ASME
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Performance Monitoring Guidelines for Steam Power Plants

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
United Engineering Center
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Performance Monitoring Guidelines for Steam Power Plants

PERFORMANCE TEST CODES

ASME PTC PM–1993

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FOREWORD

(This Foreword is not part of ASME PTC PM-1993.)

This document represents a departure from the traditional efforts of ASME to serve the needs of industry in performance testing of power equipment. In the past the Performance Test Codes (formerly called Power Test Codes) were largely oriented toward acceptance testing of new equipment. With few exceptions, the codes gave no attention to on-going performance monitoring.

From the reports and discussion generated in a series of EPRI-sponsored workshops starting in 1978 it became clear that there was a need for documentation of an authoritative source on monitoring of equipment to determine performance trends throughout its lifetime. The ASME Performance Test Codes were identified as the logical choice for the development of the required work. The ASME Board on Performance Test Codes was immediately notified.

On December 12, 1984 the ASME Board on Performance Test Codes commissioned a study of the feasibility of developing performance monitoring guidelines. This study resulted in a comprehensive report containing positive recommendations for this undertaking. On June 6, 1985 the ASME Board on Performance Test Codes approved proceeding with the development of Performance Monitoring Guidelines. As a result, an Object and Scope were drafted and then approved by the Board on March 4, 1986. A technical committee of knowledgeable engineers was organized and held its first meeting in June 1986.

Since the initial meeting many more meetings of the committee were held, resulting in the present document. It is seen as the first of a series of publications dealing with methods of monitoring the performance of power plant equipment. It is necessarily broad in scope; other documents will concentrate on specific items of power plant equipment. As documents dealing with monitoring of one item of equipment are developed, future editions of the present document will contain less detail on that equipment and become more the "umbrella" document for all power plant apparatus, particularly for that not covered by any PTC publication.

This document was approved by the Board on Performance Test Codes on June 25, 1993.

The Committee invites comments from users of this first performance monitoring document for consideration when making future revisions. These should be addressed to Director, ASME Performance Test Codes, United Engineering Center, 345 East 47th Street, New York, New York 10017.

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SECTION 0 — INTRODUCTION

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

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

0.3 These guidelines provide methods and procedures to monitor steam cycle performance, validate and process the data, and analyze it to improve or optimize the following:

(*a*) unit/plant thermal efficiency
(*b*) capacity
(*c*) economic dispatch
(*d*) operator awareness
(*e*) cycle component diagnostics

and provide information for:

(*f*) engineering studies
(*g*) preventive or predictive maintenance

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SECTION 1 — OBJECT AND SCOPE

1.1 OBJECT

The Object of the 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 monitoring.

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 Measurement Uncertainty, and the related Performance Test Codes.

1.2 SCOPE

The Scope includes fossil-fueled steam plants and the balance-of-plant portion 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.

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SECTION 2 — OVERVIEW

2.1 DEFINITION OF PERFORMANCE MONITORING

Performance monitoring is an overall long-term effort to measure, sustain , and improve the plant/unit thermal efficiency, capacity, dispatch cost, and maintenance planning. The program can be implemented for multiple reasons. A decision to implement a performance monitoring program should be based on local needs, organization, economics, and 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 steam power plants. They seek to advise plant personnel on how to effectively monitor the efficiency and mechanical condition of their 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 monitoring needs beyond the traditional Performance Test Code function of contract compliance of individual items of equipment.

The guidelines are intended to be used only to the extent that it is practically feasible in power plant performance monitoring. The value of implementing the guidelines will vary significantly between plants which are crucial to operations and those which are not. The remaining life of the plant, size of the plant staff, and other resources already available will influence the degree to which these guidelines are employed.

The guidelines are arranged by Section in the logical order of program development and use. Following the Definitions, Section 3, the order of the Sections are as follows:

(*a*) Planning — Section 4
(*b*) Instrumentation — Section 5
(*c*) Implementation — Section 6
(*d*) Interrelationships — Section 7
(*e*) Diagnostics — Section 8
(*f*) Optimization — Section 9
(*g*) Incremental Heat Rate — Section 10

Other available guidance for performance monitoring includes short courses by consultants, universities, professional engineering societies, and industry research arms such as the Electric Power Research Institute. Helpful papers and texts are referenced at the end of most Sections.

2.2 PURPOSE OF PERFORMANCE MONITORING

Performance monitoring programs involve the collection and analysis of process data for various cost/benefit purposes such as:

(*a*) providing instantaneous operator feedback on controllable losses. See Section 6.

(*b*) tracking controllable losses over long time periods. See Section 6.

(*c*) establishing unit heat rate for fuel accounting and regulatory records or for performance comparisons. See Section 6.

(*d*) determining cycle component contribution to total unit performance. See Sections 6 and 7.

(*e*) diagnosing component condition for establishing overhaul schedule and scope and to improve ordering of parts requiring long lead times. See Section 8.

(*f*) optimizing individual cycle component optimization. See Section 9.

(*g*) determining input/output characteristics for economic/incremental loading. If determined by non-fuel methods, the complexity is significantly increased. See Section 10.

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2.3 PERIODIC VERSUS CONTINUOUS MONITORING

A decision which significantly characterizes a program is whether to monitor periodically, continuously, or some combination of the two. The additional benefits of continuous or "on-line" monitoring include the following:

(*a*) Ability to accumulate heat rate over time for official record

(*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

(*e*) Opportunity to dispatch the unit based on current cost

(*f*) Ability to track controllable losses over time

(*g*) Opportunity for continuous optimization

The evaluation of periodic versus continuous monitoring should include a comparison of the higher one-time capital and on-going 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 these two extremes 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 important decision. The joint-use option is most economically beneficial when incorporating a monitoring program into the design of new capacity or major modifications.

2.4 FACTORS CRITICAL TO SUCCESSFUL PROGRAMS

Some considerations which contribute to the success of a performance monitoring program are the following.

(*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 which 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 require increased quantities of instrumentation. A major plateau is sufficient instrumentation to allow the calculation of a 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 exhaust flow, turbine shaft work, LP turbine efficiency, turbine cycle heat rate and flow factors and the further cycle analysis. An accurate flow balance requires isolation (shut off) of all flows not measured or calculated.

(*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 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) 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 may be more beneficial than if their incremental costs cause them to operate fully loaded. See Section 10, Incremental Heat Rate.

2.5 TYPICAL PLANT ENERGY DISTRIBUTION

One of the keys to establishing an effective performance monitoring program is to allocate resources to the areas which 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 2-1 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%, respectively, of total heat input.

Figure 2-2 shows the constituents of the steam generator losses which account for 11% of total heat input to the cycle for a coal-fired unit. Figure

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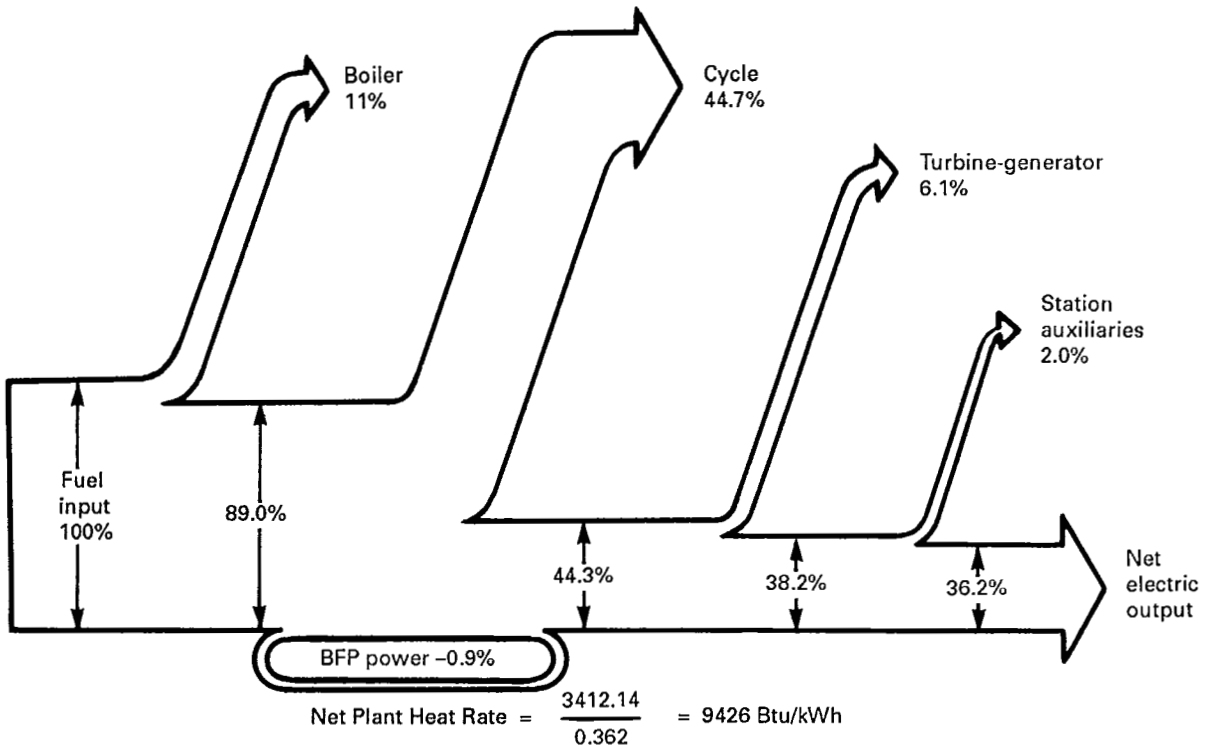

FIG. 2-1 TYPICAL PLANT LOSSES
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2-3 shows the variables which make up the total losses of 45% of heat input defined as cycle losses, including feedwater cycle and condenser. The breakdown of turbine-generator and station auxiliary power losses which account for 6% and 2% of total heat input, respectively, is shown in Fig. 2-4.

The typical losses shown in Tables 2-1 and 2-2 and in Figs. 2-1, 2-2, 2-3, and 2-4 are not to be applied indiscriminately. The losses for a particular unit need to be determined.

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 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.

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. 2-5 and 2-6.

Figure 2-7 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 on Fig. 2-8. This chart was developed at Morgantown during the EPRI Project 1681. It shows the effect of several steam generator related parameters. Each line can rotate 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.

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 Section 4, Program Planning.

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TABLE 2-1 OFF-DESIGN CONDITIONS APPROXIMATE EFFECT ON ACTUAL HEAT RATE

Main Steam Temperature	−10°F	+10 Btu/kWhr
Main Steam Pressure	−10 psig	+3 Btu/kWhr
Reheat Temperature	−10°F	+10 Btu/kWhr
Reheat Spray	+1% (Throttle Flow)	+10–15 Btu/kWhr
Back Pressure	+0.1 in. HgA	+20 Btu/kWhr
Excess O_2	+1% O_2	+30 Btu/kWhr
Flue Gas Temperature	+10°F	+20 Btu/kWhr

TABLE 2-2 VALUE OF TURBINE SECTION EFFICIENCY LEVEL IMPROVEMENT ON A UNIT HEAT RATE OF 10,000 BTU/kWHR

One Percentage Point On	Percent Effect On Turbine Cycle Heat Rate	
High Pressure	0.2% Heat Rate	= −20 Btu/kWhr
Intermediate Pressure	0.2% Heat Rate	= −20 Btu/kWhr
Low Pressure	0.5% Heat Rate	= −50 Btu/kWhr

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

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2.6 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 conducted with the knowledge and care necessary to assure 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 Section 8, Component Diagnostic Techniques. Some considerations which help to assure 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.

A logical function of the more experienced performance personnel that would allow their advancement while remaining in the performance area of an organization is direct involvement in the cost benefit and field implementation of the potential savings determined by performance monitoring. 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 Supplements and Sections 4 and 5 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 Section 5.

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

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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.

2.7 ENSURING VALID DATA

One measure of the validity of data is the statistical sufficiency of data for the process conditions under which it was collected and the variation in the data as displayed by its statistical parameters. See Section 6.

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 Section 6. 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.

2.8 MAKING THE PROGRAM COST BENEFICIAL

In order for a monitoring program to be cost beneficial 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 their man-hours 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 & 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.

2.9 ADDITIONAL BENEFITS OF PERFORMANCE MONITORING

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

Operating practice input includes:

(*a*) Capacity increases are possible 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 Section 7, Cycle Interrelationships, and Section 9, Performance Optimization.

(*b*) The procedure for starting, loading, and stopping major unit auxiliaries such as pulverizers, fans, pumps, and precipitators over the load range can be optimized for power consumption.

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

(*d*) Test instrumentation can serve as an audit of normal plant instrumentation.

(*e*) 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 can be clarified via performance monitoring.

Typical design revision decisions which can be influenced by performance monitoring include:

(*a*) turbine control valve operating modes and throttle pressure set point selection;

(*b*) boiler surface ratio adjustment needs and benefits can be established. Ratio adjustments can improve steam temperature and reduce spray flow.

(*c*) sootblower additions;

(*d*) variable speed motor drives;

(*e*) feedwater heater replacement performance specifications;

(*f*) condenser tube replacement specifications.

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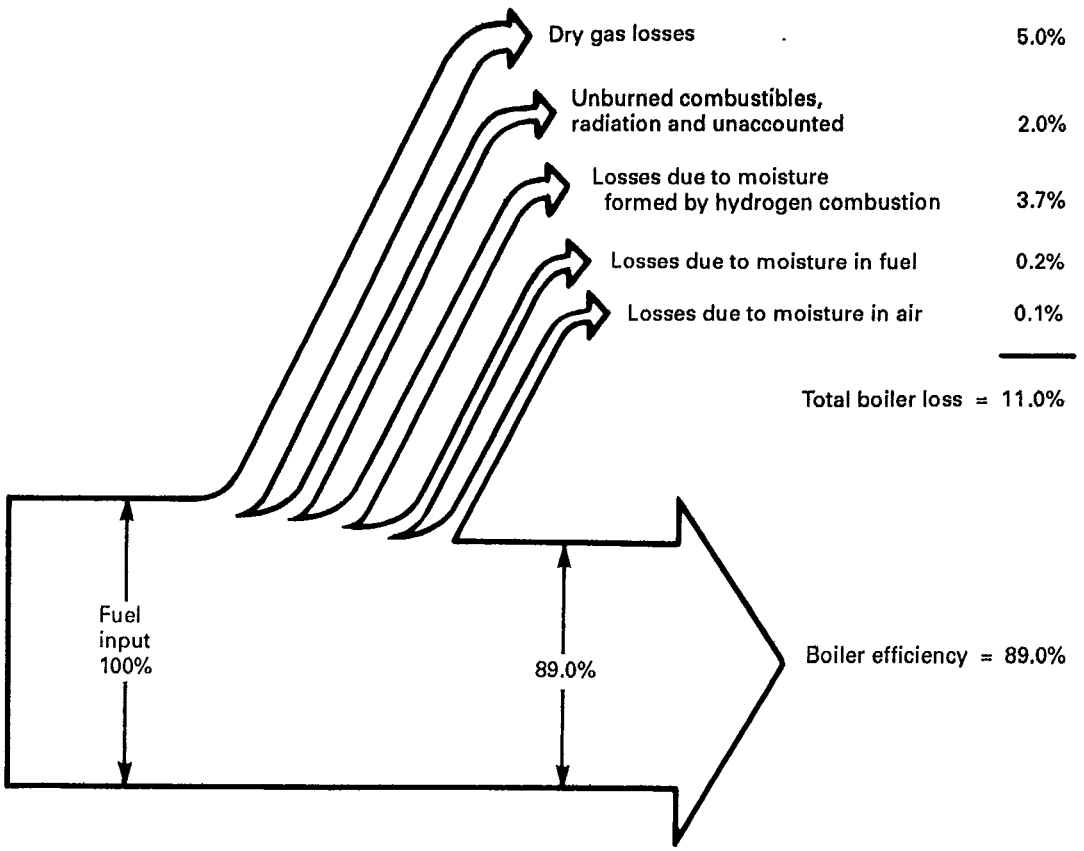

FIG. 2-2 TYPICAL BOILER LOSSES
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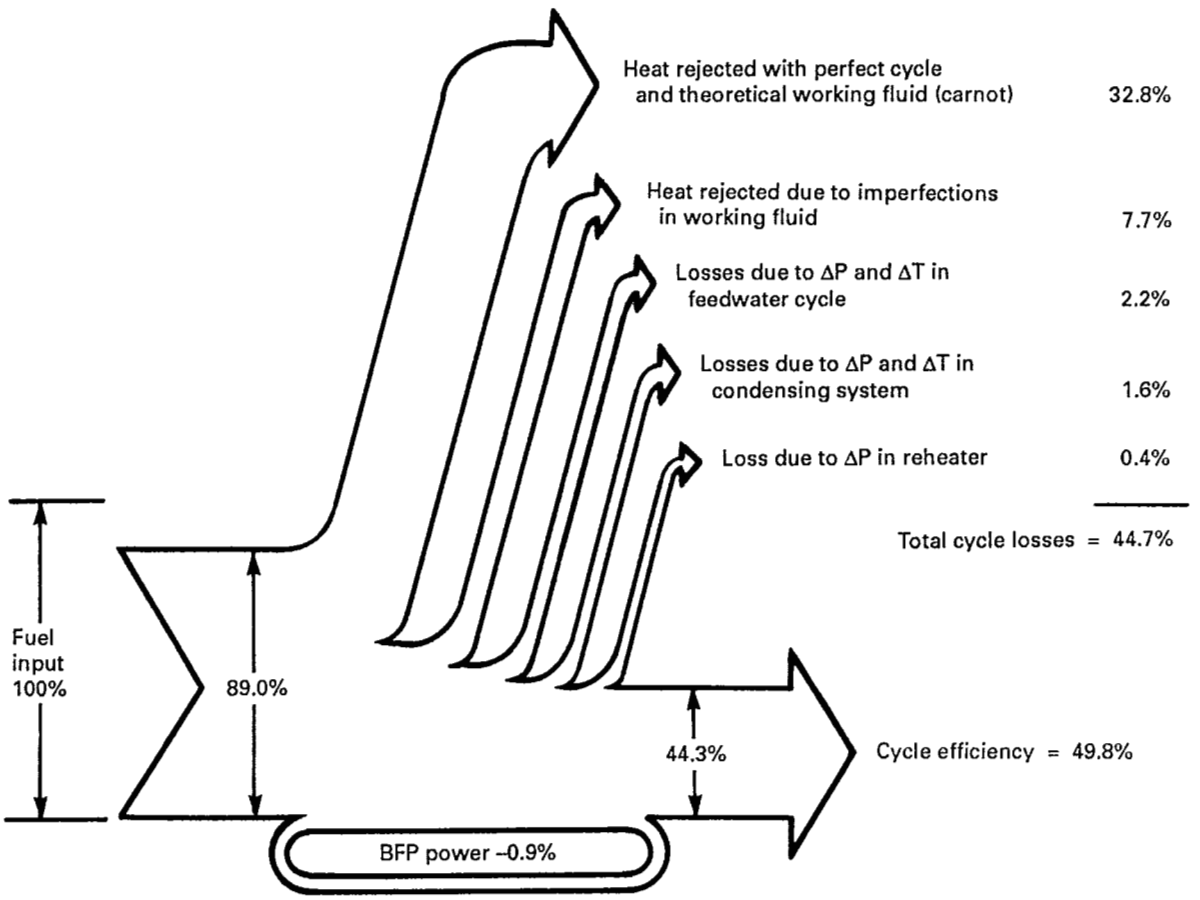

FIG. 2-3 TYPICAL CYCLE LOSSES
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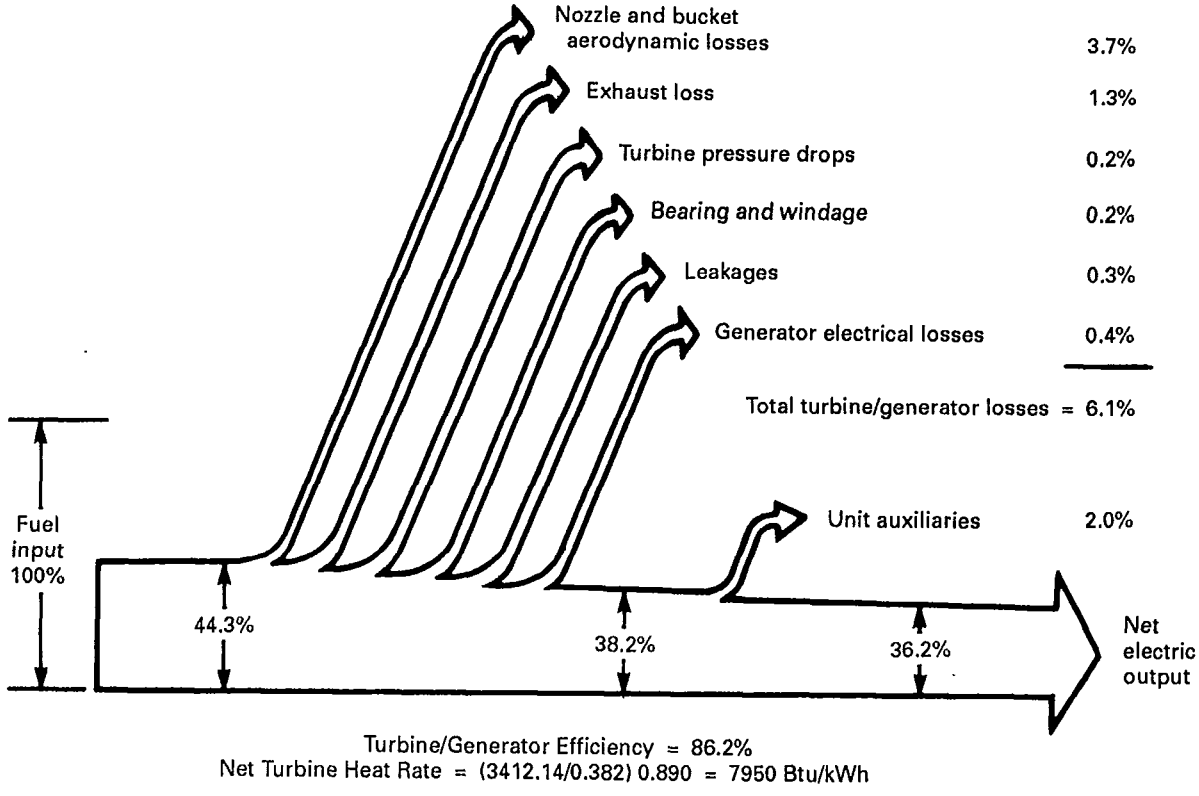

FIG. 2-4 TYPICAL TURBINE/GENERATOR LOSSES
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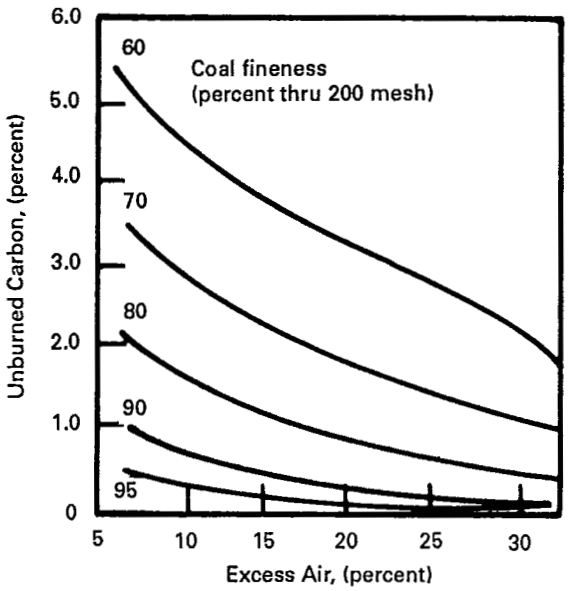

FIG. 2-5 COMPUTED VARIATION OF UNBURNED CARBON WITH EXCESS AIR
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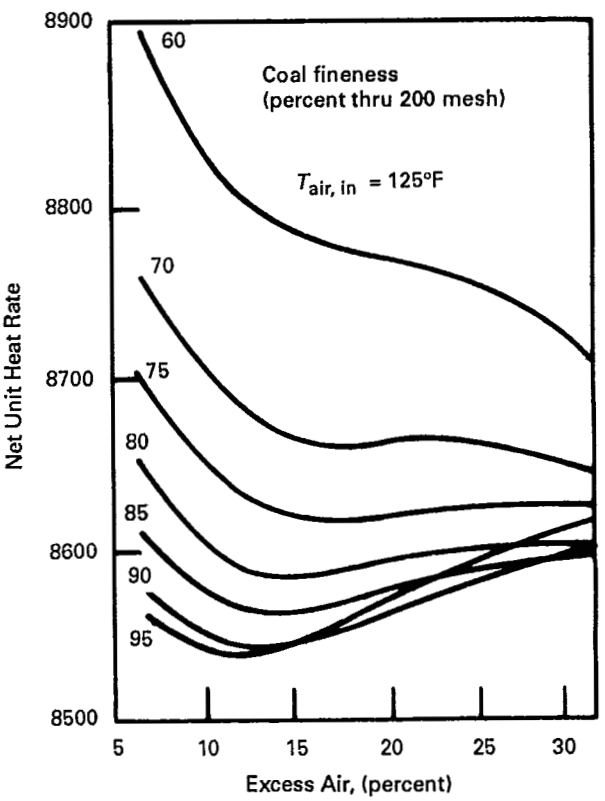

FIG. 2-6 EFFECT OF O_2 AND COAL FINENESS ON UNIT HEAT RATE
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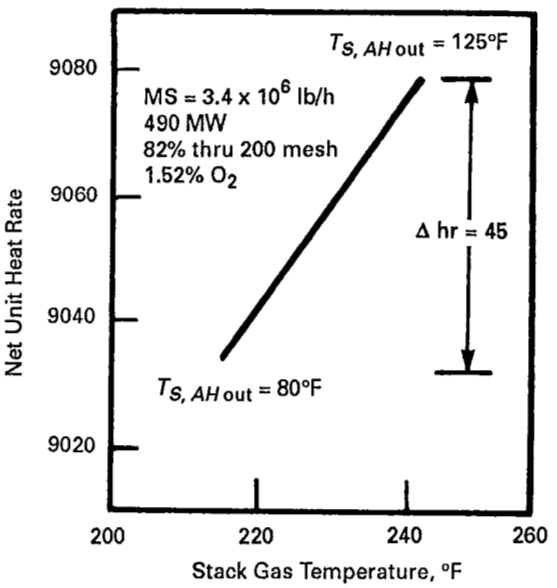

FIG. 2-7 EFFECT OF STACK GAS TEMPERATURE ON UNIT HEAT RATE
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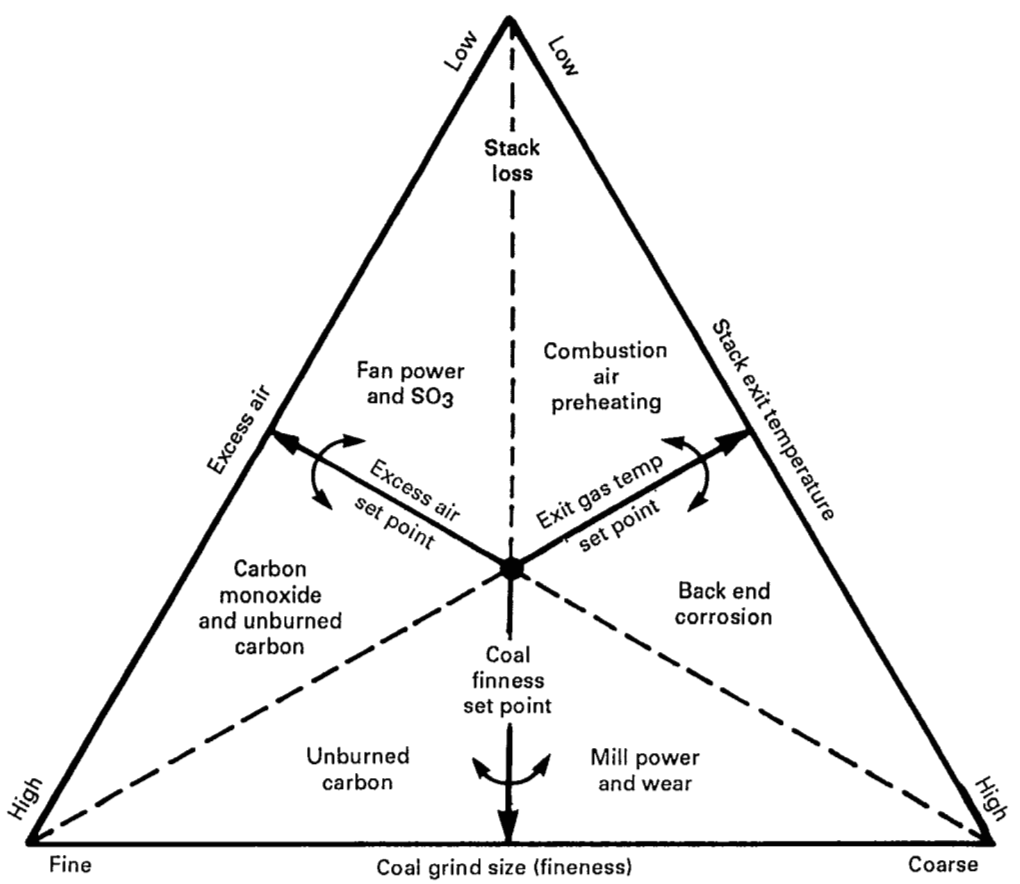

FIG. 2-8 BOILER LOSS OPTIMIZATION

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SECTION 3 — DEFINITIONS AND DESCRIPTION OF TERMS

The intent of this Section is to include terms used in these guidelines as well as additional ones of a general nature specific to performance monitoring. They are 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 which 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*

boiler air-inleakage — 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

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

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-inleakage — 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.

