers can be of great benefit to plant operation and performance, with them come several issues that might detract from their usefulness unless properly addressed.

The optimizer must gain the trust of the operators. Without the operator's willingness to keep the device in closed-loop operation, optimal performance will not be continuously achieved. That level of trust is built via several channels. First, the constraints must be realistically set or the optimizer might drive the plant's performance into an unsafe, unstable, or otherwise unwanted regime. Second, beyond orientation on the operation of the optimizer, operators should be trained on how to best use it, restore it, and keep it in service. Documentation should accompany the training and be readily available in the control room. Third, the rationale on why the optimizer has been installed must be clear and plant leadership support must not waiver. Operators should understand that the optimizer is not replacing them for any reason, e.g., their prior performance or other labor issues, and that they are still responsible for the safe and efficient operation of the plant.

The optimizer must be maintained, much like other pieces of plant equipment. A site champion or champions, typically an engineer or lead operator, should be assigned both the responsibility and authority to keep the optimizer in service and operating correctly. The instruments supplying critical parameters must be maintained and calibrated or the optimizer will base its decisions on erroneous inputs. With age, as software is updated and DCSs are upgraded, the connections to the optimizer and its operation should be verified.

The constraints on the optimizer's operation should be reviewed periodically. As plant operation changes, due to fuel switching, lower load, off-design operation, or other reasons, the optimizer should return to learning mode and its constraint reset to realistic values. If the optimizer is permitted to operate outside knowledge base, its behavior might be unpredictable and undesirable. The tradeoff of reliability versus efficiency must be addressed in addition to ensuring proper and up-to-date constraints.

The last issue is that of the unmeasured parameters. These parameters are not available continuously and therefore do not feed into the logic and decision process internal to the optimizer. For boilers, as an example, unburned carbon and water wall wastage cannot be measured in real time. They should be periodically monitored by plant engineers and the effect on them by the optimizer should be understood.

2-5.7 For Further Reference

2-5.7.1 General References

[1] Heat Rate Improvement Conference and Workshop, October 1985, San Francisco, California, Proceedings, EPRI CS-4736.

[2] EPRI Power Plant Performance Monitoring and System Dispatch Improvement Workshop, November 1986, Washington, DC, Proceedings, EPRI CS/EL-5251-SR.

[3] EPRI Seminar on Fossil Plant Improved Heat Rate and Availability, December 1987, San Diego, California, Proceedings, EPRI RP-1403.

[4] EPRI Heat Rate Improvement Conference, May 1988, Richmond, Virginia, Proceedings, EPRI GS-6635.

[5] EPRI Heat Rate Improvement Conference, September 1989, Knoxville, Tennessee, Proceedings, EPRI RP-1711-10.

[6] EPRI Heat Rate Improvement Conference and Workshop, May 1991, Scottsdale, Arizona, Proceedings.

[7] "Electrostatic Precipitator Power Conservation," Kaltenbac, R. P., et al, Burns and McDonnell Engineering Company, 1985, Proceedings of the American Power Conference, Chicago, Illinois, Vol. 47. Illinois Institute of Technology, Chicago, Illinois.

[8] "Sootblowing Operation Lowers Total Cost of Recovery Boiler Operation," Slejko, D. F., et al, Bailey Controls Company, 1985, ISA Transactions, Volume 24.

[9] "Combustion Optimization of a Coal-Fired, Supercritical 575 MW Boiler," Wong, F. K. L., et al, EPRI, 1985, 8th International Coal and Solid Fuels Utilization Conference, Pittsburgh, Pennsylvania, Industrial Publications Inc., Houston, Texas.

[10] "Application of a Performance Diagnostic System to Improve Boiler Turbine Operation," Barkan, J. L., et al, Babcock and Wilcox Company, 1983, Proceedings of the American Power Conference, Chicago, Illinois, Volume 45. Illinois Institute of Technology, Chicago, Illinois.

[11] "Steam Turbine Performance Profile," Ellison, R. B., Duke Power Company, 1990, Joint ASME/IEEE Power Generation Conference, Boston, Massachusetts. ASME, New York, New York.

[12] "Southern Company Service's Cost Effective Approach to Turbine Cycle Performance Testing," Johnson, R. A., et al, Southern Company Services, Inc., 1989, Joint ASME/IEEE Power Generation Conference, Dallas, Texas. ASME, New York, New York.

[13] "Improving the Thermal Performance of Older Steam Turbines," Schofield, P., et al, General Electric Company, 1990, International Joint Power Generation Conference, Boston, Massachusetts. ASME, Power, Volume 10, New York, New York.

2-5.7.2 Specific References. See General References [1] through [6] for 1A through 6W.

[1A] "Power Plant Efficiency Improvements," Wanklyn, S. V., Florida Power and Light Company.

[1B] "Best Buy in Heat Rate Recovery—Turbine Cycle Isolation Maintenance," Legg, J. K., et al, Southern Company Services, Inc.

[1C] "Operator Awareness (With a Twist)," Mancini, R., et al, New York State Electric and Gas Company.

[1D] "Heat Rate Improvement Program," Shipp, J., et al, East Kentucky Power Company.

[1E] "Heat Rate Improvement Program Techniques and Philosophy," Hill, D. L., Public Service Company of New Mexico.

[1F] "Energy Management Standards Program at PSE&G," Case, R. C., Public Service Electric and Gas Company.

[1G] "Component Efficiency and the Optimum Repair Cycle Period," Sabol, S. B., Virginia Electric Power Company.

[1H] "On-line Condenser Performance Monitoring System," Peterson, S. N., San Diego Gas and Electric Company.

[1I] "Heat Rate Improvement Through Retrofit of Advanced Technology," Touchton, C. L., EPRI.
[1J] "Applying Today's Technology to Improved Turbine Efficiency," Evans, D. H., et al, Westinghouse Electric Company.
[2A] "Turbine Control Valve and Position Monitoring Procedures," Williams, S., Potomac Electric Power Company.
[2B] "Advantages and Peculiarities of Valve Point Loading in Unit Dispatch," Clelland, P., et al, Philadelphia Electric Company.
[2C] "Optimization of Unit Heat Rate Through Variations in Fireside Parameters," Levy, E., et al, Lehigh University; and Williams, S., et al, Potomac Electric Power Company.
[3A] "Boiler and Turbine Retrofit For Variable Pressure Operation," Kimel, E., Burns and Roe.
[3B] "Experiences and Operating Results of Conversion of a Supercritical Once-Through Unit from Constant Pressure to Variable Pressure Operation," Maruyama, I., IHI; and Matsumoto, S., Hokuriku Electric Power Company.
[3C] "Burner Retrofit For Reduced NO_x, Excess Air and Heat Rate," Brunton, L., Kansas Power and Light Company.
[3D] "Startup/Standby Boiler Feed Pump Retrofit For Availability and Maintenance Improvements," Hinshaw, B., Southern Company Services.
[3E] "Retrofit Installation of a Boiler Feed Pump Hydraulic Coupling," Taylor, S., et al, Potomac Electric Power Company.
[3F] "Improved Heat Rate Through Cooling Tower Retrofits," Bartz, J., EPRI; and Johnson, B., Battelle-Northwest.
[4A] "Performance Monitoring of Steam Turbines," Campbell, W. A., Philadelphia Electric Company.
[4B] "Performance Monitoring of Fossil Steam Generators," Davidson, P. C., Black and Veatch.
[4C] "Performance Monitoring of Pumps," Beekman, W., Peabody Floway.
[4D] "Repowering and Heat Rate Enhancement Using Fluidized Bed Boilers," Jarboe, T. B., Bechtel Eastern Power Company.
[4E] "Heat Rate Improvement Theory and Implementation at Southern California Edison," Babb, J. A., Southern California Edison.
[4F] "Use of On-line Monitoring System to Improve Station Heat Rate and Availability," Wilkens, I., et al, Montana-Dakota Utilities Company; and Blakely, J. E., EI International.
[4G] "Real-time Optimization of Boiler Operation for Heat Rate Improvement," Swindell, D. K., et al, Technology for Energy Corporation.
[4H] "Modifications of Secondary Air Ducts and Field Balancing of Secondary Air Flows for Heat Rate Improvement," Boroway, W. I., et al, Virginia Electric Power Company.
[4I] "A Methodology to Minimize Unaccountable Heat Rate Losses," Interess, E., et al, A. D. Little Company; and Sabol, S. S., Virginia Electric Power Company.
[4J] "Use of Root Cause Analysis Techniques to Improve Operating Heat Rate," Tome, A. E., IT-Delian Corporation; and Legg, J. K., Southern Company Services.
[4K] "Recent Developments in Instrumentation for Optimizing Coal Combustion," Sotter, J. G., Energy and Environmental Research Corporation.
[4L] "An Integrated Performance Monitoring System for Fossil Power Plants with On-line and Off-line Applications," Smith, L., TRAX Corporation.
[4M] "Basin Electric's Btu Alert Program Enhanced with On-line Performance Monitoring," Eulinger, R., Black and Veatch.
[4N] "A Systematic Approach to Performance Testing and Monitoring Program," Jones, J. D., Dayton Power and Light Company; and Diaz-Tous, I. A., ENCOR-AMERICA.
[4O] "On-line Performance Monitoring at San Diego Gas and Electric Company," Peterson, S., San Diego Gas and Electric Company.
[4P] "Impact on Heat Rate of Improved Cooling System Performance," Bartz, J. A., EPRI.
[4Q] "Heat Rate Improvements from CEGB Research on Turbines, Condensers and Cooling Systems," Moore, M. J., et al, Central Electric Research Laboratories Board, England.

[4R] "Heat Rate Improvement Through Targeted Chlorination," Mussalli, Y., Stone and Webster; Chow, W., EPRI; and Freddo, W., New England Power Company.
[4S] "Impact of Boiler Feed Pump Turbine Performance on Unit Heat Rate," Fruchtman, I., Ebasco Services; Burkhart, D., et al., Southern California Edison; and Poe, G. G., EPRI.
[4T] "On-line Condenser Performance Monitoring," Reddington, I., Power Technologies, Incorporated; and Leyse, R. H., EPRI.
[4U] "On-line Controllable Parameters Monitoring for Heat Rate Improvement," Freels, J. D., Technology for Energy Corporation; and Missimer, J. R., Power Diagnostics, Incorporated.
[4V] "Operator Training as Part of a Heat Rate Improvement Program," Kenien, D. L., New York State Electric and Gas Company; and Nasal, I., General Physics Corporation.
[4W] "Heat Rate Improvement and Monitoring at TVA," Keyse, R. L., et al, Tennesse Valley Authority.
[4X] "Boiler Tuning to Optimize Performance," Peeples, J. D., TU Electric.
[4Y] "Turbine Cycle Performance Optimization," Silvestri, G. J., Westinghouse Electric Company.
[4Z] "Expert System for Steam Turbine Performance Evaluation," Kindl, F. H., et al, Encotech, Incorporated; and McKinley, J. H., Southern California Edison.
[5A] "Heat Rate Improvement Methodology for an On-line Thermal Information System," Jagde, R., Combustion Engineering; and Snyder, C. E., Wisconsin Public Service Corporation.
[5B] "Heat Rate Improvement at New England Power Company," Lundquist, D. C., et al, New England Power Company; and Batson, R. D., et al, Stone and Webster Engineering Corporation.
[5C] "Heat Rate Awareness Program," Nasal, J. R., General Physics Corporation.
[5D] "The Kingston Fossil Plant's Microcomputer Based On-line Heat Rate Monitoring System," Carter, J. C., et al, Tennessee Valley Authority.
[5E] "Computer Efficiency Monitors for Power Plant Operations and Their Role in Improving System Efficiency," McClintock, M., General Physics Corporation; and Jesikiewicz, T., New York State Electric and Gas Company.
[5F] "On-line Monitoring of Unburned Carbon," Eskenazi, D., et al, Lehigh University; Cramer, D., et al, Potomac Electric Power Company; and Valverde, J., Electric Power Research Institute.
[5G] "On-line High Pressure Turbine Steam Seal Performance Monitor," Elderkin, M. J., Pacific Gas and Electric Company.
[5H] "Operational Improvements and Experiences of Real-time PC-based Controllable Parameters Monitoring System at OMU," Conn, S. K., Owensboro Municipal Utilities.
[5I] "Pulverized Coal Mill Fuel/Air Ratio Testing," Jones, C. R., Texas Municipal Power Agency.
[5J] "Installation and Testing of a Redesigned Boiler Feed Pump Turbine Steam Supply System at the O.B.G.S.," Burkhart, D. M., et al, Southern California Edison Company; and Fruchtman, I., Ebasco Services Incorporated.
[5K] "Heat Rate Improvement at TU Electric's North Lake Unit 2," Warren, S., Sargent and Lundy.
[5L] "Power Plant Heat Rate Improvement Through a ThermoEconomic Analysis," Tsatsaronis, C., et al, Tennessee Technological University.
[5M] "Use of Computer Models for Boiler Performance Improvement Monitoring and Simulation," Bauer, F. W., et al, Stone and Webster Engineering Company.
[5N] "A Method for Approximating the Cost/Savings of Extraction Steam Air Preheating," Price, C. M., General Physics Corporation.
[5O] "Optimizing the Operations of TVA's Kingston Fossil Plant," Ciles, J. E., et al, Tennessee Valley Authority.
[5P] "Development of an On-Line Expert System: Heat Rate Degradation Advisor," Sopocy, D. M., et al, Sargent and Lundy; and Divakurani, S. M., et al, EPRI.
[5Q] "Economically Based Performance Improvement," Kindl, F. H., Encotech, Inc.
[5R] "Performance Optimizing of Pulverized Coal Boilers," Levy, E. K., et al, Lehigh University; Crim, H. G., Potomac Electric Power Company; and Leyse, R. H., EPRI.
[5S] "Future Improvement in Plant Design Utilizing Higher Efficiency Auxiliary Equipment," Winch, R., Houston Lighting and Power.

[5T] "Troubleshooting Feedwater Heaters: A Case History Approach," Stalnaker, G. K., et al, Public Service Company of Oklahoma.
[5U] "Feedwater Heater Performance Prediction Calculation Procedure," Tsou, J., EPRI.
[5V] "Advanced Predictive Performance Modeling for Condensers and Feedwater Heaters," Mussalli, Y., Stone and Webster; and Didsbury, R., et al, Heat Transfer and Fluid Systems Service.
[6A] "Station Thermal Awareness, Results and Recognition at Eddystone Station," Wusinich, D., Philadelphia Electric Company; and McCreight, M., Robert H. Schaffer and Associates.
[6B] "Heat Rate Improvement Methodology for an On-Line Thermal Information System," Synder, G., Wisconsin Public Service; and Jagde, R., Asea Brown Boveri Power Automation.
[6C] "Performance Driven Plant Management," Missimer, J., et al, Environmental Systems.
[6D] "Effect of Maintenance Awareness Training on Plant Heat Rate and Reliability," Kenien, D., et al, New York State Electric and Gas; and Sheikh, S., General Physics Corporation.
[6E] "Improved Sootblowing Through the Application of Expert Systems Technology," Jesikiewicz, T., et al, New York State Electric and Gas; and Bangham, M., et al, General Physics Corporation.
[6F] "TVA's On-Line Boiler Performance Monitoring System," Carter, J., et al, Tennesse Valley Authority.
[6G] "Fuel Consumption Index for Proper Monitoring of Power Plants," Lang, F., Exergetic Systems; and Horn, K., Pacific Gas and Electric.
[6H] "The Development of On-Line Thermal Performance Monitors in Nuclear Electric Company Stations," Conner, A., Nuclear Electric Company.
[6I] "Computer-Aided Performance Monitoring Program at Diablo Canyon," Nelson, T., et al, Pacific Gas and Electric; and Kessler, T., Expert- EASE Systems.
[6J] "The Effect of Burner Tilt Angle on Unit Performance at PEPCO's Morgantown Unit 2," D'Agostini, M., et al, Lehigh University; and Fernandes, J., et al, Potomac Electric Power Company.
[6K] "Cape Canaveral Unit 2 Superheater Spray System Improvements," Chang, P., Florida Power and Light Company; and Fitzgerald, D., Foster Wheeler.
[6L] "The Uses of Continuous Unburned Carbon Measurement," Leopold, D., Potomac Electric Power Company; and D'Agostini, M., Lehigh University.
[6M] "Search for Efficiency," Lindsey, R., Duke Power Company.
[6N] "Steam Turbine Retrofits and Operational Procedures for Improved Heat Rates," Martin, H., et al, Westinghouse.
[6O] "Heat Rate Improvement at PG&E Hunter's Point Unit 4 Condenser Through a Circulating Water Pump Performance Test," Virdee, A., Pacific Gas and Electric Company; and Diaz-Tous, I., et al, ENCOR-AMERICA.
[6P] "Improving Performance and Improving Costs of Cooling Towers," Bartz, J., EPRI.
[6Q] "Heat Rate Recovery Through Cycle Isolation Diagnosis and Correction," Branco, M., Long Island Lighting Company.
[6R] "Performance Monitoring Key to Improved Maintenance Practice at Montour 1 & 2," Fuge, D., Pennsylvania Power and Light.
[6S] "Economizer Optimization and Boiler Performance Studies—A Mathematical Model Application," Bauer, F., Stone and Webster; and White, W., Potomac Electric Power Company.
[6T] "Verifying and Predicting Boiler Performance Using an Energy Balance Computer Program," Kettenacker, W., Performance Engineering; and Browers, B., Minnesota Power.
[6U] "A Performance Monitoring Model for Steam Driven Boiler Feed Pumps," Kona, C., Power Technologies, Inc.
[6V] "Extraction Steam Air Preheating: Boiler Credits vs. Turbine Cycle Debits," Nissen, A., Intermountain Power Service; and Nasal, J., General Physics.
[6W] "An Economic Model for Determining the Optimum Operating Strategy for a Regenerative Air Preheater," D'Agostini, M., et al, Lehigh University; and Robb, P., Potomac Electric Power Company.
[14] "Application of Diagnostic Performance Programs in Power Plants," Moradian, M. A., et al, Westinghouse Electric Corporation, 1987, Joint ASME/IEEE Power Generation Conference, Miami Beach, Florida. ASME, New York, New York.

[15] “A Review of Sliding Throttle Pressure for Fossil Fueled Steam Turbine Generators,” Silvestri, G. J., et al, Westinghouse Electric Corporation, 1972, American Power Conference, Chicago, Illinois.

[16] “Retrofit to Variable Pressure Operation to Increase Efficiency and Availability of Fossil Fired Power Plants,” Touchton, G., et al, EPRI, 1988, Joint ASME/IEEE Power Generation Conference, Philadelphia, Pennsylvania. ASME, New York, New York.

[17] “Improved Steam Condenser Gas Removal System,” Kubik, W. J., et al, Graham Manufacturing Company, 1988, ASME Winter Annual Meeting, Chicago, Illinois. ASME, New York, New York.

[18] “Modernize Your Cooling Tower,” Burger, R., Burger and Associates, Inc., 1990, Chemical Engineering Progress, Volume 86.

[19] “Development and Implementation of a Feedwater Heater Testing Program,” Thompson, N., Alabama Power Company, 1990, International Joint Power Generation Conference, Boston, Massachusetts. ASME, Power, Volume 12, New York, New York.

[20] “Optimizing Feedwater Heater Replacements for a Supercritical Fossil Fired Unit,” Fusco, J. A., et al, Pacific Gas and Electric Company, 1988, Joint ASME/IEEE Power Generation Conference, Philadelphia, Pennsylvania. ASME, New York, New York.

[21] “Steam Trapping System Optimization,” Lehman, G., 1986, 9th World Energy Engineering Conference, Strategies for Energy Efficient Plants and Intelligent Buildings. Association of Energy Engineers, Prentice-Hall, Inc., Englewood Cliffs, New Jersey.

[22] “How to Cope with Fan Capacity Problems,” Tallant, P. E., PowerFlow Engineering, Inc., 1988, Power Engineering, Volume 92, Barrington, Illinois.

[23] “Design Possibilities and Performance of Combined Cycle Operation of Converted Steam Power Plants,” Linnemeijer, M. J. J., et al, Stork Boilers, 1988, ASME, Amsterdam, Netherlands, 88-GT-178. ASME, New York, New York.

[24] “Mark 1 Performance Monitoring System at PEPCO Morgantown No. 2 Unit,” Crim, H. G., et al, Potomac Electric Power Company, 1989, Joint ASME/IEEE Power Generation Conference, Dallas, Texas. ASME, New York, New York.

[25] “Performance Monitoring Systems Can Reduce Operating Costs,” French, P. D., Kansas City Power and Light Company,” 1989, Power Engineering, Volume 93, Barrington, Illinois.

[26] “Optimization of Advanced Steam Condition Power Plants,” Silvestri, G. J., et al, Westinghouse Electric Corporation, 1988, Joint ASME/IEEE Power Generation Conference, Philadelphia, Pennsylvania. ASME, New York, New York.

[27] “Role For Exergy Auditing in a Thermal Power Station,” Yasni, E., et al, Electricity Corporation of New Zealand, 1987, ASME Winter Annual Meeting, Boston, Massachusetts. ASME, New York, New York.

[28] “Optimization of Multi-Plant Cogeneration System Operation Including Electric and Steam Networks,” Moslehi, K., et al, Power Automation, Inc., 1991, IEEE Transactions on Power Systems, Volume 6.

[29] “Recommended Practice for Monitoring and Instrumentation of Turbine Generators,” Corkins, R. J., Detroit Edison Company; Gray, R. F., Westinghouse Electric Corporation; and Nippes, P. I., N.P.A. Incorporated; 1993, IEEE, P1129.

Section 3
Case Studies/Diagnostic Examples

3-1 AIR HEATER PLUGGING DUE TO FAILED SOOTBLOWER

3-1.1 Summary

Exit gas temperatures on a coal fired generating unit having two rotating-basket regenerative air preheaters were observed to be slowly diverging from one another over a 2-week period. The "A" heater gas temperature was trending upward, while the "B" heater was trending downward (see Fig. 3-1.1-1). Air heater *X*-ratios and gas-side efficiencies were also noted to be diverging. However, the average of the two *X*-ratios was near normal, indicating a flow imbalance between the two heaters. Gas-side efficiencies, when averaged, showed a decrease, indicating fouling. An internal inspection during a weekend outage revealed a ruptured elbow in the air heater sootblowing piping resulting from extensive erosion along the outer radius, likely due to water entrainment from malfunctioning steam traps upstream. This resulted in ineffective cleaning of the "A" air heater, gradual plugging, and a biasing of gas flow to the relatively clean "B" heater.