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 offdesign or non-standard conditions

cycle isolation — 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

excess air — the air in excess of the stoichiometric requirements

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excess oxygen — the oxygen present in the products of combustion

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 which 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 on a wet or dry 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, net — the ratio of the total energy input to the unit to the net electrical generation

heat rate, 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

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

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

loss due to unburned carbon — heat loss expressed in Btu per pound 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 losses

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 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 which 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 which can be measured or calculated which 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 which 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.

preventive maintenance — maintenance activities which are performed on a scheduled basis, usually following manufacturer recommendations

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 encoun-

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tered in the reheat section of the boiler including piping

resolution — the smallest observable increment of measurement

sequential valve (partial arc control) — the operation by which the steam flow arc control into a turbine is governed by opening one or more control valves sequentially

single valve (full arc control) — the operation by which the steam flow into a turbine is governed by opening all control valves simultaneously

sliding pressure — see *variable pressure*

station electrical power — total electrical power used at the station. This includes auxiliary equipment electrical power and power used by support facilities (i.e., 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.

throttle flow — steam flow at the turbine inlet

turbine choke point — the operating condition at which further reductions in pressure at the turbine exhaust flange result in no increase in turbine output for a given set of upstream conditions

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.

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

unburned carbon — carbon in the fuel which has not changed to CO or CO_2 during the combustion process

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 which 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

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SECTION 4 — PROGRAM PLANNING

4.1 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 insure the proper assignment of those resources — both financial and human. It must pinpoint roles, functions, and responsibilities. It must establish reasonable and realistic goals and schedules. It must also be flexible, able to accommodate changes in direction, 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 heat rate logic tree
(*f*) monitoring requirements
(*g*) data acquisition
(*h*) instrumentation
(*i*) uncertainty analysis
(*j*) data archival and retrieval
(*k*) results reporting
(*l*) budget allocation
(*m*) cost benefit analysis

The plan needs to be carefully thought out in advance at both the general and the detailed levels. The program plan must be geared toward accomplishing the identified objectives. Exactly what approach will best serve the user will be a function of the objectives, the user's time frame and available resources, and other relevant factors. Information contained in this Section 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 actually being used.

4.2 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 having 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 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*) maintenance planning prior to outages;
(*g*) maintenance evaluation following maintenance activities.

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

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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.

4.3 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 heat rate teams, or an integrated corporate-wide program. Either type should match existing utility organizational structures. It may also be appropriate to use a combined approach. A heat rate 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 given to 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 may be a natural tendency in performance monitoring work to concentrate on mechanical matters of equipment and units. However, there are extremely important people-related issues which 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 which are party to large-scope technical undertakings is not attempted herein. However, certain key human elements are listed which should be taken into consideration in the planning, conducting, and managing of the program. Some of these include:

(*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, will increase the benefits attained, and will help in avoiding pitfalls and traps that may not be recognized through purely engineering evaluations and management assessments.

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

4.4 AVAILABLE INFORMATION

Another activity that needs to be accomplished early in the planning stage is to determine what performance information is already available. 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*) as-built heat rate information

(*c*) equipment modifications that have altered as-built heat rate

(*d*) differences between design criteria and current parameters such as fuel analysis, ambient conditions, etc., that affect heat rate

(*e*) results of heat rate tests

(*f*) observations of knowledgeable personnel

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

4.5 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 at startup. Determining the level of as-built performance may consist of reviewing acceptance test data, simplified baseline test data, operational startup data, or 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 lesser known 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.

4.6 CONSTRUCTION OF HEAT RATE LOGIC TREE

Performance parameters which contribute to heat rate deviations can be identified with the aid of heat rate logic tree diagrams [2, 3]. The heat rate logic tree is intended to be a diagnostic tool for identifying the root cause of heat rate degradation (see Section 8, Diagnostic Techniques). The logic tree is structured to provide a set decision process by which the person using the tree can determine the cause of a problem by successively narrowing the problem scope based on available information. The 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.) which 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 which 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 $<$ 1000 degrees 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/graphs of parameter values versus plant conditions, checklists of the status of various equipment, or references to tests which 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.

4.7 MONITORING REQUIREMENTS

During the records review, information will be collected which identifies specific areas within the plant that are contributing the most 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
(*b*) Turbine
(*c*) Cycle Heat Rejection
(*d*) Feedwater System
(*e*) Auxiliary Electric Power
(*f*) 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.

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4.7.1 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, could easily exceed one hundred. It is therefore unrealistic to expect to include all possible cycle interrelationships in any performance monitoring program, even the most sophisticated. Performance engineers are thus faced with the decision of which ones to accommodate in their ongoing programs.

A monitoring envelope concept (see Section 7, Interrelationships) is used to visualize the existence of cycle interrelationships. The envelope involves imaginary, but defined, boundary lines which surround the equipment component, system, or unit being monitored. Cycle interrelationships may be viewed as those interactions which 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 insuring that appropriate measurement of and compensation for any important interactive effects are included in the monitoring approach.

4.7.2 Diagnostics. During the planning stage one must look ahead and consider how best to analyze all the data which will be collected. The goal of diagnostics is to discover the root causes for performance degradation. With the end in mind the program planner can envision what inputs, instruments, software and hardware, and data acquisition equipment is required.

4.7.3 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 Section 9 during the planning stage.

4.7.4 Planning-Stage Questions. A group of questions appears below, the answers to which will lead the program planner to sound 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?

It will be recognized that 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.

4.8 DATA ACQUISITION

An important part of program planning includes determining the method of data acquisition that will be used. The data acquisition method chosen should allow for upgrading as new equipment and techniques become available. It 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 sys-

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tem continually updating a display in the control center. However, information for the results person may be needed periodically and 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 be conducted to determine the number of points to be measured and the method of data acquisition.

4.8.1 Using the Control System. It may be possible to implement a complete or partial performance monitoring system using existing plant control systems and/or plant computer systems. There are both advantages and disadvantages to this approach.

Using existing systems may or may not be less expensive, depending on the particular existing system under consideration, and the type of performance monitoring and calculations being contemplated. It might offer a reliability advantage, since great emphasis is placed on control system reliability (if the control system goes down, so may the plant). Additional displays might not be required for the control room operators, if the existing displays are used instead of new, dedicated displays.

Disadvantages may include lack of adequate computing resources required for the calculations, or the possibility of slowing down the control system response by overloading the processors. There may be fewer qualified personnel with the necessary knowledge for this approach. Development time may be very high, or the program may be difficult to modify. Certain commercial products would not be possible to implement under this approach. Graphic display resolution may be lower for existing plant control systems when compared to new graphics hardware.

It may be cost-effective or convenient to extract certain performance signals from existing control systems for use in a separate performance monitoring computer. 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. This option might not be available for older systems which have been since upgraded by the manufacturer, or may no longer be in production.

The interface hardware and software can represent a significant cost for certain types of systems. The level of support that is required from the manufacturer should be considered, along with the expertise of in-house personnel. Interface hardware and software may be in different stages of development or revision as new systems evolve. This may be a key consideration in the economic analysis of the justification for a performance monitoring system, or may impact the type of system that is purchased or developed.

On analog systems, use of isolation or buffer amplifiers is recommended to assure control security and to allow independent calibration adjustment. It is not recommended to share transmitters between control and monitoring functions on critical measurements such as throttle temperature and pressure, feedwater flow, etc. A separate monitoring transmitter, properly tubed and valved, can be calibrated without interruption of the control signal, thereby providing a check on the critical measurement between routine calibrations.

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, A/D converters, data loggers, 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.

If performance monitoring calculations are to be performed on the data acquisition hardware, data files may have to be created for transferring plant data to the calculation algorithm. If performance monitoring calculations are to be performed on a separate computer, communications protocol between the data acquisition system and the performance monitoring computer needs to be established. The physical connection is typically a serial interface or a high speed networking interface. Direct memory access may be used between the latest family of distributed control systems and a performance monitoring computer. On digital systems, a value fully converted to engineering units can be accessed from system memory. If an on-line monitoring system is to be used, it should also be determined whether the data acquisition system can be interfaced with the performance monitoring computer. Documentation relative to the data acquisition system's software programming libraries and the availability of a communications port should be reviewed.

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4.8.2 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 which 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. The input signal usually needs to be amplified to bring it up to a usable level. 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 analog output signal. Output current is normally 4–20 mA, and output voltage is normally between 0 and 10 volts DC.

An A/D converter is a device that converts an analog signal to a digital signal. The most important consideration with an A/D converter is the number of A/D converter data bits contained in the count register used to represent the actual binary data value of the measured parameter. A minimum of 12 bits is recommended; most A/D converters have between 14 and 17 bits.

The final step in the measurement process is converting the binary number to engineering units with a data processor.

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

4.8.3 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 dumb (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 main frame system.

All forms of remote systems offer the advantage of reduced cabling cost, reduced exposure to noise/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 determines 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, Section 6.

4.8.4 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.

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4.9 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 plant instruments. This list will be used many times 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 nozzle coefficients are being used in the calculation procedure.

Program planning in the area of instrumentation should also include developing an adequate calibration plan. This 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 reoccurring instrument problems. There are commercial software products available for storing calibration data and the calibration dates. An automated system for tracking instrument maintenance would help in maintaining adequate data quality.

Instrument installations need to be checked for correct installation practices. Instrument maintenance and calibration practices should be reviewed. These are discussed in Section 5.

4.10 UNCERTAINTY ANALYSIS

An uncertainty analysis can be performed to determine the overall uncertainty of calculating net plant heat rate 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. Conducting an uncertainty analysis consists of determining the influence 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 1 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 such as 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 which 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 percent rule [15]. Those instruments having an effect of at least 20 percent of the highest instrument should be considered for improvement.

4.11 DATA ARCHIVAL AND RETRIEVAL

With an on-line system, enormous amounts of measured and calculated data can be stored on hard disk drives, magnetic tape, optical storage devices,

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and others. Historical data is the most frequent performance picture that engineers and managers observe to make decisions about plant improvements. Steps must be taken during program planning to ensure that data, at an appropriate frequency, is being stored.

Most commercially available performance monitoring software packages have automated archival procedures and do not require user interaction at a terminal. Automated procedures should be established and implemented on any self-developed monitoring program.

All measured and calculated data for a specified time period should be saved at a user defined frequency such as once a minute, in a unique file, and time tagged. An example would be to save one day's worth of data in a unique file thus providing a convenient organization for data retrieval and off-loading data from the system to backup media. The file structure should provide retrieval of discrete data points for plotting versus time or any other variable.

Data compression techniques can be utilized to reduce disk storage requirements. These techniques include assigning a deadband to variables and only storing a value when the band width is exceeded. However, data resolution will become somewhat diluted depending on the size of the deadband. Whether or not data compression is used, all data should preferably be saved at the same frequency. Storing data at different time intervals creates more complex software and may misrepresent the condition of the plant at a discrete point in time.

4.12 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. Amounts of information needed and the frequencies of reports will vary widely, but universally in successful programs information is provided to and evaluated by management at appropriate levels.

Data should be placed in the hands of the people who can act on it. Plant operators need on-line data relative to operator controllable parameters. Results engineers need raw and calculated thermodynamic data and performance parameters, primarily historical information. The maintenance department needs the information to assist in setting priorities on maintenance activities. Management needs reports consisting of trends, monthly summaries, percent improvement, and cost savings.

A good example for illustrating the need for interchange of common performance data is coal and fuel oil inventories. The case of coal is most interesting. The fuel buyer needs to determine end-of-year inventories. He usually purchases coal that is weighed at the point of origin. Coal consumed at the plant is normally measured via belt scales and coal feeders. Fuel inventories can be calculated by taking into consideration transportation losses and scale readings or determined by coal pile surveys.

Fuel oil can present its own set of problems with respect to measuring fuel consumption. Fuel oil measurement is discussed in Section 5. Natural gas meters are subject to AGA 3 calculations. The performance engineer should consult published literature prior to using natural gas meters as the primary measurement of fuel consumption.

With the addition of an on-line performance monitoring system, the performance engineer can calculate fuel consumption via a number of methods. These include the heat-loss and output-loss methods. The problem occurs when the calculated values do not agree with the measured values. The performance engineer needs to determine the criteria to use to account for differences. During the planning stage he needs to discuss alternatives with all the people involved and establish mutually agreeable criteria for determining fuel inventories.

4.12.1 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. Report format may include tabular listings, graphs, bar charts, pie charts, etc.

The program plan should determine whether reports are to be produced manually or automatically by computer. Data reduction methods are discussed in Section 6; 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. Some of these reports may exist as real-time displays for unit operators; some may be printed information for operator use; some may be real-time, computer-accessed, or printed reports for management review and control.

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Reporting is simply the providing of information in various forms to the different groups with technical and management functions. Some reporting is needed in real time directly to the people with hands-on roles in the work. This is definitely the case with operators and at times with engineers in the operational optimization process. Other people have need for 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. Ultimately this information may be reduced to fundamental assessments of whether the program objectives are being met, how effective is progress toward them, are the defined approaches working or do they need to be modified, what performance changes are occurring, what are the values of those changes, how much are they costing to achieve and how many people are being committed to the process.

4.12.2 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 manpower, 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 with the monitoring process which are absolutely 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 actually 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 the taking of 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, 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.

4.13 BUDGET ALLOCATION

A budget should be prepared during the planning stage to serve as a guideline for a complete review of the program requirements. The resources required to meet the established goals of the project need to be determined. These resources include primarily the people 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. This budget should reflect the difference between the available resources and the resources needed to satisfy the requirements of the program plan.

Financial resources are needed to support the monitoring efforts. Cost outlays are often necessary to obtain larger financial returns. There are significant cost considerations in the area of mechanical optimization, where the anticipated value of performance changes may be great, but the outlays to obtain them may also be great.

It is necessary to face the issue of program cost at the same time as anticipating savings, since net program worth has to consider both sides of the equation. The matter of costs in many, if not all, cases will be a top priority concern, one that may establish some of the most pressing constraints under which the program must function.

It is recommended that program costs be highlighted for management consideration in the early stages of planning. This will prevent such costs from coming as a surprise when they actually arise, and will allow for them to be considered, planned for, and allocated in advance of the actual need. Of further help in securing necessary financing for specific expenses would be the development of approval guidelines on benefit-to-cost ratios and

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payback periods. This would assist not only the organization's management in evaluating specific funding requests, it would also assist those directly engaged in the work by providing a defined structure of minimum acceptable returns on performance monitoring investments.

4.14 COST BENEFIT ANALYSIS

Finally, a cost benefit analysis may be required to provide justification for the project. This is the most important phase of program planning and also the most difficult. It is difficult because prior estimates of performance improvements are not always quantifiable. Several steps can be taken to produce credible numbers.

The first step is to establish as-built performance parameters such as heat rate. This activity has been demonstrated at several EPRI-sponsored heat rate improvement projects [16, 17]. This can be a long and arduous task owing to the usual shortage of historical plant data.

The next step is to determine best achievable performance by considering the effect on performance of differences between current operating parameters and design parameters. The result is the current best achievable performance. Next, the current unit performance should be measured. The difference between current and best achievable performance is a measure of potential improvement.

Improvements in performance can be accomplished by either operational changes or 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. The cost associated with operational changes might include the cost of switching fuel. 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 [18].

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 resources are being allocated — both human and financial — 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 the value of those performance levels, and to assess the total cost of the program itself.

It may be 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 to cope with the improvement projects will be justified together with the projects. This will minimize the pitfalls of justifying performance monitoring programs which are too ambitious and over-optimistic.

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 actually 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 which are hopefully tied to very high levels of performance already being achieved.

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SECTION 5 — INSTRUMENTATION

5.1 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 Section provides guidelines for the selection, use, and calibration of instrument equipment for performance monitoring. This Section references Performance Test Code documents for the 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.

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 good instrument accuracy and repeatability are critical to a performance monitoring program.

5.1.1 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.

5.1.2 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. 4.10, 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 Sections 4 and 5.

5.2 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 really involves a system.

This system includes the generator, instrument transformers, connecting wires, all other devices in the same circuit, and, finally, the wattmeter. All these elements can affect the indicated power, thus greatly complicating power measurement. In addition, there are two different methods of connections

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for a three-phase generation system which 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.

5.2.1 Unit Watthour Meters. Three element watthour meters used for unit generation accounting generally have a ±0.50 to ±1.0 percent 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 1/6 of a disk rotation.

5.2.2 Power Transducers. High accuracy two- and three-phase transducers that indicate kW or integrate kWhr are available with uncertainties of ±0.2 percent. 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.

5.2.3 Indicated Electrical Output. This type of electrical output measurement is usually the least accurate [maybe ±5 percent]. 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.

5.2.4 Power Factor. Power factor is needed along with hydrogen pressure and purity to calculate electrical losses in the generator. Power factor is important if it is very different from design.

5.2.5 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 five amperes 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.

5.2.6 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 19.6-1955 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.

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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.

5.3 MEASUREMENT OF STEAM AND WATER FLOW

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.

5.3.1 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 one percent error in flow causes a one percent error in calculated heat rate.

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. If the primary flow element is located at the boiler inlet (final feed), generally only a few instruments around the last feedwater heater are required to calculate heat rate. 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 PTC 6.1-1984 (para. 4.35 and Fig. 4.3A).

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 verses 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 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 very 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.

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5.3.2 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.

5.3.3 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.

5.3.4 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.

5.3.5 Circulating Water Flow Measurements (PTC 18). Pitot tubes can be used to accurately measure circulating water flow, provided there is a straight accessible section of circulating water pipe which 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, it 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 then it can 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.

5.3.6 New Technologies. New technologies offer other flow measurement options. Ultrasonic flow meters offer a non-intrusive 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 them. 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 which 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.

5.3.7 Additional Considerations. 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.

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 taken only at 5 min intervals. Remember that most flow calculations are based on psid, not inches of water. Great care must be exercised when using water manometers. The psi 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 psid.

While properly instrumented heat balances around feedwater heaters provide excellent calculated mass flow determination (para. 5.5.3), 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

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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 thermowell can a cause large error in the calculation.

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

5.3.8 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.

5.4 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 a same fluid as in a differential pressure measurement, ambient pressure as in gage pressure, or zero pressure as in absolute pressure. Figure 5-1 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 5-2 displays the general accuracy of commonly used instruments. The following list of instruments can be used to measure pressure in performance monitoring.

5.4.1 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).

5.4.2 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 are not easily automated.

5.4.3 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 percent accuracy readings are acceptable; however, they must be calibrated in place to obtain this uncertainty.

5.4.4 Deadweight Gages. The deadweight gage is an accurate pressure measurement device which 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 which could cause the piston to stick. Its main advantage over other instruments is its accuracy of up to ±0.1 percent 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.

5.4.5 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

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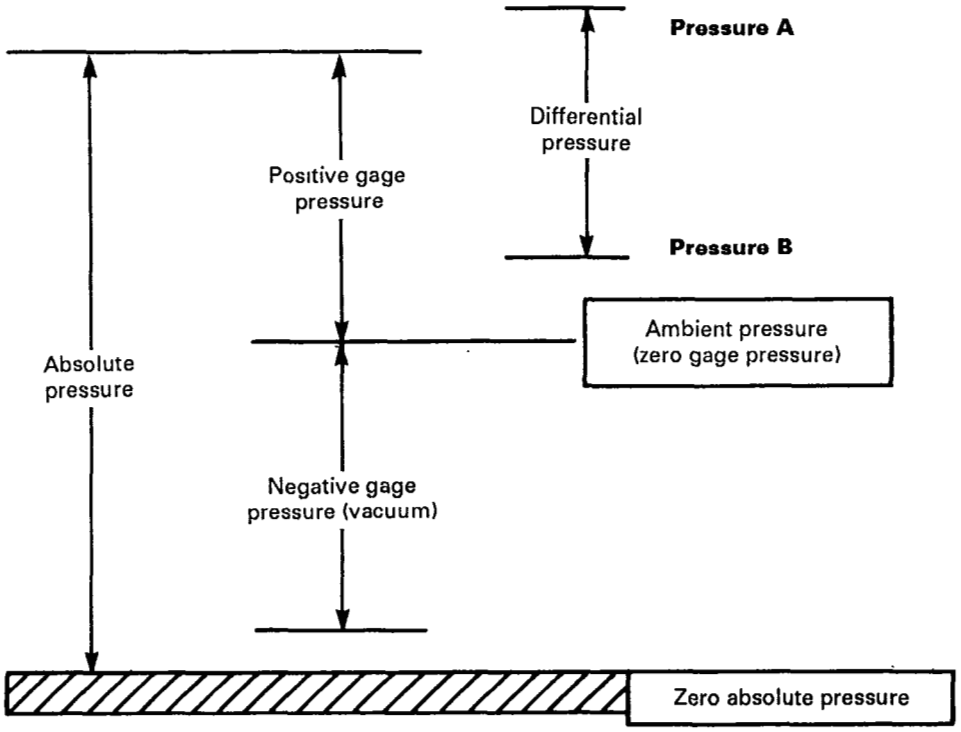

FIG. 5-1 BASIC PRESSURE TERMS FROM ASME PTC 19.2

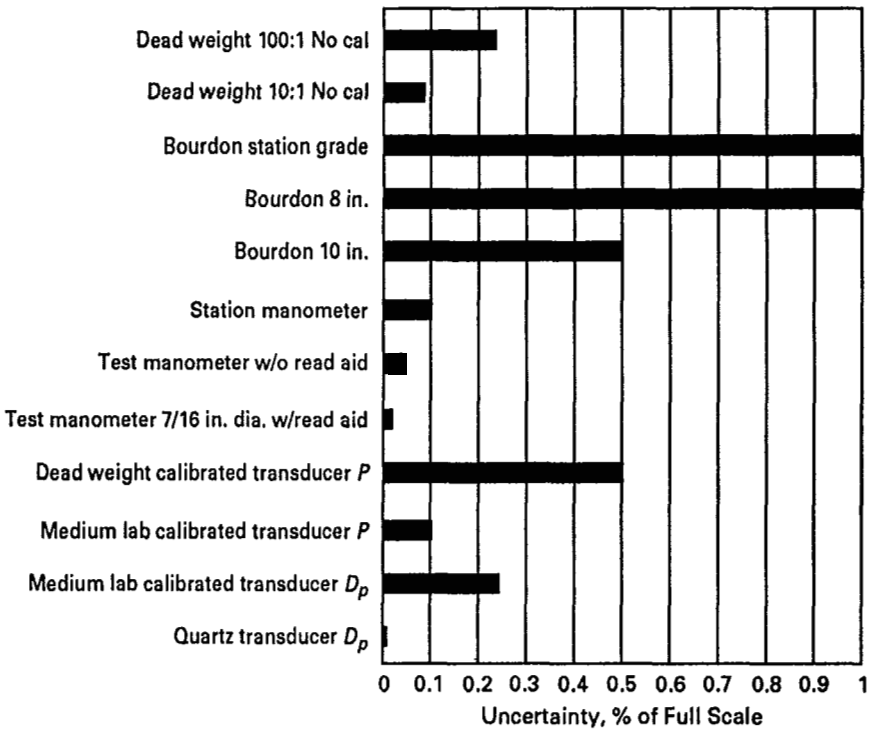

FIG. 5-2 GENERAL UNCERTAINTIES OF PRESSURE MEASURING DEVICES FROM PTC 6 REPORT

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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. 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. In instances where the instrument is below the source, the correction, known as a water leg correction, is subtracted from the reading. The water leg correction is added if the instrument is above the source, provided the existence of the water leg is determined. In general, liquid-filled lines should be routed from the source to the instrument in a manner such 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 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 lines which necessitates frequent manual or automated purging.

Consideration of the goals of the monitoring program will yield useful pressure test points. 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: lower flow yields a larger rise.

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 is shown in Fig. 5-3 of a high pressure (HP) turbine with a base efficiency of 83% at VWO. The bias lines are examples of efficiency change due to a 10 psi shift in both throttle and a 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 percent shift while a ±10 psi error causes a ±1.2 percent change.

In Fig. 5-4 there is a similar analysis of an intermediate pressure (IP) turbine. Notice that for the same bias amount ±10 psi is now worth ±3 percent while the ±10 psi error in exhaust pressure is now worth ±4.6 percentage points. This indicates that identical pressure measurement errors and bias can triple the influence on IP turbine isentropic efficiencies as they do on an HP turbine.

5.4.6 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 recommended calibrations methods.

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, para. 9.3.1). 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.

5.5 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 tem-

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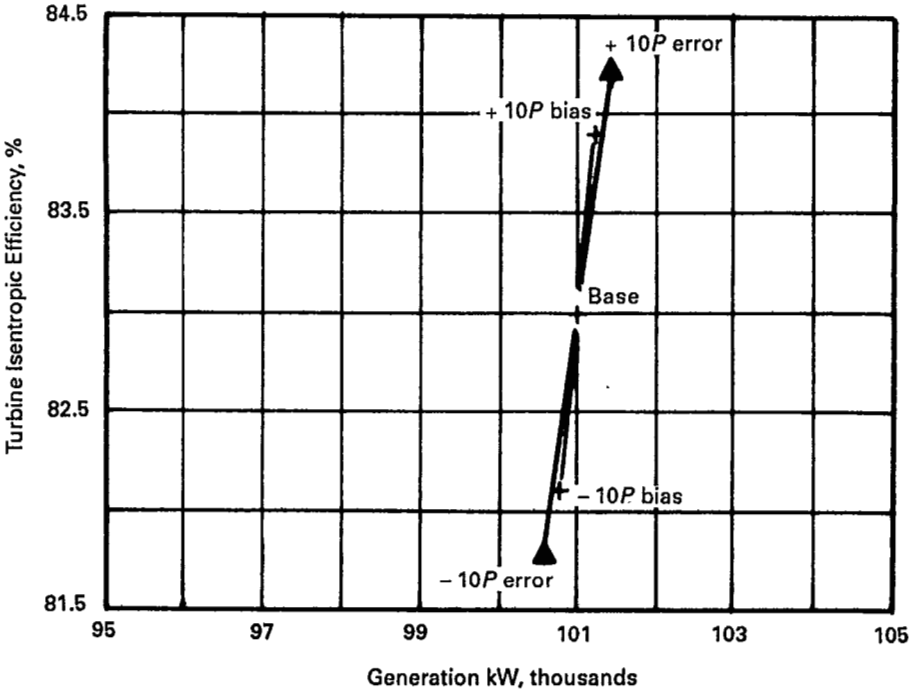

FIG. 5-3 EFFECT OF PRESSURE AND BIAS ERRORS ON HP TURBINE EFFICIENCY

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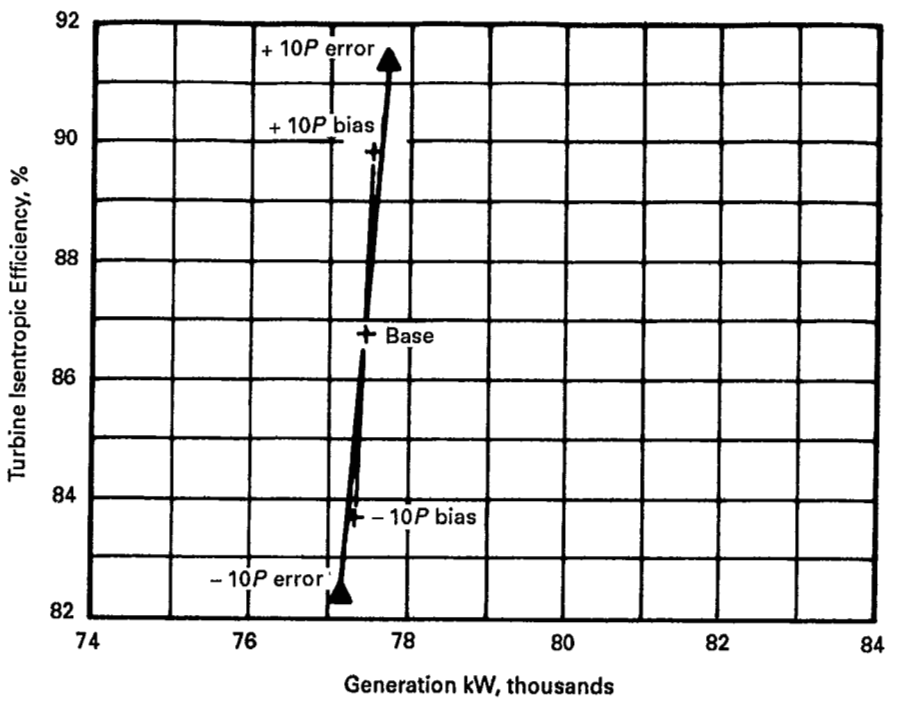

FIG. 5-4 EFFECT OF PRESSURE AND BIAS ERRORS ON IP TURBINE EFFICIENCY

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perature instrumentation commonly used in power plants for performance monitoring. Figure 5-5 displays the general accuracy of commonly used temperature instruments.

5.5.1 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, 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 read-out systems where thermocouple signals are jumped or paralleled can be done with care but are not recommended. Note that some types of thermocouples corrode in certain environments, and some thermocouples have very low or no signal at low temperatures. Types E or K are preferred because of their higher signals.

Thermocouples can be used over a very wide range of temperatures and can withstand more abuse than RTDs (para. 5.5.2). 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 can be obtained with leads inside a heavy stainless steel sheath which 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 that the couple is exposed (see Fig. 5-6).

5.5.2 Resistance Temperature Devices. A resistance temperature detector (RTD) is a conductor of a known resistance which 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, 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 5-7 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. PTD 19.3-1974 indicates that a 20 percent 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 212°F, thermistors are very sensitive and typically have temperature coefficients of -2 ohms/°F.

5.5.3 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, 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 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.

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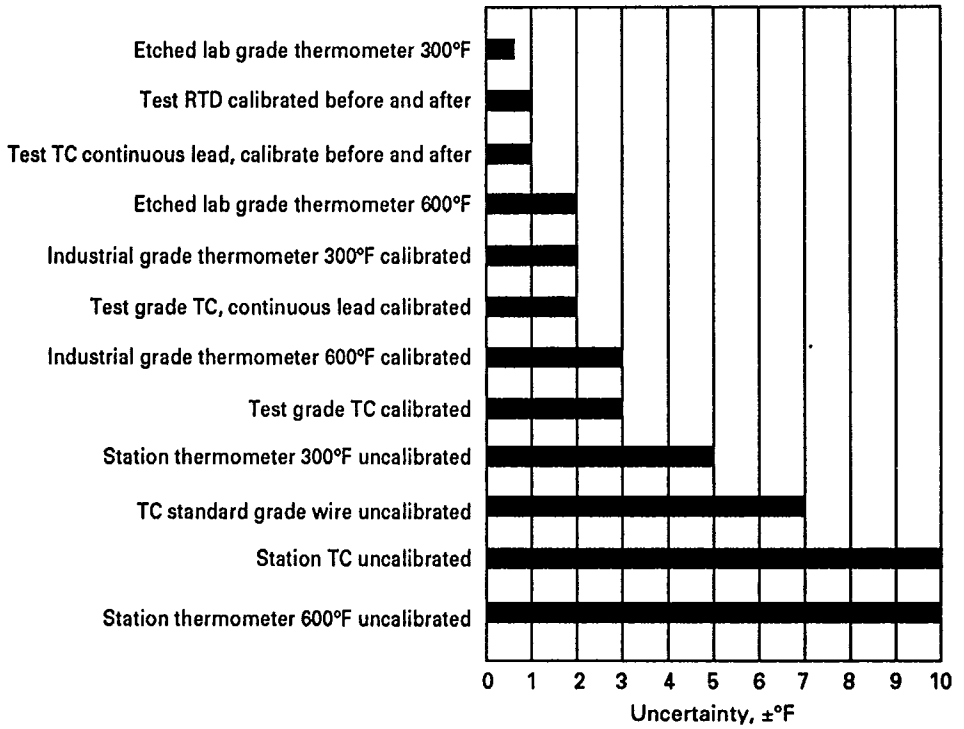

FIG. 5-5 TEMPERATURE MEASUREMENT DEVICE UNCERTAINTIES FROM ASME PTC 6 REPORT

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 only a few degrees change over time which can be obscured when temperatures are combined.

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 an HP turbine is shown in Fig. 5-8 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 a 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 percent shift while a ±10°F error causes a ±3 percent change.

Figure 5-9 displays similar analysis of an IP turbine. Notice that the same bias about ±10°F is now worth ±4 percent while the ±10°F 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).

5.5.4 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.

5.5.4.1 Thermocouple Calibration. It is recommended that sample thermocouples from each

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lot be checked against reference tables (see PTC 19.3, Section 3, paras. 25 and 40). 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.

5.5.4.2 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 IPTS-68 (International Temperature Scale of 1968) corrections. For more details on the Calendar equation see PTC 19.3.

5.6 MEASUREMENT OF AIR AND FLUE GAS FLOW

Air or flue gas flow rates are needed for stackloss, 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 which are functions of other flow measurements. The most accurate methods require multiple-point, three-dimensional traverses of ductwork. 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 would allow boiler efficiency calculations without fuel flow or heating value measurements.

5.6.1 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 addition will allow an in-service calibration with a pitot traverse (para. 5.6.2).

5.6.2 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. However, it is a complicated traverse. A five-hole probe can resolve the total velocity vector [see ASME PTC 11].

5.6.3 Inferred Measurements. ASME boiler heat loss method as described in PTC 4.1 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.

5.7 MEASUREMENT OF FUEL FLOW

Accurate fuel flow measurement is difficult. Whether the fuel is gaseous, liquid, or solid, measurement of each type of fuel has its own set of different problems.

5.7.1 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. Beside 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 those 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 their virtually total electronic determination of mass flow. They

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are a viable alternative for new or retrofit installations where operational, accounting, or performance purpose fuel measurement is required.

5.7.2 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 AGA method of calculation of supercompressibility of hydrocarbons. This method requires knowledge of the specific gravity of the flowing fluid.

5.7.3 Coal Flow Measurements. There are three 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 Section.

5.7.3.1 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, then it 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 the after 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 Section 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.

5.7.3.2 Conveyor Belt Scales. Conveyor belt scales are used to measure both received and consumed coal for utility boilers, and oftentimes are 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:

(*a*) certified for basis-of-payment

(*b*) non-certified

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 the NIST HB44. All conveyor belt scales should be calibrated using a weighed material load test whenever possible. However, this method of calibration is required by the NIST HB44 for certified conveyor belt scale systems (NIST HB44,

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Section 2.21. N.11). A material test consists of preweighing or post-weighing a quantity of material, to be passed across the belt scale, on a certified static reference scale (NIST HB44, Section 2.21. N.2 & N.3.2 thru 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 shall be conducted. On subsequent verifications, at least two individual tests shall be conducted (NIST HB44, 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 which will relate the results of simulated load test to the results of the material test (NIST HB44, 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 41°F (NIST HB44, 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 HB44, 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, static weights, or electronically, depending on the scale manufacturer's recommendations and user preference (NIST HB44, 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 generally a calibration adjustment is 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 which 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 HB44, 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%; adjust 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.

5.7.3.3 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 frequency and with the same methods as other belt scales whenever possible. Accuracies of gravimetric feeders are normally ±0.5%, and the user should consult the feeder vendor for actual accuracy value.

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 which must be converted to mass flow rates based on periodic calibration.

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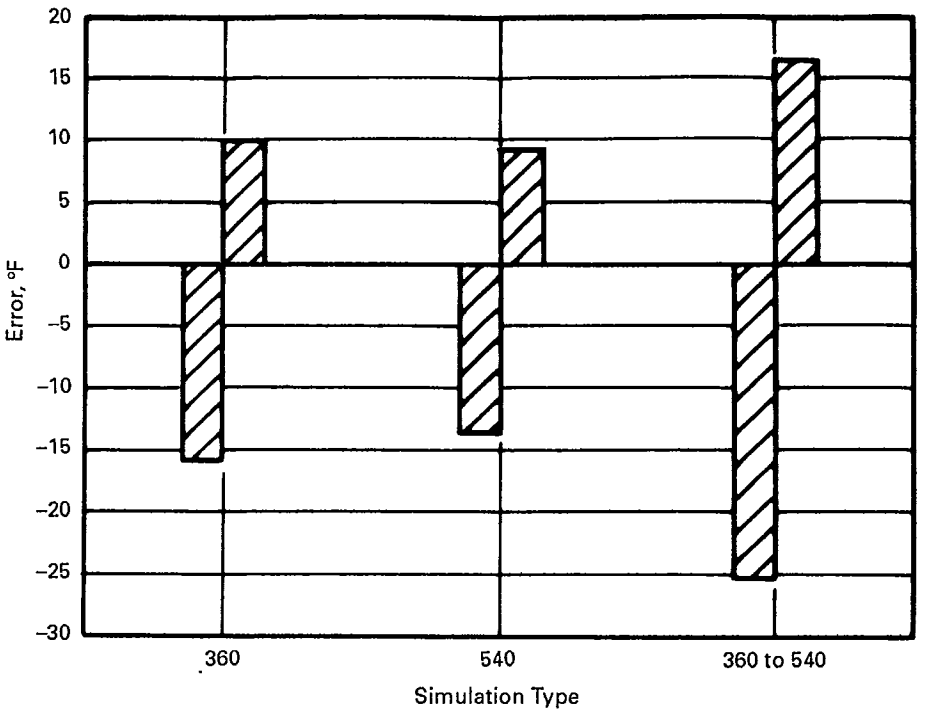

FIG. 5-6 TC DRIFT STUDY OF 6 THERMOCOUPLES CYCLED 210 TO 300 DAYS

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Obviously, coals 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.

5.8 MEASUREMENT OF FLUE GAS COMPOSITION

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, 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.

5.8.1 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 paramagnetic sensor, wet electrochemical or a zirconium oxide cell. The zirconium oxide cell must operate at a high temperature which makes it a natural choice of in situ measuring systems. The paramagnetic sensor and wet electrochemical cell require a cooled and dried sample (see 5.8.3). The wet electrochemical cell is lightweight and is rugged which means that it is a good choice for a portable analyzer also. The electrochemical cells have a limited life and must be replaced on a routine basis. The paramagnetic sensor is a very delicate device which 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 is more appropriate for the monitoring of total combustibles. Infrared absorption analyzers must have dried and cooled samples. They are quite accurate and require no cell replacement.

5.8.2 Orsat Analyzer. One commonly used instrument for flue gas analysis is the Orsat analyzer. The Orsat consists of a measuring burette and three reagent pipettes which 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.

5.8.3 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 Section 6 for 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. Beware that some gases interfere with the desired measurement (i.e., CO_2 is read as CO). Also consider that expanding the range of a CO analyzer to include upset conditions (>1000 ppm) will diminish the accuracy of the analyzer in lower useful range (50 to 400 ppm) where the high resolution is needed.

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

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 true average value. A multipoint grid must be used to obtain good flue gas measurements.

5.8.4 Calibration. Electronic gas analyzer requires 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.

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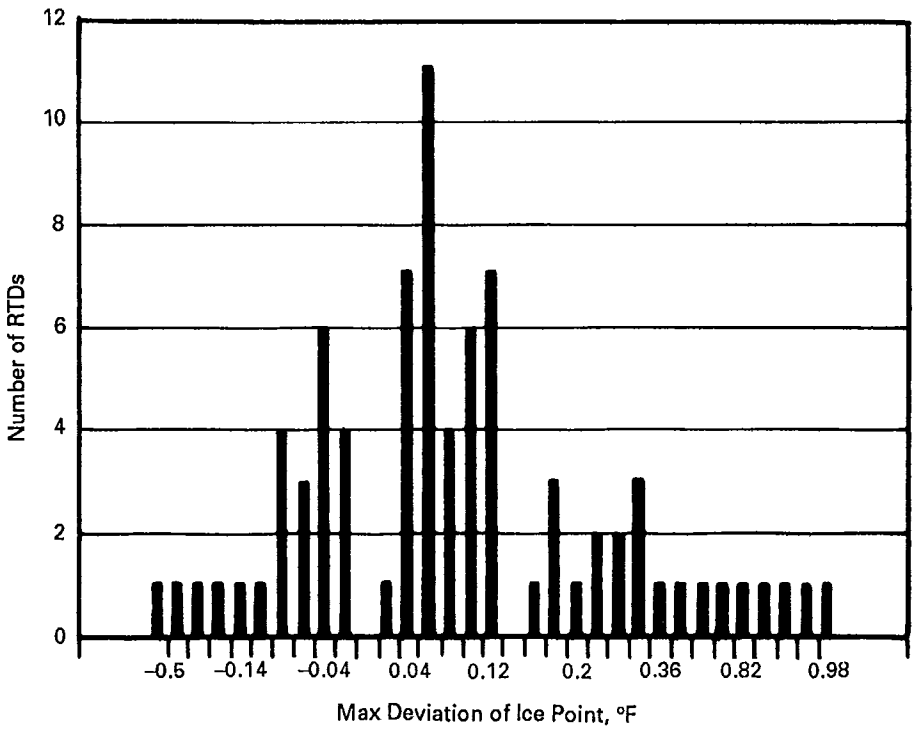

FIG. 5-7 DRIFT OF ICE POINT RESISTANCE OF 102 RTDs CYCLED 810 DAYS

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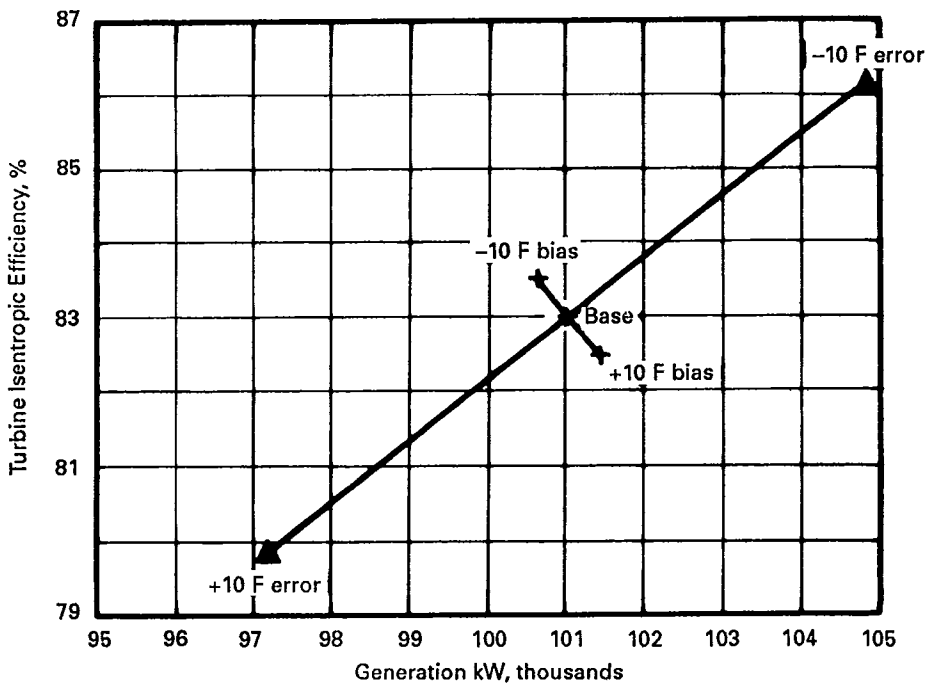

FIG. 5-8 EFFECT OF TEMPERATURE BIAS AND ERROR ON HP TURBINE EFFICIENCY

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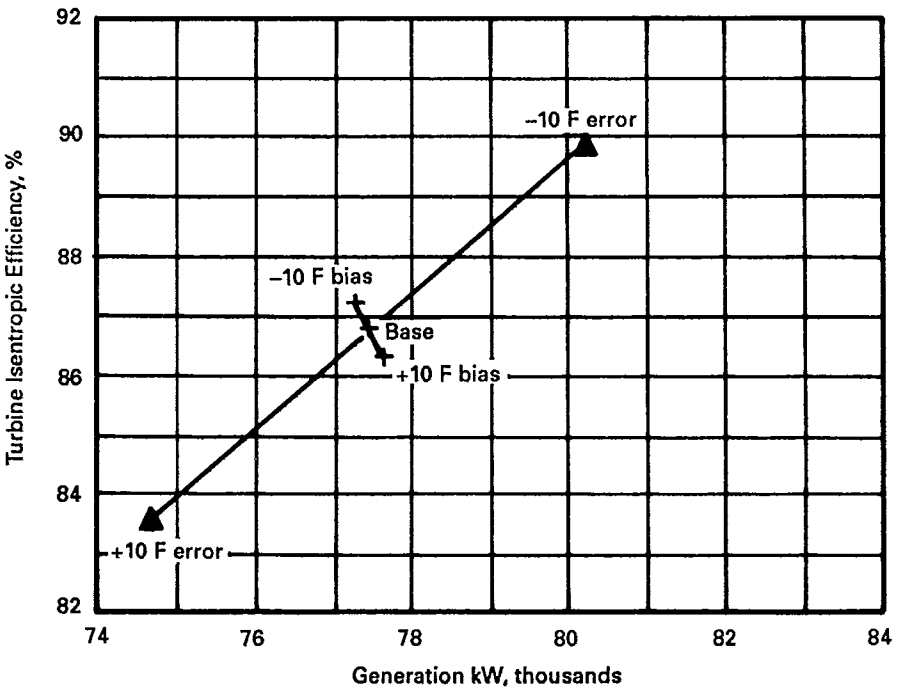

FIG. 5-9 EFFECT OF TEMPERATURE BIAS AND ERROR ON IP TURBINE EFFICIENCY

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SECTION 6 — PERFORMANCE MONITORING IMPLEMENTATION

6.1 GENERAL

Implementation of a performance monitoring program requires an understanding of performance parameters and their effects on equipment and unit efficiency. In addition, identification of performance degradation, testing, data validation, calculations, and results reporting are all essential parts of a performance monitoring program. This Section covers the basic information needed to implement a performance monitoring program for turbine, boiler, and balance of plant equipment.

6.2 TESTING OVERVIEW

Performance testing of plant cycles, subcycles, and component equipment should be performed to ensure information of sufficient detail and accuracy to properly identify degradation. In addition to providing a more in-depth review of the operation of the unit and component equipment, testing will identify and resolve more complex problems before they have a significant detrimental effect on performance. Expensive and time-consuming major equipment overhaul programs can also be evaluated to ensure proper improvement and scheduling frequency. Testing may range from contractual acceptance/baseline testing for indexing expected levels of performance (new equipment, rebuild, or overhaul) to routine testing under actual operating conditions.

Testing should be performed at regularly scheduled intervals. Good engineering judgement should be used when determining frequency, bearing in mind the need to increase frequency when sudden or abrupt performance deterioration is suspected.

6.2.1 Test Procedure. Testing should be conducted using guidelines established by the Performance Test Codes. Prior to testing, a test procedure should be developed and issued to all participating parties. The procedure should address such items as the purpose of the test, the expected results, instrumentation, test schedule, operating conditions, estimated cost, potential safety risks, etc.

Calibration of instruments should be performed prior to the test and checked for drift after the test. NIST traceable calibration standards are recommended for calibration. Pressure sensing instruments should be calibrated at several increments in the expected range of readings. Temperature sensors should be calibrated at different immersion depths and at sufficient intervals of temperature range for the desired accuracy. The ASME PTC 19 Series should be consulted for calibration requirements applicable to the desired type of instrumentation.

6.2.2 Setup of Test Conditions. Preparations for testing should be as thorough as possible to prevent the occurrence of subtle procedural problems. Unlike instrument problems that are more obvious and can be resolved during the testing period, problems resulting from poor testing procedures are not likely to be revealed until after the results have been calculated and it is too late to rectify.

In setting up the system(s) or component equipment for testing, all conditions influencing the test results should duplicate as close as possible those conditions specified in designing the system. Flows not indicated by design conditions should be shut down and isolated. Ambient conditions prevailing during test should be as close to design as possible. Controllable steam conditions should be adjusted to levels as close as possible to design values.

Testing of a noncontinuously monitored unit will require more preparatory activities (sensor troubleshooting, measurement of water legs, etc.) than one which is continuously monitored. However, both modes of monitoring will require cycle isolation for accuracy and repeatability. For turbine cycle testing, an unaccounted for water balance loss of less than 0.5 percent of maximum turbine throttle flow is required to provide acceptable test accuracy. Computation of the loss is determined by measurement of water storages in the condenser hotwell, deaerator, boiler drum, and other points within the

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cycle. Extraneous flows leaving the cycle (such as pump gland sealing flows) should be measured using a timepiece and applicable container (55-gallon drum, etc.).

Isolation checklists should be developed for use during testing when required. The list of valves should be compiled from piping diagrams and detailed piping drawings by a knowledgeable plant person, and should be revised to incorporate existing piping changes. Valves located in piping, whose leakage can result in measurement error, should be given priority over those of lesser impact. Turbine cycle examples include feedwater heater alternate drains to the condenser, boiler drains to sumps, bypass valves, etc., as given in para. 6.2.1.2. The identified valves should be physically located and tagged prior to testing.

Manual, motor operated, and pneumatically operated valves should be checked for closed position and leakage downstream by either a pyrometer, hand-feel, or tell-tale valves. Safety valves are checked by observing the vent outlets. Steam traps and float valves should be checked with a stethoscope to assess normal cyclical operation. An ultrasonic valve detector should be used to determine the leakage intensity of a hot valve. During the isolation walkdown, flange or packing leakage should be noted on any unit valve, not just the valves listed on the isolation checklists.

Caution should be exercised when isolating a system or subsystem so as not to endanger the safety of the unit. For example, isolated turbine cycle systems presenting a high risk of unit trip and which should be monitored closely include turbine water induction, boiler feed pump (BFP), minimum flow, auxiliary BFP, high pressure steam to BFP turbines, feedwater heater alternate drains, and high pressure supply gland sealing steam. The unit should never be put at risk and subsequent to each testing activity all "isolated" valves should be returned to their normal operating mode.

Once test conditions have been set, a stabilization period is required to assure consistency and validity of test results. Defined as that time period prior to the test commencing with the final setting of test conditions and extending until steady state operation is attained, this phase of testing is a matter of judgement. All available data (control room test data acquisition, etc.) should be consulted to provide a sound basis in determining when the system(s) or component equipment is properly stabilized.

6.2.3 Test Data Collection. After all parties to the test are satisfied that the stabilization criteria has been met, the collection of test data is ready to begin. A trial test run should be performed to exercise and modify procedures and instrumentation as required. This trial run should be conducted at the highest load condition achievable, particularly for tests with multiple load points. All conditions, procedures, and instrumentation should be in force as if an actual test run was being conducted.

The number of tests to be conducted depends primarily on the scope and objective of the test effort. A sufficient number of tests should be conducted to fully ensure that the test data represents the true condition of the tested equipment, system, etc., and that repeatable data exists. Data collection sampling frequency and acquisition time intervals should be established and agreed upon by all parties to the test and should adhere as closely as possible to the data sufficiency requirements described in para. 6.4.1. Every effort should be made to run the tests under specified design conditions, or as close to specified operating conditions as possible in order to avoid the application of corrections to test results, or to minimize the magnitude of the corrections.

A stability criteria of less than ±0.5 percent in the coefficient of variance for each averaged test value is recommended. If during a test run, stability criteria are exceeded or other test data values vary significantly, the data collection for that run should stop. Data collection should be resumed only when the fluctuation has stabilized, and all previous data for that run should be disregarded. If the fluctuation does not stabilize within a reasonable time frame, the test run should be discontinued and rescheduled for a later date.

The permissible rapid fluctuation from the average test value, where rapid fluctuation is defined as one whose frequency is twice the frequency of the readings, is dependent on the type of test being conducted. Typical values recommended for turbine cycle testing are ±4°F for superheat and reheat steam temperatures (27°F–50°F superheat), ±0.25 percent for generator output, etc., as contained in Table 3.1 in ASME PTC 6. Each user should establish the permissible fluctuation necessary to ensure accurate results.

6.2.4 Scope of Testing. The scope of testing necessary to adequately identify areas of performance impairment should be decided by the user based on the quality, quantity, and cost of information desired. The scope should be limited only to the extent of resources available, with the real-

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ization that greater confidence can be placed in information gathered from the most accurate and repeatable data.

6.3 TRENDING

As an aid in identifying degraded areas of performance, trending of pertinent contributors should be developed. Unit indices to be trended should include heat rate, generation, capacity factor, efficiency factor, etc. Cycle and/or component contributors to be trended can be categorized according to whether they are collected on a continuous real-time basis or determined periodically. This list of parameters to be trended is to be formulated by the user from the measured and calculated parameters listed herein.

These parameters are normally maintained as close to optimum (design) as possible. The impact of operating with some or all of these parameters deviating from their optimum condition is discussed in detail in Section 9, Performance Optimization.

Prior to trending, all indices/contributors obtained at a condition different from a predetermined reference condition must first be normalized to that standard reference. Examples include the following:

Index/Contributor	Normalized Condition
Heat Rate	Group 1/2 Turbine Cycle Corrections
Generation	Group 1/2 Turbine Cycle Corrections
First Stage Pressure	Design Throttle Pressure
Throttle Steam Flow	Design Inlet Steam Conditions
Turbine Extraction Pressures	Design Inlet Steam/Hot Reheat Steam Conditions
Air Heater Exit Gas Temperature	Design Air Heater Leakage and Inlet Air Temperature

Although contributors trended on a periodic basis are feasibly easier to normalize, it is highly recommended that continuously trended parameters include software algorithms necessary to perform the required normalizations. Otherwise, erroneous engineering/operator actions to correct the apparent deficiency may result.

Parameters (indices/contributors) would be trended against time and against dependent parameters when functions thereof. It is most important that when trending a parameter against time, the condition existing at the time when the parameter was obtained be consistent with the condition that existed at the previous data collection time interval. The following table illustrates the consistency needed for several different contributors:

Parameter	Consistent Data Collection Condition
Heat Rate	Generator Output
HP Turbine Efficiency	Governor Valve Position
IP Turbine Efficiency	Generator Output
First Stage Pressure	Governor Valve Position
Turbine Extraction Pressures	Governor Valve Position
Air Heater Effectiveness	X-Ratio
MSR TTD	Governor Valve Position

When contributors cannot be obtained at conditions consistent with historical data, then the collected data must be converted to the same basis (condition) as the historical data. This will then result in a valid trend comparison between contributors at the different time intervals. A good example of this is the collection of generator output for a nuclear turbine-generator. Given two different time periods of data collection, a degradation in MSR TTD will result in a change of the governor valve position (more throttling) to maintain the same 100 percent *MWt* heat input. To adequately compare the generator output for both time periods, some method of correcting for the different valve position should first be performed.