3-1.2 Key Performance Indicators

The key performance indicators for this example are air heater exit gas temperature, air heater differential pressure, gas-side efficiency, and *X*-ratio. Air heater exit gas temperature is a measured parameter and a general indicator of overall boiler efficiency. Air heater differential pressure is a measured parameter and is a useful indicator of fouling/plugging due to ash build-up. Gas-side efficiency, a relative indicator of air heater heat transfer effectiveness, is the gas temperature drop expressed as a percentage of the temperature head. *X*-ratio, expressed as the ratio of the gas temperature drop to air temperature rise, is an indicator of air flow relative to gas flow.

Fig. 3-1.1-1 Air Heater Exit Gas Temperature 2-Week Trend (Courtesy General Physics Corp.)

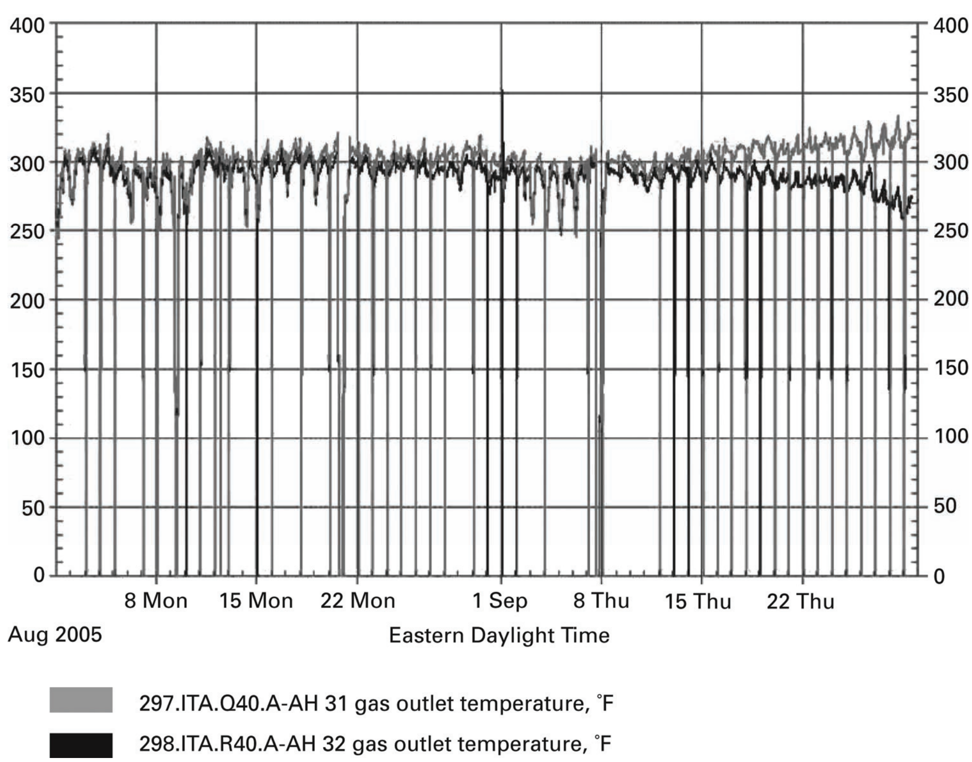
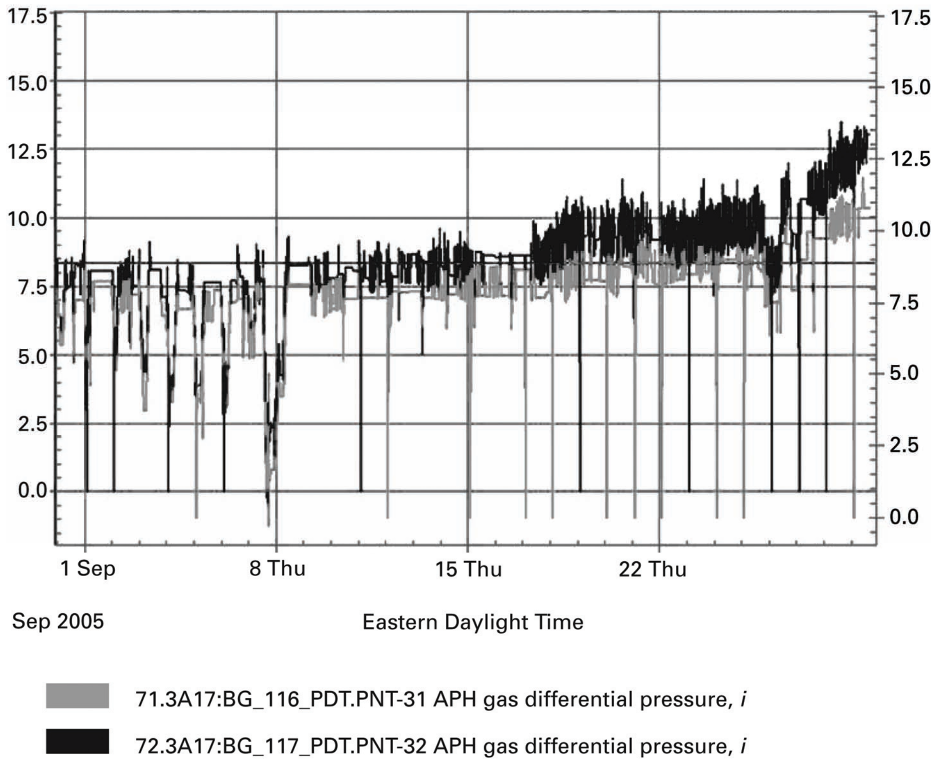


Fig. 3-1.3-1 Air Heater Differential Pressure 2-Week Trends (Courtesy General Physics Corp.)



3-1.3 Initial Observations and Corroborating Evidence

The problem was initially detected by trending the measured air heater exit gas temperatures using the plant's on-line performance monitoring system (see Fig. 3-1.1-1). *X*-ratios for both heaters were also observed to be trending away from one another, indicating a potential imbalance in gas distribution between the two heaters. The *X*-ratios, when averaged, were close to normal, which was consistent with a flow imbalance. Corroborating evidence included trends of air heater differential pressures that showed increases in both heaters, but at different rates (see Fig. 3-1.3-1). The "B" heater showed a gradual increase, consistent with increased gas flow, while the "A" heater showed a sudden increase consistent with excessive plugging. The provisional diagnosis was excessive plugging on the "A" heater, resulting in the flue gas taking the path of least resistance through the relatively clean "B" heater.

3-1.4 Physical Inspection

An internal inspection during a weekend outage revealed a ruptured elbow in the air heater sootblowing piping. Extensive erosion along the outer radius, likely due to water entrainment from malfunctioning steam traps upstream (a common problem), lead to a thinning of the pipe wall and eventual failure. The elbow was replaced and the sootblowing system returned to service. An air heater wash was performed at a later date to return the "B" heater to normal effectiveness.

3-1.5 Capacity, Efficiency, and Reliability Impacts

The primary efficiency loss in this example is decreased boiler efficiency (–0.2%) due to increased boiler exit gas temperature (+8°F). This corresponds to an approximately 21 Btu/kWh heat rate penalty. Although no capacity shortfalls were noted at the time, continued operation without the repair would have likely resulted in a derating due to the flow restriction in the air heater and an increasing inability to remove products of combustion.

3-1.6 Financial Analysis

The increase in fuel consumption is equal to the decrease in boiler efficiency, which may be determined following the calculation procedures in ASME PTC 4 or using any commercial software program suitable for the purpose. For the example here, the boiler efficiency decrease, and thus fuel consumption increase, was estimated to be 0.21%. Since the repair was considered necessary from both safety and reliability viewpoints, an ROI analysis was not deemed necessary.

3-2 BOILER EXAMPLE

3-2.1 Summary

This subsection briefly describes the problem scenario, the key performance indicators (KPIs) used to detect and analyze the problem, and the problem diagnosis.

A large utility boiler had performance monitoring data over a 10-yr period, which indicated a gradual increase in the gas temperatures leaving the two air preheaters. This resulted in reduced boiler efficiency, increased unit heat rate, and decreased precipitator collection efficiency. An evaluation by the boiler OEM was performed to determine possible causes. A number of possibilities were evaluated, the data and testing results led to the conclusion that uncontrolled infiltration air into the backpass of the boiler affected the air preheater performance, and this alone accounted for as much as a 37°F increase in the air heater exit gas temperatures. Other contributing but less important factors included the large amounts of tube shields covering the economizer and horizontal reheater heat transfer surfaces, cold tempering air leakage into the pulverizers, and increasing excess air levels. The primary recommendation was to replace the peg finned backpass walls with a welded wall design to reduce the infiltration of ambient air into the boiler. Lower cost recommendations and operational recommendations included improved pulverizer maintenance, air heater testing and inspections, and improved sootblower maintenance and optimization. The installation of additional economizer surface was also proposed as a means of further improving boiler efficiency and heat rate.

3-2.2 Key Performance Indicators

This subsection reviews the specific KPIs used for detecting and diagnosing the root cause of the problem, including appropriate references to pertinent sections containing KPI definitions and formulae.

The key indication of changes in boiler performance was the measured gas temperatures leaving the two air preheaters. Paragraph 2-3.8 of this document discusses boiler monitoring and diagnostics, and includes discussion of the boiler, pulverizers, and air heaters. Paragraph 2-3.8.6.3 discusses air preheater calculations. However, temperature measurements around the air preheaters do not by themselves provide complete information concerning air preheater performance, the air and gas flows must also be determined.

3-2.3 Initial Observations and Corroborating Evidence

This subsection describes the pattern in KPIs that indicated the presence of the problem, any additional parameters taken into consideration, and the provisional diagnosis.

The plant computer historical information trending the air heater gas temperatures showed an increasing trend over a 10-yr period, although downward changes were also evident in this data.

A series of tests were performed by a testing contractor to determine the amount of backpass infiltration. These tests were conducted by measuring the oxygen levels at different locations within the boiler backpass, using high temperature water cooled sampling probes. These tests confirmed an unusually large increase in the oxygen level between the furnace outlet plane and the economizer outlet plane. Testing performed after some backpass wall repairs also indicated a decrease in backpass leakage.

This plant fires a Texas lignite coal; over many years their coal had a trend of an increasing percentage of relatively erosive ash. The plant maintenance records revealed that a large number of tube shields had been installed in many of the boiler sections to prevent tube failures from the thinning of the tubes due to fly ash erosion. Laboratory testing by the boiler OEM had shown that tube shields cause a certain amount of reduction in heat transfer rates. This would not normally result in a measurable change in boiler performance; however, the very large number of tube shields was calculated to result in a 5% to 10% change in the heat transfer of the shielded economizer and reheater sections. The calculated effect was an additional 12°F increase in air heater exit gas temperatures as a result of tube shields.

3-2.4 Physical Inspection

This subsection describes the results of a further tests or physical inspections used to determine the root cause of the problem.

Inspections by the plant revealed that the backpass walls were in deteriorated conditions, with displaced tubes and gaps in the refractory between the tubes. The expansion joints to the air heaters were in good condition; these are often a source of infiltration air. The air heater baskets and seals were inspected and were in good condition.

3-2.5 Capacity, Efficiency, and Reliability Impacts

This subsection describes impact of the problem on plant capacity, both for current conditions and peak operating periods (e.g., warmer weather). It also describes the additional fuel consumption that may occur as well as long-term and short-term reliability considerations.

The main effect of the increasing exit gas temperatures was a 1.0% decrease in boiler efficiency, an increase in the required coal flow, and an increase in plant heat rate. Other effects included an increase in I.D. fan horsepower, and decreased precipitator performance leading to increase stack opacity. Although the tube shields were judged to have an averse effect on efficiency, the plant seldom had a forced outage due to tube leaks caused by fly ash erosion damage.

3-2.6 Financial Analysis

This subsection describes the basis of determining the financial impact of the problem scenario in terms of additional fuel consumption and reduced capacity. It also considers the cost of repair, replacement, or operating change to determine ROI, if applicable.

The 1.0% calculated decrease in boiler efficiency was estimated to approximately equal a 100 Btu/kWh increase in heat rate. This was estimated to be a cost of $500,000 per year. The installed cost of the backpass repairs was estimated at $800,000, and would require an outage of 4 weeks.

3-3 TEMPERATURE CALIBRATIONS

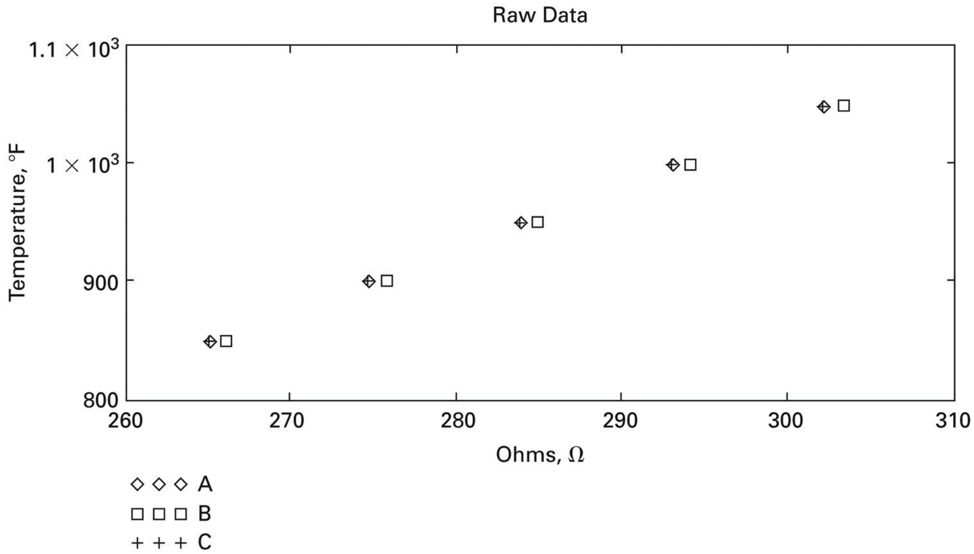
3-3.1 Problem

Performance tests were run on a reheat turbine with test grade instrumentation RTDs (resistance temperature detectors). When the data were analyzed, the throttle and reheat temperatures did not change during the test. The field check of data showed reasonable temperatures.

3-3.2 Analysis

The three temperatures in question were calibrated RTDs. Three RTDs, A, B, and C, had about 18 samples collected at five different temperatures between 850°F to 1,050°F (see Fig. 3-3.2-1).

Fig. 3-3.2-1 Three RTDs: Readings Collected at Five Temperatures



Fit data with third order polynomial. Note that the standard error of estimate values (SEEs) are very low

SEE 0.1049

$$a := \begin{pmatrix} 799.026306 \\ -8.355192 \\ 0.045241 \\ -4.917516 \cdot 10^{-5} \end{pmatrix}$$

SEE 0.1283

$$b := \begin{pmatrix} 994.931213 \\ -10.380441 \\ 0.052104 \\ -5.702374 \cdot 10^{-5} \end{pmatrix}$$

SEE 0.1168

$$c := \begin{pmatrix} 893.779663 \\ -9.346064 \\ 0.048698 \\ -5.315742 \cdot 10^{-5} \end{pmatrix}$$

$$\text{pol a }(x) := a_0 + a_1 \cdot x + a_2 \cdot x^2 + a_3 \cdot x^3$$

$$\text{pol b }(x) := b_0 + b_1 \cdot x + b_2 \cdot x^2 + b_3 \cdot x^3$$

$$\text{pol c }(x) := c_0 + c_1 \cdot x + c_2 \cdot x^2 + c_3 \cdot x^3$$

The fit looks good (Fig. 3-3.2-2). But look at a histogram of A of residuals (Fig. 3-3.2-3).

Note that distribution of errors is not balanced around the mean (Fig. 3-3.2-4).

These RTDs were calibrated at 0%, 50%, 100%, 75%, and 25% of calibration span. This produces a better calibration for high accuracy, but the fit does not bisect the hysteresis loop because heating and cooling points are not balanced in number (50 vs. 75 and 25) and because they are not replicated (50 up and then 50 down).

What would happen with an open circuit or 0 Ω? Figure 3-3.2-5 describes the situation.

Note that with B that zero ohms would indicate 1,000°F if the RTD was not plugged in. Note that RTDs calibration data are very linear.

Fig. 3-3.2-2 Fit of RTD Data

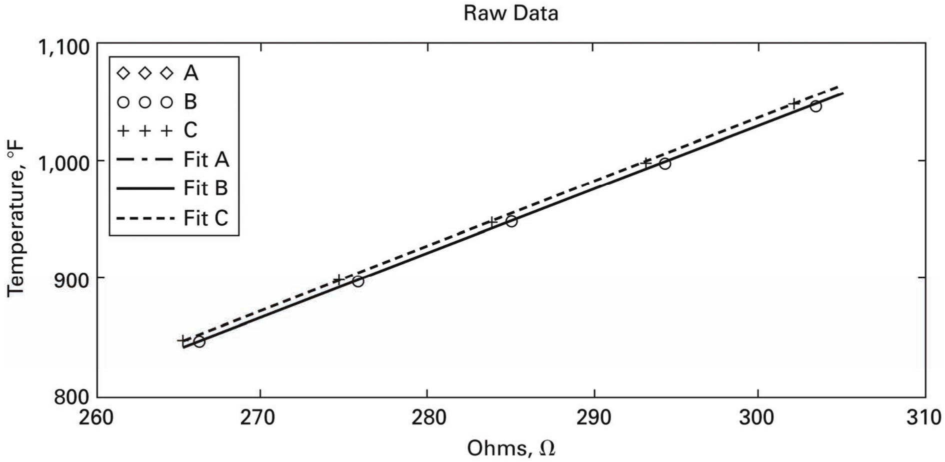


Fig. 3-3.2-3 Histogram of RTD A

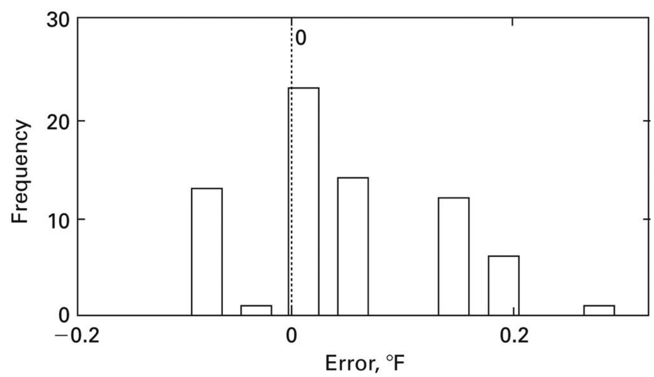


Fig. 3-3.2-4 Distribution of Errors for the Three RTDs

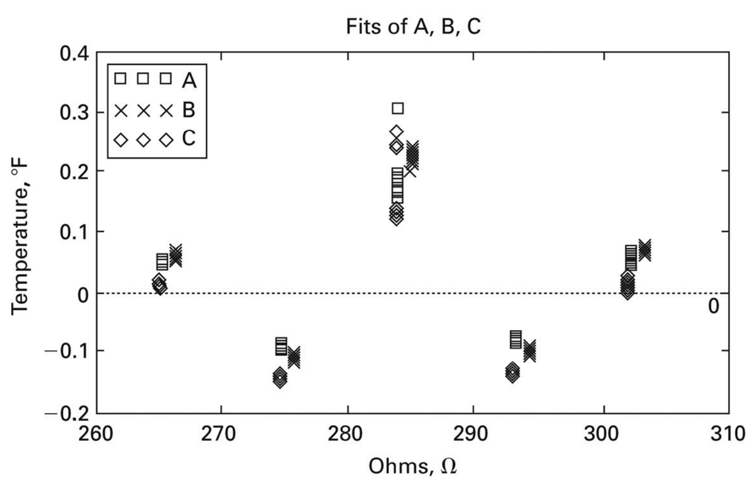


Fig. 3-3.2-5 Fits of RTDs A, B, and C in Open Circuit

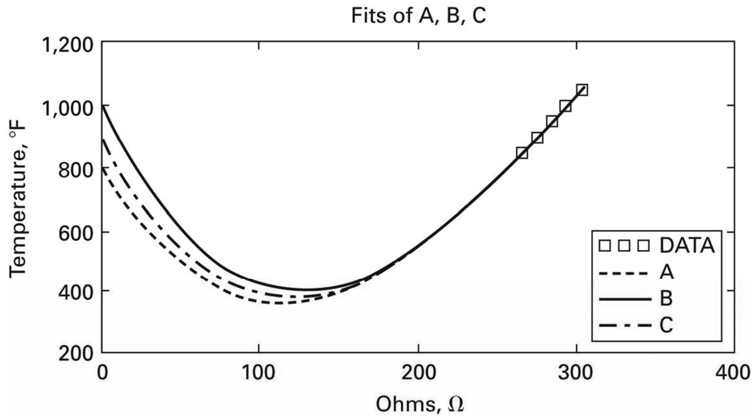
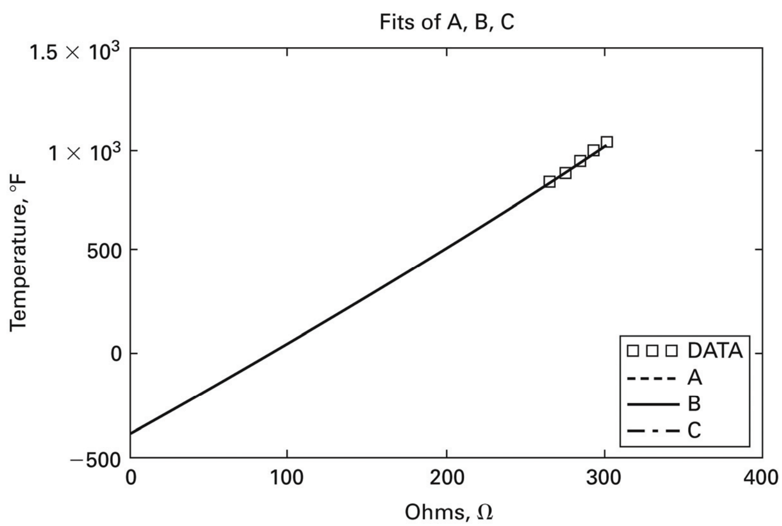


Fig. 3-3.2-6 Fits of RTDs A, B, and C Using the Calendar–Van Dusen Eq. (3-3.2) for Calibration



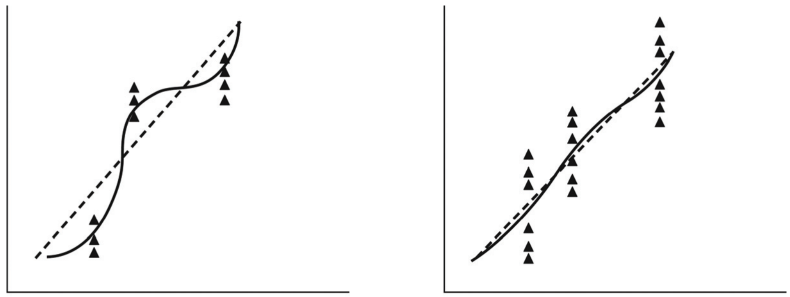
What if the Calendar–Van Dusen Equation was used to fit the calibration data? Refer to eq. (3-3.2).

$$\mathrm{Tcal}\left(R_t,\ \delta,\ R_0,\ R_{100}\right)$$

$$:=\left[\left[\frac{50}{\delta}\cdot(\delta+100)-\left[\frac{50}{\delta}\cdot\sqrt{\frac{400\cdot\delta}{R_{100}-R_0}}\cdot\sqrt{\left(\frac{1}{400}\cdot\delta+\frac{25}{\delta}+\frac{1}{2}\right)\cdot\left(R_{100}-R_0\right)+R_0-R_t}\right]\right]\cdot\frac{9}{5}+32\right] \qquad (3\text{-}3.2)$$

The fit looks good (Fig. 3-3.2-6).