One example of trending is the plotting of HP and IP turbine parameters against time for enthalpy-drop efficiency testing. HP and IP turbine efficiency, first stage, hot reheat, and LP crossover pressures, and reheater pressure drop — when trended against time (i.e., months) — not only aid in the diagnosis of turbine degradation but are also very useful for detecting errors in test data. ASME PTC 6S Report-1988 is an excellent source of trending information for monitoring turbine degradation.

Another example of trending is the plotting of unit indices such as load and heat rate. Due to seasonal fluctuations in heat rate and generator output, an accumulative year-to-date weighted average of these indices is usually more meaningful to the user than the monthly instantaneous value.

It is recommended that the trend plot for each index/contributor include the "design" or "target" value. Ideally the unit will be operating at the optimum condition when each contributor is at its design value. Displaying of the design value will also enable the performance engineer to identify those contributors with the largest impact (deviation from design) on performance.

6.4 DATA VALIDATION AND SUFFICIENCY

Data validation is a required step in performance monitoring to assure that results are consistent with expectations based on performance characteristics

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of the equipment being monitored. Performance engineers need to first know how the equipment they are monitoring operates, the interrelationships between a piece of equipment and the cycle, and how dependent variable performance varies as a function of independent variables. Once this is known, a performance engineer can then ascertain whether the data being collected is reasonably correct and ready for further detailed evaluation based on some fundamental relationships.

Data validation not only includes the more traditional techniques such as statistical analysis of data, but also covers and emphasizes those special techniques which test the conformance of data and results with the physical laws of the process such as thermodynamics, fluid dynamics, and conservation of energy.

Only after data and results have passed the above tests to the satisfaction of the performance engineer should an attempt be made to reach conclusions regarding the performance of a system or the internal condition of its components.

Data validation techniques are given in subsequent sections of this document for various equipment.

Many of the techniques noted are documented in the Performance Test Codes such as PTC 4.1 and PTC 6, and the I&A Supplements such as PTC 19.22. Where fully explained in existing codes, the techniques will not be repeated in this document except to the extent necessary to assure that the user places the technique in the proper context relative to the overall validation task and to establish a single reference source.

A performance monitoring program also requires the engineer to collect statistically sufficient data to properly evaluate performance and take corrective action. Several influences on performance, such as ambient temperature effects on circulating water temperature, and load changes must be accounted for in analyzing performance data. Guidance in assessing data sufficiency may be found in ASME PTC 19.22, Digital Systems Techniques.

6.5 TURBINE

6.5.1 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
(*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*) Hydrogen pressure
(*r*) Power factor

6.5.2 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

6.5.3 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
(*2*) LP feedwater heater alternate drains
(*3*) Deaerator auxiliary overflow to condenser

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(*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*) Main steam dump valves (nuclear)
(*q*) Heater drain tank alternate drains to condenser (nuclear)
(*r*) Reheater drain tank alternate drains to condenser (nuclear)
(*s*) Steam generator pump recirculation drains to condenser (nuclear)
(*t*) MSR steam scavenging vent chamber drains to condenser (nuclear)

6.5.4 Turbine Testing

6.5.4.1 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 and PTC 6.1 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 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 PTC 6.1 test method, employing the use of an inspection port final feedwater nozzle as its primary flow, 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.

6.5.4.2 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.

6.5.4.3 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 percent of full load throttle steam flow for an acceptance test and 0.5 percent for a routine heat rate test. It is recommended that the 0.5 percent criterion be applied for performance monitoring purposes.

For nuclear-steam cycles, the same procedure

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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 its effect on unit performance.

6.5.5 Turbine Cycle Test Data Validation

6.5.5.1 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 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 is expected, such as a difference between enthalpy at a turbine extraction and heater inlet, should be suspect.

6.5.5.2 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.

6.5.5.3 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 they introduce 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 and correction process.

6.5.5.4 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 (IP dummy) flow on combined HP/IP turbine elements.

6.5.6 Turbine Cycle Data Normalization. Group 1 corrections to heat rate and generation apply to variables which primarily affect the regenerative feedwater cycle, and are discussed in detail in PTC 6. A set of equations for using Group 1 correction curves is listed in PTC 6.1. 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.

Turbine stage and shell pressures require correction for departure from reference steam conditions. The first-stage pressure on the HP section of a reheat turbine, or the stage or shell pressure for any stage of a non-reheat turbine shall be corrected to reference conditions by the equation:

$$P_c = P_o \times (P_d/P_t)$$

where:

P_c = corrected pressure for plotting, psia
P_o = test stage or shell pressure, psia
P_d = design, or reference, throttle pressure, psia
P_t = test throttle pressure, psia

This equation also applies to an intermediate stage extraction in the HP turbine section.

For stage or shell pressures at or following the inlet to the reheat turbine section and for the exit

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from the last stage of the HP section, the correction equation becomes:

$$P_c = P_o \sqrt{\frac{P_d}{P_t} \times \frac{V_t}{V_d} \times \frac{V_{dr}}{V_{tr}}}$$

where:

V_t = test throttle specific volume, ft^3/lbm

V_d = design or reference throttle specific volume, ft^3/lbm

V_{dr} = specific volume at test temperature and test pressure at reheat value, ft^3/lbm

V_{tr} = specific volume at design reheat temperature and test pressure at reheat value, ft^3/lbm

Deviation from design turbine inlet steam conditions will affect the efficiency of the HP turbine section. Although this correction is insignificant for normal variations at full throttle pressure, it can have a significant effect under extreme operating conditions such as variable pressure operation and throttle steam temperature drop.

Unless the tested throttle steam conditions are exactly as specified by the design turbine cycle, correction must be made to the throttle steam flow at a constant control valve opening. This is the flow that would exist with design steam conditions at the tested control valve opening. The corrected flow is obtained by multiplying the tested flow by the square root of the products of design throttle pressure divided by test throttle pressure and test throttle specific volume divided by design throttle specific volume.

$$M_c = M_t \sqrt{\frac{P_d}{P_t} \times \frac{V_t}{V_d}}$$

where:

M_c = corrected throttle steam flow, lbm/hr

M_t = test throttle steam flow, lbm/hr

Test generator output is corrected for off-design power factor and hydrogen pressure and purity. Normalization of the resultant generator output to a standard reference condition is then accomplished utilizing the Group 1 and Group 2 turbine cycle corrections.

6.5.7 Turbine Cycle Performance Calculations

6.5.7.1 Turbine Cycle Heat Rate (Test and Corrected). 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.

6.5.7.2 HP Turbine Efficiency. 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.

6.5.7.3 IP Turbine Efficiency. 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 higher than expected due to excessive N2 (IP dummy) packing gland leakage. Mixing of the cooler gland leakage with the hotter reheat bowl steam flow results in a cooler IP exhaust condition, and therefore a 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 by EPRI and General Electric. (Ref. [4], Section 8)

6.5.7.4 LP Turbine Efficiency. 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 very 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.

6.5.7.5 Turbine Shaft Leakoff Flows. Turbine shaft gland leakoff flows are measured and/or calculated. If measured, high pressure gland leakoffs

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(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.

6.5.8 Turbine Cycle Effect On Unit Performance

6.5.8.1 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 percent change in heat rate and +0.25 percent change in generation for a +1 percent change in HP turbine efficiency at full load.

6.5.8.2 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 percent change in heat rate and +0.16 percent change in generation for a +1 percent change in IP turbine efficiency at full load.

6.5.8.3 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 percent change in heat rate and +0.50 percent change in generation for a +1 percent change in LP efficiency.

6.5.8.4 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. 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 (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 non-combined 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.

6.5.8.5 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 are found in the turbine vendor's thermal kit.

6.5.8.6 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

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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.

6.5.8.7 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. Please refer to the cycle thermal kit for quantifying the effect on performance.

6.5.8.8 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 thermal kit as 10 percent. However, most fossil units operate below this level, depending on the reheater design. Modifications to reheater surface area will change the percent 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 this percent drop by changing the flow through the reheater at constant steam conditions.

6.5.8.9 Auxiliary Turbine Steam Flow. A change in auxiliary turbine steam flow of 1 percent of main unit throttle flow has a corresponding 0.5 percent to 0.75 percent 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. Figures A3 and A4 in PTC 6.1-1984 should be consulted when determining this effect on performance.

6.5.8.10 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 0.02 percent effect on heat rate for a spray flow of 1 percent of throttle flow. The effect on generation is somewhat more at 0.07 percent. If the desuperheater flow originates downstream of the highest pressure feedwater heater (final feedwater), there is no effect on performance (heat rate or generation).

6.5.8.11 Reheater Desuperheating Water Flow. Reheater desuperheating water flow is used to control reheater outlet steam temperature. Originating from the boiler feed pump, the effect on performance of a spray flow of 1 percent of throttle flow is a 0.6 percent to 0.7 percent change in heat rate and generation. 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.

6.5.8.12 Cycle Isolation. The following performance effects of cycle isolation are intended to aid the performance engineer in his 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.

6.5.8.13 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 percent and a generator output decrease of 0.5 percent is realized for a 50 percent diversion of normal drain flow to the condenser. This same performance effect also holds true for the second highest pressure heater diverting 50 percent of its normal drain flow to the condenser. Diversion of 5 percent of these same drain flows results in a performance penalty of 0.1 percent in heat rate and generator output for the same type unit.

6.5.8.14 HP Steam to Auxiliary Turbines. On most steam driven auxiliary turbines such as BFP,

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an auxiliary source of high pressure steam (usually from main steam) is supplied for start-up purposes and low load operation. 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 the 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 percent intentional use of main steam flow or equivalent leakage will result in a heat rate increase of +0.3 percent with a corresponding decrease in generator output of 0.3 percent.

6.5.8.15 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 therefore causing a decrease in power output.

Typical performance effects are a 0.2 percent change in heat rate and power output for a 5 percent extraction steam leakage of the second highest pressure feedwater heater, and a 0.3 percent change in heat rate and power output for a 5 percent 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 percent (5 percent extraction leakage) with insignificant performance effects for the remaining low pressure feedwater heaters.

6.5.8.16 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 usually 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 percent increase in heat rate for removal of the highest pressure feedwater heater to a +0.4 percent effect for removal of the lowest pressure heater. In a 5-heater reheat cycle arrangement, these effects increase to +1.3 percent (highest pressure heater) and +0.7 percent (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. The applicable turbine vendor instruction book should be consulted before proceeding with this mode of operation.

6.5.8.17 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. 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 two hours), 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 percent loss of water out of a typical fossil turbine cycle will result in a +0.2 percent change in heat rate and a −0.2 percent 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 contained in Fig. A7 of PTC 6.1.

6.6 STEAM GENERATOR EQUIPMENT

6.6.1 Measured Performance Parameters

6.6.1.1 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)

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(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 desuperheater spray flows, temperatures, and pressures

(d) SH and RH tube metal temperatures

(e) Boiler exit gas temperature and flue gas analysis

(f) Drum pressure

(g) Drum level

(h) Boiler circulation pumps amps, differential pressure

(i) Unburned carbon in ash

(j) Fuel analysis (ultimate and proximate)

(k) Furnace gas pressures and differential pressures

(l) Blowdown flow

(m) Sootblower operations log

6.6.1.2 Pulverizers and Burner Equipment

(a) Number and location of mills in service

(b) Mill outlet temperatures

(c) Mill motor amps

(d) Tempering air flow

(e) Burner register settings

(f) Fuel nozzle tilt position

(g) Number and location of gas and oil burners in service

(h) Auxiliary fuel flow

(i) Atomizing steam flow

(j) Windbox/furnace differential pressure

(k) Windbox air damper positions

(l) Overfire air damper positions and nozzle tilt

(m) Mill fineness

6.6.1.3 Air Heaters

(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

(f) Primary and secondary air outlet flows

(g) Air and gas pressure differentials

6.6.1.4 Fans. FD/PA/ID/GR Fan: Inlet and outlet pressures, temperatures, flows, damper positions, motor amps, ambient air temperature, barometric pressure, humidity.

6.6.2 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, cleanliness, X-ratio, etc.)

(f) Air heater exit gas temperature corrected to the zero leakage condition

(g) SH and RH spray flows

(h) Boiler surface cleanliness

6.6.3 Steam Generator Equipment Performance Degradation

6.6.3.1 Pulverizers and Burner Equipment. Combustible loss-on-ignition (LOI) data from fly ash samples should be collected, and causes of excess carbon carryover identified. Air flows at mill inlet 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, 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.

6.6.3.2 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.

6.6.3.3 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. Minimization of air heater leakage should be implemented through seal repair. Tubular and heat pipe air heaters should also be checked for leaks due to corrosion, erosion, or mechanical damage.

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

6.6.4 Steam Generator Equipment Testing

6.6.4.1 Pulverizer Testing. Pulverizers are se-

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lected to provide a required grinding capacity, based upon a set of design criteria which 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 and the rank of the coal. It is also a function of the type of the 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 6-1 shows the relationship 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.

Clean air curves are a plot of mill differential versus stationary pitot tube differential with no coal flow. It represents the pressure drop through the mill due to its internal components. As such, it gives an indication of wear by comparing clean air curves over time. The number of hours of operation of a pulverizer, 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 which 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 minutes from mills which 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 new International Organization for Standardization (ISO) method. Figure 6-2 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.

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 is available with the PTC 4.2 method. Additional sampling points can be an advantage when there is roping of the coal in the pipe.

With collection complete, the pulverized coal samples from each mill are 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 individual screens plot as a straight line on a Rosin and Rammler Probability Chart, Fig. 6-3. If the percentages do not fall on a straight line, either improper sampling has occurred or this may indicate mechanical problems within the mill. Investigation should continue until the results fall on a straight line.

6.6.4.2 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 4.1 should be consulted for the proper procedures and calculations for the input-output and heat loss boiler

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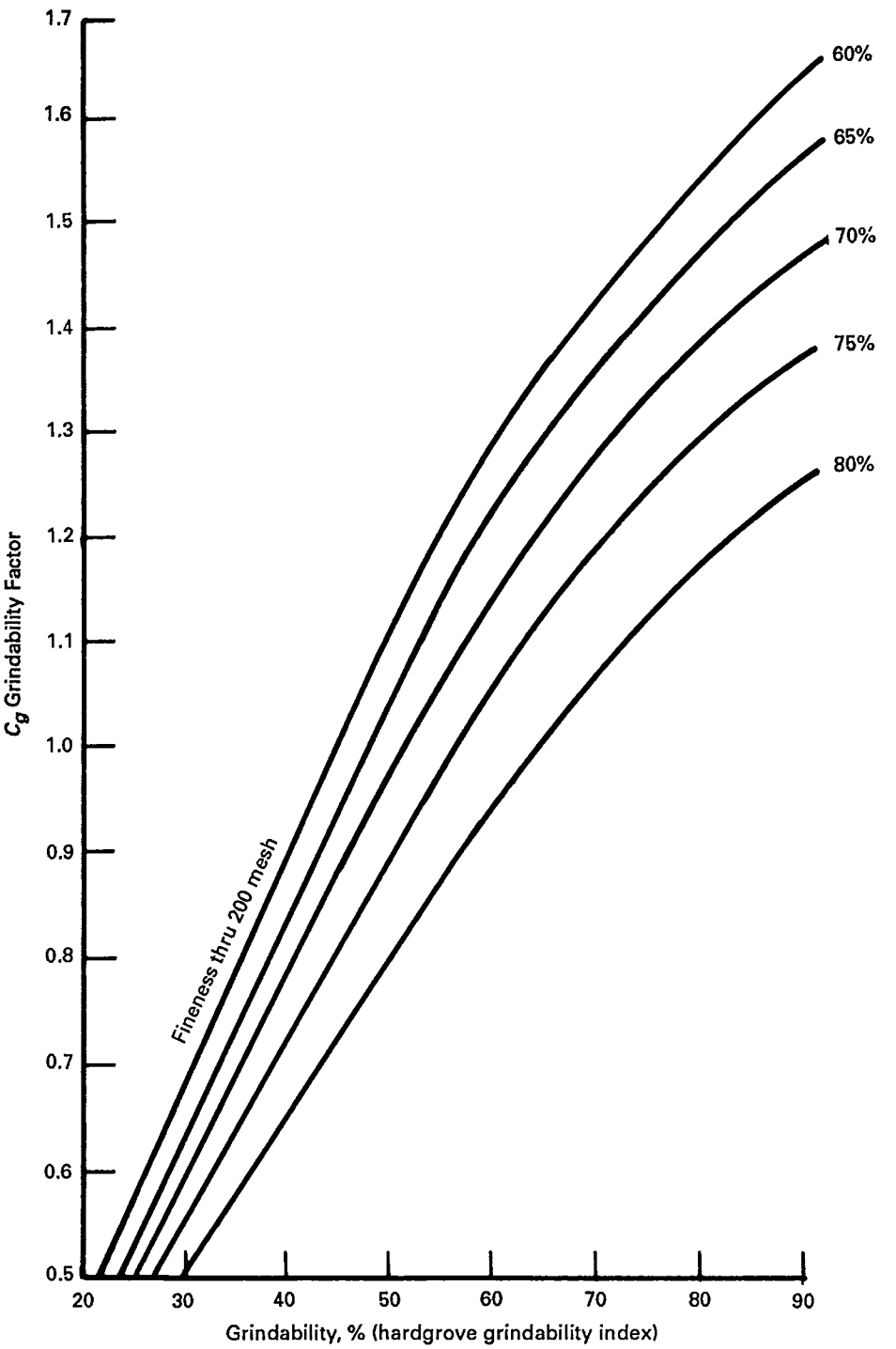

FIG. 6-1 PULVERIZER CAPACITY CURVE
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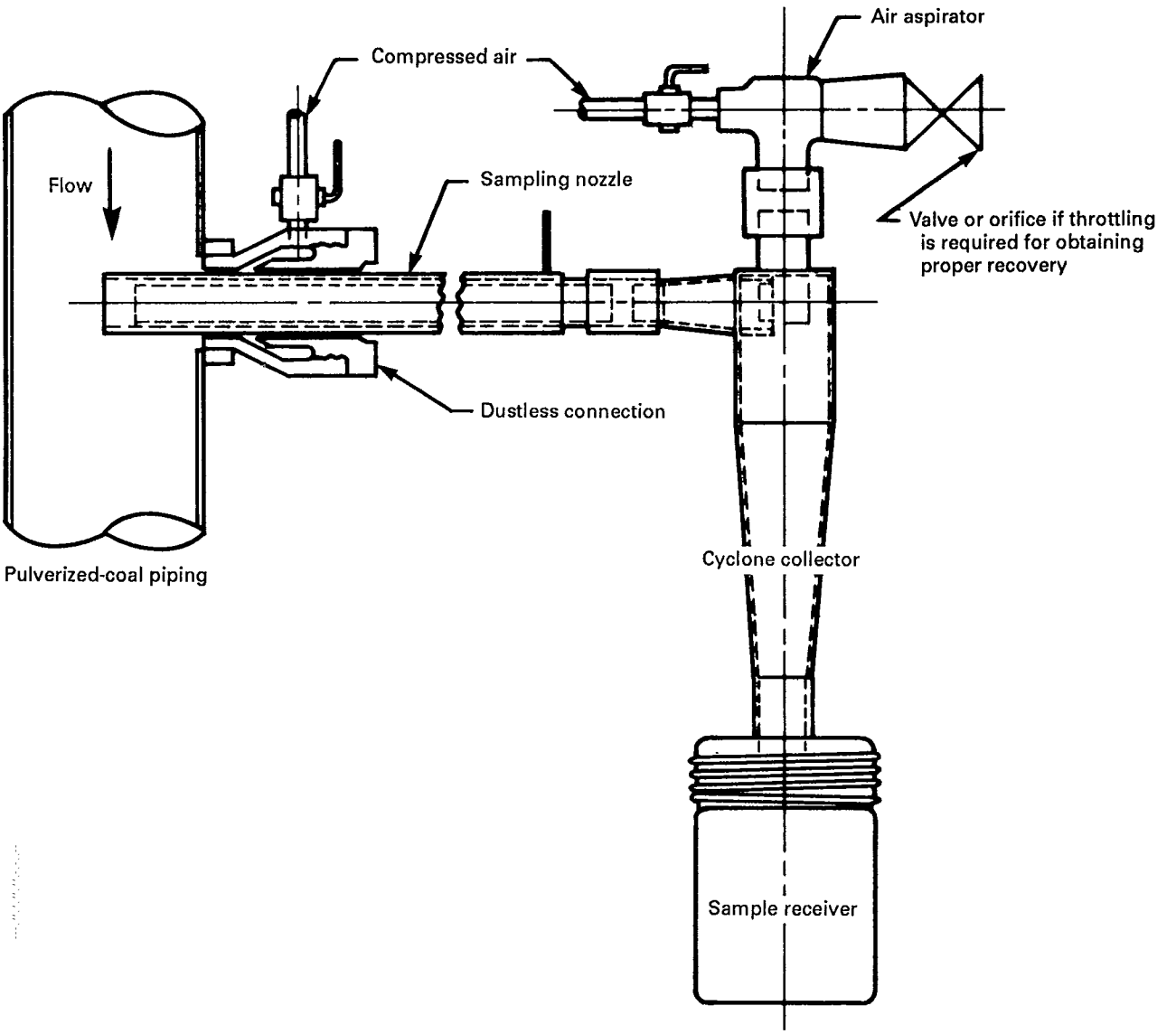

FIG. 6-2 ARRANGEMENT FOR SAMPLING PULVERIZED COAL
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efficiency tests. The heat loss method is preferred for performance monitoring.

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 there is a sufficient number of them 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 (multi-point 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 to 60 minutes prior to taking data. The data necessary to evaluate performance of the boiler and heater can then be taken during the next 30 minutes, at 5 minute 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.

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 4.1 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 4.1 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 4.1, the list of measurement and calculation parameters can be reduced to include only the major losses. Referring to PTC 4.1 Figs. 1 and 2, a review of steam generator boundaries, heat inputs, and heat outputs can be made to establish specific monitoring goals.

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 4.1 for the effect of measurement errors on efficiency. In addition, the losses are quantified under loss categories which will aid the operations and performance people to locate and reduce 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

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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 4.1 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 which 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 multi-point 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.

Losses to accurately determine on a regular basis:

(*a*) dry gas loss

(*b*) water from combustion of H_2 in fuel loss

(*c*) water from H_2O in fuel loss

(*d*) unburned carbon in ash

Other losses to calculate or estimate are:

(*a*) moisture in air loss — use typical percent moisture in air or actual values if available.

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

(*c*) radiation loss from hot-side precipitators — estimate from historical data.

(*d*) radiation loss — estimate using ABMA standard radiation loss chart as shown in PTC 4.1.

6.6.4.3 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. All heat transfer surfaces should be washed down to eliminate pluggage. Care should be taken to get a thorough washdown so 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 will offset air heater performance. Changes in entering air temperature will result in a change in temperature head, which directly affects outlet gas temperature. For example, a 10°F increase in entering air temperature will increase the exit gas temperature by 10°F times the gas side efficiency (decimal) of the air heater. The same is true for changes in entering gas temperature. A 10°F increase in 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 performance is poor 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. 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 percent and 8 percent on pulverized coal fired units, and 10

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percent to 12 percent 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.

6.6.4.4 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, simplifying assumptions and test methods may be employed.

For the measurement of fan flow, a grid of probes designed to measure the fluid total pressure is 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 square feet, but for the purposes of performance monitoring, this requirement may be relaxed to every 3 to 4 square feet. When measuring the flow of particulate laden gas, hole pluggage is a concern and periodic purging of the probes may be required.

Static pressure, 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 203].

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 percent of all velocity pressure readings are greater than one-tenth the maximum reading of velocity pressures. Multiplying the root mean square average of the readings by the area of the traverse plane will determine the volumetric flow rate. Fan static pressure rise across the fan and the inlet velocity pressure should be calculated 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.

PTC 11 should be consulted for recommended procedures to be used in conducting rigorous fan tests. A rough approximation of fan performance can be made by simply measuring fan power static pressure rise across the fan with inlet vanes wide open. A combustion calculation using fuel flow rate, typical ultimate analysis of fuel, unit heat rate, and generation will yield flow rate which can be plotted on a manufacturer's curve to determine if there is a problem which needs to be investigated further through more accurate testing.

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6.6.5 Steam Generator Equipment Data Validation

6.6.5.1 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.

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. 6-3, as detailed in para. 6.6.4.1.

6.6.5.2 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, econimizer inlet versus feedwater leaving last heater.

A heat balance around attemperators using temperature data can be used to validate spray flow measurement; however, the performance engineer should be familiar with limitations of the heat balance method based on other considerations as described in para. 5.3.7.

O_2 and CO_2 should, when plotted, fall on the combustion line. This correlation can only be checked if CO_2 is measured. Appendix A describes the method for drawing the radial line on the combustion chart.

When Orsat analysis is used to determine flue gas composition, a small error in CO_2 determination will result in an error in the calculated leakage. This is especially true if the CO_2 value is low. For example, a difference of 0.1 in reading CO_2 percentage can produce a 1 percent difference in calculated air heater leakage.

The reliability of CO_2 analysis can be established if, in addition to CO_2, each gas sample is also analyzed for O_2 content. The measured value of CO_2 and O_2 when plotted on a volumetric combustion chart will fall on a predetermined radial line. The radial line can be drawn based on the ultimate analysis of the fuel (see Appendix A for details).

6.6.5.3 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.

6.6.6 Steam Generator Equipment Performance Calculations

6.6.6.1 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

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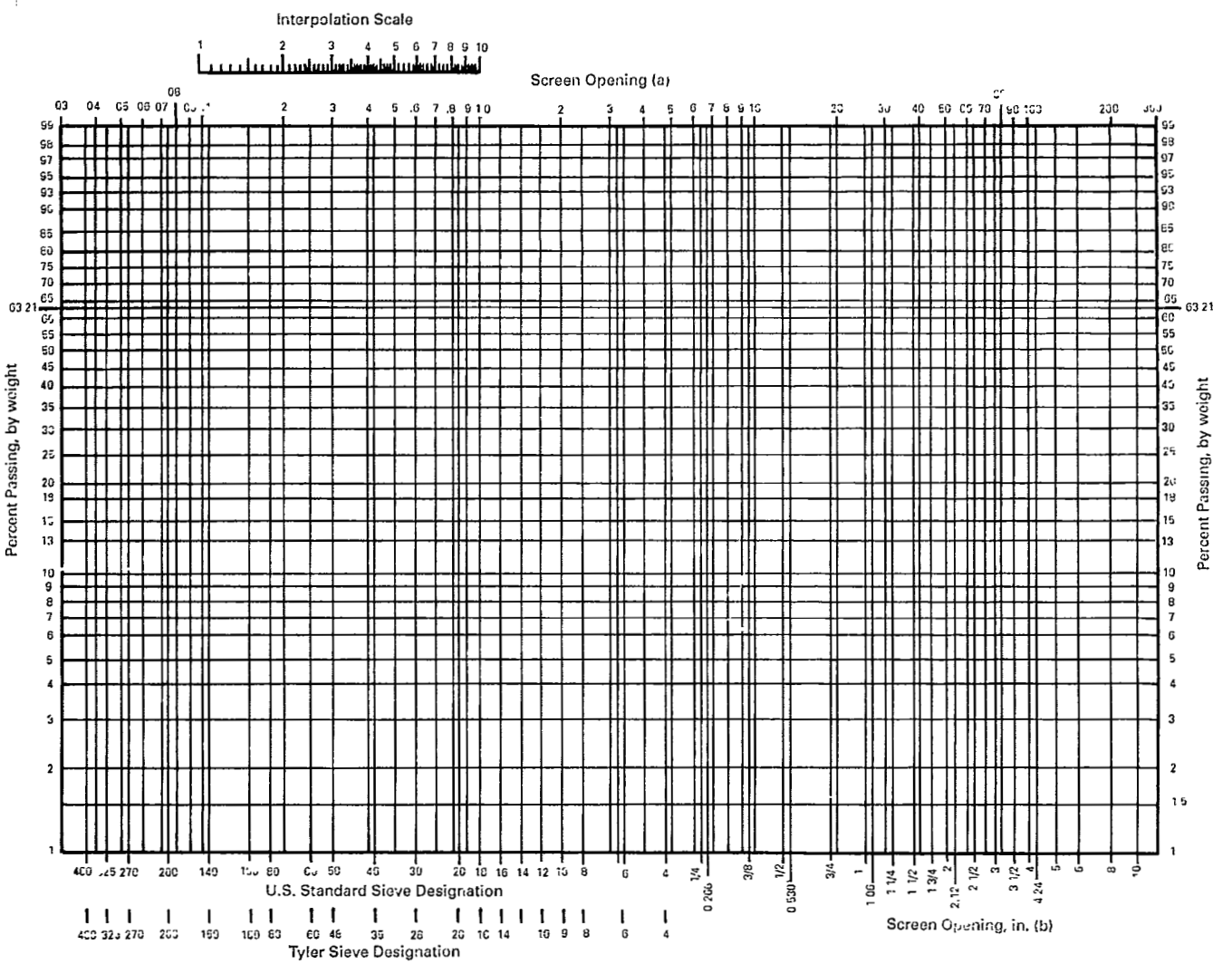

NOTES:
(a) Any scale. If in millimeters, coincides with lower scale.
(b) Square hole if used as continuation of fine series.