Fig. 3-3.3-1 Fits With and Without Replicate Data



(a) Fit Without Replicate Data (b) Fit With Replicate Data

3-3.3 Calibration Method

An SPRT (standard platinum resistance thermometer) was used as a transfer standard. All we know about the SPRT is δ*, R_0, and R_{100}. These constants substituted into eq. (3-3.2) can be reduced to the simple polynomial forms shown in eq. (3-3.3)

$$\text{ohms}(t) = a + b^*t + c^*t^2 \tag{3-3.3}$$

The details are very complicated, but the critical point is the SPRT has only three “knowns” as a function of temperature. When a calibration is run, these calibration “knowns” transfer these temperature knowns to the test RTD. So if a third order fit is used, there is one more “known” or the fit has another degree of freedom. It is believed that this additional degree of freedom has allowed the false conclusion that reduced span or range of calibration actually improved the fit. Reduced range neither changes the magnitude of the output of the SPRT, nor does it increase the accuracy of the SPRT. Most likely the model (fits) specific calibration run variation. The fit can wiggle more to pass through the data hysteresis. These questions need laboratory work to determine the best approach. Fig. 3-3.3-1 shows that with proper replicate data the third-order fit would have been a “better” calibration because it would include the balanced high and low hysteresis deviations.

3-3.4 Conclusions

(a) RTDs have hysteresis.

(b) Span adjustment and statistical grading of the quality of the fit without additional study can show a false improvement in a fit.

(c) Using the calendar eq. (3-3.2) would have prevented the false readings when the device was unplugged.

(d) RTDs were not plugged in during the test.

3-4 CAPACITY LOSS INVESTIGATION DUE TO FOULING OF FEEDWATER FLOW NOZZLE (NUCLEAR PLANT)

3-4.1 General

Nuclear power plant maximum capability can be affected by several components and systems. A logic tree may be constructed showing the effect on capacity of these various components and systems and their performance parameters.

3-4.1.1 Logic Tree

Figure 3-4.1.1-1 is the logic tree for investigating capacity losses for this case study (1100-MWe nuclear power plant). The tree has been divided into main areas listed below. Each area shows the constituent components/systems.

(a) Heat Source. The following should be noted regarding the heat source:

(1) The reactor/nuclear steam supply system (NSSS) thermal power cannot be measured directly.

(2) The NSSS thermal power is computed through an energy balance using measurements of

(a) main steam pressure and quality

(b) feedwater flow, pressure and temperature

(c) steam generator blowdown flow, pressure, and temperature.

(3) The reactor power is computed through an energy balance using

(a) the calculated NSSS thermal power

(b) recirculation pumps power

(c) reactor water clean-up system power

(d) radiation losses

(4) Typically, the biggest source of capacity loss associated with the heat source is inaccurate determination of the reactor/NSSS thermal power. Specifically, fouling of the feedwater flow nozzles causes a falsely high differential pressure indication. Consequently, both the calculated feedwater flow and NSSS thermal power will also indicate greater than reality. This leads to limiting the indicated reactor power at the set license level with attendant loss of capacity.

(b) Prime Mover

(c) Heat Sink

(d) Regenerative Cycle

(e) Other

The final level consists of performance parameters and other information that are measured, calculated, or estimated.

3-4.1.2 Decision Tree

Figure 3-4.1.2-1 shows the decision tree for investigating capacity loss associated with fouling of the feedwater flow nozzle.

(a) Compare the feedwater flow indication from the differential pressure measurement across the flow nozzle to the measured main steam flow, if available.

(b) If the measured main steam flow is unavailable or found to be lower than the feedwater flow indication, compare the measured high-pressure (HP) turbine first stage shell pressure to that expected for that flow rate.

(c) If the actual HP turbine first stage shell pressure is low, compare the measured generator output to expected for that reactor power level.

(d) If the measured generator output is low, fouling of the feedwater flow nozzles can be suspected.

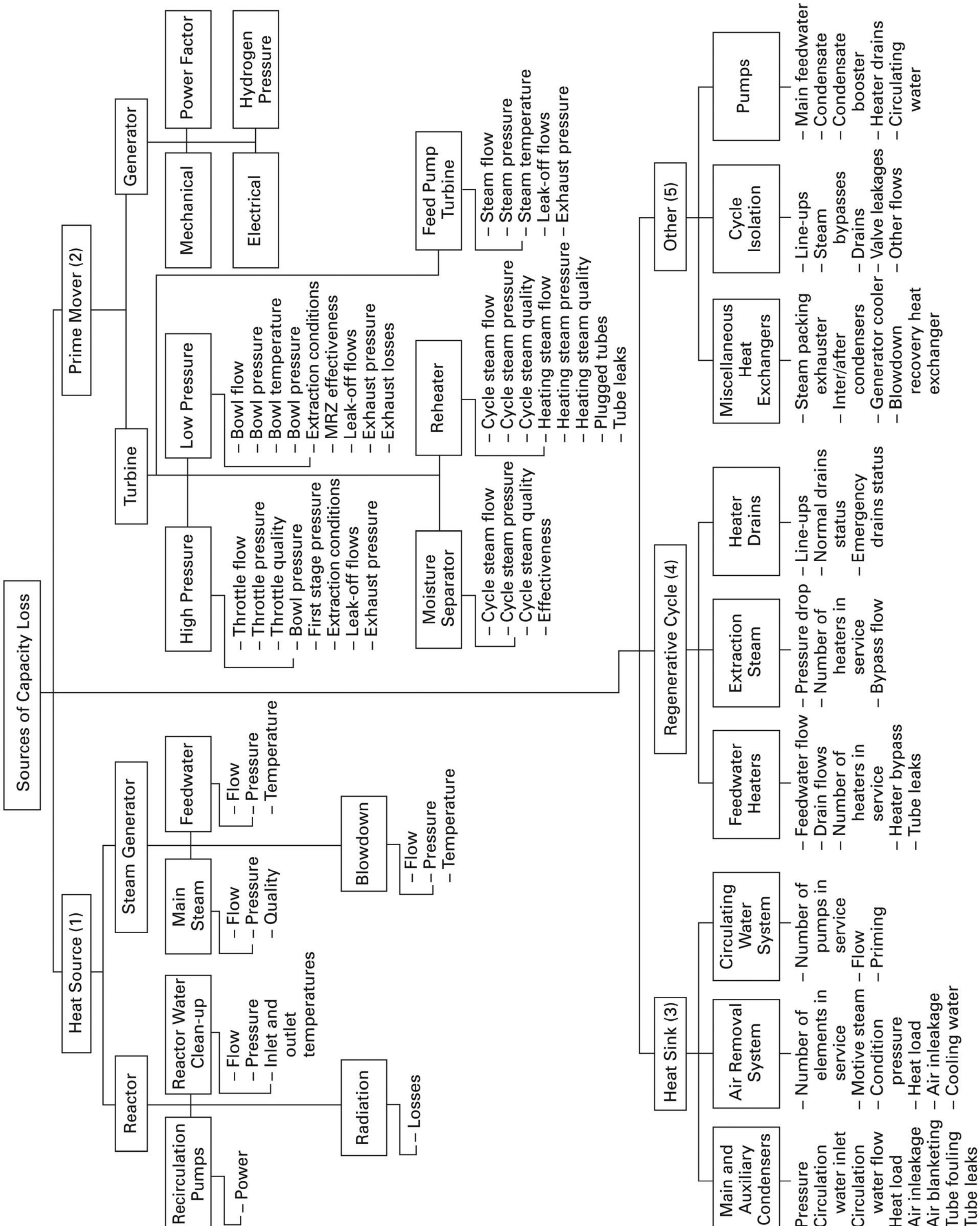
3-4.1.3 Heat Balance Diagram. Figure 3-4.1.3-1 shows the heat balance diagram for the case study generated using a performance modeling tool.

If fouling of the feedwater flow nozzle is suspected and the feedwater flow indication is higher than expected, for each 1% increase, the calculated key performance parameters will vary as follows:

(a) The reduction in main steam flow as well as condensate flow is approximately 1%.

(b) The reduction in final feedwater temperature is approximately 1.0°F.

Fig. 3-4.1.1-1 Logic Tree for Case Study: Capacity Loss Investigation (Courtesy Power and Energy Systems Services)



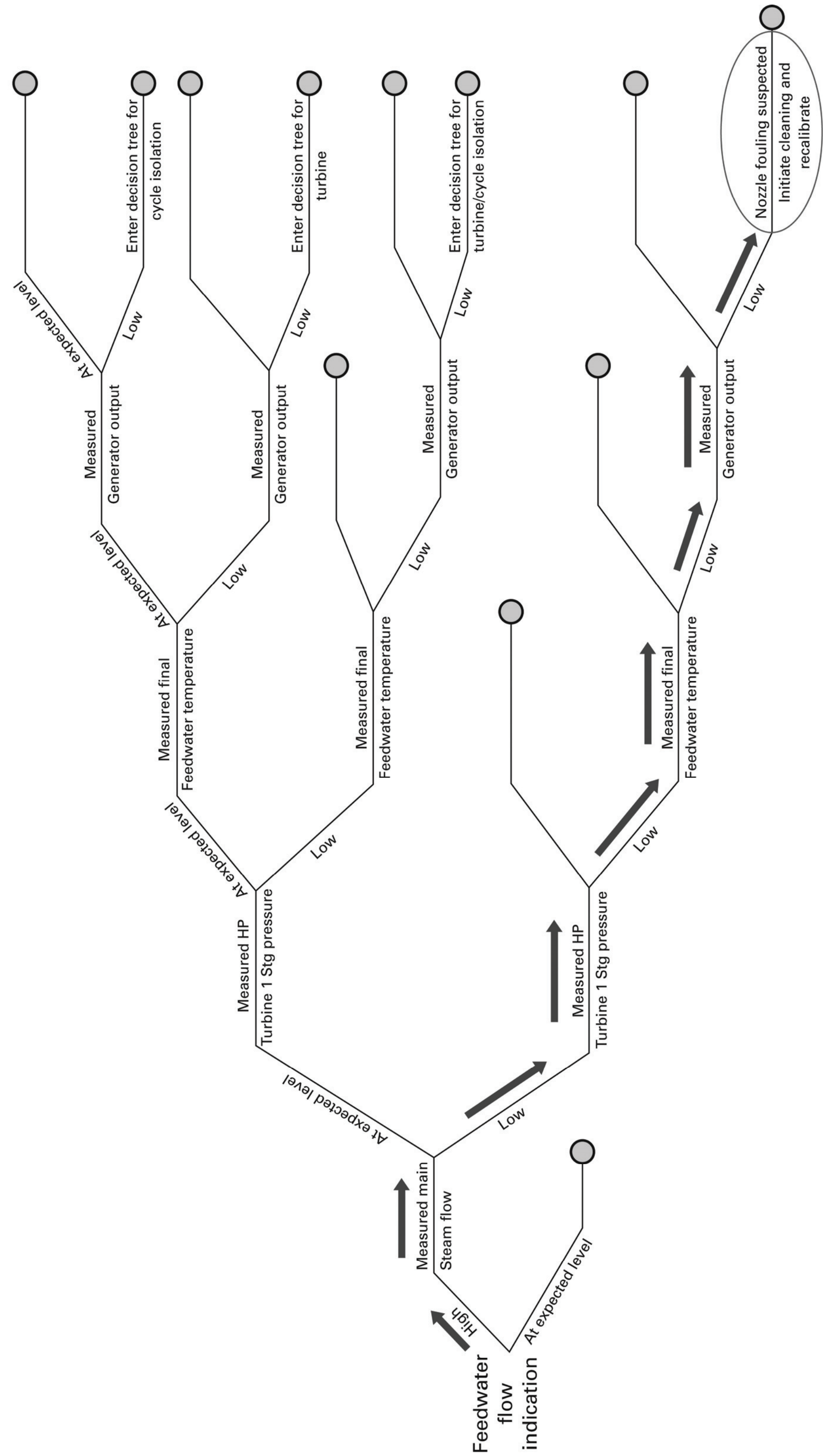
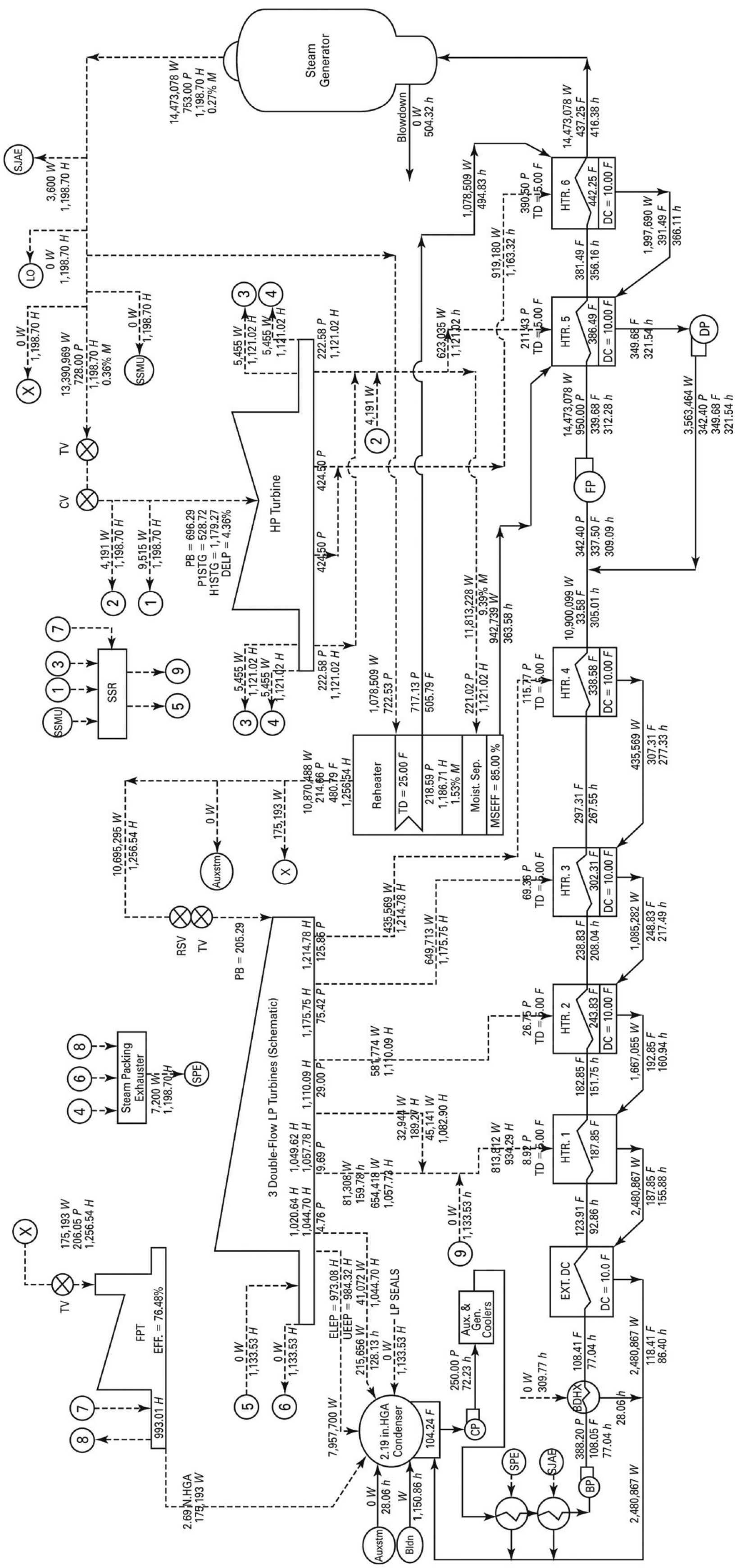


Fig. 3-4.1.2-1 Decision Tree for Capacity Loss Due to Suspected Fouling of the Feedwater Flow Nozzle (Courtesy Power and Energy Systems Services)

Fig. 3-4.1.3-1 Power Design Heat Balance for Case Study

NSSS power = 3,318.3 MWT
Generator output = 1,105.99 MW



(c) The reduction in HP turbine first stage shell pressure is approximately 6 psi (1%).

(d) The reduction in HP turbine exhaust pressure is approximately 2 psi (1%).

(e) The reduction in generator output is approximately 0.8% (9 MWe).

The measured parameters are main steam flow, feedwater flow from differential pressure measurement across the flow nozzle, final feedwater temperature, HP turbine first stage shell pressure, and generator output. Obviously, with a greater number of reliable performance parameters monitored, it will be easier to diagnose problems, quantify/verify the magnitude of the losses, take corrective action, and incorporate lessons learned/feedback into the performance monitoring program.

It can be noted from Fig. 3-4.1.2-1 that the capacity loss may be also due to other sources such as cycle isolation or component performance degradation, depending upon the values of the measured parameters in relation to their expected levels. The appropriate decision tree has to be entered to best determine the cause.

3-5 UNIT CAPACITY AND ID FAN CAPACITY DUE TO AIR HEATER LEAKAGE

3-5.1 Problem

In addition to being an example of how to diagnose this particular problem, the case outlined in this section highlights the need to verify root causes in order to take appropriate corrective action. It also demonstrates the potential to misdiagnose a problem with inadequate instrumentation.

In the summer of 2004, Unit 1 had to derate its maximum capacity by 15 MW due to the induced draft (ID) fan damper position nearing 100% open. Initially, the plant staff assumed the root cause to be air heater ash plugging. High air heater pressure drop had been a repeating problem in the recent past, requiring off-line water washes. Instead, this turned out to be a problem with air heater leakage.

3-5.2 Diagnosis

First, the assumed root cause (ash plugging) was disproved by studying the measured differential pressure across the gas side of the air heaters. A long-term trend showed that the summertime differential pressure was not much higher than normal, and certainly was lower than when water washing was required.

Next, other typical causes of ID Fan capacity limitations were considered, including air leakage into the flue gas. Total air leakage, as calculated[1] from the stack CEM data, had increased from 20% to an abnormal 45%. However, note that no leakage increase was seen in either of the Unit 1 air heaters, when calculated[2] from the air heaters' gas outlet oxygen probes. In fact, the Unit 1 air heater leakage appeared "normal" and slightly better than on the identical Unit 2, which did not have an ID Fan capacity problem. (See Table 3-5.2-1.)

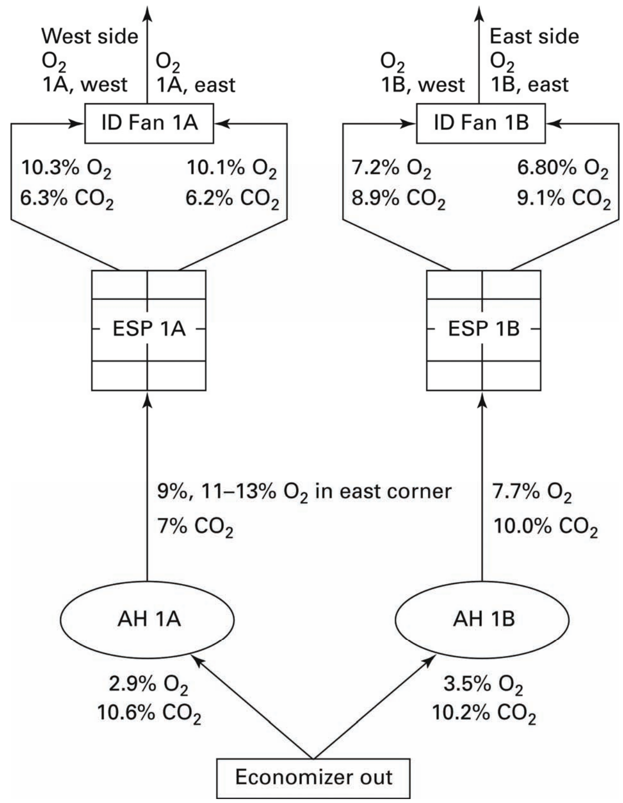
Table 3-5.2-1 Air Heater Leakage (Courtesy Black and Veatch Corp.)

Unit	Unit 1		Unit 2	
Air heater	AH 1A	AH 1B	AH 2A	AH 2B
Air heater leakage	**22%**	**26%**	**26%**	**31%**
Total leakage (from CEMS data)	**43%**		**29%**	

[1] The total leakage calculation compares the oxygen concentration at the economizer outlet versus the carbon dioxide concentration at the stack, where air leakage causes dilution of the carbon dioxide concentration.

[2] The air heater leakage calculation compares the oxygen concentration at the economizer outlet versus the oxygen concentration at the air heater gas outlet. A simplified air heater leakage calculation is available in ASME PTC 4.3

Fig. 3-5.2-1 Flue Gas Analyzer Measurements at Locations Along the Gas Path (Courtesy Black and Veatch Corp.)