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

(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.)

FIG. 6-3 GRAPHICAL FORM FOR REPRESENTING DISTRIBUTION OF SIZES OF BROKEN COAL

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between 90%–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. 6-4. The sampling time at each point should be approximately 5 seconds, and the sampling location should be in a vertical pipe 7 to 10 diameters from preceding bends, changes in cross section, or valves.

6.6.6.2 Steam Generator Efficiency (Heat Loss Method). Calculation of boiler efficiency by the heat loss method is determined using the calculation forms in PTC 4.1. 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:

(*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

Appendix B is an example of boiler efficiency calculation using nomographs and graphs to determine the breakdown of losses. Refer to PTC 4.1 for the detailed equations to calculate boiler efficiency.

6.6.6.3 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:

$$\text{AH gas side efficiency} = \frac{\text{Gas Drop}}{\text{Temperature Head}} \times 100 = \frac{T_{g1} - T_{g2}}{T_{g1} - T_{a1}} \times 100$$

where:
T_{g2} = gas temperature entering air heater
T_{g2} = gas temperature leaving the air heater corrected to the no leakage condition
T_{a1} = air temperature entering the air heater

The gas side efficiency by itself is not very useful as a diagnostic tool. 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. 6.6.4.3. A change in the 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:

$$\text{X-ratio} = \frac{T_{g1} - T_{g2}}{T_{a2} - T_{a1}} = \frac{W_a \text{Cp}_a}{W_g \text{Cp}_g}$$

where:
T_{g1}, T_{g2}, and T_{a1} are as defined above
T_{a2} = air temperature leaving air heater
Cp = 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, leakage from the setting, 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.

Air and gas side pressure drop should be monitored and trended to idenfity degradation.

The tested air heater gas side efficiency should be compared with its expected efficiency at the tested X-ratio, 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 again on the level of accuracy desired and the specific data collected.

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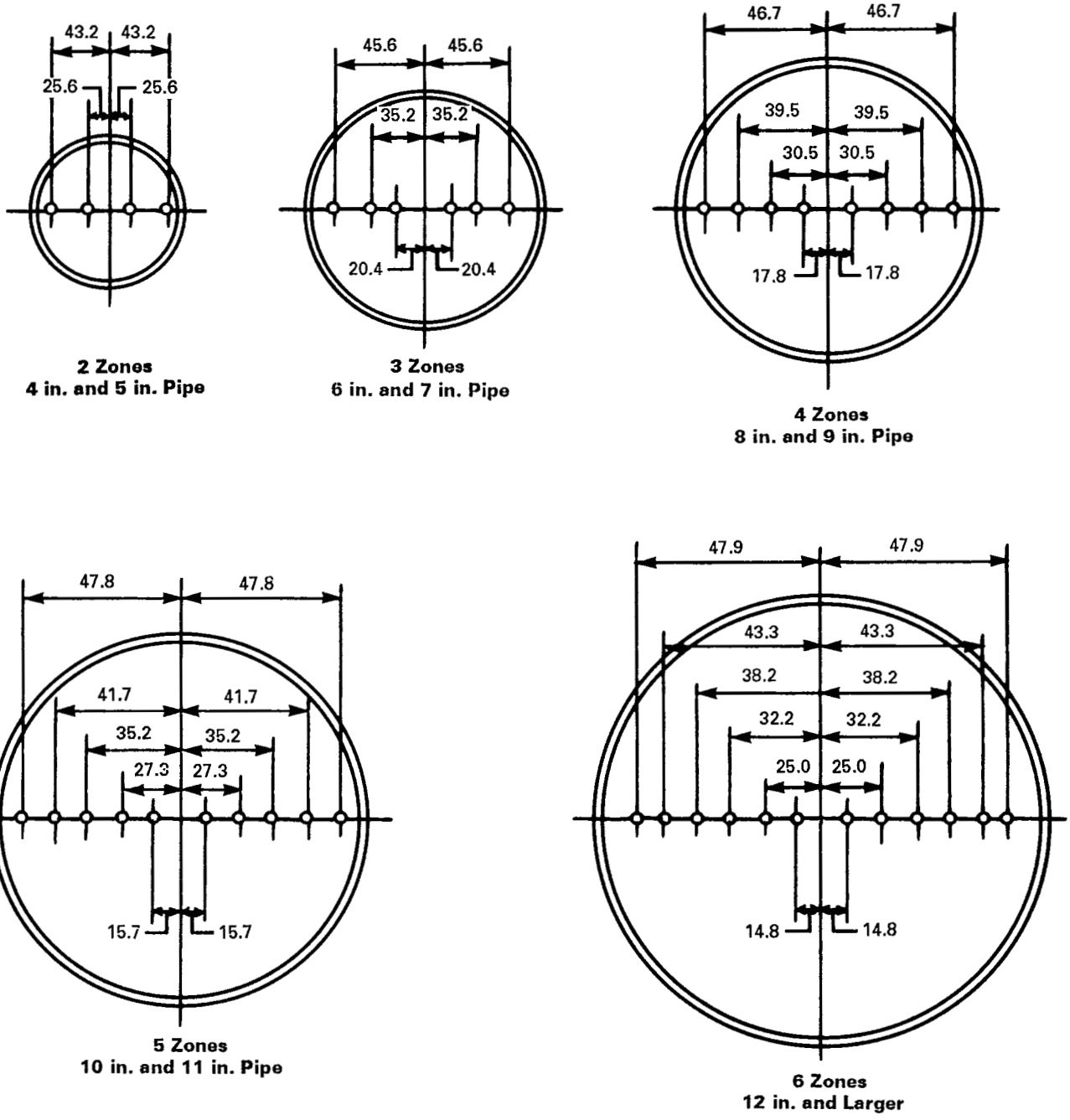

FIG. 6-4 SAMPLING DIRECT-FIRED PULVERIZED COAL-SAMPLING STATIONS (DIMENSIONS ARE "PERCENT OF PIPE DIAMETER")
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If O_2 analyzers are used, air heater leakage is approximated by the following equation:

$$\% \text{ Leakage} = \frac{(\%O_2 \text{ out} - \% \ O_2 \text{ in})0.9*100}{(21 - \% \ O_2 \text{ out})}$$

It is imperative that the O_2 basis is the same at the air heater inlet and outlet, wet or dry. 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 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 them 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 (plus or minus one percentage point) to leakage determined on a weight basis.

6.6.6.4 Fan Calculations. The efficiency of a fan is used as a measure of fan performance. Fan efficiency is calculated using the equation:

$$\text{Eff} = \frac{\text{cfm} \times \text{Static Pressure Rise (in. } H_2O)}{6356 \times \text{Fan Brake Horsepower}}$$

The above equation shows that fan brake horsepower is directly proportional to flow rate. Fan brake horsepower is difficult to measure. Fan motor input power can be used, but the calculation of fan efficiency will include fan motor efficiency, normally in the 90 to 95 percent range depending on motor size.

6.6.7 Steam Generator Effects on Performance. In analyzing the heat losses mentioned previously, the losses due to unburned combustible material and flue gases are the controllable ones.

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 percent or less. Carbon may appear in either the fly ash or bottom ash, but more frequently in the fly 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.

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 will result in an increase of unburned carbon carrying over to the boiler exit. Coarse coal may also cause furnace slagging. If slagging conditions occur while burning a coal which 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:

$$L_G = (W_G)(Cp_g)(T_e - T_r)$$

where:

$W_G = \dfrac{\text{lb of dry gas}}{\text{lb of fuel (as fired)}}$

Cp_g = mean specific heat of dry gas

T_e = flue gas exit temperature, °F

T_r = inlet air temperature, °F

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. The operator must, however, be informed on just how low he can carry excess air. 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,

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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 which is arbitrarily applied to account for small losses not elsewhere accounted for, such as sensible heat in ash discharged from the unit. The unaccounted loss is taken as approximately 1.5 percent.

6.7 BALANCE OF PLANT (CONDENSERS, COOLING TOWERS, HEATERS, AND PUMPS)

6.7.1 Balance of Plant Measured Parameters

6.7.1.1 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

6.7.1.2 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

6.7.1.3 Feedwater Heaters

(*a*) Heater inlet/outlet feedwater temperatures
(*b*) Extraction inlet temperature and pressure
(*c*) Cascade drain temperature
(*d*) Drain cooler exit temperature
(*e*) Tubeside feedwater/condensate flows
(*f*) Feedwater heater level
(*g*) Tubeside pressure drop

6.7.1.4 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)

6.7.1.5 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

6.7.2 Balance of Plant Calculated Parameters

6.7.2.1 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

6.7.2.2 Cooling Tower

(*a*) Deviation from design cold basin temperature

6.7.3 Balance of Plant Degradation Identification

6.7.3.1 Condenser. The condenser is the heat sink for the power cycle. 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 non-condensible gases which collect or otherwise 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 three areas of focus: (1) the demand being placed on the condenser in terms of the size of the total heat load; (2) the ability of the CCW system to provide adequate CCW flow rate and inlet temperature; and (3) the resistance to the heat transfer attributable to any and all of the various contributors. Per-

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formance 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 within limits, 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 flow rates can be caused by either partially open valves in the system, closed valves in the system, or condenser waterbox debris.

As a rule-of-thumb, 1–2 scfm air in-leakage for each 100 MW of installed capacity is considered an acceptable level of air in-leakage. Sources 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. Non-condensible flow rate from the condenser should be checked. The flow rate should not exceed 25 percent of the capacity of the air removal equipment, or unit performance can be expected to deteriorate. Drain lines to the condenser are normally designed with a fitting (valve, etc.) within 2 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.

The waterbox air removal system should be checked by examining the water level in the water boxes. 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 couple of degrees of subcooling should typically be found from the calculated saturation temperature to the condensate temperature. Excessive subcooling may indicate such problems as too much circulating water flow, too low inlet temperature, or excessive makeup for a given heat rejection rate. 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.

6.7.3.2 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 and in turn again rejected to the atmosphere in the cooling tower.

Evaluation of the cooling tower may be divided into three areas of focus: (1) the demand being placed on the cooling tower in terms of the size of the total heat load; (2) the ability of the CCW system to provide adequate CCW flow rate; and (3) 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.

6.7.3.3 Feedwater Heaters. In the absence of tube leakage or tube rupture, changes in feedwater heater performance are relatively slow. However, variations in operating conditions causing dirty heat transfer surfaces, leaks from erosion/corrosion of the division plates, and inadequate venting of noncondensable gases 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. Abnormal heater levels and fluctuating drain control valve positions may be caused by increasing heat duty from a flow unbalance or upstream heater bypassed, on a tube leak. 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.

6.7.3.4 Boiler Feed Pump. BFP performance may not have a significant impact on unit heat rate,

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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).

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.

For turbine driven boiler feed pumps, the boiler feed pump determines the amount of power required by the boiler feed pump turbine. Knowing 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.

6.7.3.5 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: (1) the moisture separator, and (2) the reheater sections. The moisture separator section consists of plates, screens, chevrons, or other apparatus which 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 two degrees 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, as the total cycle steam inlet moisture content would be less at the higher HP turbine expansion endpoint.

Other parameters to monitor include the cycle

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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.

6.7.4 Balance of Plant Testing

6.7.4.1 Condensers. Most power plants routinely monitor condenser performance on a weekly or monthly basis. The frequency is dependent on the rate of performance degradation experienced by the condenser. Both thermal and hydraulic performance of the condenser should be evaluated. Conditions that can be identified include:

(*a*) condenser tube fouling
(*b*) waterbox pluggage
(*c*) low waterbox level
(*d*) air in-leakage.

The most widely accepted procedure for conducting a condenser thermal performance test is PTC 12.2. This 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; and

(*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- and labor-intensive.

The parameters that must be measured or calculated, at a minimum, to determine an average thermal transmittance for the condenser are:

(*a*) the total surface area of the condenser tubes
(*b*) the entering and exiting circulating water temperature
(*c*) the condensing steam temperature
(*d*) the heat load on the condenser.

Most condenser performance tests do not utilize extensions of tubes through the waterbox which is a deviation 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.

6.7.4.2 Cooling Tower Performance. Most 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

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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 — less than 10 mph

(*b*) wet bulb temperature — 3°F above to 7°F below design

(*c*) cooling range — 20 percent of design

(*d*) water rate — 10 percent of design

(*e*) heat load — 20 percent 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 one hour after steady state conditions have been established.

6.7.4.3 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 in which tubeside feedwater flow, tubeside temperature rise, heater level, cascade drain flow and state conditions, and shell/drain flows and state conditions are determined. For dual heater arrangements where measurement of individual feedwater flows are not possible, a 50% flow split can be assumed. For a three parallel heater arrangement, a 33% split is assumed. Determination of shell inlet, drain outlet, and cascade drain flows may in some instances require the instrumentation of adjacent heaters for mass and energy balance computations.

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 percent of the rated heater flow. The turbine cycle should be operating at normal conditions with the condition of the turbine 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 contains the instrumentation requirements and procedures to be consulted for conducting feedwater heater tests. Formal acceptance testing of feedwater heaters should be conducted using PTC 12.1.

6.7.4.4 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 seal water injection, balancing drum leak-off, 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

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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 very similar to those used in determining low pressure turbine efficiency. As mentioned in para. 6.3.7.4, 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.

6.7.4.5 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. ["Metering of steam-water two phase flow by sharp edged orifices," by Russell James, Proc. Inst. Mech. Engineers, 1965–1966, Vol. 180, No. 93, Pages 549–566].

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.

6.7.5 Balance of Plant Data Validation and Normalization

6.7.5.1 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.

6.7.5.2 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, which is useful for monitoring cooling tower condition over time.

6.7.5.3 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.

6.7.5.4 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

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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.

6.7.6 Balance of Plant Calculations

6.7.6.1 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

$$U\text{actual} = \frac{Q[ln((T_S - T_{cwi})/T_s - T_{cwo}))]}{A(T_{cwo} - T_{cwi})}$$

Q = heat transferred, Btu/hr

A = heat transfer surface area of the condenser, ft^2

T_S = condenser inlet steam temperature, F

T_{cwi} = circulating water inlet temperature, F

T_{cwo} = circulating water outlet 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:

$Q = W_{cw}Cp(T_{cwo} - T_{cwi})$, Btu/hr

W_{cw} = mass flow rate of the circulating water, lbm/hr

Cp = 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.

6.7.6.2 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.

$$\text{Test Range} = (\text{Hot Water Temp}) - (\text{Cold Water Temp})$$

$$\text{Capability (\%)} = \frac{\text{Adjusted test water rate}}{\text{Predicted test water rate}} \times 100$$

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.

6.7.6.3 Feedwater Heater TTD and DCA. Feedwater heater terminal temperature differences (TTD) and drain cooler approaches (DCA) should be calculated.

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6.7.6.4 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.

Hydraulic HP

$$= \frac{\text{lbm of Fluid Per Min.} \times \text{Total Developed Head (Feet)}}{33{,}000}$$

The equation for pump power input for a motor driven pump (brake horsepower, bhp) is

$$\text{bhp} = \left(\frac{\text{kW}}{0.746}\right) \eta_g$$

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.

6.7.6.5 Moisture Separator Reheater TTD, Shell Pressure Drop, and Moisture Carryover. 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.

6.7.7 Balance of Plant Effects on Performance

6.7.7.1 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 percent 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.

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 percent change in heat rate and −0.2 percent change in load at 100 percent throttle flow for 1 percent 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

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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.

6.7.7.2 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.

6.7.7.3 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 percent increase in heat rate and a 0.4 percent increase in generator output. The change in TTD for the top heater is reflected in the temperature of the final feedwater.

6.7.7.4 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.

6.7.7.5 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 & 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 phenomena 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 percentage 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 percentage point increase in carryover, the increase in heat rate is approximately 25 Btu/kWhr and the reduction in generator output is about 1700 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 percent moisture, 510°F inlet steam conditions at a turbine exhaust pressure of 2.5 in. HgA.

6.8 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 (i.e., unit losses, system losses) in a prioritized manner according to performance impact and/or the cost to correct 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

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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 no less than on 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. Sections 8 and 9 should be consulted for information on these techniques.

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APPENDIX A

The purpose of the dry flue gas volumetric combustion chart, Fig. A-1, is to validate the measurement of CO_2 and O_2 in the combustion flue gases as depicted in para. 6.6.5.2.

The amount of O_2 in the flue gases is significant in defining the status of the combustion process. Its presence always means that more oxygen (excess air) is being introduced than is being used. Assuming complete combustion, low values of O_2 in the flue gases reflect moderate (nearly correct) excess air and reduced heat losses to the stack, while higher values of O_2 mean needlessly higher stack losses. The quantitative determination of total air (total air = 100 + percent excess air) admitted to an actual combustion process requires a complete flue gas analysis of CO_2, O_2, CO, and N_2 (by difference) or the direct measurement of the air supplied by a suitable fluid meter.

A formula that has long been used for approximating the percent excess air from an Orsat analysis is:

$$\% \text{ Excess air} = 100 \times \frac{O_2 - CO/2}{0.264\ N_2 - (O_2 - CO/2)}$$

The results are reasonably close for the flue gases from hydrocarbon fuels of low nitrogen content; the error increases with increase in nitrogen content in the fuel.

Figure A-1 is another helpful chart for rapid determination of excess air when a series of gas analyses are taken. From any Orsat analysis, it indicates the kind of fuel being burned. It also indicates relationships between fuels. The various types of fuel are indicated by the reference letters at the upper right of Fig. A-1; these are identified in Table A-2.

Each Orsat analysis may be plotted as a single point on this chart. For any particular fuel, every valid analysis yields a point which falls on a single straight line drawn from the pivot point to the point on the "zero oxygen" line ($O_2 - CO/2 = 0$), where the CO_2 coordinate equals the maximum CO_2 value (percent by volume) obtainable with the particular fuel.

One good Orsat analysis will define the fuel, i.e., the result is plotted as a point on Fig. A-1, and a straight line through the pivot and this point is extrapolated to the zero oxygen line. The point of intersection is the maximum CO_2 value possible for the particular fuel, which is identified by the reference fuel letter adjacent to the zero oxygen line.

The maximum CO_2 value can be checked from the following formula:

$$\text{Max } CO_2 = \frac{21.0\ CO_2}{21.0 - O_2}$$

where chemical symbols represent volume percentages of dry flue gas. Percentages on the right side of the equation are from the Orsat analysis.

Maximum CO_2 value can also be computed from the formula:

$$\text{Max } CO_2 \left(\begin{array}{l}\% \text{ by volume,} \\ \text{dry flue gas}\end{array}\right) = \frac{31.3C + 11.5S}{1.504C + 3.55H_2 + 0.56S + 0.13N_2 - 0.45O_2}$$

where chemical symbols other than CO_2 represent weight percentages of each material in the ultimate as-fired analysis of the fuel.

Both of the above equations assume complete combustion. For incomplete combustion with CO and CO_2 (assuming no carbon loss—all carbon burned to either CO or CO_2), the maximum CO_2 may be computed from the Orsat analysis as follows, where chemical symbols represent volume percentages of dry flue gas:

$$\text{Max } CO_2 = \frac{100\ (CO_2 + CO)}{100 - 4.78\ O_2 + 189\ CO}$$

Figure A-1 should not be used for fuels with more than 6% CO, nor for fuels which contain high

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TABLE A-2 MAXIMUM CO_2 RANGES FOR COMMON FUELS

Fuel	% CO_2 by Volume	Reference Letter (Fig. A-1)
Coke oven gas (CO 6%)	9.23–10.6	A
Natural gas	11.6–12.7	D
Oil	14.25–16.35	F
Tar and pitch	17.5–18.4	H
Bituminous coal	17.7–19.3	I
Charcoal	18.6	J
Black liquor without saltcake	18.6	J
Lignite, coke	19.2	L
Anthracite coal	19.3–19.85	M
Tan bark	20.1	O
Wood	20.1–20.5	P
Bagasse	20.65	R
Pure carbon	20.9	S
Blast furnace gas	24.6–25.3	T

amounts of CO_2. However, blast furnace gas may be used, but note it requires the different family of percent-total-air curves included on the left side of Fig. A-1.

REFERENCE

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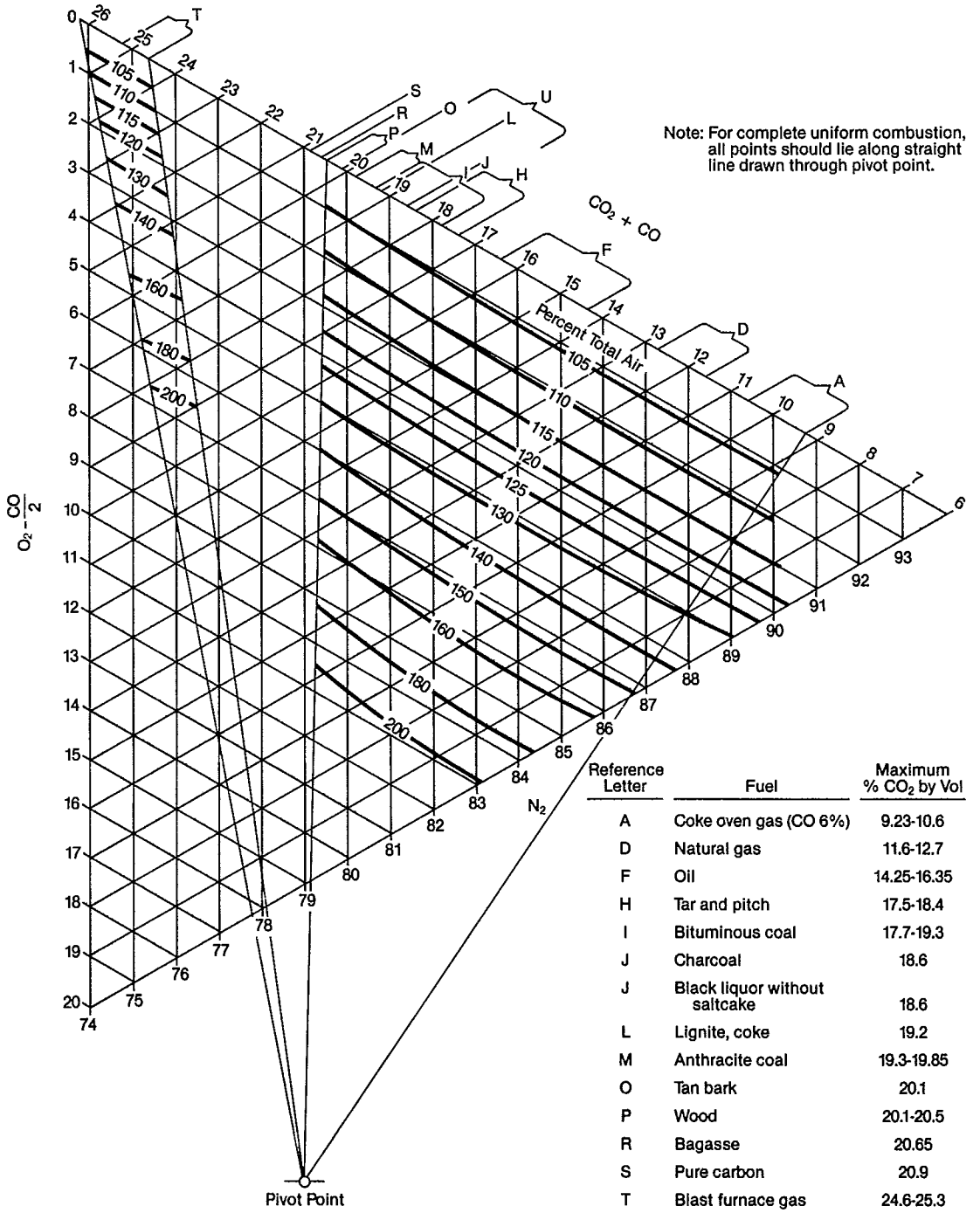

Reference Letter	Fuel	Maximum % CO_2 by Vol
A	Coke oven gas (CO 6%)	9.23-10.6
D	Natural gas	11.6-12.7
F	Oil	14.25-16.35
H	Tar and pitch	17.5-18.4
I	Bituminous coal	17.7-19.3
J	Charcoal	18.6
J	Black liquor without saltcake	18.6
L	Lignite, coke	19.2
M	Anthracite coal	19.3-19.85
O	Tan bark	20.1
P	Wood	20.1-20.5
R	Bagasse	20.65
S	Pure carbon	20.9
T	Blast furnace gas	24.6-25.3
U	MSW/RDF	18.0-20.0

FIG. A-1 DRY FLUE GAS VOLUMETRIC COMBUSTION CHART
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APPENDIX B

The purpose of Appendix B is to provide the fundamental tools necessary for calculating boiler efficiency quickly and easily. The analysis is performed through the use of charts and nomographs and is an acceptable method for approximate performance results. A detailed and rigorous method for determining boiler efficiency may be found in PTC 4.1.

The following method of obtaining the efficiency of a boiler unit is based on the heat loss method. It uses standardized charts, calculations, and nomographs for determining the various heat losses which affect boiler efficiency. Moisture and ash in the fuel as fired, combustible in the fly ash and ash pit refuse, flue gas analysis, average steam (or feedwater) flow, flue gas exit temperature, and forced draft fan inlet air temperature are the only data required.

The following nomenclature will be used throughout Appendix B:

For measured values:

M = total moisture in fuel as fired, %
A = ash in fuel as fired, %
CO_2 = carbon dioxide in flue gas at boiler or air heater outlet, %
O_2 = oxygen in flue gas at boiler or air heater outlet, %
CO = carbon monoxide in flue gas at boiler or air heater outlet, %
T_e = flue gas exit temperature, °F
T_r = inlet air temperature, °F

For chart and calculated values:

H = heating value of fuel as fired, Btu/lb
H_C = heating value of combustible in moisture and ash-free fuel, Btu/lb
VM = volatile matter in fuel as fired, %
FC = fixed carbon in fuel as fired, %
W_m = weight of water from moisture in fuel, lb/MKB (MKB = million Btu)
W_H = weight of water from combustion of hydrogen in fuel, lb/MKB
W_C = theoretical weight of products of combustion, excluding moisture in fuel and air, lb/MKB
W_G = weight of dry flue gas, lb/MKB
W_A = weight of dry air, lb/MKB
L_C = loss due to unburned combustible, %
L_M = loss due to moisture and hydrogen in fuel, %
L_G = loss in dry gases, %
L_A = loss due to moisture in air, %
L_R = loss due to radiation, %
L_U = loss unaccounted for, %

I. L_C, heat loss due to unburned combustible: If the proximate analysis of coal lists only the heating value on a dry, ash-free basis, then calculate H from formula: $H = (1 - A/100 - M/100) * (H_C)$. Obtain percentage refuse distribution to ash pit and fly ash from refuse distribution sheets. Determine L_C, unburned carbon loss, (a) in ash pit, and (b) in fly ash from Fig. B-1, as explained thereon. Add the above losses (a) and (b) for the total loss due to unburned combustible. (Any loss due to unburned CO in the flue gas is not included.)