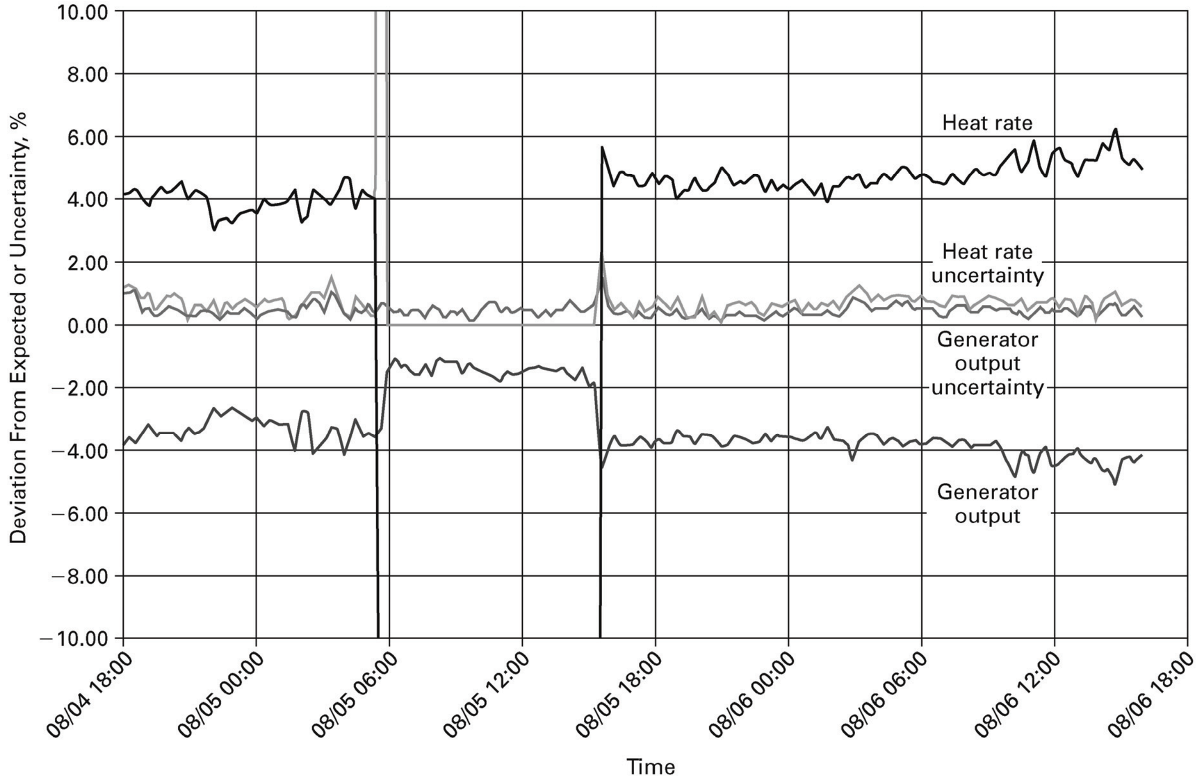
Therefore, the Unit 1 ID Fan capacity problem seemed to be narrowed down to a large air leak located somewhere in Unit 1 flue gas path, downstream of the air heaters but upstream of the ID Fans. That left a rather large physical area to investigate for the leakage. No permanent instrumentation was available to further locate the leak, but sample ports were available to perform manual traverses with a portable flue gas analyzer, at various locations along the flue gas path. Measurements were taken at variable depths at each sample-port location, with resulting averages as shown on Fig. 3-5.2-1. Elevated oxygen in the traverse readings clearly indicate the air leak was located on the West side A rather than the East side B. They also indicate the air leak was actually located inside the Air Heater 1A, because the high oxygen readings were found upstream of the ESP, albeit very stratified in one corner of the duct.

These are trisector regenerative air heaters, so each air heater has both secondary air and primary air flowing through it. Long-term trends showed that the PA fan power had increased in that time, while the FD fan power had remained constant. Therefore, the air leakage appeared to be from the high pressure primary air into the lower pressure flue gas.

3-5.3 Resolution

Internal inspection during a short unit outage found a large corrosion hole in the plenum wall separating the primary air from the flue gas. Repairing that hole restored the leakage and the unit capacity to normal.

Fig. 3-6.3-1 Generator-Output and Heat Rate Deviation (Courtesy Encotech, Inc.)



3-6 LOSS OF EXTRACTION FLOW

3-6.1 Summary

During normal operations, a performance monitoring system tracks key performance indicators and alerts the operations staff to changes that may need to be investigated. In this case, the investigations led to the identification of an inadvertently closed valve, which was negatively impacting unit heat rate and capacity.

3-6.2 Key Performance Indicators

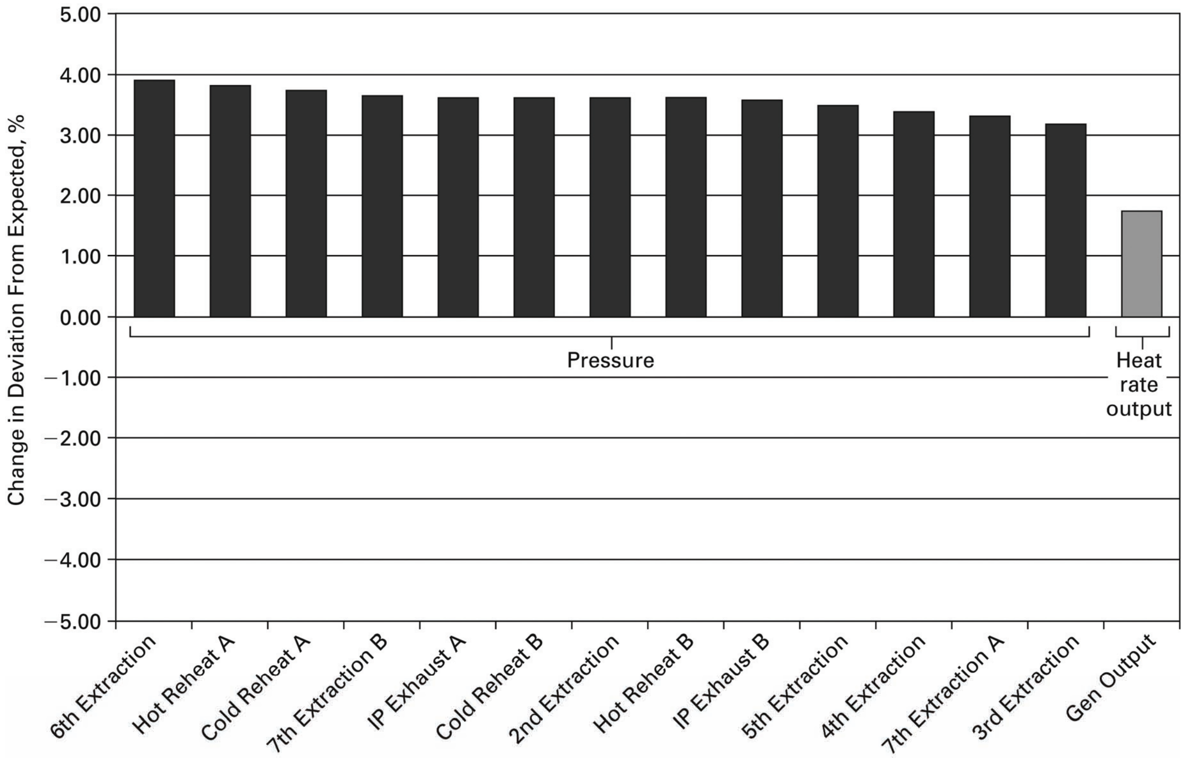
The key performance indicators during this investigation were the indicated deviations in heat rate and output from expected values at the current operating conditions.

3-6.3 Initial Observations and Corroborating Evidence

During normal operating conditions of a 650 MW unit, the performance monitoring system indicated a step change in both heat rate and generator output deviation from expected. The value for generator output deviation from expected changed in a step-wise fashion by more than two percentage points, and the heat rate increased to a level above the range of the graph, as shown in Fig. 3-6.3-1.

The performance monitoring program was also able to develop a profile to show the significant changes in parameters of interest from before to after the step change in performance. The profile developed is shown in Fig. 3-6.3-2.

Fig. 3-6.3-2 Change in Performance Profile Over Significant Cycle Positions (Courtesy Encotech, Inc.)



As shown in Fig. 3-6.3-2, all the turbine pressures, with the exception of first stage shell pressure, increased approximately the same amount as the unit output dropped. This was originally thought to be due to a change in extraction flow to the highest pressure heaters. The control room was contacted, and it was revealed that an operator had been sent to tag out the level control on the B high pressure heater, but nothing had been done to change extraction flow.

Further examination revealed that the feedwater temperature leaving the B high pressure heater was about the same as the inlet temperature, which again indicated that this heater was not receiving adequate extraction steam flow. A second call to the control room was made, and a physical inspection of the location where the level control had been tagged out revealed that the operator tagging out the level control had inadvertently tripped the extraction line nonreturn valve closed that had shut off the extraction flow to the B heater. Reopening of the nonreturn valve restored the system to normal operation.

3-6.4 Capacity, Efficiency, and Reliability Impacts

The loss in extraction flow to the high pressure heater caused significant changes in measured heat rate, as was indicated by the performance monitoring system, and shown on the trace in Fig. 3-6.3-1. The performance monitoring system also indicated the loss of the extraction to the B side high pressure heater resulted in a more than a 2% loss in unit capacity.

3-7 QUESTION AND ANSWER SESSION: A NUCLEAR PLANT DIAGNOSTIC PROBLEM

Question (1): Can you describe the project briefly?

Answer (1): They had a puzzling situation. During the last operating cycle (since the last refueling) the control system had been gradually opening the turbine control valves, and the generator output was increasing. This, of course, could be explained if the thermal power was increasing, since this would be the normal reaction. However, all the measurements of thermal power indicated that it was remaining constant.

The amount we are talking about over a roughly 9-mo period was less than 0.5%, which does not sound like much. These people, however, are very conscientious about being sure that they are measuring thermal power correctly and that they are not exceeding their authorized output. They had convened a special task force to examine their procedures for calculating thermal power that had recalibrated instruments, reviewed calculation procedures, examined computers used in the processing of data, etc.; and still could find nothing wrong with their procedures for measuring reactor thermal power. The only possible conclusion seemed to be that in some manner the secondary cycle efficiency was improving, but exactly how was unclear.

Question (2): What was your role in this project?

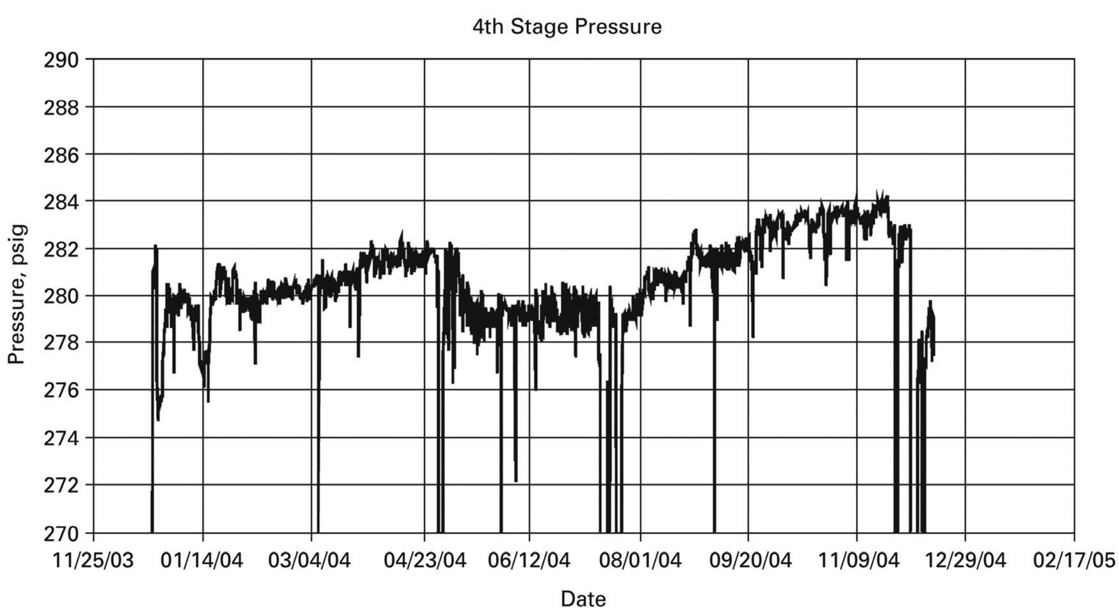
Answer (2): They asked me to review their evaluation to see if I could find any errors in the logic, and also to determine if there was some logical explanation for why the secondary cycle efficiency was improving.

Question (3): Did you identify any problems with their analysis and/or measuring techniques?

Answer (3): No; their analysis looked pretty sensible to me and so I decided to try and identify what might be going on in the secondary cycle.

Engineers at the plant had noticed that there were some unexplained changes in the various turbine pressures — first stage, extraction, casing exhaust, etc. An example for the fourth stage extraction is shown in Fig. 3-7-1. Since the plant runs at a constant thermal power, you would normally expect these pressures to be pretty constant.

Fig. 3-7-1 Variations of Fourth-Stage Pressure (Courtesy Encotech, Inc.)



A closer look at the first stage shell pressure and the second and sixth stage extraction pressures showed similar patterns, with the deviations from normal being greater on the fourth and sixth stage extractions than the higher pressure locations.

The next question is what could be causing these pressures to vary in the manner shown. One possibility is water-soluble deposits. The rises in pressure are gradual, which is typical of deposits. There were also some short-term shutdowns that could have resulted in washing some of the deposits off and would explain the pressure reductions that occurred. Deposits, of course, are not known for improving cycle efficiency, but maybe there were some offsetting phenomena taking place that were not yet obvious.

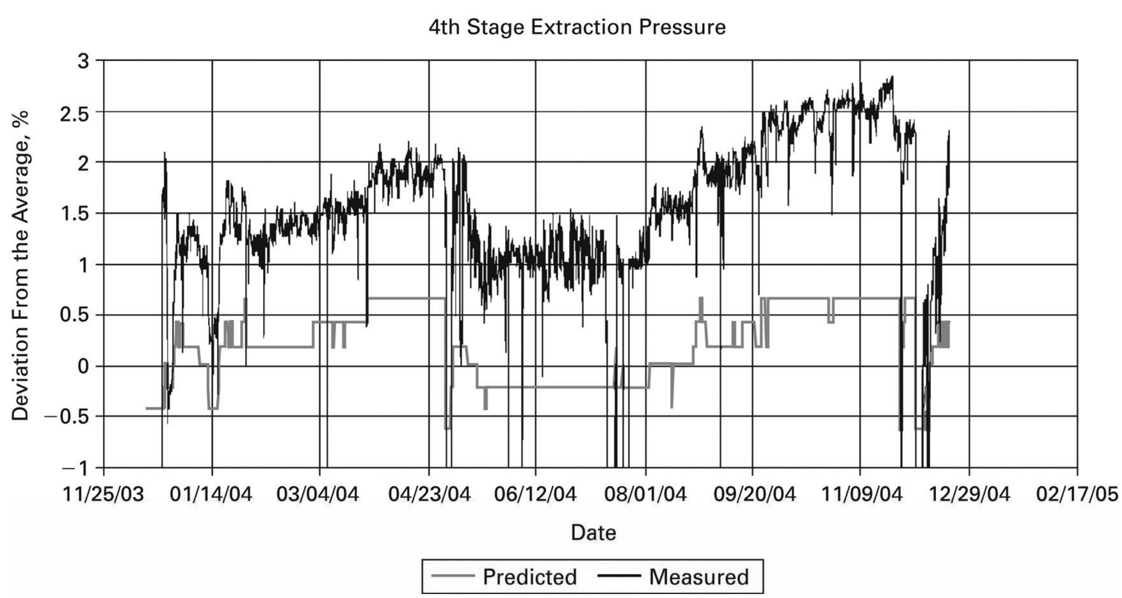
Fortunately, I was able to participate in a meeting with plant personnel that covered a wide range of subjects and performance data. A graph that was shown during the meeting plotted the number of moisture separator drain pumps that were in service during the current operating cycle. It immediately caught my attention because it showed a pattern that looked a lot like the pattern of variations for the various turbine pressures.

Question (4): Please tell us what moisture separator drain pumps have to do with turbine pressures; and why were they turning them on and off anyway.

Answer (4): Let's take your questions one at a time. There are six moisture separators at this installation, and the drains flow to drain tanks and then are pumped to the heater fed by the sixth stage extraction. These drains are relatively hot and so, when these drain pumps are operating, this thermal energy helps to heat the feedwater flowing through the heater and reduces the need for extraction flow from the turbine. This raises the extraction pressure and sends higher temperature feedwater on to the next higher pressure heater, which in turn, needs less extraction flow since the feedwater coming to it is hotter. This effect attenuates as it proceeds to higher pressure heaters but nevertheless is felt all the way up to the first stage pressure, which is only one stage above the highest pressure extraction.

I was able to make use of our program to verify that turning these pumps on and off would have the expected effect on the turbine pressures. I ran the program with various numbers of the pumps on and off and then plotted the expected changes in pressure during the times that records showed various numbers of pumps running. That graph is shown next for the fourth stage extraction in Fig. 3-7-2.

Fig. 3-7-2 Similarities Between Predicted and Measured Pressure Changes (Courtesy Encotech, Inc.)



As you can see, there is a strong similarity between the predicted and measured pattern of pressure changes. Note that the average pressure for the measured values was a little lower than for the predicted ones because it included short periods when the unit was at low loads or temporarily shut down. This is the reason for the general shift in "deviation from the average" values for the measured as compared with the predicted values.

Question (5): Why doesn't the plant just leave these pumps running instead of causing all this confusion?

Answer (5): When the pumps are running the drain flow enters the sixth stage heater and the drains from this heater are pumped into the main feedwater stream; If the pumps are shut down the moisture separator drains flow directly to the condenser. The plant was experiencing some water cleanup problems during this operating cycle and letting the drains flow to the condenser sends them through the condensate polisher more quickly and speeds up the water clean-up process.

Question (6): One last question: Early on you said that the control valves were gradually creeping open and the plant power output was increasing. Does turning these pumps on and off explain those phenomena as well?

Answer (6): Yes it does. First of all, note that, while pumps were turned on and off, there was a general trend during the operating cycle to increase the number of pumps in service and for longer periods of time. As more pumps are put in service, the temperature of the feedwater leaving the heaters goes up, including the final feedwater temperature from the highest pressure heater. This increase in final feedwater temperature tells the thermal power monitor that the thermal power is going down and so it opens the control valves to bring operation back up to the licensed thermal power.

Another way to look at it is to consider the change from letting the moisture separator drains go to the condenser as compared with pumping them to the sixth stage heater. The drains from the moisture separator are relatively hot. If they flow to the condenser that thermal energy is lost to the cycle because it is absorbed by the condenser cooling water. On the other hand, pumping the drains to the heater uses that thermal energy for heating feedwater and improves the efficiency of the secondary cycle, resulting in more generator output for the same thermal power input to the cycle.

Question (7): Does this conclude the analysis satisfactorily?

Answer (7): I think it does, as far as the impact of turning the moisture separator drain pumps on and off. However, there is still a very slight drift in control valve movement that needs some further investigation to understand. It is a low level priority, however, and looks like it may have to wait for a lowering of concern for other problems before it will get much attention.

3-8 APPLICATION OF TURBINE TEST DATA FOR PROBLEM IDENTIFICATION

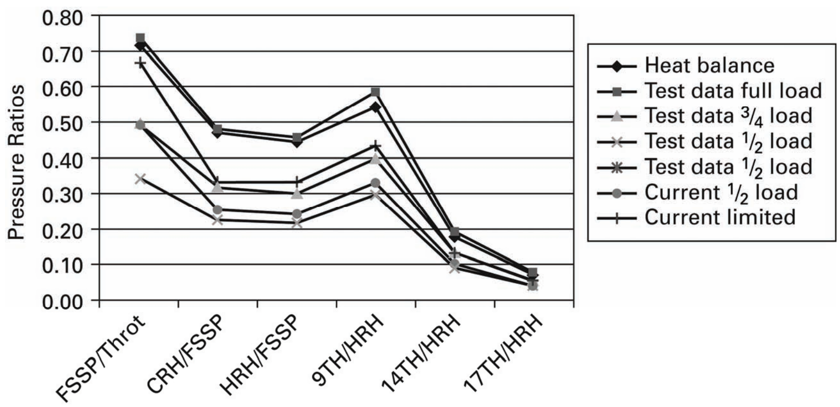
3-8.1 Summary

A 40-yr-old unit, which is frequently cycled, became load limited. A preliminary investigation led to an outage for a more in-depth inspection of the steam turbine. The results of that inspection indicated a need to replace the first stage buckets in order to regain lost performance.

3-8.2 Key Performance Indicators

The key performance indicator in this case was the maximum available output of the unit, as compared to the heat balance expectations. Pressure ratios across each steam turbine section were also used to narrow down the location of the lost performance within the unit.

Fig. 3-8.3-1 Turbine Pressure Profiles



3-8.3 Initial Observations and Corroborating Evidence

This case concerns a 40-yr-old unit that is frequently cycled and used in load-following operation. The unit suddenly became load limited. Maximum output was reduced to approximately 80% of normal capacity when the turbine values were wide open. There were no observed increases in any bearing vibrations.

To determine the cause of this lost capacity, the current turbine pressure profile was charted relative to the design heat balance profile and the profiles measured during new and clean test operation at various load points. The pressure profiles created are shown in Fig. 3-8.3-1.

As shown in Fig. 3-8.3-1, the two curves established from current data do follow the design curve shape for all points except for the first stage shell pressure point. This is indicative of a closing of the first stage flow area, restricting steam flow through that point.

3-8.4 Physical Inspection

The turbine was opened for inspection and troubleshooting. The first stage buckets were found to be severely damaged, closing the flow area by approximately 50%.

3-8.5 Capacity, Efficiency, and Reliability Impacts

After the buckets were replaced, the unit was able to again reach full power, an improvement in maximum capacity of approximately 20%.

3-9 CONDENSER TUBE FOULING PROBLEM

3-9.1 Problem

Plant A consists of two identical 600 MW units with multipressure condensers. During normal operation with both units at 600 MW and one vacuum pump running, it was observed that the condenser pressure on the Unit 2 low-pressure (LP) shell was 3.20 in. HgA compared to 2.10 in. HgA for the Unit 1 LP shell. Similarly, the condenser pressure for the high-pressure (HP) shell was 3.80 in. HgA for Unit 2 versus 2.90 in. HgA for Unit 1. Both units take their inlet circulating cooling water suction from the same source, which was at 80°F. With the inlet water temperature increasing as the summer months approached (up to 90°F), plant engineering initiated a study to determine the cause of the higher condenser pressure on Unit 2 versus Unit 1.

3-9.2 Procedure

Consideration was given to the following possible reasons for the much higher condenser pressure on Unit 2:

(a) inaccurate condenser pressure readings

(b) air binding due to poor vacuum pump performance

(c) air inleakage

(d) fouled condenser tubes

The item in (a) was ruled out, since these instrument transducers had just been recalibrated. For the item in (b) the accuracy of the vacuum pump gauges at the pump proper and at the condenser (HP shell) was found to be correct, and the vacuum at the pump was lower than the vacuum at the condenser. (The reverse would indicate a possible restriction in the vacuum-pump suction line.) To assess vacuum pump performance, the seal water inlet temperature was checked and found to be approximately 5ºF higher than the temperature of the cooling water to the cooler (70°F versus 65°F on this day). This indicated adequate cooling of the seal water for proper operation. To confirm adequate pump performance and check for air inleakage [the item in (c), a second vacuum pump was put into service to see if it reduced the condenser vacuum significantly. It did not as the LP shell pressure decreased only 0.05 in. HgA, and the HP shell pressure remained virtually unchanged.

The item in (d) was investigated by first observing the circulating water pumps' discharge pressures on Unit 2, which were 11.0 psig versus 9.0 for Unit 1. Three years earlier, the Unit 2 discharge pressure had averaged 8.0 psig subsequent to a condenser tube cleaning. The increased discharge pressures therefore indicated a larger restriction to flow than was evident after the tube cleaning 3 yr earlier. Thus, the next step involved the verification of tube fouling by calculating the actual "cleanliness factors" for both the LP and HP shells.

3-9.3 Calculations

(a) LP Condenser Shell Cleanliness Factor

Given: Tube Material: 90-10 CuNi
Tube Gauge: 19
Tube OD: 1.0 in.

Np (No. of tube passes in condenser)	= 1.0
Nt (No. of condenser tubes)	= 13,416
Vt (Flow per tube @ 1 fps – HEI Table)	= 2.065 GPM
C (Constant C – HEI Table)	= 263
Ti (Inlet circulating water temperature)	= 80.0°F
To (Outlet circulating water temperature)	= 92.0°F
Tr (LP shell temperature rise)	= 12.0°F
G (Circulating water flow rate – design/test data)	= 188,400 GPM
Cp (Specific heat capacity of fresh water)	= 1.0 Btu/lb-°F
A (Total condensing tube area)	= 120,000 ft^2
BP (Condenser shell pressure)	= 3.20 in. HgA

Solve:

Tsat (Saturation temperature of condenser @ BP)	= 117.1°F
Z	$= \ln [(Tsat - Ti)/(Tsat - To)]$

LMTD (Log mean temperature difference)	= *Tr/Z* (°F)
LMTD	= 12.0/ln [(117.1 - 80.0)/(117.1 - 92.0)]
LMTD	= 30.71°F
ρ (Density of circulating water flow)	= 62.162 lb/ft^3
Vel (Tubeside velocity) – (*G* × *Np*)/(*Nt* × *Vt*) (FPS)	
Vel	= (188,400 × 1.0)/(13,416 × 2.065)
Vel	= 6.80 FPS
Ui (Ideal heat transfer coefficient)	= C √ (Vel) (Btu/hr-ft^2-°F)
Ui = 263 × √ (6.80)	= 685.8 Btu/hr-ft^2-°F
Q (Condenser heat load) (Btu/hr)	= *G* × *Tr* × (60/7.48) × ρ × *Cp*
Q	= (188,400) × (12.0) × (60/7.48) × (62.162) × (1.0)
Q	= 1,127.3 × 10^6 Btu/hr
Ud (Design heat transfer coefficient)	= *Q* / (*LMTD* × *A*) (Btu/hr-ft^2-°F)
Ud	= (1,127.3 × 10^6)/(30.71 × 120,000)
Ud	= 305.9 Btu/hr-ft^2-°F
Ct (Inlet water temperature correction factor – HEI Table)	= 1.04
Cm (Tube material correction factor – HEI Table)	= 0.935 (19 gauge, 90-10 CuNi)
CF (Cleanliness Factor) = *Ud* / (*Ui* × *Ct* × *Cm*)	
CF	= (305.9)/[(685.8) × (1.04) × (0.935)]
CF	= 0.459
CF (%)	= 45.9%

(b) HP Condenser Shell Cleanliness Factor

Given: Same data as LP shell with following changes:

Ti	= 92.0°F
To	= 103.0°F
Tr	= 11.0°F
BP	= 3.73 in. HgA

Solve:

Tsat	= 122.9°F
Z = ln [(122.0-92.0)/(122.9-103.0)]	= 0.44
LMTD = 11.0/0.44	= 25.0°F
ρ	= 62.035 lb/ft^3
Ui (same as LP Shell)	= 685.8 Btu/hr-ft^2-°F
Q	= (188,400) × (11.0) × (60/7.48) × (62.035) × (1.0)
Q	= 1,031.2 × 10^6 Btu/hr
Ud = (1,031.2 × 10^6)/(25.0 × 120,000)	= 343.7 Btu/hr-ft^2-ºF
Ct	= 1.09
CF	= (343.7)/[(685.8) × (1.09) × (0.935)]
CF	= 0.492
CF (%)	= 49.2%

3-9.4 Conclusion

The above calculations for LP and HP shell cleanliness factors of 45% and 49%, respectively, indicated that the tubes were severely fouled. As a result, both waterboxes were opened shortly thereafter and shot with scapers. The dirty, slimy residue confirmed the suspected fouling, and the “after” cleaning results indicated cleanliness factors of 85% and 82%, respectively.

Thus, Unit 2 recovered almost 1.0 in. HgA in lower condenser pressure from the tube cleaning.

3-10 FEEDWATER PARTITION-PLATE BYPASS PROBLEM

3-10.1 Problem

Plant B, Unit 1 consists of four high-pressure feedwater heaters: 6A, 6B, 7A, and 7B. During a test conducted in 1995, it was discovered that the final feedwater temperature was 5°F below the value determined during a similar test in 1989 (see Table 3-10.1-1). Further analysis revealed elevated terminal temperature difference (TTD) and reduced tubeside temperature rises for the 7A and 7B feedwater heaters. Plant engineering thus initiated a study to determine the cause of the elevated TTDs and reduced final elevations in feedwater temperature and in tubeside temperature.

3-10.2 Procedure

Consideration was given to the following reasons for these off-design conditions:

(a) inaccurate temperature/pressure readings

(b) air binding due to plugged operating air vents

(c) leaking feedwater heater tubes

(d) partition plate bypassing

Table 3-10.1-1 Test Results of Four High-Pressure Heaters

Heater	1989 Test	1995 Test	Design
Heater 7A TTD, °F	–2.2	3.3	0.0
Heater 7A DCA, °F	11.1	12.3	10.0
Heater 7A temperature rise, °F	63.5	56.9	65.0
Heater 7A level	Normal	Normal	Normal
Heater 7B TTD, °F	0.8	3.1	0.0
Heater 7B DCA, °F	12.3	9.1	10.0
Heater 7B temperature rise, °F	59.3	52.7	65.0
Heater 7B level	Normal	Normal	Normal
Final feedwater temperature, °F	491	486	490
Gross load, MW	508	502	500

The item in (a) was ruled out after the test instrumentation was recalibrated and found to be accurate. The item in (b) was ruled out after checking the operating air-vent lines and orifices and determining their unplugged condition. Also, since the DCA for the 7A/7B heaters was about 10ºF (design), excessive subcooling was not occurring, as is the case with air binding (low DCAs and high TTDs and temperature rises). The item in (c) was ruled out as a possible cause since the heaters had no trouble maintaining normal water levels without needing the alternate drains to open to the condenser. Also, low DCAs were not occurring, as is the case with leaking tubes (excessive subcooling).

To determine whether the partition plate separating the inlet tubesheet from the outlet tubesheet was leaking, the heater head had to be unbolted, and the welded seal door plate had to be gouged out for removal. It was decided to open the 7B heater first since it had the lowest temperature rise across the tubes (52.7°F versus 65.0°F design). Upon opening, it was discovered that several welds in the welded door plate had broken loose. Also, each lower corner of the partition plate had a hole that had eroded through the plate. After welding up these holes, the 7A heater was opened. Several cracks in welds were also discovered, which were also welded shut.

3-10.3 Conclusion

Verification of the suspected problem-causing situation was thus made after the heaters were opened. Although possible stratification of the outlet feedwater temperature from each heater threw some doubt onto the validity of the data, concurrent data on a similar unit (under the same conditions) supported the diagnosis of the problem.

3-11 AIR-HEATER PLUGGAGE PROBLEM

3-11.1 Problem

Plant C, Unit 1 consists of two regenerative air heaters. After an outage to clean the economizer gas passage of ash buildup, it was observed that the gas temperature leaving (corrected) the air heaters had increased by 17°F. With hot weather approaching, and the possibility of ID-fan limitations (from lower flue-gas density), and thus a possible derating, plant engineering initiated a study to determine the cause of the increased exit gas temperature.

3-11.2 Procedure

Consideration was given to the following possible reasons for the higher exit gas temperature:

(a) faulty temperature sensor

(b) incorrect air heater leakage

(c) higher gas temperature leaving the economizer

(d) decreased air heater efficiency

After replacing several of the thermocouple probes in the exit gas temperature grid, it was determined that the observed readings were correct. Air-heater leakage was then calculated to be 5.0%, according to the following formula:

$$\%\text{Leakage} = \frac{(\%O_2 \text{ Gas Entering A.H.} - \%\ O_2 \text{ Gas Leaving A.H.})}{(\%\ O_2 \text{ Gas Leaving A.H.} - 20.9)} \quad (90)$$

where

$\%O_2$ gas entering = 3.8%
$\%O_2$ gas leaving = 4.7%

This was determined to be correct as the leakage prior to the outage was 6.0% and most radial seals were replaced during the outage.

The item in (c) was dismissed as a possibility, since the economizer's heat absorption increased from the gas side cleaning, resulting in a lower gas temperature entering the air heater.

To determine the item in (d), the air-heater gas side efficiency was calculated as 60.4% according to the following formula:

$$Ng = [(Tgi - Tgoc)/(Tgi - Tai)]\ (100\%)$$

where

Ng	= gas side efficiency (%)
Tai	= air-heater inlet-air temperature, 95°F
Tgi	= air-heater inlet-gas temperature, 651°F
$Tgoc$	= air-heater gas-outlet temperature, corrected for no leakage, 315°F

$Tgoc$ was calculated according to the equation:

$$Tgoc = [(L) * Cpa * (Tgo - Tai)/(Cpg * 100)] + Tgo$$

where

Cpa	= specific heat of air, 0.239
Cpg	= specific heat of gas, 0.241
L	= % air heater leakage, 5.0%
Tgo	= measured air-heater gas-outlet temperature, 305°F

Prior to the outage, the gas side efficiency had been 63.5%. Thus the 3.1% point decrease in air-heater efficiency was the reason for the 17°F increased air-heater exit-gas temperature. It was also observed that the gas-side pressure drop (average of both air heaters) had increased from 7.2 in. H_2O prior to the outage to 12.3 in. H_2O.

3-11.3 Conclusion

Air-heater pluggage was responsible for the increase in exit-gas temperature, and was probably due to ash carryover from the cleaning of the economizer. At the next window of opportunity, the air heaters were washed, and the resulting pressure drops decreased to an average of 6.5 in. H_2O. Also, the air heaters' gas-side efficiencies increased to an average of 64%, with the exit-gas temperature decreasing by 20°F. Therefore, a heat-rate savings of about 70 Btu/kWh was realized.

3-12 DEPOSITS IN HIGH-PRESSURE TURBINE

3-12.1 Problem

Trending of turbine performance data at Plant D, Unit 2 over a 2-yr period revealed the following changes in performance:

Load:	–7.8%
First-stage pressure (corr):	–3.5%
Hot-reheat pressure (corr):	–4.5%
Crossover pressure (corr):	–4.3%
HP turbine efficiency:	–6.5%
IP turbine efficiency:	–0.6%

Table 3-12.2-1 Reconciliation of Load Change Based on Change in Performance Parameters

Performance Parameters	Change in Performance Parameter	Effect on Load
Change in flow	–4.5% × 0.94 =	–4.2%
HP turbine condition	–6.5% × 0.27 =	–1.76%
IP turbine condition	–0.6% × 0.15 =	–0.15%
Total KW accounted for		–6.11%

With a scheduled boiler outage of 4 weeks' duration coming up, plant engineering was asked by plant management if opening the turbine for an inspection, etc. was warranted.

3-12.2 Procedure

Based on the large decrease in HP turbine efficiency (and therefore degradation), it is best not to rely on the first stage pressure as a good indicator of flow. Since the hot reheat and crossover-pressure changes are in good agreement, these should be indicative of flow changes. Reconciliation of the load change will then be accomplished using the hot reheat pressure since any error in pressure measurement will have less uncertainty due to the higher level of pressure (see Table 3-12.2-1).

The "0.94" value is the percentage change in output for a 1% change in flow. A 1% change in HP efficiency will result in a 0.27% change in output, while a 1% IP efficiency change will change the output by 0.25% (for this specific turbine).

The reconciliation accounts for about 80% of the change in load. The remaining degradation is most likely in the LP turbine or the steam cycle (isolation, etc.). The main problem is in the HP turbine, and the decrease in load and pressure indicates the problem is a flow restriction. Thus deposits are indicated in the HP turbine.

Plant engineering subsequently recommended opening the HP turbine during the upcoming outage to remove the deposits.

3-12.3 Conclusion

The HP turbine, when opened, was found to be coated with heavy deposits, especially on the first stage. The deposits were analyzed and found to be copper carryover from the high-pressure feedwater heaters. A program to monitor more carefully the water chemistry was subsequently implemented by plant management.

After cleaning the turbine and returning to service, post-outage test data showed an increase in HP turbine efficiency of 5.5% and a load increase of 6.9%. Thus, a heat-rate improvement of 90 Btu/kWh was realized as a result of the deposit removal.

3-13 PULVERIZER COAL-MILL FINENESS PROBLEM

3-13.1 Problem

Plant E, Unit 3 consists of five MPS pulverizers. Coal fineness after a recent boiler outage (in which mill work was performed) averaged only 58% (passing 200 mesh) versus 72% prior to the outage. Plant engineering initiated a study to determine the reason for the low values.

3-13.2 Procedure

Consideration was given to the following sources of incorrect coal-fineness levels:

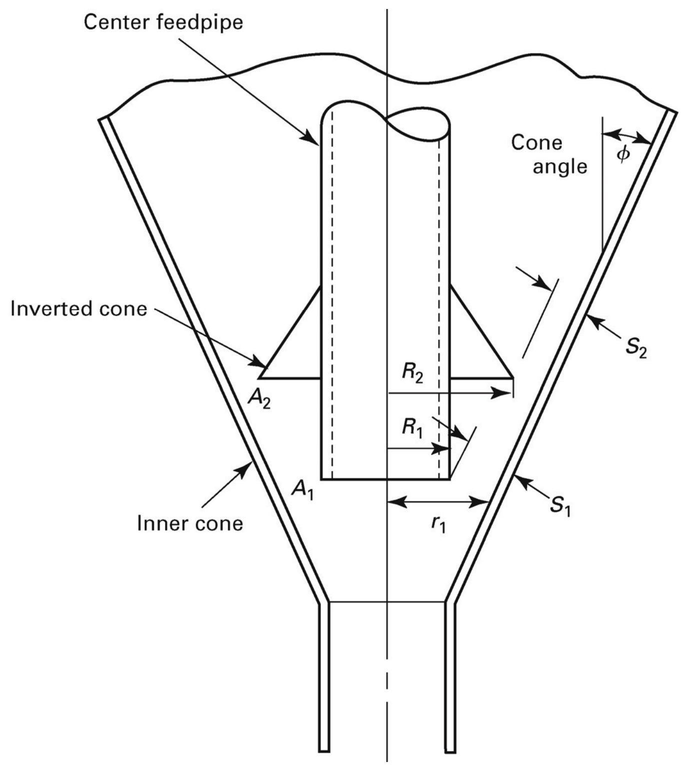
(a) classifier blades set in wrong position

(b) classifier blade assembly out of calibration

Table 3-13.3-1 Measurements Taken at the Outage

	3A Mill	3B Mill	3C Mill	3D Mill	3E Mill
Cone angle, ϕ, deg	37	37	37	37	37
Inverted cone gap, S_2, in.	2.9	4.8	5.3	3.9	4.0
Feedpipe gap, S_1, in.	4.9	6.5	7.0	5.9	6.1
Inverted cone outer gap, ($R_2 - R_1$), in.	7.0	7.0	7.1	7.0	7.3
Feedpipe diameter, ($2*R_1$), in.	16	16	16	16	16

Fig. 3-13.3-1 Adjusted Inverted Cone



(c) holes worn through inner cone

(d) incorrect inverted cone/inner-cone and feedpipe/inner-cone gap clearances

The items in (a) and (b) were ruled out since new classifiers were installed, calibrated, and set to pre-outage positions. Inspection of the mills during the outage revealed no holes in the inner cones. Post-maintenance records, however, did indicate that coal-pipe lengths had been shortened and/or lengthened, and some inverted cones had been trimmed to a shorter diameter. Consequently, it was suspected that unequal inverted cone/feedpipe areas were responsible for the lower fineness levels.

3-13.3 Calculations

At the next outage opportunity, each mill was opened, and the measurements in Table 3-13.3-1 were taken (see Fig. 3-13.3-1):

Table 3-13.3-2 Calculated Cone and Feedpipe Areas

	3A Mill	3B Mill	3C Mill	3D Mill	3E Mill
Inverted cone area, A_2, in.^{2}	290	504	570	402	423
Feedpipe area, A_1, in.^{2}	305	433	475	380	402

Table 3-13.3-3 Resulting Gap Clearances and Areas

	3A Mill	3B Mill	3C Mill	3D Mill	3E Mill
$(R_2 - R_1)$, in.	7.0	7.0	7.0.	7.0	7.0
R_2, in.	15.0	15.0	15.0	15.0	15.0
S_1, in.	7.0	7.0	7.0.	7.0	7.0
A_1, in.^{2}	475	475	475	475	475
A_2, in.^{2}	475	475	475	475	475
S_2, in.	4.5	4.5	4.5	4.5	4.5

Using these measurements, the following inverted cone areas (A_2) and feedpipe areas (A_1) were calculated and are shown in Table 3-13.3-2.

As shown by these values, only the 3A-mill areas agreed within the ±5% tolerance as recommended by the manufacturer. The 3B and 3C mills were more than 15% out of agreement. Also, for the manufacturer's recommended S_1 distance of 7.0 in. and S_2 distance of 4.5 in. (for a 30 in. diameter inverted cone), the equal areas of 475 in.^{2} ($A_1 = A_2$) only existed for the feedpipe area on the 3C mill.

To correct these gap measurements, all feedpipes were centered in the inner cone to within 0.25 in. Then based on the measured inverted cone outer-gap measurement $(R_2 - R_1)$, R_2 was calculated. With S_1 set to 7.0 in. and knowing R_1, area A_1 was calculated. Setting A_2 equal to A_1 and knowing R_2, S_2 was calculated. The resultant gap clearances and areas that were set are as in Table 3-13.3-3.

3-13.4 Conclusion

After returning the unit to service, coal fineness levels averaged 75% through 200 mesh screen, and loss-on-ignition (LOI) levels decreased by an average of 2% points. A heat-rate savings of 20 Btu/kWh was realized, and the operation of the mills improved significantly.

NONMANDATORY APPENDIX A
THERMODYNAMICS FUNDAMENTALS

Thermodynamics is the study of energy transformations between heat and work, and of the relationships among properties.

A-1 NOMENCLATURE

c = specific heat for a solid or incompressible fluid

c_p = specific heat at constant pressure

c_v = specific heat at constant volume

$e,\ E$ = energy per unit mass, and total system energy, respectively

$h,\ H$ = enthalpy per unit mass, and total system enthalpy, respectively

$i,\ I$ = irreversibility per unit mass, and total irreversibility, respectively

$\dot{I}$ = rate of irreversibility (rate of exergy destruction)

k = ratio of specific heats, c_p/c_v

$ke,\ KE$ = kinetic energy per unit mass, and total kinetic energy, respectively

m = system mass

$\dot{m}$ = mass flow rate

NSSS = nuclear steam supply system

P = pressure (subscripts: $P_{Gage},\ P_{Abs},\ P_{Vac}$)

P_o = atmospheric pressure

$pe,\ PE$ = potential energy per unit mass, and total potential energy, respectively

$q,\ Q$ = heat transfer per unit mass, and total heat transfer, respectively

$\dot{Q}$ = rate of heat transfer

$s,\ S$ = specific entropy, and total entropy, respectively

$\dot{S}_{Gen}$ = rate of entropy generation

T = temperature

T_o = environmental temperature (dead state temperature)

$u,\ U$ = internal energy per unit mass, and total internal energy, respectively

$\hat{U}A$ = product of the overall heat transfer coefficient and the heat transfer area

V	=	velocity
v, $\forall$	=	volume per unit mass, and total volume, respectively
$\dot{\forall}$	=	volumetric flow rate
w, W	=	work per unit mass, and total work, respectively
$\dot{W}$	=	power
ϕ, Φ	=	nonflow exergy per unit mass, and total nonflow exergy, respectively
η	=	efficiency
η_{II}	=	second law efficiency (exergy out/exergy in)
ρ	=	density
ψ	=	flow exergy per unit mass

Subscripts:

f	=	saturated liquid
g	=	saturated vapor
fg	=	difference in property for vaporization from liquid to vapor

A-1.1 Use of Units

Any fundamental relationship in thermodynamics is valid regardless of the unit system selected. Equations used by practicing engineers including conversions within constants should be used with extreme care. Appropriate conversion factors should always be considered, and units should be carefully tracked to ensure dimensional consistency. Unit conversion must also be considered when calculations are performed with software, as evident from NASA's $185 million mistake in 1998, which destroyed a Mars landing capsule. Documentation should be carefully reviewed and confirmed with sample manual calculations that account for units and appropriate conversion factors. In the English Engineering (EE) unit system, Newton's second law $\left(\vec{\mathbf{F}} = m\vec{\mathbf{a}}\right)$ is frequently written as a proportionality $\left(\vec{\mathbf{F}} = m\vec{\mathbf{a}}/g_c\right)$. Regardless of whether g_c is specifically written, it can be included in any calculation (including those with SI units) as a conversion factor as required for unit consistency. The value of g_c is

(U.S. Customary Units)

$$g_c = 32.2\left(lb_m \cdot ft\right)/\left(lb_f \cdot s^2\right)$$

(SI Units)

$$g_c = \left(kg \cdot m\right)/\left(N \cdot s^2\right)$$

A-2 BASIC CONCEPTS AND TERMINOLOGY

The laws of thermodynamics are applied to a thermodynamic system, which is the mass of a substance being analyzed. Formally, the **system** is the matter analyzed within a defined boundary. A **closed system** is a fixed collection of matter, while in an **open system**, a fluid is allowed to cross the system boundary. The gas contained within a piston cylinder is a closed system. The steam turbine is an open system, since the boundary cuts across the inlet and outlet piping. When analyzing a steam turbine, the system is the mass within the turbine casing. Under steady-state conditions, only the steam is included in the system, and the turbine rotor and casing may be excluded. Under transient conditions, the rotor and casing may need to be considered as part of the system, since they store thermal and kinetic energy.