II. L_M, heat loss due to moisture and hydrogen in fuel: Using H and M, obtain W_M from Fig. B-2a. Using ratio FC/VM, obtain W_H from Fig. B-2b. Using sum $(W_M + W_H)$ and difference between flue gas exit temperature and inlet air temperature $(T_e - T_r)$, obtain total loss from Fig. B-2c.

III. L_G, heat loss in dry gases: Using CO_2 and ratio FC/VM, obtain W_C from Fig. B-3a, as explained thereon. Using loss due to unburned combustible, L_C from item I, and $(W_C - W_H)$, obtain W_G from Fig. B-3b. Using W_G and difference between flue gas exit temperature and inlet air temperature, $(T_e - T_r)$, obtain loss from Fig. B-3c.

IV. L_A, heat loss due to moisture in air: Using CO_2 and ratio FC/VM, obtain W_A from Fig. B-3a, as explained thereon. Using W_A and difference between flue gas exit temperature and inlet air temperature $(T_e - T_r)$, obtain loss from Fig. B-4.

V. L_R, heat loss due to radiation: Using steam flow, obtain loss from Fig. B-5.

VI. L_U, heat loss unaccounted for: This loss is intended to correct for errors in measured losses and omission of miscellaneous minor losses. For our purposes, this shall be taken as 1.5%.

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Efficiency:

Boiler efficiency is obtained by subtracting the above heat losses from 100.

$$\text{Efficiency, \%} = 100 - (L_C + L_M + L_G + L_A + L_R + L_U)$$

SAMPLE PROBLEM

General Data:	Coal Data:
Combustible in fly ash = 30%	Ash = 10.0%
Combustible in ash pit = 15%	Moisture = 8.0%
Flue gas exit temp., T_e = 480F	H_c = 14,650 Btu/lb
FD fan inlet air temp., T_r = 80F	FC/VM = 1.65
CO_2 at ID fan outlet = 13.6%	
Steam flow = 150 Mlbm/hr	

I. L_C, Heat Loss due to unburned combustible:
   (a) Heating value of fuel as fired:
   $H = (1 - A/100 - M/100) * (H_c)$
   $H = (1 - 10/100 - 8/100) * (14650) =$ 12,000 Btu/lb
   (b) From percentage refuse distribution sheets, ash pit refuse 30% total refuse, fly ash = 70% of total refuse.
   (c) From Fig. B-1, uncorrected unburned carbon loss in ashpit refuse with 15% combustible = 2.2%
   But for 30% of total refuse, corrected unburned carbon loss in ashpit refuse = 2.2 × 0.30 = 0.66%
   (d) From Fig. B-1, uncorrected unburned carbon loss in fly ash with 30% combustible = 5.2%
   But for 70% of total refuse corrected unburned carbon loss in fly ash = 5.2 × 0.70 = 3.64%
   (e) Total Heat Loss due to unburned combustible = 4.30%

II. L_M, Heat Loss due to moisture and hydrogen in fuel:
   (a) From Fig. B-2a, W_m = 6.6 lb/MKB
   (b) From Fig. B-2b, W_H = 33.3 lb/MKB
   (c) $W_m + W_H$ = 39.9 lb/MKB
   (d) From Fig. B-2c for $(T_e - T_r)$ = (480-80) = 400°F, Heat Loss due to moisture and hydrogen in fuel = 4.90%

III. L_G, Heat Loss in dry gases:
   (a) From Fig. B-3a, for CO_2 = 13.6% and FC/VM = 1.65, W_c = 1085 lb/MKB
   (b) From Fig. B-3b for $W_C - W_H$ = 1052 and L_C = 4.30, W_G = 1006 lb/MKB
   (c) From Fig. B-3c, Heat Loss in dry gases = 9.66%

IV. L_A, Heat Loss due to moisture in air:
   (a) From Fig. B-3a, for CO_2 = 13.6 and FC/VM = 1.65, W_A = 1004 lb/MKB
   (b) From Fig. B-4, Heat Loss due to moisture in air = 0.25%

V. L_R, Heat Loss due to radiation:
   (a) Using individual boiler charts, read off % radiation loss for steam flow. For illustrative purposes, Heat Loss due to radiation = 0.65%

VI. L_U, Heat Loss unaccounted for: = 1.5%

$$\text{EFFICIENCY, \%} = 100 - (L_C + L_M + L_G + L_A + L_R + L_U)$$
$$= 100 - (4.30 + 4.90 + 9.66 + 0.25 + 0.65 + 1.50)$$
$$= 100 - 21.26 = 78.74\%$$

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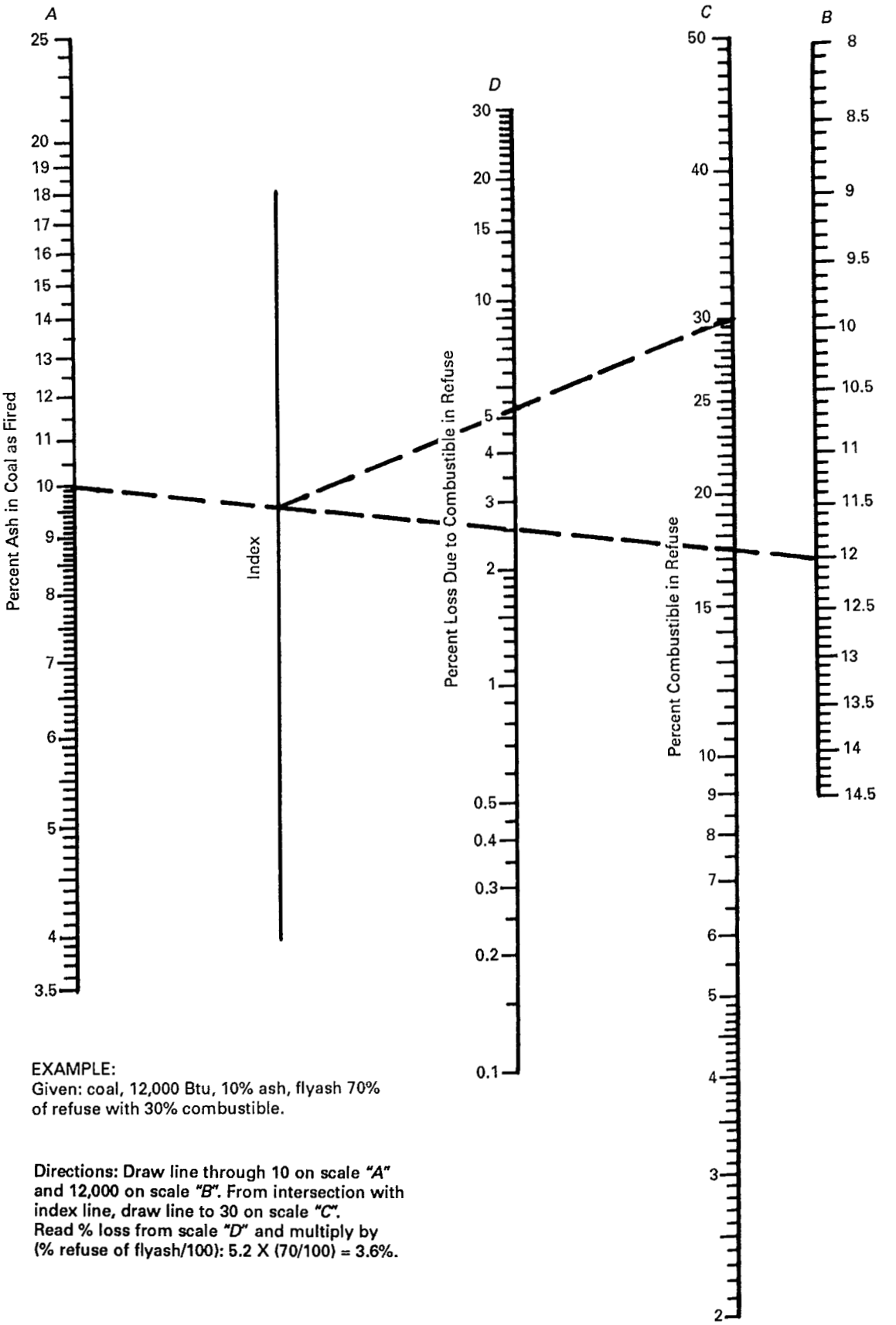

EXAMPLE:
Given: coal, 12,000 Btu, 10% ash, flyash 70% of refuse with 30% combustible.

Directions: Draw line through 10 on scale *"A"* and 12,000 on scale *"B"*. From intersection with index line, draw line to 30 on scale *"C"*. Read % loss from scale *"D"* and multiply by (% refuse of flyash/100): 5.2 X (70/100) = 3.6%.

FIG. B-1 LOSS DUE TO COMBUSTIBLE IN REFUSE
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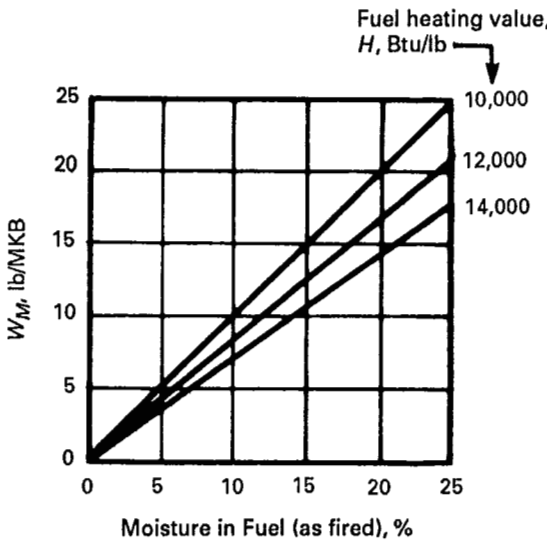

FIG. B-2a POUNDS OF MOISTURE IN FUEL AS A FUNCTION OF PERCENT MOISTURE IN FUEL AND FUEL HEATING VALUE

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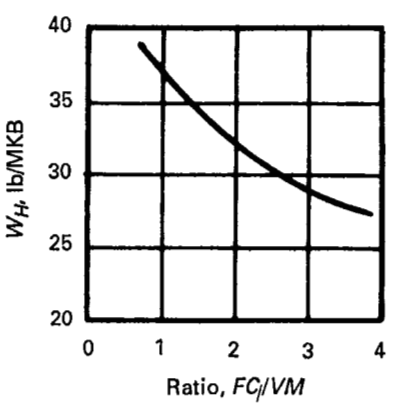

FIG. B-2b POUNDS OF HYDROGEN IN FUEL AS A FUNCTION OF FIXED CARBON TO VOLATILE MATTER RATIO

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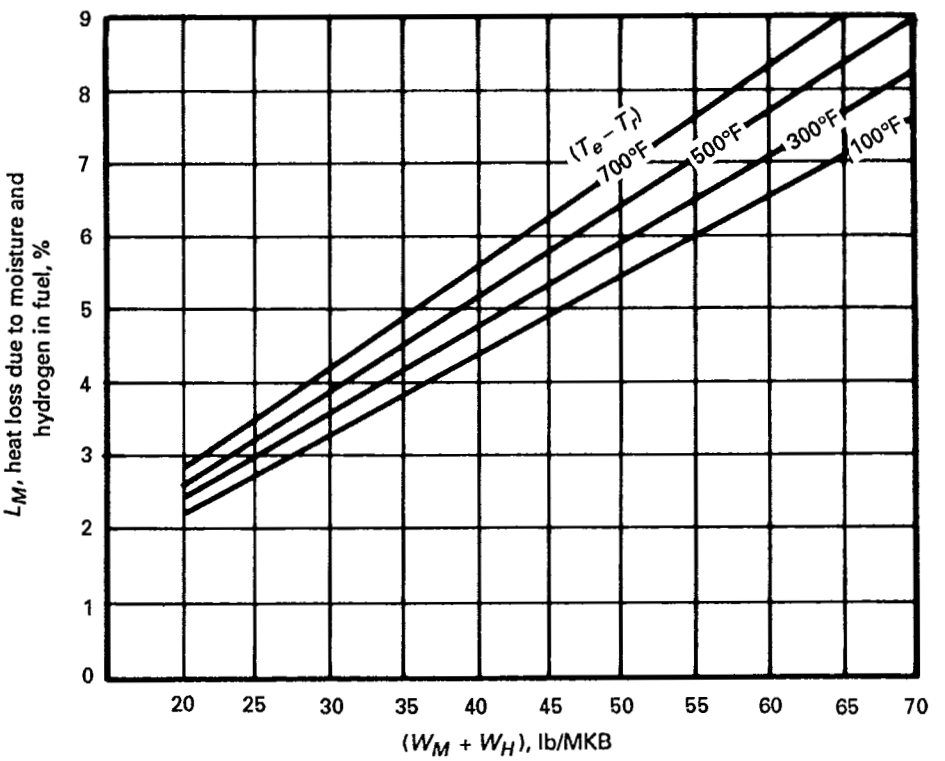

FIG. B-2c HEAT LOSS DUE TO MOISTURE AND HYDROGEN IN FUEL AS A FUNCTION OF TEMPERATURE HEAD (EXIT GAS TEMPERATURE AND AIR INLET TEMPERATURE)

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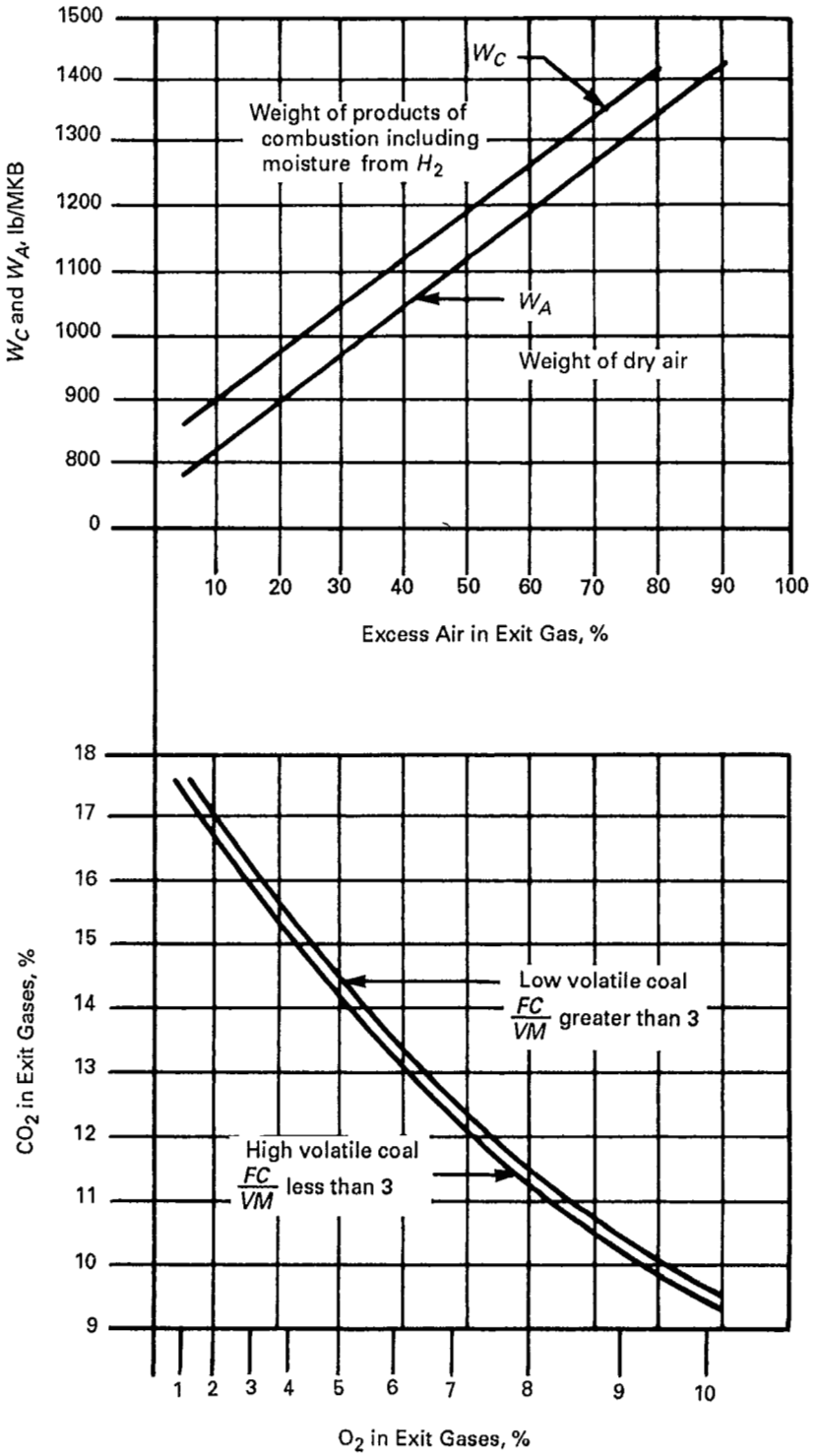

From % CO_2, move horizontally to intersection with FC/vM curve.
Move vertically to correct weight curve from which the weight is read from the vertical scale.

FIG. B-3a WEIGHT OF AIR AND FLUE GAS AS A FUNCTION OF EXCESS AIR

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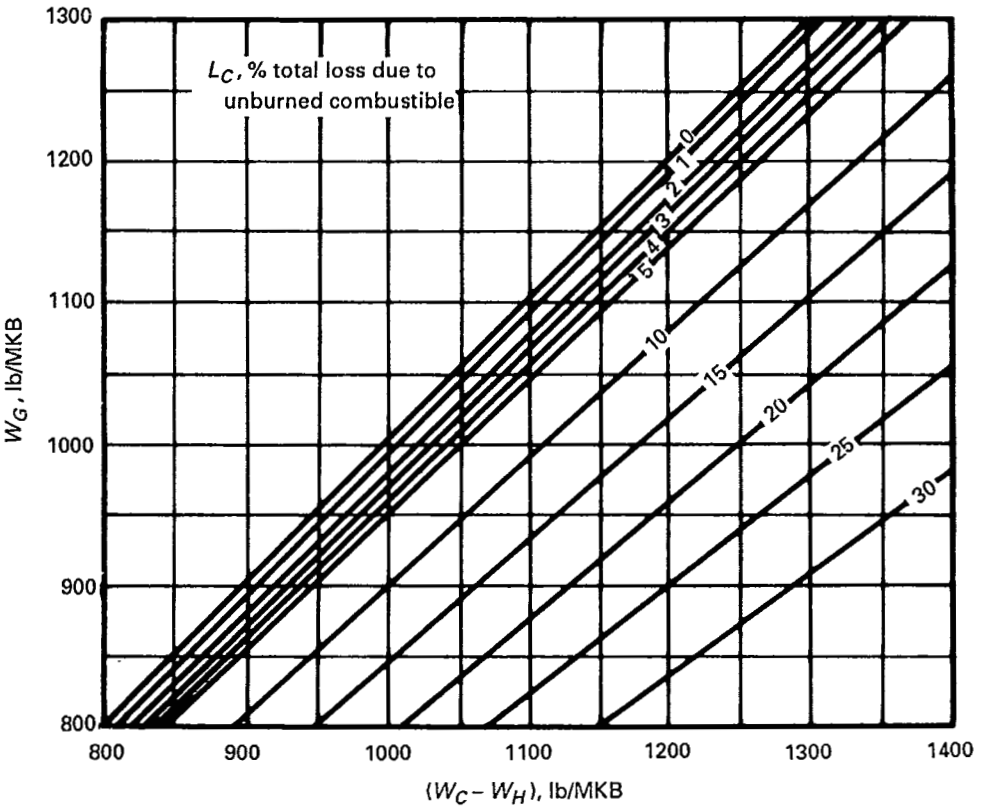

FIG. B-3b LOSS DUE TO UNBURNED COMBUSTIBLE
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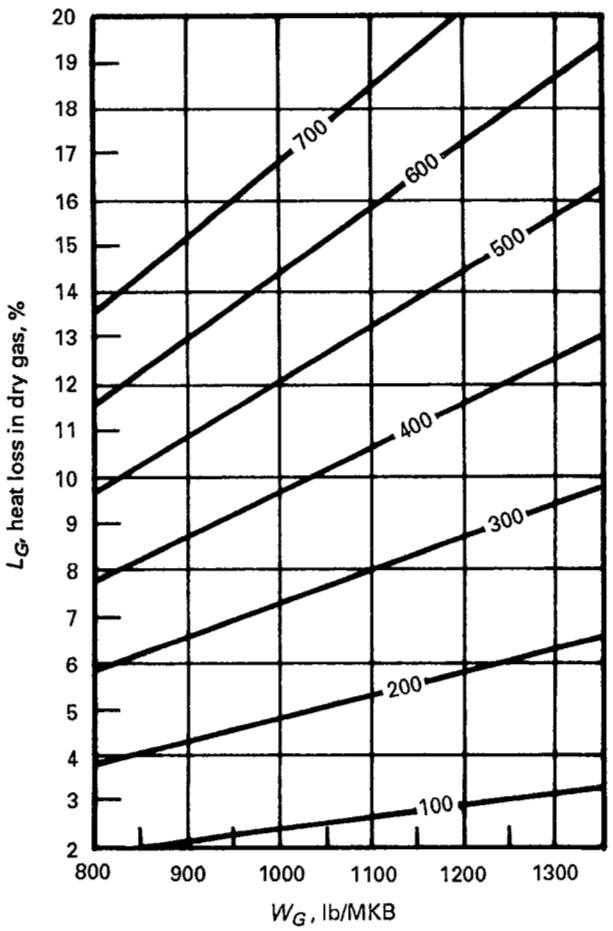

FIG. B-3c HEAT LOSS IN DRY FLUE GAS
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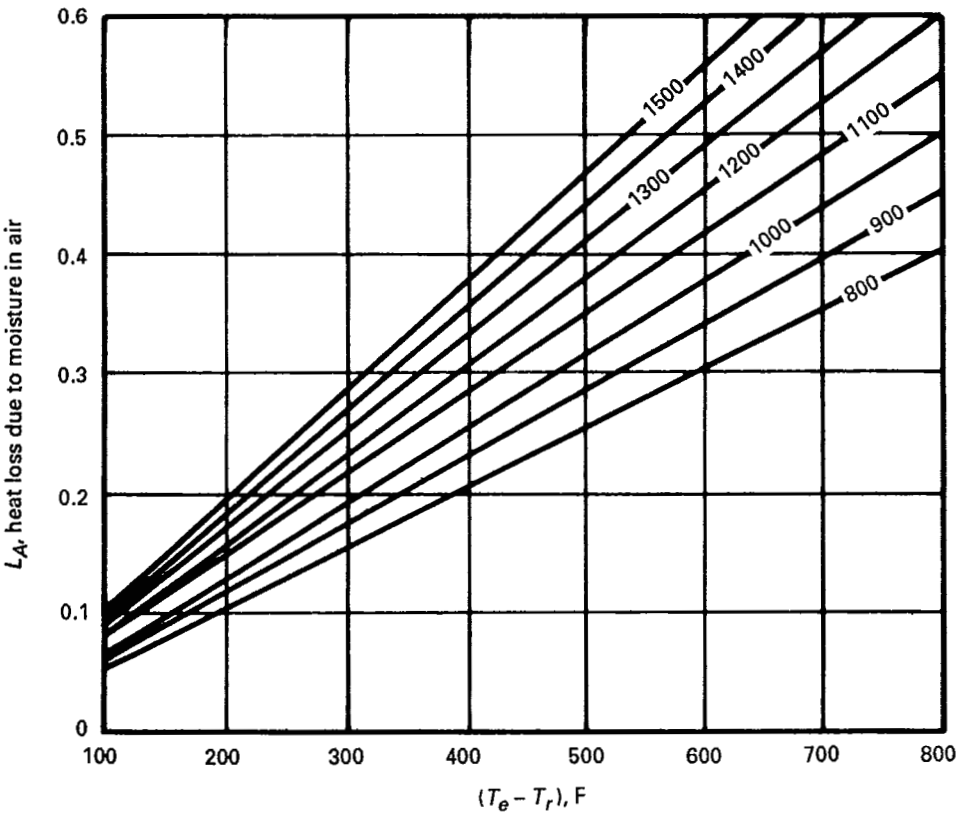

FIG. B-4 HEAT LOSS DUE TO MOISTURE IN AIR
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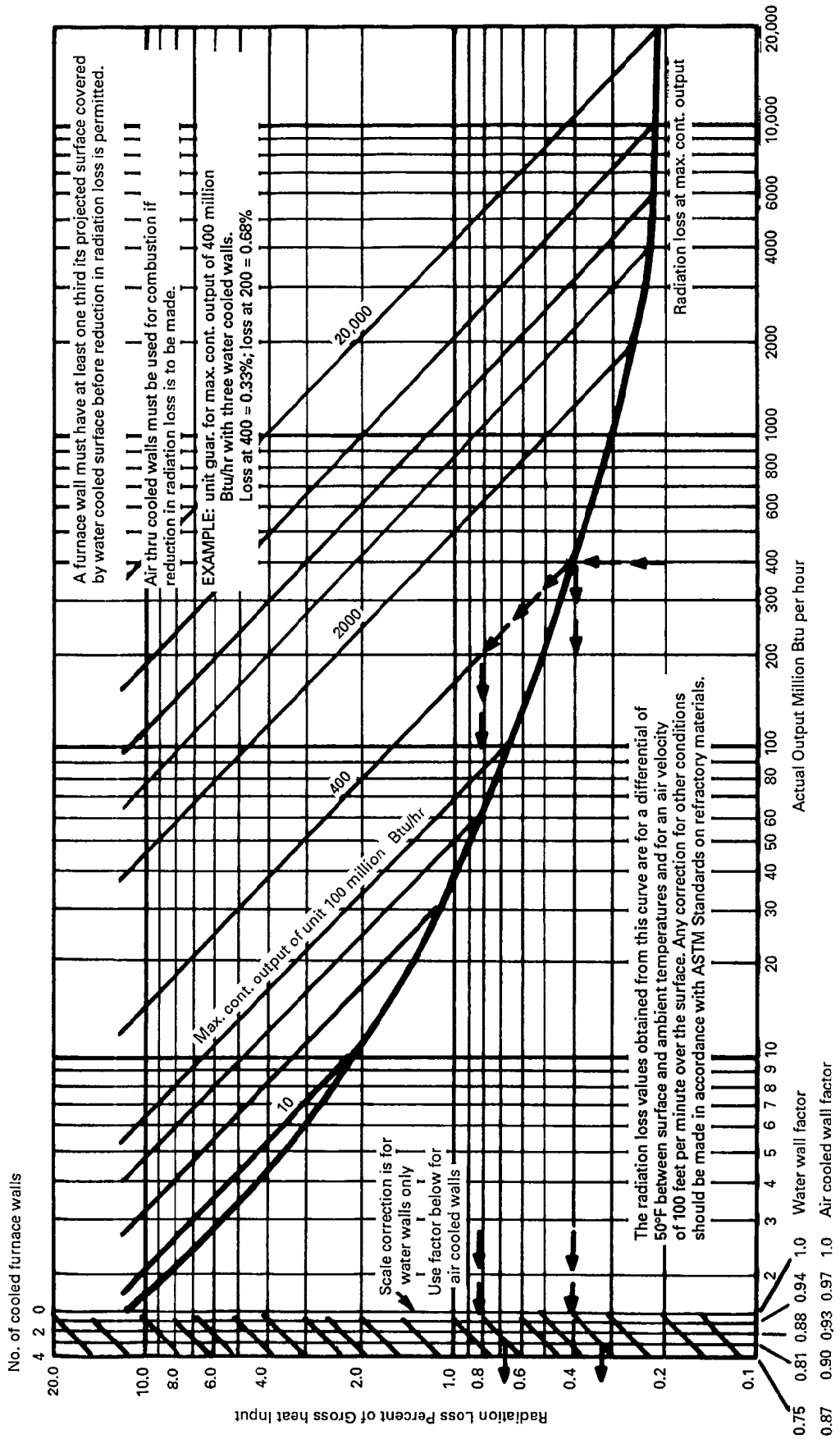

FIG. B-5 ABMA STANDARD RADIATION LOSS CHART
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APPENDIX C

The purpose of Appendix C is to demonstrate the difference between a wet and dry O_2 measurement system. 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.