The **state** of a system is defined by a set of **properties**, which are characteristics of a system (e.g., pressure, temperature, density, specific volume enthalpy, entropy, etc.). **Extensive properties** are mass dependent (e.g., total system energy and system mass), whereas **intensive properties** are independent of mass (e.g., temperature and pressure). Extensive properties become intensive properties if divided by the system mass, e.g., enthalpy per unit mass (or specific enthalpy), $h = H / m$. Calculations are most conveniently performed by using intensive properties and multiplying by the total mass or mass flow rate to determine the total work or power.

Gages measure **gage pressure**, which is the difference in pressure between the system and the surrounding atmosphere. In analyzing thermodynamic properties, the **absolute pressure** must be determined by adding the local atmospheric pressure.

$$P_{abs} = P_{gage} + P_{atm}$$

System pressures that are below atmospheric pressure are referred to as **vacuum**.

$$P_{abs} = P_{atm} - P_{vac}$$

For precise calculations, the atmospheric pressure should be determined from a barometer. If a high degree of accuracy is not required, standard atmospheric pressure for the local elevation above sea level may be used. At sea level, standard atmospheric pressure is

$$P_{\substack{\text{Std}\\\text{Atm}}} = \begin{cases} 1 \text{ atm} \\ 14.696 \text{ psi} \\ 29.92 \text{ in. Hg} \\ 1.01325 \text{ bar} = 101.325 \text{ kPa} \end{cases}$$

Temperature scales are relative $({}^{\circ}C, {}^{\circ}F)$ or absolute $(K, {}^{\circ}R)$. In any calculation involving temperature changes or temperature differences, a relative temperature scale may be used. Any calculation involving a single temperature (such as the ideal gas law) or ratio of temperatures must be completed using an absolute temperature scale. Absolute temperatures are determined from relative scales with the following relationships:

(Kelvin)

$$T(K) = T({}^{\circ}C) + 273.15$$

(Rankine)

$$T({}^{\circ}R) = T({}^{\circ}F) + 459.67$$

Rankine to Kelvin conversions are determined by the relationship $1K = 1.8^{o}R$. Fahrenheit to Celsius conversions may be performed with the relationships

$$T({}^{\circ}F) = 1.8\,T({}^{\circ}C) + 32$$

$$T({}^{\circ}C) = \left(T({}^{\circ}F) - 32\right) / 1.8$$

A-3 MASS CONSERVATION

Mass is conserved; thus, for an open system

$$\begin{bmatrix}\text{The rate of mass}\\ \text{flow into a system}\\ \text{from all inlets}\end{bmatrix} - \begin{bmatrix}\text{The rate of mass}\\ \text{flow out of a system}\\ \text{though all outlets}\end{bmatrix} = \begin{bmatrix}\text{The time rate of}\\ \text{change of mass stored}\\ \text{within the system}\end{bmatrix}$$

or, written symbolically

$$\sum_{inlets} \dot{m} - \sum_{outlets} \dot{m} = \frac{dm}{dt}$$

For a system at steady state

$$\sum_{inlets} \dot{m} = \sum_{outlets} \dot{m}$$

The mass flow rate is related to the density, velocity, and flow area, or the density and volumetric flow rate

$$\dot{m} = \rho VA = \rho \dot{\cancel{V}}$$

A-4 WORK AND HEAT

Work is a mechanical energy transfer that equals the product of the applied force and the displacement in the direction of the force.

$$W = \int \mathbf{F} \cdot d\mathbf{x}$$

The definition can be generalized to include rotating shaft work and electrical work by defining work as any energy transfer that can completely result in the raising of an external weight. Alternatively, work is any energy transfer that has zero entropy associated with it. Excluding irreversibilities, work and other forms of mechanical energy are 100% convertible. In contrast, in accordance with the second law of thermodynamics, heat energy can only be partially converted into work, with the balance rejected at a lower temperature. By convention, the term “shaft work” includes all forms of work excluding boundary work and flow work. Gravitational work is excluded from work terms and instead accounted for as a potential energy term.

The sign convention for equations representing the first law is to define work done by the system as positive work. For a work consuming device (pump or compressor), an implied sign convention is that work on the system is positive.

Specific work is the work per unit mass in the system or the work per unit mass that passes through the system.

$$w = \frac{W}{m} = \frac{\dot{W}}{\dot{m}}$$

A-4.1 Boundary Work

Boundary work is the work done by a system as it expands its boundary.

$$W = \int P d\cancel{V} \text{ or } w = \int P dv$$

In the case of an expanding boundary, a portion of the work is expended by forcing against the atmosphere. The useful boundary work is the total boundary work minus the work done on the atmosphere.

$$W_{Useful} = \int \left(P - P_{atm}\right) d\cancel{V}$$

Under steady-state conditions or for any system with a rigid boundary such as a pump or turbine, the boundary work is zero.

A-4.2 Flow Work

Flow work is performed by a fluid as it flows across a system boundary $\left(w_{flow} = Pv\right)$. In open systems, flow work appears so frequently, that the property **enthalpy** is defined as the sum of the internal energy and the flow work $\left(h = u + pv\right)$.

A-4.3 Power

Power is the time rate of work and is related to specific work and the mass flow rate through an open system.

$$\dot{W} = \dot{m}w$$

A-4.4 Heat

Heat is the transfer of thermal energy due to a temperature difference. By convention, heat added to the system is positive. Analogous to power and work, net specific heat transfer and heat rate are defined

$$q = \frac{Q}{m} \qquad \dot{Q} = \dot{m}q \qquad \dot{Q} = \frac{\delta Q}{dt}$$

A-5 FIRST LAW OF THERMODYNAMICS

The first law states that energy is conserved, although it can be changed in form. The first law of thermodynamics only considers the total quantity of energy and provides no requirement for the direction of processes. From a thermodynamics standpoint, it is convenient to treat energy released from nuclear fission, radioactive decay, or particle capture as a heat generation source term. For a system that undergoes a transient

$$\begin{bmatrix}\text{The net rate}\\ \text{that heat is}\\ \text{transfered in}\end{bmatrix} - \begin{bmatrix}\text{The power}\\ \text{produced by}\\ \text{the system}\end{bmatrix} + \begin{bmatrix}\text{The net rate that energy}\\ \text{is brought into the system}\\ \text{by mass flows}\end{bmatrix} = \begin{bmatrix}\text{The net rate}\\ \text{change of energy}\\ \text{in the system}\end{bmatrix}$$

The power produced by the system can be in the form of shaft power (including electrical) and the rate of boundary work $\left(P\,d\cancel{V}/dt\right)$. The rate of boundary work is zero for either steady-state conditions or any system with a rigid boundary such as a pump or a turbine. Any flow that enters the system brings with it its total energy $\left(e = u + pe + ke\right)$ and also performs flow work $\left(pv\right)$ on the system. Flow at the outlets removes energy and its flow work. Substituting the definition of enthalpy $\left(h\right)$ results in a general expression for the first law of thermodynamics

$$\dot{Q} - \dot{W}_{shaft} - P\frac{d\cancel{V}}{dt} + \sum_{Inlets} \dot{m}\left(h + pe + ke\right) - \sum_{Outlets} \dot{m}\left(h + pe + ke\right) = \frac{d}{dt} m\left(u + pe + ke\right)$$

For a closed system for an interval of time, the first law becomes

$$q - w = \Delta e = \Delta u + \Delta pe + \Delta ke$$

In most cases in the analysis of closed systems, potential and kinetic energy are not significant to the application.

$$q - w = \Delta u$$

In steady flow systems, boundary work and the energy storage term are eliminated.

$$\dot{Q} - \dot{W} = \sum_{out} \dot{m}\left(h + pe + ke\right) - \sum_{in} \dot{m}\left(h + pe + ke\right)$$

For a single inlet and outlet, the mass flow rate may be divided out, producing the steady flow energy equation for single inlet and single outlet systems

$$q - w = \Delta h + \Delta pe + \Delta ke$$

Applications of the steady flow energy equation are shown in Table A-5-1 where the terms "pump work" and "compressor work" imply that work is positive.

Table A-5-1 Applications of the Steady Flow Energy Equation

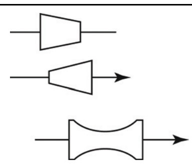
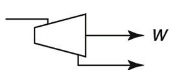
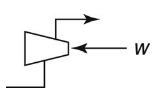
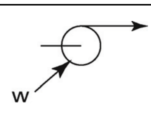
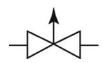
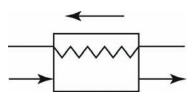
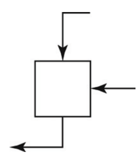
Device	Symbol	Assumptions	Equations
Nozzle diffuser		Adiabatic Insignificant potential energy changes	$\dot{m}_1 = \dot{m}_2$ $\Delta ke = -\Delta h$
Turbine		Adiabatic Insignificant potential and kinetic energy changes	$\dot{m}_1 = \dot{m}_2$ $w = h_1 - h_2$ Turbine Efficiency: $\eta_T = \dfrac{\text{Shaft work}}{\text{Isentropic work}} = \dfrac{h_1 - h_2}{h_1 - h_{2S}}$
Compressor		Adiabatic Insignificant potential and kinetic energy	$\dot{m}_1 = \dot{m}_2$ $w_{in} = h_2 - h_1$ Compressor Efficiency: $\eta_C = \dfrac{\text{Isentropic work}}{\text{Shaft work}} = \dfrac{h_{2s} - h_1}{h_2 - h_1}$
Pump		Adiabatic Insignificant potential and kinetic energy changes	$\dot{m}_1 = \dot{m}_2$ $w_{in} = h_2 - h_1 = \underbrace{u_2 - u_1}_{\text{Work that Raises Temp}} + \underbrace{v\left(P_2 - P_1\right)}_{\text{Ideal or "Hydraulic" Pump Work}}$ Pump Efficiency: $\eta_P = \dfrac{\text{Hydraulic work}}{\text{Shaft work}} = \dfrac{v\left(P_2 - P_1\right)}{h_2 - h_1}$

Table A-5-1 Applications of the Steady Flow Energy Equation (Cont'd)

Device	Symbol	Assumptions	Equations
Throttle		Adiabatic Insignificant potential and kinetic energy changes	$\dot{m}_1 = \dot{m}_2$ $h_1 = h_2$
Heat exchanger (closed)		Adiabatic, except for heat exchange between fluids Insignificant potential and kinetic energy changes	For each stream: $\dot{m}_1 = \dot{m}_2$ Energy: $\dot{Q}_{\substack{Cold\\Stream}} = \dot{Q}_{\substack{Hot\\Stream}}$ $\dot{m}\left(h_{out} - h_{in}\right)\Big\|_{\substack{Cold\\Stream}} = \dot{m}\left(h_{in} - h_{out}\right)\Big\|_{\substack{Hot\\Stream}}$
Mixing chamber		Adiabatic Insignificant potential and kinetic energy changes	$\sum_{In} \dot{m} = \sum_{Out} \dot{m}$ $\sum_{In} \dot{m}h = \sum_{Out} \dot{m}h$

For any device operating on a thermodynamic cycle

$$q_{Net} = w_{Net}$$

A-6 ENTROPY AND THE SECOND LAW OF THERMODYNAMICS

A-6.1 Formulations of the Second Law

In addition to the requirements of the first law, no process will occur unless it also satisfies the second law of thermodynamics, which restricts the direction of processes; that is, some processes are irreversible. For example, heat travels from high temperature to low temperature; it is impossible for heat to be transferred from low temperature to high temperature without some other effect on the environment. There are two commonly cited qualitative statements to the second law.

(a) Kelvin-Planck Statement: It is impossible for any device that operates in a cycle to receive heat from a single thermal reservoir and convert it entirely into work.

(b) Clausius Statement: It is impossible to construct a device that operates in a cycle that produces no other effect on the environment other than the transfer of heat from a low temperature reservoir to a higher temperature reservoir.

Both the Kelvin-Planck and Clausius statements of the second law can be proven from the **increase in entropy principle**, which states that the total entropy change for any process is greater than or equal to zero. Entropy for a system can decrease when heat is transferred out, but the increase in entropy outside that system will be greater than the entropy reduction within the system.

$$\Delta s\Big|_{Total} \geq 0$$

A-6.2 Entropy

The formal definition of "entropy" is the integral of the heat added, divided by the temperature along any internally reversible path

$$\Delta S = \int \frac{\delta Q}{T}\bigg|_{\substack{Int\\Rev}}$$

Fortunately, it is not normally necessary to carry out this integration. Physically, entropy is a measure of the amount of energy that is not usable, and unlike energy, entropy is not conserved. Work and mechanical energy are fully usable and therefore have no associated entropy. Processes that generate entropy are irreversible. In the case where a system is isothermal, the entropy associated with heat transfer to or from a heat source or heat sink at a temperature T is

$$\Delta S = \frac{Q}{T\left({}^{o}R \; or \; K\right)}$$

Since heat transfer occurs from high temperature to low temperature, entropy increases and the usefulness of the heat energy decreases. The entropy balance for a system is

$$\begin{bmatrix} \text{The net rate} \\ \text{that entropy is} \\ \text{transfered in with} \\ \text{heat transfer} \end{bmatrix} + \begin{bmatrix} \text{The} \\ \text{rate of} \\ \text{entropy} \\ \text{generation} \end{bmatrix} + \begin{bmatrix} \text{The net rate that} \\ \text{entropy is brought} \\ \text{into the system} \\ \text{by mass flows} \end{bmatrix} = \begin{bmatrix} \text{The net rate} \\ \text{change of} \\ \text{entropy} \\ \text{in the system} \end{bmatrix}$$

Symbolically

$$\sum_j \frac{\dot{Q}_j}{T_j} + \dot{S}_{Gen} + \sum_{Inlets} \dot{m}s - \sum_{Outlets} \dot{m}s = \frac{dS}{dt}$$

In the discussion of exergy below, the generation of entropy is directly equated with the irreversibility or lost available work. One way to optimize efficiency is to minimize entropy generation.

A-7 EVALUATION OF THERMODYNAMIC PROPERTIES

A-7.1 The State Postulate

According to the state postulate, in order to define the state of a simple compressible system (SCS), it is necessary to specify two independent properties. (An SCS is a fluid that can change shape or volume but does not undergo chemical reaction and is not affected by motion, gravity, surface tension, or other force fields.) All other properties are then fixed.

If a system has some other interaction, such as gravity, then an additional property is required to define the state for each different interaction.

As an example, consider determining the total flow energy of steam. If the steam is considered to be a simple compressible system, then two independent properties are sufficient to determine all other properties (including the flow energy, in this case, equal to enthalpy). For superheated steam, pressure and temperature suffice. For wet steam, pressure and temperature are not independent and therefore another independent property is required (for instance, pressure and quality or pressure and entropy). If other work interactions are considered, then additional parameters are required to determine the state, such as elevation or velocity for gravity and unbalanced forces, respectively.

On the basis of the state postulate, a simple compressible system may be fully defined by two independent properties. Hence, all other properties are dependent. A relationship among *P, v,* and *T* (for example, the ideal gas law; see para. A-7.6.1) is commonly referred to as an equation of state.

A-7.2 Thermodynamic Process

A thermodynamic process is a change of state. Most processes have some entropy generation associated with them and are irreversible. Sources of irreversibility include mechanical friction, fluid viscous friction, mixing fluids of different composition or temperature, heat transfer across a temperature difference, pressure variation in expansion or compression, inelastic deformation, electrical resistance, and conversion of chemical to thermal energy, among others.

(a) isothermal process: temperature is constant.

(b) isobaric process: pressure is constant.

(c) isochoric (isometric) process: volume is constant.

(d) adiabatic: without heat transfer.

(e) isentropic process: entropy is constant. (The term "isentropic" in common usage implies adiabatic and reversible.)

(f) throttling (constant enthalpy that reduces pressure): process that conserves energy, but generates entropy; thus, the ability to perform work is reduced.

A process in which pressure and temperature remain uniform throughout a system is referred to as "internally reversible."

A-7.3 Properties of Water/Steam and Other Real Substances Using Tabulated Data (e.g., Steam Tables)

Water exists in a gas or vapor phase only at sufficiently high temperatures and low pressures. For certain ranges of the properties, water exists in a single phase only. In other regions, two phases exist simultaneously in equilibrium, and along a line called the triple line, all three phases coexist. Where liquid and vapor exist in equilibrium, the vapor is called "saturated vapor," and the liquid is called "saturated liquid." The temperature and pressure of a saturated fluid are called "saturation temperature" and "saturation pressure," respectively.

When evaluating properties of real substances using printed tables, the first step is to determine the phase(s) of the substance (i.e., subcooled liquid, saturated liquid, wet vapor, saturated vapor, or superheated vapor). This is conveniently done using saturation tables. If pressure and temperature are known, then a substance is wet or saturated if these values correspond. A substance is subcooled liquid if $T < T_{sat}$ for the corresponding pressure or $P > P_{sat}$ for the corresponding temperature, and a substance is superheated if $T > T_{sat}$ or $P < P_{sat}$. If pressure or temperature and entropy (or specific volume or enthalpy) are known, then the substance is wet or saturated if the entropy is bracketed by s_f and s_g. If $s < s_f$, then it is a subcooled liquid, and if $s > s_g$, then it is a superheated vapor.

The most widely recognized tables and charts for steam and water properties are those included in references [1–3].

For most calculations, it is recommended that the user refer to the most recent revision. However, when comparing results with previous calculations for long-term trend analysis, it may be necessary to refer to previous revisions. Regardless of which source is used, for a given calculation, all values should always be taken from a single source.

A diagram that is commonly used in plant performance work is the enthalpy–entropy diagram or the Mollier diagram as shown in Figs. A-7.3-1 and A-7.3-2.

Fig. A-7.3-1 Enthalpy–Entropy Diagram for Water; Frequently Designated as Mollier Diagram (U.S. Customary Units)

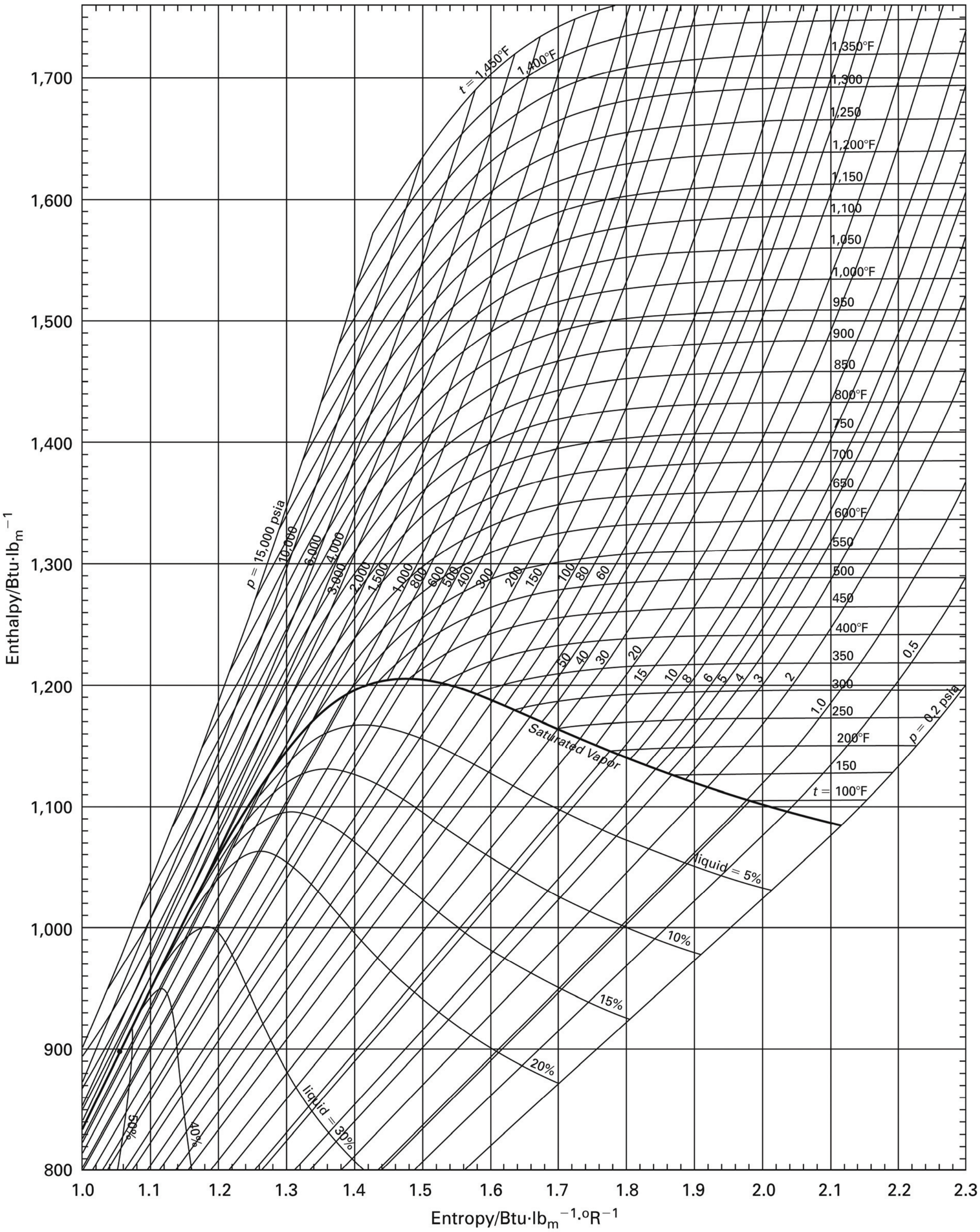
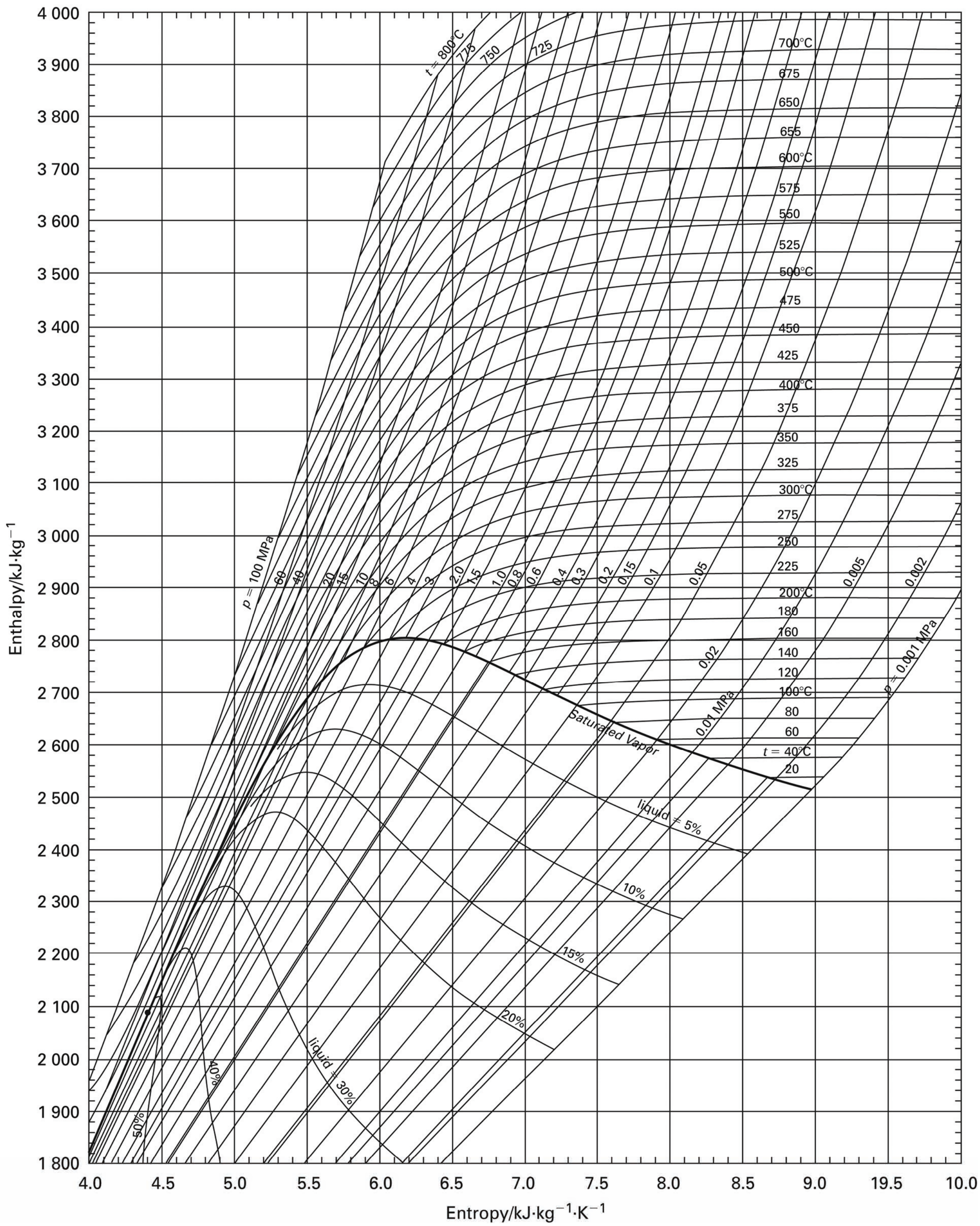


Fig. A-7.3-2 Enthalpy–Entropy Diagram for Water; Frequently Designated as Mollier Diagram (SI Units)



A-7.4 Properties of Wet Vapors

$$s = s_f + Xs_{fg} \text{ (likewise for } h, u, \text{ and } v\text{)}$$

where

$$s_{fg} = s_g - s_f$$

X = quality of the fraction that is vapor

A-7.5 Subcooled (Compressed) Liquids

For the most part, pressure has little effect on the properties of liquids and properties to be approximated as saturated liquids.

$$u = u_{f@T} \qquad s = s_{f@T} \qquad v = v_{f@T}$$

For enthalpy, a slight pressure correction improves the accuracy. This amounts to adding an ideal pump work term to account for the enthalpy added due to increased pressure.

$$h_{scl} = h_{f@T} + \underbrace{v_{f@T}\left(P - P_{sat@T}\right)}_{\substack{\text{Ideal pump work} \\ \text{(Very small correction)}}}$$

A nonideal pump will increase enthalpy due to both the pressure increase and viscous heating of the fluid.

$$\Delta h = c\Delta T + v_{f@T}\left(P - P_{sat@T}\right)$$

If more accurate data are required, particularly in high-pressure and high-temperature systems, printed subcooled liquid tables or, preferably, computerized tables account for compressibility effects. The importance of liquid compressibility increases with both pressure and temperature.

The change in entropy for incompressible liquids can be approximated in cases where the variation in specific heat is small.

$$\Delta s = c \ln\left(\frac{T_2}{T_1}\right)$$

A-7.6 Ideal Gas Properties and Processes

A-7.6.1 Basic Formulations. The ideal gas equation of state (ideal gas law) is $Pv = RT$, or $P\mathit{V\!\!\!/} = mRT$, where R is the gas constant for the gas of question (R is the universal gas constant divided by the molecular weight). The most common ideal gas in thermodynamic cycles is air, although there is increasing interest in other gases for use in high-temperature gas-cooled nuclear reactors acting as the heat source for a closed system Brayton cycle.

According to Joule's Law, internal energy and enthalpy are only functions of temperature: $u = u(T)$, and $h = h(T)$. For this reason, ideal gas properties are tabulated as a function of temperature only.

Entropy for an ideal gas is a function of both pressure and temperature. Because ideal gas properties are tabulated as a function of temperature only, it is necessary to correct entropy for pressure variation.

$$s(T,P) = s^o(T) - R\ln\left(\frac{P}{P^o}\right)$$

where

s^o = the entropy at the reference pressure P^o, normally 1 atm

Thus entropy changes are calculated

$$s_2 - s_1 = s_2{}^o - s_1{}^o - R\ln\left(\frac{P_2}{P_1}\right)$$

Assuming constant specific heats, the following simplified relations may be used:

$$u_2 - u_1 = c_{v,ave}\left(T_2 - T_1\right) \qquad h_2 - h_1 = c_{p,ave}\left(T_2 - T_1\right)$$

$$s_2 - s_1 = c_{v,ave}\ln\left(\frac{T_2}{T_1}\right) + R\ln\left(\frac{v_2}{v_1}\right) = c_{p,ave}\ln\left(\frac{T_2}{T_1}\right) - R\ln\left(\frac{P_2}{P_1}\right)$$

A-7.6.2 Isentropic Processes for Ideal Gases

A-7.6.2.1 Variable Specific Heats (Air Standard Analysis)

$$\left.\frac{P_2}{P_1}\right|_{s=const} = \frac{P_{R2}}{P_{R1}} \quad ; \quad \left.\frac{v_2}{v_1}\right|_{s=const} = \frac{v_{R2}}{v_{R1}}$$

A-7.6.2.2 Constant Specific Heats (Cold Air Standard Analysis)

$$\left.\frac{P_2}{P_1}\right|_{s=const} = \left(\frac{v_1}{v_2}\right)^k \quad ; \quad \left.\frac{T_2}{T_1}\right|_{s=const} = \left(\frac{v_1}{v_2}\right)^{k-1} \quad ; \quad \left.\frac{T_2}{T_1}\right|_{s=const} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

A-8 MEASURE OF PERFORMANCE

A-8.1 Plant Efficiencies

Overall efficiency is a measure of the utilization of the energy supplied to the plant in generating an electrical output.

$$\eta_{OA} = \frac{\dot{W}_{Electrical}}{\dot{Q}_{Fuel}}$$

Overall efficiency is the product that includes boiler or combustion efficiency, thermal efficiency of the cycle, and electric generating efficiency as appropriate.

$$\eta_{OA} = \eta_{Combustion} \cdot \eta_{th} \cdot \eta_{Generating}$$

Thermal efficiency is the efficiency of the thermodynamic cycle in converting a heat input into a mechanical work output.

$$\eta_{th} = \frac{\dot{W}_{net}}{\dot{Q}_{Supplied}} = 1 - \frac{\dot{Q}_{Rejected}}{\dot{Q}_{Supplied}} = \frac{w_{net}}{q_{Supplied}} = 1 - \frac{q_{Rejected}}{q_{Supplied}}$$

A-8.2 Heat Rate

Heat rate is a dimensional reciprocal of efficiency.

$$HR = \frac{Q_{Supplied}\left(Btu\right)}{W_{Elec}\left(kW - hr\right)} = \frac{3\,412.14}{\eta}$$

Heat rate definitions vary with application. In making comparisons between units or trends of units, it is important to be consistent in the definition. With all of the possible variations, it is impossible to list all of the possibilities here, but the following general definitions are used.

boiler/turbine heat rate: based on the heating value of the fuel and the electricity generated at the generator terminals. It does not account for auxiliary loads external to the turbine cycle such as emissions equipment, cooling water pumps, fuel conveyers, etc.

turbine cycle heat rate: based on the heat supplied to the working fluid (i.e., $m_{steam}\Delta h_{steam}$) and the electrical output at the generator terminals. The heat supplied to the turbine cycle may be normally determined from a measurement of the feedwater flow and pressures and temperatures. This may be further analyzed, depending upon whether motor-driven, auxiliary turbine-driven, or main turbine shaft-driven boiler feed pumps are used in the cycle.

unit heat rate (or plant heat rate for multiple units): based on the chemical potential in the fuel supplied and the electrical power that leaves the unit or plant on the transmission lines.

$$HR = \frac{Q_{Fuel}\left(Btu\right)}{W_{\substack{Elec\\Output}}\left(kW - hr\right)} = \frac{3412.14}{\eta_{OA}}$$

For a nuclear cycle, heat rates may be defined in terms of the reactor thermal power or nuclear steam supply system (NSSS) thermal power.

Unlike fossil power plants, nuclear power plants operate at constant reactor thermal power (heat input), and any improvement in heat rate is reflected in a comparable improvement in output.

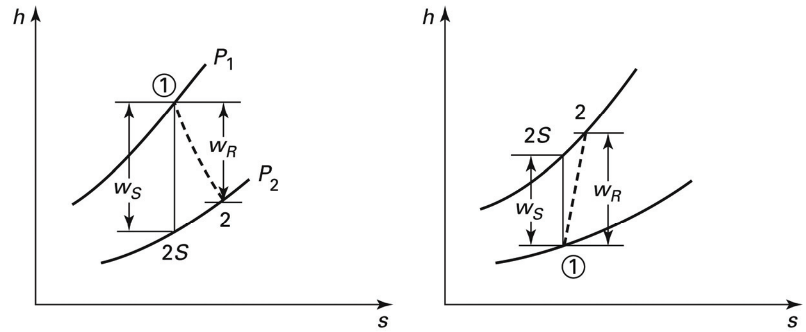
Heat rate and efficiency fail to account for value added by using process steam. For comparative purposes, evaluating the value of process steam can only be meaningfully accomplished with exergy analysis, which is discussed later.

A-8.3 Component First Law Efficiencies

A-8.3.1 Turbine Efficiency (see Fig. A-8.3.1-1)

$$\eta_T = \frac{\text{Actual Work}}{\text{Ideal (Isentropic) Work}} = \frac{w_R}{w_s} = \frac{h_1 - h_2}{h_1 - h_{2s}}$$

Fig. A-8.3.1-1 Terminology of Component First Law Efficiencies



A-8.3.2 Compressor Efficiency (See Fig. A-8.3.1-1)

$$\eta_c = \frac{\text{Ideal (Isentropic) Work}}{\text{Actual Work}} = \frac{w_s}{w_R} = \frac{h_2 - h_{1s}}{h_2 - h_1}$$

A-8.3.3 Pump Efficiency

$$\eta_c = \frac{\text{Hydraulic (Isentropic) Work}}{\text{Actual Work}} = \frac{h_{2s} - h_1}{h_2 - h_1} \cong \frac{v\left(P_2 - P_1\right)}{h_2 - h_1}$$

where the actual work is the measured shaft work.

For completely incompressible liquids, the ideal pump work equals $v\Delta P$; however, for applications requiring greater accuracy, use of subcooled liquid data is recommended. The magnitude of the error in assuming constant specific volume increases with pressure and temperature.

A-9 RANKINE VAPOR CYCLE

A-9.1 Ideal Rankine Cycle

See Fig. A-9.1-1.

The process between points 1 and 2 is isentropic pumping. The process between points 2 and 3 is isobaric heat addition. The process between points 3 and 4 is isentropic expansion. The process between points 4 and 1 is isobaric heat rejection.

Point 1 is as near to a saturated liquid as possible.

Point 3 is generally superheated to increase the efficiency, with the exception of nuclear plants that produce saturated steam.

A-9.2 Analysis

Heat: $q_{net} = \cancel{q_{12}} + q_{23} + \cancel{q_{34}} + q_{41}$

Processes 2-3 and 4-1:

$$q - \cancel{w} = \Delta h + \cancel{\Delta ke} + \cancel{\Delta pe}$$

Fig. A-9.1-1 Rankine Vapor Cycle

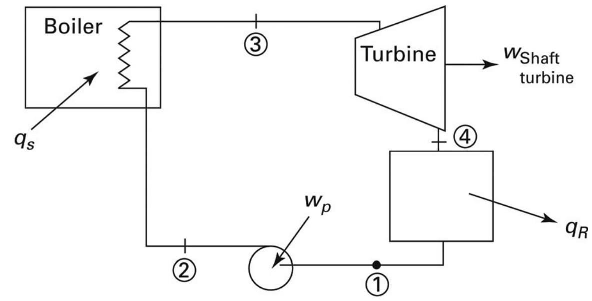
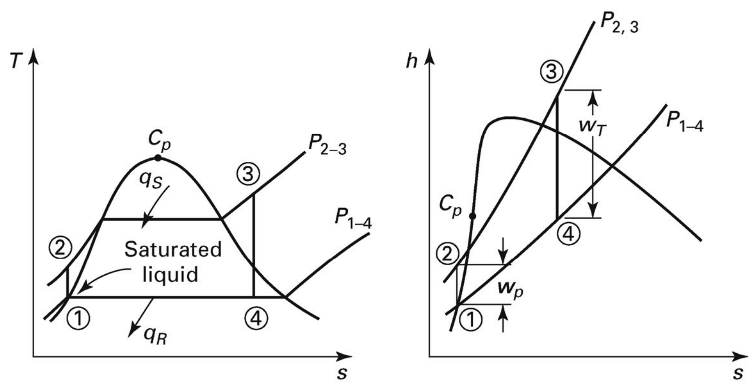


Fig. A-9.3-1 *h-s* and *T-s* Diagrams for an Ideal Rankine Cycle



Work: $w_{et} = w_{12} + \cancel{w_{23}} + w_{34} + \cancel{w_{41}}$

Pump: $w = h_2 - h_1 \cong v\left(P_2 - P_1\right)$

Turbine: $w = h_3 - h_4$

A-9.3 Process Diagrams

Process diagrams (especially *h-s* and *T-s*) are very useful in visualizing and analyzing cycles. In the case of gas cycles, the *h-s* and *T-s* diagrams look the same, although the physical interpretation is different. Figure A-9.3-1 shows *h-s* and *T-s* diagrams for an ideal Rankine cycle. As noted in the *T-s* diagram, heat transfer is the area under a process line for any internally reversible process. On the *h-s* diagram, vertical distances are heat transfer of work magnitudes.

A-9.4 Deviations From the Ideal Cycle

Real Rankine cycle steam plants see efficiency reduced from inherent irreversibilities, including turbine efficiency, compressor efficiency, subcooled point 1, and system heat losses to ambient, and pressure drops. The effect of these losses is shown on the *h-s* diagram in Fig. A-9.4-1.

A-9.5 Improvement With Feedwater Heaters

Real insight into the value of feedwater heaters is better understood by examining a steam plant from an exergy standpoint, but in short, feedwater heaters improve steam plant efficiency by reducing the temperature difference in the heat addition process. To accomplish this, feedwater heaters preheat the feedwater entering the boiler with steam extracted in the turbine.

Fig. A-9.4-1 Losses in Real Rankine Cycle Steam Plants

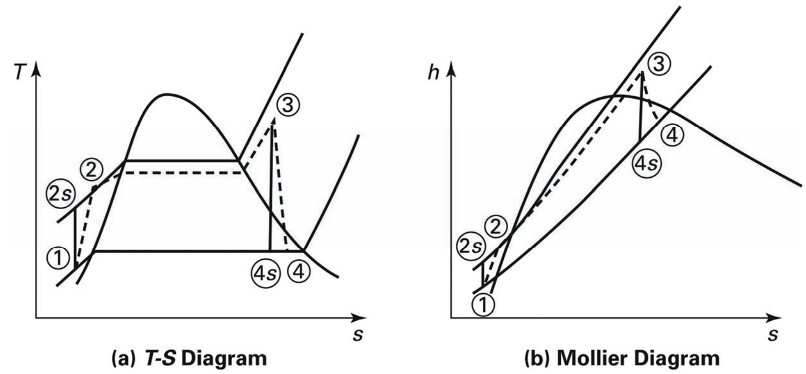
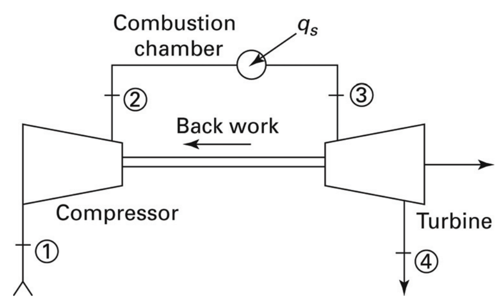


Fig. A-10.1-1 Brayton Gas Cycle



A-9.5.1 Open Feedwater Heaters. Open feedwater heaters extract steam in from the turbine and mix it directly with the feedwater. They are also used to help remove air from the feedwater. In analyzing open feedwater heaters

$$\sum_{in} \dot{m} - \sum_{out} \dot{m} = 0 \quad \text{and} \quad \sum_{in} \dot{m}h - \sum_{out} \dot{m}h - \dot{Q}_{Amb} = 0$$

where

$\dot{Q}_{Amb}$ = heat loss to ambient (normally negligible)

A-9.5.2 Closed Feedwater Heaters. Closed feedwater heaters extract the steam from the turbine and heat feedwater that passes through tubes in contact with the extracted steam. These are used in high pressure applications. For extracted steam being used to preheat liquid feedwater

$$\dot{m}_{Ext}\left(h_{Ext,in} - h_{Ext,out}\right) - \dot{m}_{Feed}c_{p,Feed}\left(T_{Feed,out} - T_{Feed,In}\right) - \dot{Q}_{Amb} = 0$$

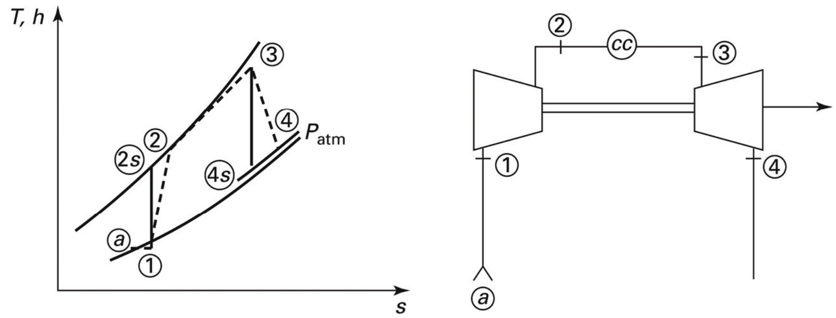
A-10 BRAYTON GAS CYCLE

A-10.1 Ideal Brayton Cycle

See Fig. A-10.1-1.

The process between points 1 and 2 is isentropic compression. The process between points 2 and 3 is isobaric heat addition. The process between points 3 and 4 is isentropic expansion. The process between points 4 and 1 is isobaric heat rejection.

Fig. A-10.4-1 Deviations From an Ideal Brayton Gas Cycle



A-10.2 Air Standard Assumptions

Air standard assumptions assume that the working fluid is air and that it behaves as an ideal gas, that all processes are internally reversible, that the combustion process is replaced with a heat addition, and that the exhaust and fresh air intake are replaced with a heat rejection. Because air is 80% nitrogen, the physical properties of the gas are largely unchanged by the slight change in chemical composition, and the air standard assumptions provide reasonable answers. Tabular values for pure air are used $\left[h = h(T)\right]$ and isentropic processes are analyzed using

$$\left.\frac{P_2}{P_1}\right|_{s=const} = \frac{P_{R2}}{P_{R1}}$$

A-10.3 Cold Air Standard Assumptions

Cold air standard assumptions, in addition to the air standard assumptions in para. A-10.2, use constant specific heats with the values of c_p and c_v evaluated at 25°C or 77°F $\left(\Delta h = c_{p,ave}\Delta T\right)$. Isentropic processes are evaluated with

$$\left.\frac{T_2}{T_1}\right|_{s=const} = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}}$$

Cold air analysis of Brayton cycle plants will result in errors on the order of 15% due to increases in specific heats as temperature increases.

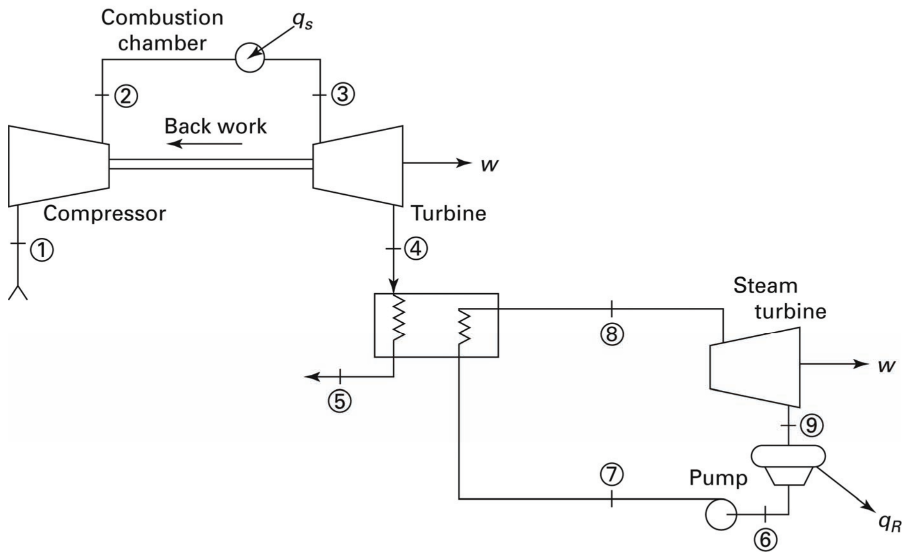
A-10.4 Deviations From the Ideal Cycle

Real gas turbine engines see reduced efficiencies that result from inherent irreversibilities including turbine efficiency, compressor efficiency, and pressure drops throughout the system. The effect of these losses is shown on the *h-s* diagram in Fig. A-10.4-1.

A-10.5 Combined Cycle Plant

Combined cycle plants have become increasingly popular, especially in smaller power plants. Most combined cycle plants burn a fossil fuel in a conventional gas turbine engine. The turbine exhaust is then passed through a boiler where the heat extracted is used in a Rankine cycle (see Fig. A-10.5-1).

Fig. A-10.5-1 Combined Cycle Plant



A-11 EXERGY ANALYSIS

A-11.1 Introduction

Exergy analysis is a relatively simple extension of first law analysis that considers the balance of the useful work potential, termed "exergy." Exergy is formally defined as the amount of useful work that can be performed by reducing a system to the dead state. In determining the exergy, it is necessary to specify the environmental conditions or the dead state.