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

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

(2) Assuming 100 pounds 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
Sulphur	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)	

(3) 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
Sulphur	$S + O_2 = SO_2$	$0.05 \cdot 1.0$	= 0.05 moles
Oxygen	$O_2 = O_2$	$0.11 \cdot (-1)$	= −0.11 moles

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

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

(5) 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

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

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

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

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

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

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 temp. = 80°F
$P_{saturation}$ = 0.5073 psia, using steam tables

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

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

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

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

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Reaction: $6.00\ C + 2.20\ H_2 + 0.05\ N_2 + 0.05\ S$

$+ 0.11\ O_2 + 0.44\ H_2O + 7.04\ O_2 + 26.47\ N_2 + 0.71\ H_2O$

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

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

Total moles $= CO_2 + N_2\ (fuel) + SO_2$

$+ H_2O\ (from\ H_2\ in\ fuel) + H_2O\ (from\ H_2O\ in\ fuel)$

$+ H_2O\ (air) + N_2\ (air)$

$= 6.00 + 0.05 + 0.05$

$+ 2.20 + 0.44 + 0.71 + 26.47$

$= 35.92$

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

Theoretical O_2 requirement $= 1.10 \cdot 7.04 = 7.74$

Moles of $N_2 = 7.74 \cdot 3.76 = 29.12$

Total moles of dry air = Moles of O_2 + Moles of N_2

$= 7.74 + 29.12 = 36.86$

Moisture in air $= 0.0211 \cdot 36.86 = 0.78$

Total moles flue gas $= CO_2 + N_2\ (fuel)$

$+ SO_2 + H_2O\ (from\ H_2\ in\ fuel)$

$+ H_2O\ (from\ H_2O\ in\ fuel) + H_2O\ (air)$

$+ N_2\ (air) + O_2\ (excess)$

$= 6.00 + 0.05 + 0.05 + 2.20 + 0.44 + 0.78$

$+ 29.12 + (7.74 - 7.04)$

$= 39.34$

$$\%\ O_2\ \text{by volume, wet} = \frac{\text{moles } O_2}{\text{total moles of wet flut gas}} \times 100$$

$$= \frac{(7.74 - 7.04) \cdot 100}{39.34}$$

$$= 1.78\%$$

$$\%\ O_2\ \text{by volume, dry} = \frac{\text{moles } O_2}{\text{total moles of dry flue gas}} \times 100$$

$$= \frac{(7.74 - 7.04) \cdot 100}{39.34 - (2.20 + 0.44 + 0.78)}$$

$$= 1.95\%$$

(11) 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 $= 40.22 \cdot 0.0211 = 0.85$

Total moles flue gas $= CO_2 + N_2\ (fuel) + SO_2$

$+ H_2O\ (from\ H_2\ in\ fuel) + H_2O\ (from\ H_2O\ in\ fuel)$

$+ H_2O\ (air) + N_2\ (air) + O_2\ (excess)$

$= 6.00 + 0.05 + 0.05 + 2.20 + 0.44$

$+ 0.85 + 31.77 + (8.45 - 7.04)$

$= 42.77$

$$\%\ O_2\ \text{by volume, wet} = \frac{\text{moles } O_2}{\text{total moles of wet flue gas}} \times 100$$

$$= \frac{(8.45 - 7.04) \cdot 100}{42.77}$$

$$= 3.30\%$$

$$\%\ O_2\ \text{by volume, dry} = \frac{\text{moles } O_2}{\text{total moles of dry flue gas}} \times 100$$

$$= \frac{(8.45 - 7.04) \cdot 100}{42.77 - (2.20 + 0.44 + 0.85)}$$

$$= 3.59\%$$

(12) 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.

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SECTION 7 — CYCLE INTERRELATIONSHIPS

7.1 GENERAL

For performance monitoring purposes, cycle interrelationships are the operational and mechanical interactions which exist between the many systems and components in a power plant. The individual parameters cited for measurement and calculation in Section 6 are usually directly related to the equipment performance but in some instances they may be a result of the interaction or influence of another piece of equipment. This Section alerts the performance engineer to be aware of interactions and relationships between the many systems and components in the power plant. Failure to recognize these interrelationships may lead to erroneous diagnostics.

Recognizing and understanding these relationships provide a basis for using performance as indicators of equipment physical condition. It is necessary to quantify the more significant interrelationships in order to compensate for them in the interpretation of the performance monitoring results. Two concepts are presented to help the performance engineer understand interrelationships. One concept is the monitoring envelope for defining boundaries. The other is a matrix showing the influence of operational parameters on performance.

7.2 MONITORING ENVELOPE CONCEPT OF INTERRELATIONSHIPS

There are many interrelationships in effect at all times in a power plant. These have significant influence on the operation and performance of the unit. Since it would be difficult for a performance monitoring program to accommodate all possible cycle interactions, the performance engineer is faced with determining which interactions to include.

In order to define which interrelationships to consider in a performance monitoring program, it is important to establish a boundary, or envelope, around the unit or equipment being monitored. The envelope involves imaginary but defined boundary lines which surround the equipment or unit such that all energy that crosses the boundary can be identified. The temperature, pressure, and mass flow of the fluid crossing the envelope boundary and the work entering or leaving the envelope must be taken into account so the influence, or interaction, can be determined. Referring to Fig. 7.1, it can be seen that several envelopes can be established, such as unit, cycle, or equipment envelopes. The envelopes depicted on Fig. 7.1 are one possible selection and the user is free to define envelopes which best represent a particular cycle.

If the focus of the performance monitoring program is unit heat rate, the envelope is drawn around the entire unit. The ambient air and water temperatures, fuel flow and quality, electric sendout, and connections to other units (physical and electrical) must be measured and the effect should be determined. Ambient conditions affect the boiler and heat rejection cycles; fuel quality affects the boiler efficiency and heat input; low load operation affects the turbine cycle efficiency; and steam and electrical ties to other units remove energy from the unit.

If the focus of the monitoring program is the turbine cycle, the steam conditions entering the turbine cycle from the boiler affect the turbine efficiencies, feedwater heater performance, and generator output. Similarly, the condenser performance affects turbine backpressure which affects the turbine steam flow and generator output.

If the focus of the program is the steam generator cycle, final feedwater and cold reheat steam conditions, ambient air temperature, and fuel conditions would be measured and quantified.

When the monitoring program is on the equipment level, such as feedwater heaters, then the performance of the steam turbine must be considered when analyzing the performance of the heaters because steam crosses the boundary around the feedwater heaters. Now, if the performance engineer were analyzing the performance of the steam generator, the engineer would be concerned about the performance of the final feedwater heater and the cold reheat steam conditions, because feedwater and steam cross the envelope boundary. Air heater,

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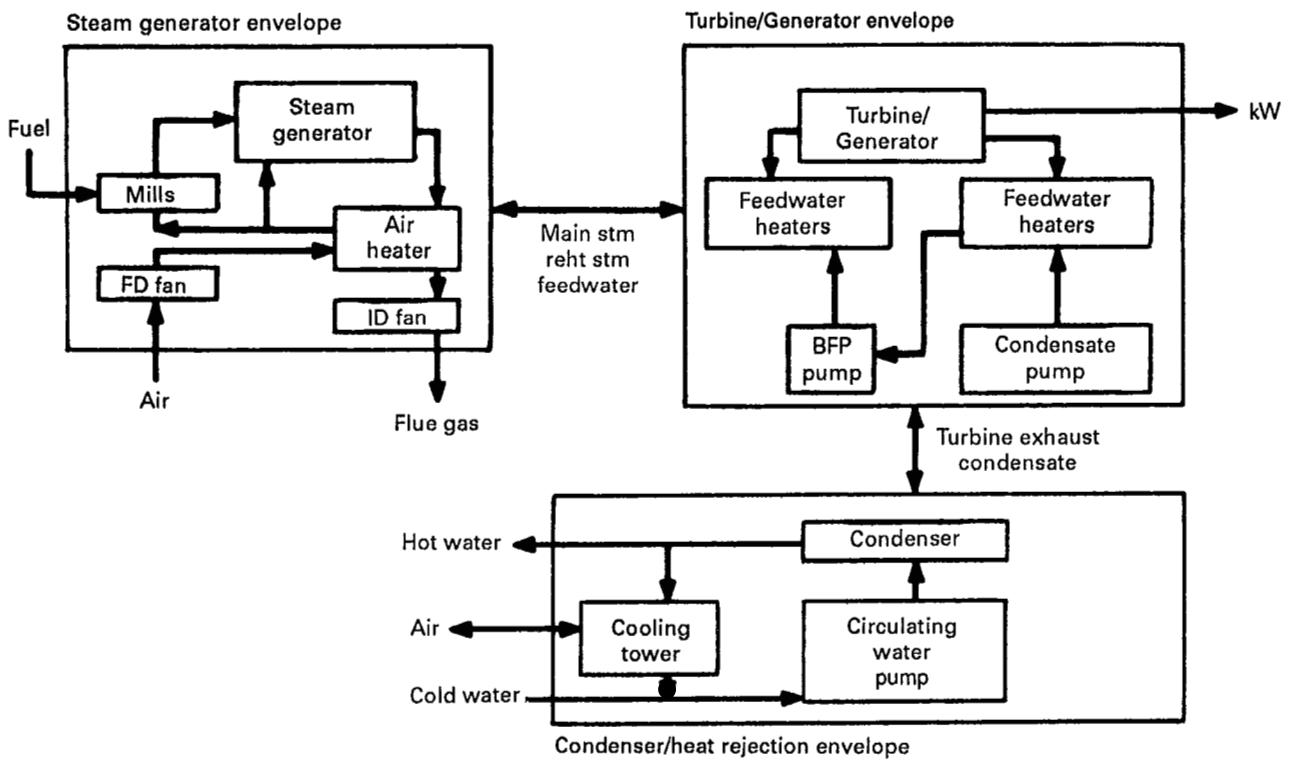

FIG. 7-1 CYCLE INTERRELATIONSHIPS

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fan, and pulverizer performance also affect the steam generator.

Applying the monitoring envelope concept in the design and conduct of the performance monitoring program will assist the performance engineer in better meeting the objectives established in Section 4. 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. The following paragraphs make the distinction between operational and mechanical interrelationships.

7.3 OPERATIONAL INTERRELATIONSHIPS

Operational interrelations include those interactions which 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.

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*) 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.

(*e*) 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 Section 9. 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.

7.4 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 in-

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corporating that knowledge into the interpretation of the results is vital to the performance monitoring program.

7.5 MATRIX OF CYCLE INTERRELATIONSHIPS

A matrix identifying interrelationships which may occur in a typical fossil-fueled electric generating plant is shown in Table 7.1. The vertical axis lists thirteen 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 which 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.

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. Further, 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.

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 which 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 which 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 which 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 which 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 which influence the net electric output.

(*f*) *Fuel Burn Rate.* This is the results category identifying the interactions which 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 of which the user wishes to pursue the problem. The matrix is intended to convey the 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 thirteen general categories.

The following matrix is intended as a guide only. All the interrelations on the horizontal axis are valid only within the context of the thirteen individual categories. Vertical comparison under each performance category should not be interpreted on a relative basis.

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TABLE 7.1 MATRIX OF CYCLE INTERRELATIONS

	Performance Results					
Cycle Parameters	Boiler Operation & Performance	Turbine Operation & Performance	FW Heaters Operation & Performance	Cond./C.T. Operation & Performance	Electric Output	Fuel Burn Rate
1. Boiler Fluid Outputs Main Steam Flow Main Steam Pressure Main Steam Temp.	XXX	XXX	XX	XXX	XXX	XXX
2. Reheater Fluid Inputs Cold RH Steam Flow Cold RH Steam Pressure Cold RH Steam Temp. 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 Temp.	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	- - -
5. Feedwater Heater, Extraction Steam, Drainage FWH Ext. Steam Flows FWH Ext. Steam Pressure FWH Steam Temp. FWH Drainage Pressure FWH Drainage Temp.	X	XXX	XXX	X	XX	X
6. Other Turbine Cycle Fluid Extractions (all F., P., T.) Aux. Steam House Htg. Steam Air Prehtg. Fluid Process/Extnl. System	XX	XX	XX	XX	XX	XX
7. Condenser Steam Side Conditions Condenser Pressure Hotwell Condens. Temp. Air In-Leakage/Removal Rate	XX	XXX	X	XXX	XX	XX

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TABLE 7.1 MATRIX OF CYCLE INTERRELATIONS (Cont'd)

	Performance Results					
Cycle Parameters	**Boiler Operation & Performance**	**Turbine Operation & Performance**	**FW Heaters Operation & Performance**	**Cond./C.T. Operation & Performance**	**Electric Output**	**Fuel Burn Rate**
8. Condenser, Cooling Tower Heat Sink Conditions Circulating Water Flow Circulating Water Inlet Temperature Cooling Tower Air Flow Barometric Pressure Ambient Air Temp. Ambient Air Relative Humidity	XX	XXX	X	XXX	XX	XX
9. Condensate, Feedwater System Conditions L.P. FWH Temp. Rises BF Pumps Temp. Rises FWH Pressure Drops (Tube Side) H.P. FWH Temp. Rises	X	- - -	XX	- - -	X	X
10. Boiler Fluid Inputs Final Feedwater Flow Final Feedwater Pressure Final Feedwater Temp. Main Steam DSH Flow	XXX	X	XXX	- - -	XX	XXX
11. Boiler Combustion Air Inputs Pri., Sec., Ter. Air Flows Barometric Pressure Ambient Air Temp. Relative Humidity	XX	- - -	- - -	X	X	XX
12. Boiler Incidental Inputs Seal Air Sys. Flow Sootblowing System Boiler Air In-Leakage	X	- - -	- - -	- - -	- - -	X
13. Boiler Heat Inputs Fuel Flows Fuel Temperature Fuel Analyses Atomizing Steam Supply Air Preheating System	XXX	X	X	X	XX	XXX

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SECTION 8 — DIAGNOSTIC TECHNIQUES

8.1 INTRODUCTION

The purpose of this Section is to provide guidance on converting performance results into equipment physical condition. This information is typically expressed in terms of performance characteristics of the equipment. This information will be used by operations and maintenance personnel in performance optimization.

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

(*a*) operational changes
(*b*) maintenance actions
(*c*) system or equipment modifications
(*d*) new or replacement equipment.

8.2 DIAGNOSTIC METHODOLOGIES

The diagnostic techniques presented in this Section 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. It is important to be able to determine whether changes in a component's parameters originate from local conditions or conditions upstream of the process, passed down as a result of interaction with another component. Interactions between systems or components affect values of common boundary interface parameters. The subject of system interactions (interrelationships) was discussed in Section 7.

Any of the following conditions can exist at any given time:

(*a*) A portion of the equipment is operating inefficiently or off-design

(*b*) A portion of the equipment or system is operating in a non-standard configuration

(*c*) A portion of the equipment is not operating at all

A situation is created where a portion of the system parameters will have off-design values. In such cases, superficial problems may mask the actual root problem. Cases such as this may require testing to determine the cause-and-effect relationships between the superficial problems and the actual root problem. For example, many articles and papers are available to guide in power plant diagnostics, and a list of a few of these is provided at the end of this Section. However, a thorough fundamental understanding of one's own system is essential in applying this guidance to problems at hand.

8.2.1 Generic Performance Curves or Diagrams. Performance curves provide a means of diagnosing general trends in performance. Target and as-operating curves may be plotted and compared to determine operation outside of typical design or reference conditions. Off-target conditions may be caused by changes in the operating mode, equipment malfunctions, or equipment failures. Figure 8-1 is an example of a generic performance curve. Performance curves are usually qualitative in nature and offer the fundamental principles which can be customized to unit specific conditions. This provides a simple and cost-effective means of characterizing equipment performance through a deductive process.

8.2.2 Performance Diagnostic Tables. In comparison with performance curves, diagnostic tables provide a more direct way of identifying root problems. The problems, their causes, and resulting effects are usually clearly delineated, thus requiring less interpretation. Tables 8-1 and 8-2 [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 the root cause. In some cases, special actions such as suggested testing may be required to isolate the root cause.

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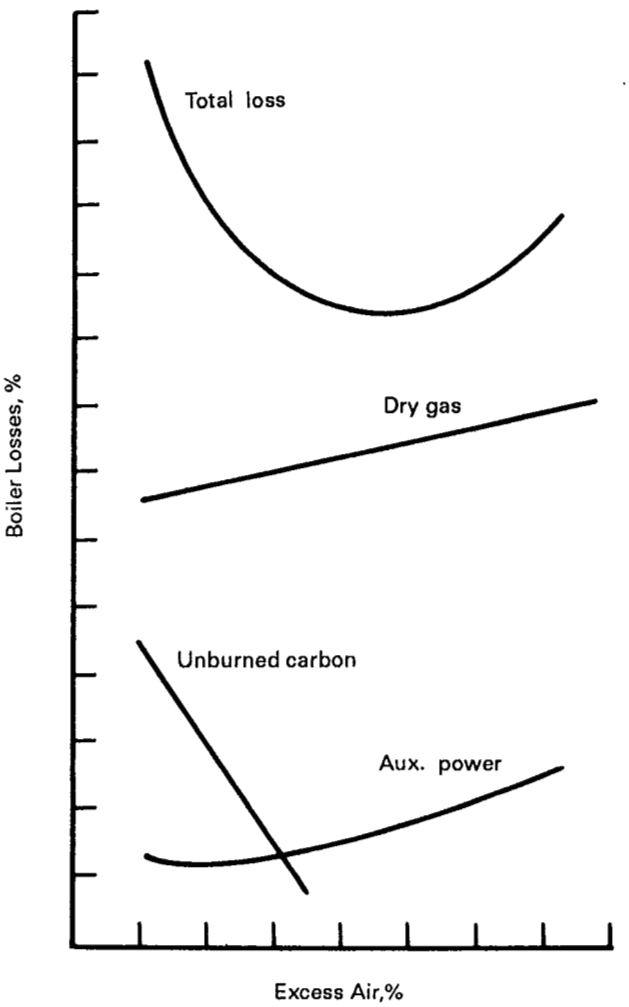

FIG. 8-1 PERFORMANCE CURVES TO CHARACTERIZE BOILER LOSSES — EXAMPLE FOR A COAL-FIRED UNIT

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TABLE 8-1 DIAGNOSTIC CHART OF TURBINE LOSS CHARACTERISTICS

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 valve
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:	Decrease 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

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TABLE 8.2 STEAM SURFACE CONDENSER DIAGNOSTICS

Cause/Measured Parameter	Condenser Pressure	Cooling Water Temperature Rise or Pressure Drop	Terminal Temperature Difference
Reduced Cooling Water Flow			
(1) Debris on tubesheet/tubes (2) Deficient siphon loop vacuum pump (3) Throttled CW valves	Increase dependent on severity	Increase proportional to CW flow reduction	Increase dependent on severity
Reduced Surface Area			
(1) Tube plugging (2) Debris on tubesheet (3) Low waterbox level	Increase dependent on severity	Increase proportional to CW flow reduction	Increase dependent on severity
Tube Fouling			
(1) Deposits/growth on tubes (2) Air in-leakage/inadequate air removal	Increase dependent on severity	Slight increase due to increased cond. duty	Increase dependent on severity

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8.2.3 Performance Logic Trees. Logic trees present successive levels of information with increasing detail as the tree expands [35]. Figure 8-2 is an example of a typical heat rate logic diagram. Beginning with the top events of the logic tree and progressing through to the other events, a set of performance parameters are identified at each branch which acts as a road map 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 the logic trees are decision trees. Whereas the logic tree does not require decisions at branch points, the decision tree does. Focusing on plant symptoms that are utilized in the operation and control of a unit, decision trees are models that provide guidance to determine the root cause of deviations in performance parameters [35, 36]. Their structure is such to enhance 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 8-3 is an example illustration of the decision tree concept for investigating deviations of main steam temperatures from target values.

8.3 DIAGNOSTIC PROCESS

The diagnostic methodologies discussed in para. 8.2 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, but yet formal, procedure for isolating the root cause of a problem. The process presented entails the following criteria which are necessary for the development of an efficient diagnostic process [23].

(*a*) The process should be deductive, calling for a step-by-step approach.

(*b*) The process should encourage the diagnostician to focus on the observed symptoms.

(*c*) The process should be flexible in the type of symptoms it may address.

(*d*) The process should provide for establishing testing and analysis programs to facilitate efficient diagnostics.

(*e*) The process should provide for making periodic judgements as to the cost-effective pursuit of the root cause.

The root cause analysis should be carried out either to a level of detail which 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 suggested diagnostic process, consisting of seven steps, represents a gradually narrowing scope of the problem [23].

Step 1 — Identify the components which 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

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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 — 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 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 which 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.

8.4 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. At each step key performance parameters are evaluated that describe the control volume being 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.

8.4.1 Unit Level Diagnostics. Unit level diagnostics should focus on variations in unit heat rate. Investigations into the causes of an increase in heat rate are often initiated by a query from plant management with no more specifics than 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 individual terms in the accounting algorithms used to determine the heat rate. The first question to answer is whether the apparent increase in the heat rate represents an actual degradation in the unit performance or a problem with one of the parameters used to calculate heat rate.

Many plants average heat rate data over periods ranging from several days to a month. This long-term averaging was originally instituted as an accounting tool for use in tracking fuel inventory. Although it has merits as an accounting tool for performance tracking, it tends to dilute equipment-specific performance information. Shorter time periods are recommended for averaging, such as over a few minutes.

8.4.1.1 Unit Generation. With respect to unit generation, a review of the data should look for both changes in the level of generation as well as changes in the pattern of generation. Units run higher heat rates at part load operation. Average numbers in a routine report, such as average load, or total numbers, such as total load generated over the period, do not give the reader the nature of the variations in the load during that period. At a minimum, the instantaneous data should be examined to determine whether there have been changes in the loading of the unit in question and whether the changes, if present, may have contributed to the observed rise in the heat rate. On units that are load cycling, it is additionally important to know the effects of part load operation on heat rate penalties. The relative significance of some parameters is much greater at part load.

8.4.1.2 Auxiliary Electric Power. Minimization of auxiliary electric power requirements not only addresses the need to secure equipment when not in use, but also addresses the need to optimize the operating combination of equipment so as to minimize the auxiliary electric power requirements. This includes such things as being knowledgeable of the optimum pair of feed pumps to run if only two of three pumps are required for a particular load condition. Likewise, instantaneous data on auxiliary electric power may show changes in the percent auxiliary electric power (auxiliary electric power/generated load). This is often a result of a change in the operational configuration of some of the auxiliary equipment.

8.4.1.3 Fuel Characteristics. Examinations of fuel characteristics and usage during the period in question is also in order. If the fuel usage reporting

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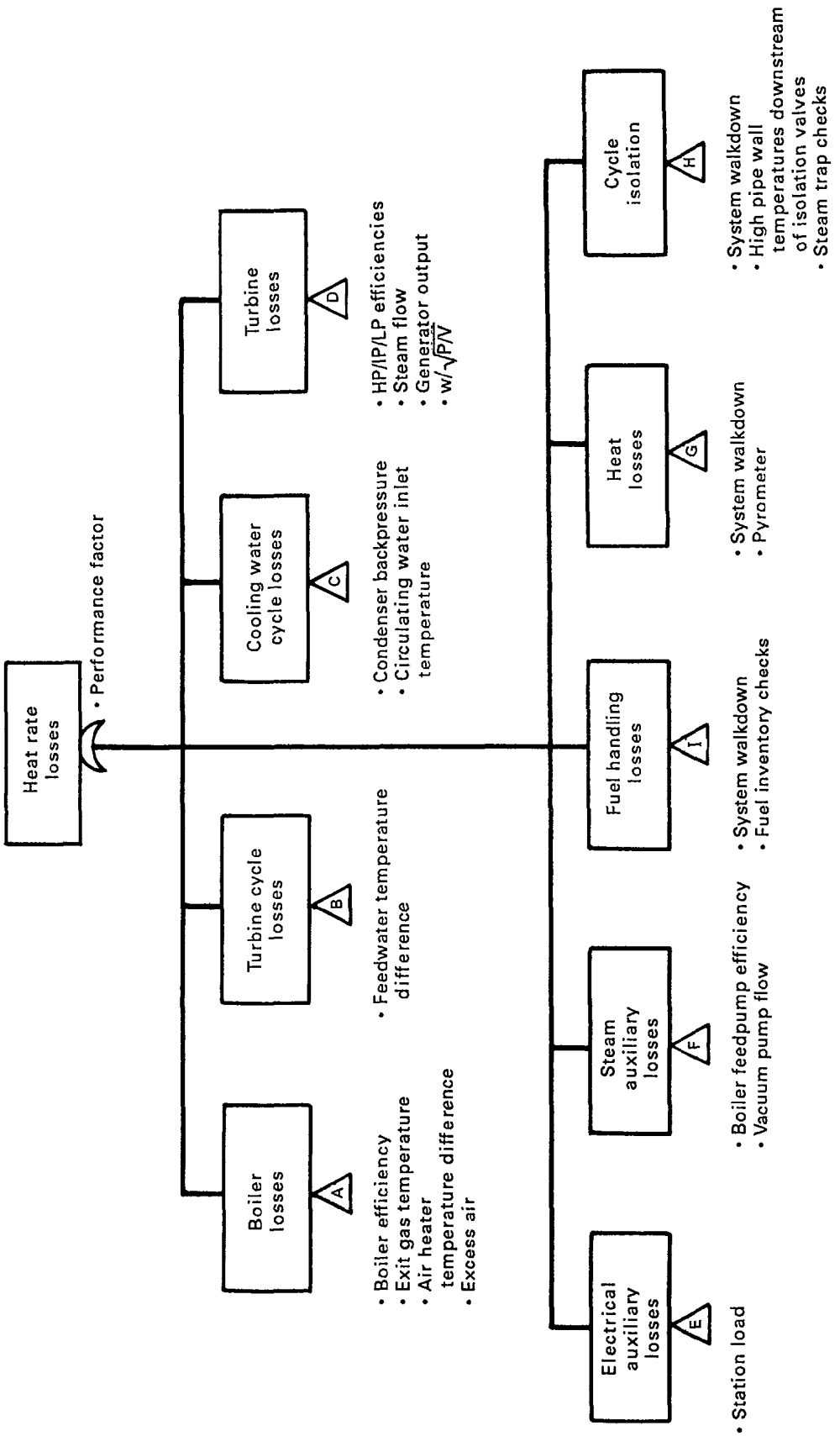

FIG. 8-2 HEAT RATE LOGIC TREE — MAIN DIAGRAM
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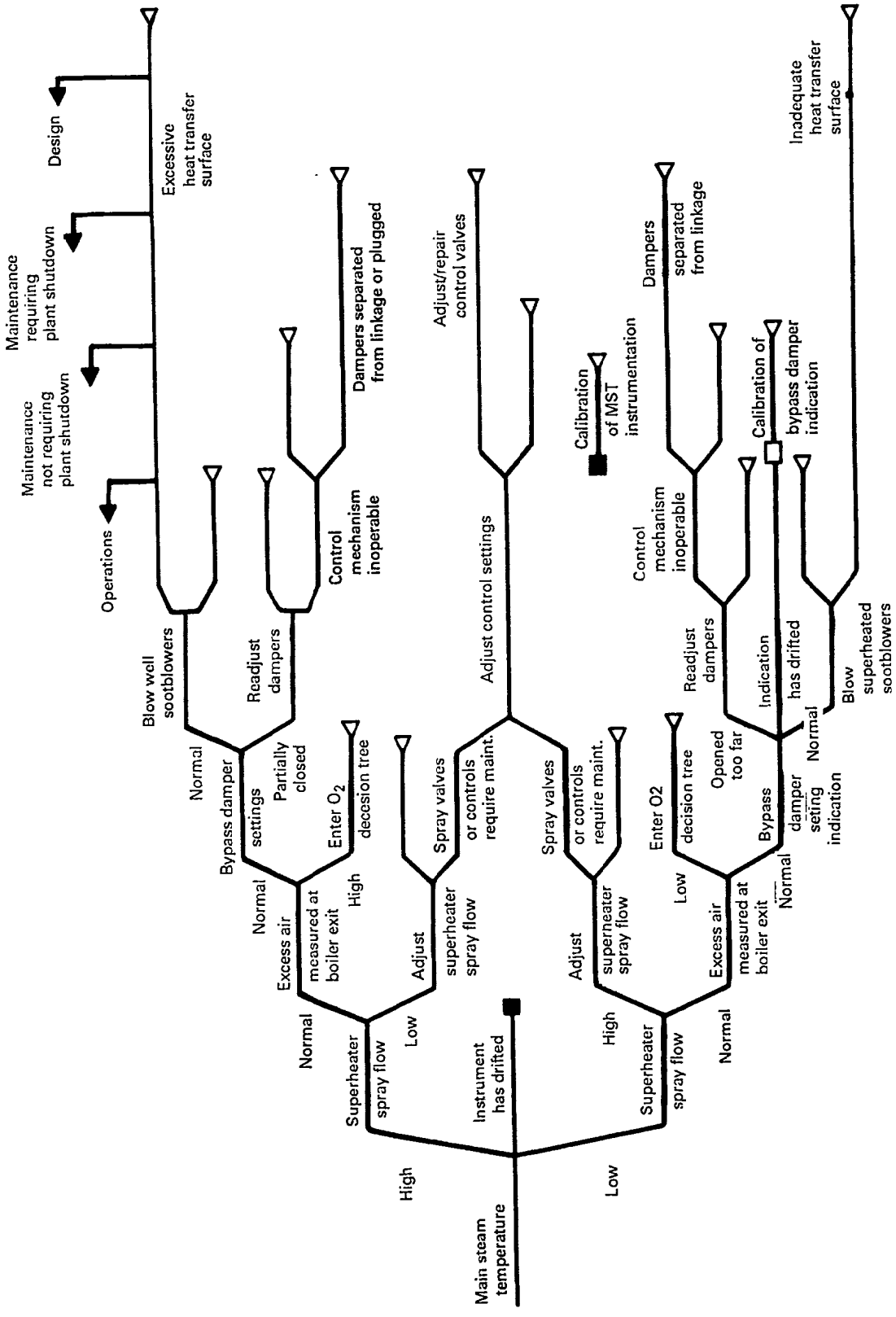

FIG. 8-3 ILLUSTRATION OF DECISION TREE CONCEPT FOR INVESTIGATING PERFORMANCE PARAMETER DEVIATIONS

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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. The first group of techniques includes many 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 assure representative samples of the fuel as it enters the boiler. The second group of techniques include on-line elemental or constituent analysis. These techniques are capable of analyzing larger samples, but 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 which 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.