Although first law analysis is relatively simple to master and easy to conceptualize, it has the disadvantage of considering only the magnitude of the energy transfer and assuming that all forms of energy have the same quality or usefulness. In reality, experience tells us that large quantities of low temperature heat have very little potential to perform useful work and that many processes that are very efficient from a first law standpoint, such as using electrical energy to heat low temperature water, are actually very wasteful. In contrast, second law analysis (exergy analysis) quantifies for the useful potential of energy and clearly identifies the source of irreversibility (lost available work). In that manner, it is possible to appropriately focus attention on components that have the greatest impact on performance.

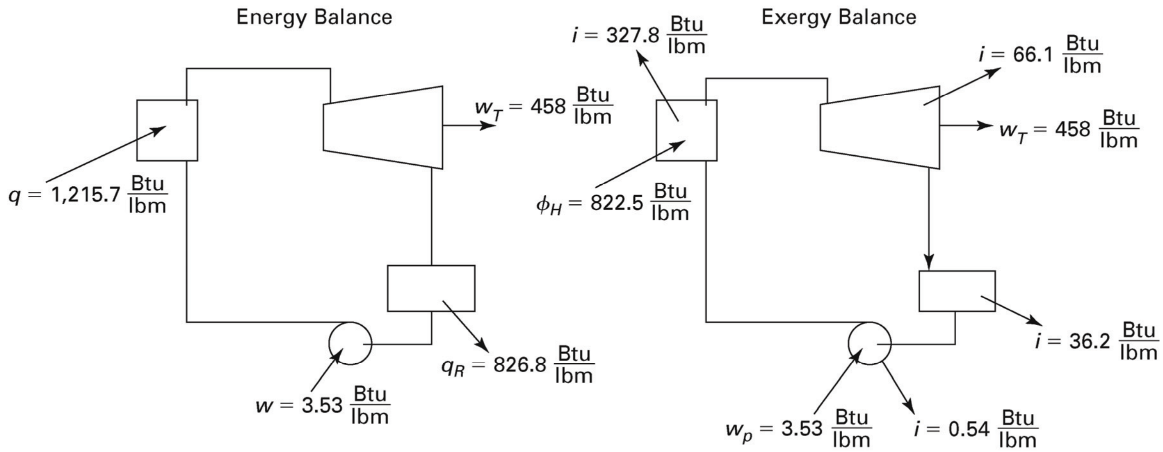
The principle of exergy analysis can be illustrated by considering a source of heat, Q, from a source at a temperature, T. If an engine can reject heat to the environment at T_o, then the maximum work potential (or exergy) of the heat source is the magnitude of the heat supplied multiplied by the Carnot efficiency

$$W_{\max} = Q\left(1 - \frac{T_o}{T}\right)$$

If the heat is transferred by conduction to a lower temperature, then even though the total quantity of energy is conserved, the exergy of the heat is reduced and entropy is generated.

In a practical example, consider a steam plant operating on a simple Rankine cycle. Boiler steam is produced at 650°F and 1,400 psi and condenser pressure is 1 psi. Assume pump and turbine efficiencies are both 85%. For simplicity, consider that boiler first law efficiency is 100%, heat source is available at 1,200°F, and the environmental dead state is at 77°F (537°R). Figure A-11.1-1 summarizes the energy and exergy balances. While first law analysis gives a measure of the overall plant efficiency, it fails to accurately reflect the sources of plant inefficiency, and it points to the condenser as the biggest loss. However, as shown by

Fig. A-11.1-1 Summary of the Energy and Exergy Balances



second law analysis, the heat rejected in the condenser has minimal work potential (or exergy), and the greatest source of inefficiency is the steam generator (100% efficient from a first law standpoint). Armed with this information, the engineer can seek to reduce boiler irreversibility by decreasing the large heat transfer temperature difference by inserting feedwater heaters.

In Brayton cycle plants, second law analysis of a gas turbine demonstrates that the turbine exhaust contains an enormous work potential, driving the engineer to consider heat recovery boilers or recuperation. For plants that supply process steam, second law analysis provides quantitative means of evaluating the value of the heat removed. Finally, for heat engine cycles that harness lower temperature heat sources, such as geothermal units or waste heat recovery units, it is possible to compare the performance on a comparable basis using first law analysis.

A-11.2 Definitions

exergy (also called availability and available energy): the maximum energy that is available to perform useful work by reducing a system to the dead state. The **dead state** is when the system reaches equilibrium with the environment.

(a) In the **restricted dead state**, a system is in thermal and mechanical equilibrium $\left(T_o, P_o\right)$; chemical equilibrium is not considered.

(b) The **standard dead state** is 537°R (298.15K), 14.696 psi (1.01325 bar), with zero velocity and zero potential energy.

irreversibility: destruction of exergy or the loss of useful work potential and is related to the generation of entropy.

$$I = T_o S_{Gen}$$

A-11.3 Calculations and Analysis

A-11.3.1 Exergy Balance Equation. The exergy balance equation is very similar to the first law of thermodynamics with the exception that useful energy is considered instead of total energy. An important difference is that exergy is not a conserved quantity; it is destroyed as entropy is generated.

$$\begin{bmatrix}\text{The net rate}\\ \text{that exergy is}\\ \text{transfered into}\\ \text{by heat transfer}\end{bmatrix} - \begin{bmatrix}\text{The useful}\\ \text{power}\\ \text{produced by}\\ \text{the system}\end{bmatrix} + \begin{bmatrix}\text{The net rate that}\\ \text{exergy is brought}\\ \text{into the system}\\ \text{by mass flows}\end{bmatrix} - \begin{bmatrix}\text{The net rate that}\\ \text{exergy is}\\ \text{destroyed by}\\ \text{irreversibility}\end{bmatrix} = \begin{bmatrix}\text{The net rate}\\ \text{change of}\\ \text{exergy}\\ \text{in the system}\end{bmatrix}$$

Symbolically

$$\Phi_{Heat} - \dot{W}_{\substack{Net\\ Useful}} + \sum_{Inlets} \dot{m}\psi - \sum_{Outlets} \dot{m}\psi - T_o\dot{S}_{Gen} = \frac{d\Phi}{dt}$$

where the nomenclature is defined in the following paragraphs.

A-11.3.2 Simplified Exergy Balances

Closed System

$$\Phi_{Heat} - W_{Useful} - T_o S_{Gen} = \Phi_2 - \Phi_1$$

Analogy: $Q - W = E_2 - E_1$

Single Inlet, Single Outlet, Steady State

$$\phi_{Heat} - w - T_o s_{Gen} = \psi_2 - \psi_1$$

Analogy: $q - w = h_2 - h_1$

A-11.3.2.1 Exergy of Work and Mechanical Energy Terms

$$\phi_{KE} = k.e. = \tfrac{1}{2}V^2 \qquad \phi_{PE} = p.e. = gz \qquad \phi_{Shaft} = w_{shaft} \qquad \phi_{Elec} = w_{elec}$$

A-11.3.2.2 Exergy of a Heat Source. The exergy of a heat source at a temperature, T, is determined by multiplying the magnitude of heat transferred by the Carnot efficiency. It is a function of the temperature of the dead state.

$$\Phi_{Heat} = Q\left(1 - \frac{T_o}{T}\right)$$

where all temperatures are given in absolute units (^{o}R or K).

A-11.3.2.3 Useful Work Associated With Boundary Work Interactions. Useful work is that portion of the work that can be harnessed.

NOTE: Boundary work required against the atmosphere is not useful work.

$$W_{\substack{Boundary\\ Useful}} = \int P d\cancel{V} - \left[P_o\left(\cancel{V}_2 - \cancel{V}_1\right)\right]$$

A-11.3.2.4 Irreversibility. For any process, the loss of useful work potential is proportional to the entropy generated.

$$I = T_o S_{Gen}$$

Irreversibility is often easier to calculate based on an exergy balance of a system.

A-11.3.2.5 Closed System Exergy

$$\Phi = (E - U_o) + P_o(V - V_o) - T_o(S - S_o)$$
$$\phi = (e - u_o) + P_o(v - v_o) - T_o(s - s_o)$$

A-11.3.2.6 Flow Exergy. The flow exergy of a fluid is the amount of useful work that could be produced if the fluid were reduced to the dead state. Flow exergy is a property that depends on the fluid state as well as the dead state.

$$\psi = (h - h_o) - T_o(s - s_o) + ke + pe$$

A-11.3.2.7 Second Law Efficiency. Second law efficiency is the fraction of exergy that survives a process or component

$$\eta_{II} = \frac{\text{Exergy Out}}{\text{Exergy Supplied}} = 1 - \frac{\text{Irreversibility}}{\text{Exergy Supplied}}$$

A-11.3.3 Second Law Analysis Examples

A-11.3.3.1 Resistance Heating. Electrical energy is work by definition, and is therefore fully useful. Resistance heating converts this fully usable energy into less useful heat energy and consequently destroys exergy. The irreversibility or exergy destroyed by resistance heating is

$$I = T_o S_{Gen} = Q\frac{T_o}{T}$$

A-11.3.3.2 Steam Throttling. As can be seen from the Mollier diagram in Fig. A-7.3-1, a constant-enthalpy throttling process (constant enthalpy, decreasing pressure) generates entropy. The rate of exergy destruction is

$$I = T_o S_{Gen} = T_o \dot{m}(s_2 - s_1)$$

A-12 HEAT TRANSFER

A-12.1 Definitions

Heat transfer is a thermal energy flux driven by a temperature difference. Classically, three forms of heat transfer are defined.

conduction: heat transfer due to molecular interaction in a planar geometry

$$\dot{Q} = kA\frac{\Delta T}{X}$$

where

A = the heat transfer area

k = the thermal conductivity

X = the thickness across the heat transfer path

It is frequently convenient to define a thermal conductive resistance

$$\dot{Q} = \frac{\Delta T}{R_{cond}}, \text{ where } R_{Cond} = \frac{X}{kA}$$

For cylindrical geometries in which the thickness is significant relative to the diameter, it is necessary to account for the variation in heat transfer area with radius. This can be conveniently done by defining the logarithmic mean area for heat transfer

$$A_m = \frac{2\pi L\left(r_0 - r_i\right)}{\ln\left(r_o / r_i\right)}$$

where

L = the pipe length

radii specified = the internal and external radii of the tube.

This leaves a thermal resistance for hollow cylinders

$$R_{Cond} = \frac{\ln\left(r_o / r_i\right)}{2\pi k L}$$

convection: heat transfer between a solid that results from a combination of molecular activity and bulk fluid motion. It is defined in terms of convective heat transfer coefficient $\left(\hat{h}\right)$ and may be likewise written in terms of a thermal resistance

$$\dot{Q} = \hat{h} A \Delta T = \frac{\Delta T}{R_{conv}}$$

radiation: transfer of heat from emitted radiant energy. It is proportional to the fourth power of the temperature.

The combined heat transfer in a heat exchanger from one fluid to the next is normally written in terms of an overall heat transfer coefficient $\left(\hat{U}\right)$ and area $\left(A\right)$

$$\dot{Q} = \hat{U} A \Delta T$$

The product $\hat{U}A$ results from a series of thermal resistances

$$\hat{U}A = \frac{1}{R_{Cond1} + R_{Conv1} + \ldots}$$

Typically, there is a thermal resistance associated with the convection on each side of the heat transfer surface as well as the conduction. Any corrosion, scaling, or fouling also produces additional thermal resistance.

A-12.2 Implications of the Second Law

Heat transfer across any temperature difference generates entropy and destroys exergy. The irreversibility associated with heat transfer is determined by

$$I = T_o S_{Gen} = Q T_o \left(\frac{1}{T_L} - \frac{1}{T_H} \right)$$

As a result, anything that increases the thermal resistance and temperature difference reduces plant efficiency.

A-13 DEGRADATION PERFORMANCE

Understanding of the thermodynamics involved allows the operator to interpret indication and optimize plant performance. Performance is reduced by deterioration of the plant due to many factors including

(a) fouling and scaling of heat transfer surfaces
(b) erosion of heat transfer surfaces
(c) flow restrictions
(d) steam leaks (internal and external)
(e) vacuum leaks
(f) valve leakage
(g) pump degradation

A-14 COMBUSTION

A-14.1 Definitions and Background

Combustion releases energy that results from the difference in bond energy of the chemical products.

exothermic reaction: a reaction that releases chemical energy.

$$Reactants \rightarrow Products + Heat$$

stoichiometric: a chemical equation that is in proper balance. In combustion, it generally means that the amount of oxygen is exactly the amount required to burn all of the fuel. (See also *theoretical air*.)

theoretical air: the amount of air required for the reaction equation to be in balance such that the amount of air supplied provides the precise amount of oxygen required to combust with the fuel.

A-14.1.1 Related Terms

air fuel ratio: mass of air to mass of fuel

$$AF = \frac{m_{air}}{m_{fuel}}$$

combustion: rapid reaction between a fuel and oxygen in which energy is liberated.

combustion of fuels:

complete combustion: all hydrogen and carbon in the reactants form H_2O and CO_2 in the products, respectively.

incomplete combustion and additional byproducts: in some cases unburned fuel survives combustion as a result of insufficient O_2, and poor mixing.

Other byproducts also form due to equilibrium at high temperature. For example, CO may form instead of CO_2. NO or NO_2 may form when N_2 reacts with O_2 at high temperature.

EXAMPLE: Determine the reaction coefficients for complete combustion of methane in dry air:

$$CH_4 + ???\left(O_2 + 3.76N_2\right) \rightarrow ???\,CO_2 + ???\,H_2O + ???\,N_2$$

One carbon will form 1 CO_2 and 4 hydrogen atoms will form 2 H_2O

$$CH_4 + ???(O_2 + 3.76N_2) \rightarrow CO_2 + 2H_2O + ???\,N_2$$

This requires 4 Oxygen atoms or $2O_2$

$$CH_4 + 2(O_2 + 3.76N_2) \rightarrow CO_2 + 2H_2O + 7.56N_2$$

At 150% theoretical air (50% excess air/equivalence ratio 0.67), the above reaction would be

$$CH_4 + 3(O_2 + 3.76N_2) \rightarrow CO_2 + 2H_2O + O_2 + 11.28N_2$$

dry air composition: 21% oxygen on a molar basis; 79% mostly nitrogen (some small components including 1% argon are lumped in with this). Thus there are 3.76 moles of nitrogen per mole of oxygen

enthalpy of combustion $\left(\bar{h}_C\right)$*:* enthalpy of reaction for a combustion process.

per unit mole of fuel: $\bar{h}_C$

per unit mass of fuel: h_C

NOTE: If energy is released (exothermic), the $\bar{h}_C$ is a negative number.

enthalpy of formation of a compound $\left(\bar{h}_f^o\right)$*:* the enthalpy of a compound at 25°C and 1 atm pressure.

enthalpy of reaction $\left(\bar{h}_R\right)$*:* difference between the enthalpy of formation for the products and the reactants at a given temperature. $\bar{h}_R = \bar{H}_P - \bar{H}_R$. For an exothermic reaction, $\bar{h}_R$ is negative.

equivalence ratio: ratio of actual to stoichiometric fuel: air ratios, or (actual fuel/air ratio) / (stoichiometric fuel/air ratio); for example, 200% theoretical air would have an equivalence ratio of 0.5 (i.e., 1/2x / 1/x = 0.5).

heating values: amount of energy released during complete combustion when the products are returned to the state of the reactants. A positive number indicates the magnitude of the enthalpy of combustion.

$$[\text{Heating Value}] = \left|\bar{h}_C\right|$$

higher heating value $\left(HHV,\ \overline{HHV}\right)$: energy released in combustion when all water in the products is in liquid form.

lean: excess air (equivalence ratio <1).

lower heating value $\left(LHV,\ \overline{LHV}\right)$: energy released in combustion when all water in the products is in vapor form. It is lower than the HHV because some of the energy is used to vaporize the water in the products.

$$\overline{HHV} = \overline{LHV} + \left(N\bar{h}_{fg}\right)_{H_2O}$$

where
N = the number of moles of water per mole of fuel

$$HHV = LHV + \left(mh_{fg}\right)_{H_2O}$$

where mass is the water in the products per unit mass of fuel.

molar air fuel ratio: moles of air to moles of fuel

$$\overline{AF} = \frac{N_{air}}{N_{fuel}}$$

Thus

$$AF = \overline{AF}\frac{M_{air}}{M_{fuel}}$$, where M is the **molar mass.**

percent deficiency of air: similarly used to describe reactions that have a shortage of air.

percent excess air: 200% theoretical air is also expressed as 100% excess air.

percent theoretical air: if the combustion is stoichiometric, it is 100% theoretical air. If combustion contains twice the required amount of air, it is 200% theoretical air.

rich: excess fuel (equivalence ratio >1).

standard reference state: to analyze a chemical reaction, it is necessary to assign enthalpies on a consistent basis.

All elemental enthalpies are chosen with the **reference point** that the stable elements have an enthalpy of zero at 77°F (25°C) and 1 atm pressure. This applies to the chemically stable form of the element (e.g., diatomic oxygen).

When there is no chemical reaction, the reference point of the enthalpy is immaterial, since the only concern is with changes in enthalpy.

A-14.2 First Law Analysis of Steady-Flow Reacting Systems

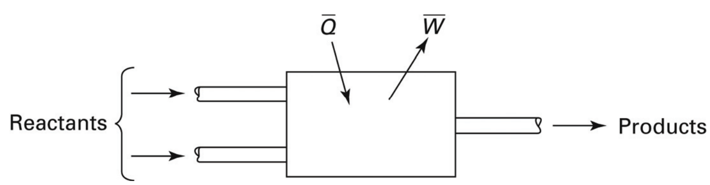
Refer to Fig. A-14.2-1 for an illustration of first law analysis of steady-flow reacting systems.

Writing the balance on a molar basis

$$\bar{Q} - \bar{W} + \sum_k N_k \bar{h}_k \Big|_{\text{Reactants}} = \sum_k N_k \bar{h}_k \Big|_{\text{Products}}$$

The heat and work terms follow the standard sign conventions (heat in, work out).

Fig. A-14.2-1 First Law Analysis of Steady-Flow Reacting Systems



A-14.2.1 Example. Consider the reaction for a hydrocarbon

$$C_aH_b + \alpha\left(O_2 + 3.76N_2\right) \rightarrow \sum_k \nu_k \pi_k$$

where

ν_k = the stoichiometric coefficient for each product component

π_k = the product.

An energy balance can be written as

$$\bar{Q} - \bar{W} + \bar{H}_R = \bar{H}_P$$

where the overbar indicates that all terms are written on a per unit mole basis. In this case, all terms are normalized by the moles of fuel.

$$\bar{H}_R = \bar{h}_{fuel} + \alpha\bar{h}_{O_2} + 3.76\alpha\bar{h}_{N_2}$$

$$\bar{H}_P = \sum_k \nu_k \bar{h}_k$$

where

$\bar{h}_k$ = the enthalpy of substance k

A-14.2.2 Evaluating Enthalpy. The enthalpy of a constituent is determined by adding the sensible enthalpy to the enthalpy at the reference point. Because enthalpy tables are not referenced to the thermochemical reference point,

$$\bar{h} = \underbrace{\bar{h}_f^o\left(T_{Ref}, P_{Ref}\right)}_{\substack{\text{Enthalpy of}\\ \text{Formation at}\\ \text{reference point}}} + \underbrace{\left[\bar{h}\left(T, P\right) - \bar{h}^o\left(T_{Ref}, P_{Ref}\right)\right]_{Tables}}_{\substack{\text{Sensible Enthalpy}\\ \text{above reference}\\ \text{point}}}$$

If the combustion enthalpy is available in the literature for the reaction in question,

$$\bar{Q} - \bar{W} = \bar{h}_C^o + \sum_k N_k\left(\bar{h} - \bar{h}^o\right)_{\substack{Tables\\ k}}\Bigg|_{\text{Products}} - \sum_k N_k\left(\bar{h} - \bar{h}^o\right)_{\substack{Tables\\ k}}\Bigg|_{\text{Reactants}}$$

A-14.3 Second Law Analysis of Steady-Flow Reacting Systems

A-14.3.1 For a Steady-State System

$$\bar{s}_R + \bar{s}_{Gen} + \sum_k \frac{\bar{Q}_k}{T_k} = \bar{s}_R$$

A-14.3.2 Entropy

$$\bar{s}\left(T, P\right) = \bar{s}_{Ref}\left(T_{Ref}, P_{Ref}\right) - \Delta\bar{s}$$

A-14.3.3 Third Law of Thermodynamics. When evaluating changes in chemical composition, the common reference point of the third law must be considered. The third law states that the entropy of a pure crystalline substance at absolute zero temperature is zero.

For tables that are not referenced with respect to the third law, the absolute entropy at a reference point must be determined and the entropy from the tables corrected by using the above equation.

A-14.3.4 Ideal Gas Entropies. The entropy of an ideal gas is dependent on its pressure and temperature.

For ideal gases, tables are generally referenced in accordance with the third law of thermodynamics.

$$\underbrace{\bar{s}(T,P)}_{\text{Absolute Entropy}} = \underbrace{\bar{s}^o(T,P_o)}_{\substack{\text{Tabulated absolute entropy}\\ \text{at T and } P_o\\ \text{referenced IAW}\\ \text{third law}}} - \underbrace{\bar{R}\ln\left(\frac{P}{P_o}\right)}_{\text{Pressure correction}}$$

Recall, P_o is normally one atmosphere and $\bar{s}_o$ is then a function of temperature only.

A-14.3.5 Ideal Gas Mixtures. Recall that the entropy for a gas mixture equals the sum of the entropies of the components. The entropy of each component is based on its partial pressure.

$$\bar{s}(T,P) = \sum_k x_k\left[\bar{s}_k{}^o(T,P_o) - \bar{R}\ln\left(\frac{P_k}{P_o}\right)\right] = \sum_k x_k\left[\bar{s}_k{}^o(T,P_o) - \bar{R}\ln\left(\frac{x_k P}{P_o}\right)\right]$$

where

$$x_k = \text{mole fraction of the } k^{th} \text{ component}$$

A-15 REFERENCES

[1] Keenan, J. H. and Keyes, F. G., “Thermodynamic Properties of Steam,” Wiley, New York, 1936 revised as Keenan, J. H., Keyes, F. G., Hill, P. G. and Moore, J. G., “Steam Tables,” Wiley, New York, 1969

[2] ASME, “Thermodynamic and Transport Properties of Steam,” American Society of Mechanical Engineers, New York, 1967.

[3] ASME “Thermodynamic and Transport Properties of Steam,” American Society of Mechanical Engineers, New York, 1997.

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