8.4.2 Unit Cycle Diagnostics. Overall unit performance diagnostics is done by comparing unit heat rate 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 the analyzing the three cycles (boiler, turbine, and condenser). The influence of the three cycles on unit heat rate is examined through the use of the boiler efficiency, the turbine cycle heat rate, and the condenser cleanliness factor and pressure.

In unit level diagnostics, the cycle interrelationships define boundary conditions impressed by one cycle on another. The interrelation between the boiler cycle, the turbine cycle, and the condenser cycle will be discussed in terms of the energy supplied by the boiler to the feedwater/steam cycle and the energy rejected by the turbine cycle to the condenser.

For a reheat unit, primary expenditures of energy by the boiler to the feedwater/steam cycle are:

(*a*) the energy supplied to feedwater;

(*b*) the energy supplied as superheat attemperation water;

(*c*) the energy supplied to cold reheat steam;

(*d*) the energy supplied to reheat attemperation water;

(*e*) the energy supplied for auxiliary requirements.

The energy rejected by the turbine cycle to the condenser is comprised of:

(*a*) LP turbine energy rejected to the condenser;

(*b*) energy rejected to the condenser from all the drains dumping to the condenser.

Some of the more commonly monitored controllable parameters for unit level diagnostics are:

(*a*) throttle steam temperature
(*b*) throttle steam pressure
(*c*) reheat steam temperature
(*d*) condenser pressure
(*e*) station electrical power
(*f*) excess air
(*g*) exit gas temperature
(*h*) superheat spray flow
(*i*) reheat spray flow

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. A word of caution is that if the unaccountable losses are larger than the sum of the accountable losses, the performance monitoring program is deficient in scope and instrumentation. The relative sizes of the accountable and unaccountable losses vary from plant to plant. Through continued probing the size of the unaccountable losses can be reduced by continually identifying new accountable losses. The point is that a generic road map can be outlined for the pursuit of heat rate, but to be truly effective in a heat rate enhancement effort requires the attainment of a good understanding of one's plant and its operation.

8.5 TURBINE CYCLE

8.5.1 Turbine Cycle Diagnostics. The overall performance of the turbine cycle can be characterized

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by comparing the unit's gross turbine cycle heat rate and net turbine cycle heat rate to expected values.

Major components in the turbine cycle are the turbines themselves (HP, IP, and LP); the low pressure and high pressure feedwater heaters; and the boiler feed pumps. These components are often thought of as having limited operator control. As such, performance enhancements to these components are thought of as outage items. With regard to the required preparation for outage items, preoutage diagnostics is very important. Such diagnostics enable improved planning including: selection and ordering of high cost, long lead time spare parts; probable outage repair procedures; modifications to operating methods; recommendations for improved instrumentation and test activities; optimization of outage timing; and scheduling on a cost/benefit basis.

A turbine cycle test may range from an ASME PTC 6 test requiring approximately 200 instruments (temperature, pressure, differential pressure, etc.) to an ASME PTC 6.1 test requiring approximately 50 instruments. For diagnostic purposes, a combination of periodic evaluation of trended variables followed by specifically targeted diagnostic tests will provide the most information. Trending of turbine performance parameters is discussed in PTC 6S. The turbine, more than any other major component, must be compared at consistent operating points (valve positions).

8.5.2 Turbine Diagnostics. An important point to be realized with regard to turbine efficiency is that operating practices can, and routinely do, influence turbine efficiency.

One of the best examples of the variability in turbine efficiency is found in the HP turbine efficiency which changes significantly over the load range based on the method used to admit the steam. Except for sliding pressure units, this is an area that the operator typically does not have a great deal of control over, but it does serve to show the point and introduce the next concept of operator influenced changes in turbine heat rate. Increases in the turbine heat rate can be the result of the operator running the throttle and/or reheat temperature below the target value. Decreases in generator output with an attendant increase in the turbine heat rate can be the result of the operator running the throttle pressure below the target value.

An examination of changes in both the efficiency and output of the turbine should include inspections of the feedwater heater operating logs. Changes that result in raising or lowering the shell pressure of the feedwater heaters will cause changes in the extraction flow rate and affect turbine efficiency and output. Localized performance can be examined by determining the efficiency of groups of adjacent stages using enthalpy-drop calculations. Measurements can be made at inlet and/or extraction points to trend the collective efficiency of the included stages. This compares the actual energy used to the available energy in an isentropic process and is most effective in detecting gradual deterioration of the included stages. This technique, as with any approximate techniques, is not without potential pitfalls. The pressures and temperatures are normally measured on the inlet or extraction lines and can, under certain circumstances, be unrepresentative of the conditions prevailing in the turbine itself. Such a case would exist if worn tip seals allowed excessive leakage flow along the shell, and this leakage flow was a nonrepresentative fraction of the flow being extracted at a downstream location. This would result primarily in a nonrepresentative extraction temperature. Long-term tracking of stage group efficiencies is therefore best accomplished using stage groups bounded by turbine inlets, major extractions, and/or turbine outlets. This is not to say that trending of stage groups which are bounded on one or both ends by minor extractions should not be done. On the contrary, these should be monitored for the diagnostic information they can provide (such as the seal wear in the previous example).

The following diagnostic techniques are discussed in more detail in Reference [6].

8.5.2.1 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 para. 6.5.6), $W_{t,c}$, before comparing to a target condition, $W_{t,t}$.

The following are possible problems associated with the different possible findings:

(*a*) $W_{t,c} > W_{t,t}$

(*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.

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

(5) flow measurement is in error

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

(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

8.5.2.2 HP Turbine Efficiency Across Load Range. For single test point evaluation, compare to VWO heat balance values directly to establish performance trends for diagnostics. For multi-point test evaluation, additional data would be available for plotting and analysis. HP efficiency should be plotted against percent valve position and throttle flow or pressure ratio. For example, 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 light 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. 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.

8.5.2.3 Turbine Flow Function Across Load Range. Information pertinent to the internal conditions of the turbine is provided using the flow factor.

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

where:

W = flow to following stage
K = flow constant
P = pressure (absolute) to the following stage
v = specific volume to the following stage

When the flow constant K is plotted against throttle flow, it is an indicator of stage nozzle area changes.

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. Application of the flow function analysis for evaluating internal turbine conditions is detailed in para. 8.5.2.4.

8.5.2.4 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; adjacent packings may be badly rubbed

(c) Other leakages or bypasses may exist.

8.5.2.5 IP Turbine Efficiency Across 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. The measurement of the IP turbine efficiency will, as a result, 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 as described in a paper by Booth and Kautzmann [4] which 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.

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Excessive N2 (IP dummy) packing leakage will cause a misleading increase in IP section enthalpy-drop efficiency. The effect is somewhat amplified at low load. Note that in addition to the N2 (IP dummy) packing leaks, additional flows to the crossover area may also be attributed to excess flow from BFP turbine exhaust or the piston ring seals.

8.5.3 Feedwater Heater Diagnostics. Feedwater heater problems generally fall into one of three categories. The first category includes deposits, both tube-side and shell-side. The second category includes internal leaks, both tube leaks and leaks in baffles, seals, and valves. The third category includes external leaks. External leaks, for heaters other than low pressure heaters located in the condenser neck, are normally identified during equipment walkdowns. The focus of this paragraph will therefore be on diagnostics associated with identification of problems falling into the first two 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 differential [6].

Deposits result in increased TTD and decreased temperature rise across the heater. Both TTD and temperature rise are functions of load, however, so 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 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 were to be operating inside the heater.

Leaks in baffles, seals, and valves constitute the remaining area requiring diagnostic procedures. Leaks in HP heater 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. 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.

8.5.4 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.

8.6 BOILER CYCLE

8.6.1 Boiler Cycle Diagnostics. The overall performance of the boiler cycle can be characterized by comparing the unit's boiler efficiency to expected values. Major components in the boiler cycle are the boiler; 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. It is a

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parameter very much influenced by both the operator's actions and equipment operation. As discussed in Section 6, there are currently three methods commonly associated with the determination of boiler efficiency.

These are:

(*a*) heat loss method

(*b*) output-loss method

(*c*) input-output method

The first and third methods are defined in PTC 4.1 for Steam Generating Units [13], while the second method is described in Reference [7].

From a diagnostics standpoint either the heat-loss 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.

Typical values of the various losses for utility boilers are given below [10, 18, 19] and also in Fig. 2-2 of this document:

	% Loss for		
	Coal	Gas	Oil
L_G (dry flue gas)	4	4	4
L_{mf} (moisture in fuel)	2	0	0
L_H (moisture from burning of hydrogen)	3	10	6
L_{UC} (unburned carbon in ash)	2	0	0

8.6.2 Boiler Diagnostics. Boiler diagnostics require the ability to separate the effects of the boiler controls, which operate on a macroscopic level, from the localized, or microscopic, effects. Both the combustion of the fuel and the heat transfer are localized in nature, being affected by local conditions; but the boiler is operated from a macroscopic perspective, the controls dealing 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.

8.6.2.1 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, if they are pertinent to the current evaluation, should be a standard step in a boiler diagnostic procedure.

The characteristics of coal, or its quality, greatly influence plant performance 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, and 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, 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.

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(*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.

8.6.2.2 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).

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 boiler only a small amount of CO (typically between 100 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 is 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 exist 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 150–250 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.

This methodology must take into account the increased requirements for excess air as the load drops off. Reductions in fuel and air flow reduces 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 startup and low loads and as a safeguard against furnace explosions [18].

8.6.2.3 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 ma-

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nipulation 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 ($\pm$20% of the average) and CO (average less than 100 ppm and no one measurement greater than 200 ppm) concentrations and temperatures ($\pm$150°F) across the furnace exit indicate good mixing within the furnace while large variations in these parameters indicate that changes in operation are 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 1500°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 second residence time is required). Fireball location will impact residence time of the coal particles in the furnace also. The higher the fireball, the less the residence time. On tangentially fired boilers, the positioning of the burner tilts will therefore affect the residence time.

8.6.2.4 Boiler Surface Cleanliness. Boiler heat transfer surfaces are soot blown 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 which 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, which 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 which trend the overall heat transfer coefficient, based on such temperature measurements, are sometimes referred to as heat flux monitors.

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 is 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,

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measure specific deposit weights, and to 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.

8.7 CONDENSER CYCLE

8.7.1 Condenser Cycle Diagnostics. The condenser cycle is the heat rejection cycle for the unit. Key components include the condenser, the condenser circulating water pumps, the air ejectors, and the heat sink (cooling tower, river, lake, pond, etc.). Cycle diagnostics should include ambient considerations as they affect the achievable cold water temperature and flow rate of the circulating water.

In condenser cycle diagnostics the turbine cycle performance and the condenser cycle performance should be separated using the heat load on the condenser as the matching boundary condition between the two. This allows the specific assignment in inefficiencies to the correct cycle. The condenser can be in effect separated from the turbine cycle analysis at the turbine-condenser flange. The key to the independent analysis is the measurement of the condenser heat load. The analysis then requires that the condenser be capable of rejecting the heat load resulting from the operation of the turbine at specified condition. The heat load will be a matching boundary condition on the separate analyses of the condenser and the remainder of the turbine cycle [24] (see Section 7, Cycle Interrelationships).

Excess heat load can be due to a loss of turbine efficiency, a feedwater heater dumping directly to condenser, an auxiliary dump open to condenser, heaters in the condenser neck with lagging off or extraction line connections leaking, or LP turbine hood sprays in service.

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.

8.7.2 Condenser Diagnostics. The process of enhancing the performance of a condenser should include the conduct of 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 (noncondensibles) removal rate

Causes of variation of these parameters from target values are not considered a part of the condenser diagnostics, but as part of the heat rejection cycle (condenser cycle) diagnostics.

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; number and location of leaking or plugged tubes; and the amount of air in-leakage.

8.7.2.1 Cleanliness Factor. The condenser cleanliness factor is commonly used in diagnostics as an indicator of thermal fouling of the heat exchange surface. The Heat Exchange Institute's (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. Problems can arise, however, using these for evaluating as-operating condenser performance. The problem arises in that in the HEI methodology, the heat exchange area is assumed to be constant. Any operating influence which 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 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 and 8

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ft/sec. Outside of this region, the accuracy of the provided heat transfer-velocity correlation is suspect.

The target cleanliness factor should also vary as a function of load. The reason is that as the heat load is reduced on the condenser, condensation occurs in the outer portions of the tube bundle. As the condensate film builds, it can eventually block the steam flow, effectively creating a no-heat transfer zone in a large portion of the inner part of the tube bundle. This will be reflected in a reduced cleanliness factor at part load conditions.

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 deposit analyses. The potential for waterside deposits and corrosion problems can be detected through the use of on-line steam/water chemistry monitors.

8.7.2.2 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 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 turbine cycle increasing the cycle 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.

The tube bundle bypass lanes are sized to produce saturated condensate at the design heat load, water temperature, and water flow. If heat load and/or water temperature is appreciably below the design value, condensate subcooling will result.

8.7.2.3 Air Binding. Decreased values of the cleanliness factor may be the result of air binding. Noncondensable 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 ejectors is within the capability of the air ejector, this should be the case. At part load conditions, it may not be possible to completely eliminate air binding.

8.7.2.4 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 fresh water snails within the condenser tubes. Large air pockets can be the result of an 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.

8.7.2.5 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.

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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 thus leaving chemical deposition.

Biological fouling is caused by microorganisms which 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 air binding and macrofouling 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.

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SECTION 9 — PERFORMANCE OPTIMIZATION

9.1 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 Section provides guidance on optimization concepts and techniques. Benefits which may be gained from effective performance optimization include:

(*a*) *Economic* — through reduction of net total production costs;

(*b*) *Technical* — through generally improved equipment conditions, by way of taking corrective actions, both operational and through mechanical work; and

(*c*) *Informational* — through revelation of previously unknown, little understood, or unquantified technical and economic information concerning units being monitored and optimized.

Sections 4, 5, 6, 7 and 8 (Program Planning, Instrumentation, Implementation, Interrelationships, and Diagnostic Techniques) are all particularly relevant to Performance Optimization. It is suggested that they be used as 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.

9.1.1 Definition and Explanation. Optimization is the process through which performance is taken to the highest levels which 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 would be sustained. In order to achieve the most value over the long term, it is necessary to maintain the optimization efforts at a level which will continue to produce the desired results.

9.1.2 Applications. Performance optimization is to be approached from both the operational and maintenance and modification (mechanical) aspects of power production. The two applications involve different concepts and different approaches. They are very much interrelated, however, since both applications strive to achieve the most cost-effective performance possible. There are further interactions between the two 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 trade-off of various operating modes. Operational optimization is normally geared to 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 at times raise capacity, improve availability, and reduce maintenance.

Operating practices to achieve best performance must at all times be governed 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 Section, the term *mechanical* will be used synonymously with mainte-

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nance 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 does operational optimization.

9.1.3 Additional Information and References. In addition to this Section of the guidelines, there are many other information sources which 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 very defined aspects of optimization. Users are referred to the list of references at the end of this Section and to Section 11.

9.2 OPERATIONAL OPTIMIZATION — EMPIRICAL TECHNIQUES

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. Appendix para. A-9.2 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 believed to be generally the most productive and the most powerful in terms of identifying the most efficient operational practices to employ.

Some of the more important performance factors which 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 Appendix para. A-9.3, Empirical Approach to Operational Optimization.

9.2.1 Boiler Cycle Equipment Evaluation. For further specific information on boiler cycle optimization, refer to this Section's reference numbers 2-C; 4-B,D,F,G,H,K,X; 5-F, I,N,R; 6-E, F,J,K,L,S,T,V,W; 7; 8; 9 and 10.

9.2.1.1 Combustion Parameters. Almost any operating actions which can be taken to improve combustion efficiency will have a net positive effect on unit performance. Exceptions include those actions which require more energy than the energy savings that 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 which improve combustion efficiency but which create greater losses elsewhere in the cycle. An example is increasing excess air in order to improve combustion and raise steam temperatures, but which 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 fuel type or types being burned, on the furnace design, and on many other factors. Some of the combustion parameters which can be evaluated empirically for optimization are listed below.

(*a*) *Boiler excess air levels* — seeking generally the lowest levels achievable which create no adverse combustion, safety, or environmental conditions

(*b*) *Variations of primary, secondary, tertiary air* — 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, evaluating various combinations of burners in and out of service to produce 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* — evaluating set-up details such as register positions, damper settings, burner tilts, tip and sprayer plate sizes for oil burners, and other variables for best combustion results. Some of these factors are important for burners out of service as well as for burners actually operating.

(*e*) *Fuel conditions for oil* — evaluating temperature, pressure, and atomizing steam flow (if used) for optimal results under various loads, numbers of burners in service, air flows, and other firing conditions

(*f*) *Fuel conditions for coal* — evaluating coal fineness for best combustion, lowest excess air requirements, cleanest stack conditions, and minimum carryover of unburned carbon into ash.

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Generally the best net efficiency will be achieved at the finest grind size of coal particles which can be produced. However, this is a function of many 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* — evaluating optimal settings of furnace draft, windbox-to-furnace differential, gas recirculating fan, and other air flow and gas flow controls for best net results

(*h*) *Pulverizer performance* — evaluating mill outlet fuel-air temperature, consistent with safe operating practice as specified by the manufacturer, for maximum safe temperature matched to the moisture and volatility of the coal being fired

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 which 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.

9.2.1.2 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. 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 where those best values are, if not at design points.

There are a large number of operating factors which 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.

(*b*) *Direct temperature controls* — such as desuperheating water systems, gas flow control dampers, sootblowing, and firing controls may all be used to maximize steam temperatures produced. Though it may seem obvious, they may also be monitored to ensure that they do not decrease temperatures inadvertently, through improper use at the wrong times.

(*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 trade-off with the opportunity for identifying the optimal combinations through empirical testing.

(*e*) *Gas flow controls* — such as gas recirculating 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 what burner arrangements will produce the optimal temperature results. Particularly at re-

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duced loads, the selection of which burners are in and out of service, and the register and damper settings for burners both in and out, may have a pronounced effect on steam temperatures produced.

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 which 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.

9.2.1.4 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, they help protect downstream equipment (ductwork, precipitator, stack liner) from corrosion, and they affect the particulate removal efficiency of cold side electrostatic precipitators. Therefore, there are a number of reasons why optimal performance of this equipment is needed. Some of the empirical considerations to its optimization are listed below.

(*a*) *Maximizing air heater efficiency* — is one of the more valuable actions that can be taken. This 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, they 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 which allow temperatures of the gas and air streams to be regulated. At least two types of these systems exist — preheating coil arrangements, supplied with steam or hot water from other sources; and bypass duct and damper arrangements, which 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 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 in the process from their heat sources, 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.

Bypass systems are not as flexible in their use as the preheating arrangements are, 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 for these control systems.

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9.2.1.5 Sootblowers. Sootblower operation throughout the boiler can be optimized through their 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 composition 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 sootblowing's advantages of improved availability, efficiency, and others.

As a general approach, sootblowing optimization may be approached through observing its impacts on boiler efficiency, on tube metal, steam and exit gas temperatures, and on 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 so dependent on the interaction of many variables which themselves may be almost continually changing. However, since the potential benefits and certain detriments are so significant, this is an aspect of boiler operation which warrants careful and ongoing attention by plant personnel.

9.2.2 Turbine Cycle Equipment Evaluation. For further specific information on turbine cycle optimization, refer to this Section's reference numbers 1B,H,J; 2-A,B; 3-A,B; 4-A,P,Q,R,T,Y,Z; 5-G,T,U,V; 6-N,O,P,Q; 11; 12; 13; 14; 15; 16; 17; 18; 19; 20; 21 and 29.

9.2.2.1 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 where a unit was originally designed with special boiler and turbine circuitry to accommodate such operation, or 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 if 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 in order to identify the optimal pressure at any given load region. There will be practical operating limits to how low the 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 percent or less of rated pressure, with potentially very large performance advantages resulting. Therefore, there is considerable potential for empirical exploration of

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sliding main steam pressure. It is not limited to very small incremental adjustments. Note that the amount of pressure reduction which 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 which 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 for 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 being 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.

9.2.2.2 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.

9.2.2.3 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 which 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.

9.2.2.4 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, to regulate levels at the optimal points, to dump high level drainage at the right times and conditions, and to bypass or to 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 perform-

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ance 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 slowly dropping condensate level in the heater. 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. Level may then be raised to a point of several inches above that where the "blow through" began. If level is set too close to the "blow through" point, the heater is potentially likely to 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 "blow through" point, 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, with drainage levels established and stable in all heaters, with drainage cascading downward from upper to lower heaters, and with 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 their 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.

9.2.2.5 Condensers. The condenser represents the major single point of unavoidable energy loss from the cycle, as well as potentially one of the 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 backpressure (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 possibly could 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. 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

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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 neutralize it, 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 step 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 their 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, then 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 where the economic gain of condenser improvement is less than the energy cost of operating the additional devices.

Operation of air removal equipment itself 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, to minimizing any restrictions on the pump discharge side, and to 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.

(*d*) *Optimized Circulating Water Flow.* Circulating water flow itself can be optimized through numbers 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 less than all pumps and not detract from condenser performance. This operating mode can be evaluated empirically, as well as through the use of computer models or through manual calculations. There are other considerations of shutting down circulating water pumps besides simply the condenser performance. These include pump motor reliability, reliability of the condenser and

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the unit itself, various tube corrosion and other factors, vibration characteristics of condenser tubes, and others. 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 which enters either through leakage or entrainment. To the extent that these priming systems are kept in good condition and that waterbox levels are full at all times, this will help to optimize overall condenser performance.

9.2.2.6 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 cost effective to do so. A limit on this would be where 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.

9.2.2.7 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 of 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 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.

(*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.

9.2.2.8 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 which 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 and temperature possible which 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

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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.

9.2.2.9 Drain, Recirculating, 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, these systems 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 of course, manual, hands-on surveillance. Simply monitoring these systems for unintended flows does nothing to correct them. The information needs to be interpreted and then 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, it 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, it is included here because operators are generally the ones to detect system problems first, and therefore will need to initiate corrective actions as required.

9.2.3 Auxiliary Equipment Evaluation. For further specific information on auxiliary equipment optimization, refer to this Section's reference numbers 4-C, S; 5-J; 6-U; and 22.

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.

9.2.3.1 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

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pumps, vacuum pumps, air compressors, and other major and minor auxiliary apparatus.

The efficiency impact 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 which will be sufficient for the output needed. In other cases it may be advantageous to operate more than are needed, having some or all of them 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, which 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.

9.2.3.2 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 should be run that are required to satisfy the process needs. 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 (c) and (d) below.

(*b*) *Performance consequences of shut-down auxiliaries* — 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, condenser performance may not degrade from the partial loss of flow, while under other 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.

9.2.3.3 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.

9.2.3.4 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.

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9.3 MECHANICAL OPTIMIZATION — GENERAL METHODOLOGY

For further information on mechanical optimization, refer to this Section's reference numbers, 1-G,I; 3-C,D,E,F; 5-S; 6-D,R; and 23.

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 para. 9.4.

9.3.1 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.

9.3.1.1 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 information which is monitoring-based 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 which occur after the initial assessment.

9.3.1.2 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 which 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 which are participants in this data base, information is available either in-house or through NERC which 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 of much value in identifying and prioritizing availability detractors on the unit.

9.3.1.3 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 it driven more by availability, by efficiency, or by 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 which may either need to be modified or else 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.

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9.3.2 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 non-economic 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 to introduce the general process to users.

9.3.2.1 Optimization Cost. For any repair or engineering solution which 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 which 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.

9.3.2.2 Performance Value. The expected performance changes that would occur with each correction option, and the probable economic value of these changes, represents the other side of the optimization equation. For actions targeting efficiency problems, expected changes may be expressed in terms of heat rate, operation or performance factor, fuel consumption, or other measure which can most readily be converted to economic value. For availability problems, expected changes may be expressed as reduced number or duration of outages, as lesser number of load reductions, as change in equivalent availability or forced outage rate or other 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 change 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 together 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 which 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 applied as an approximator when more sophisticated, model-based calculations are not available.

(*b*) *Availability Changes.* The process to convert changes in availability is quite different from 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 per MW-hr. Expected operating costs for available re-

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placement 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 approximator only. Other factors which are not represented, but which 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.

9.3.2.3 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 which are feasible and which 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 very 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.

9.3.2.4 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.

9.3.3 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.

9.3.3.1 Single Versus Multiple Unit Factors. There may exist a range of different performance problems with various correction options 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 those corrections on those units which will yield the highest net worth and the best cost-benefit ratios.

9.3.3.2 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 con-

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ditions, 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.

9.3.3.3 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 which 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 because of the technical ramifications of this level of work. Close attention needs to be given to the expected performance changes to ascertain that 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 has been 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 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.

9.4 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. Certain prerequisites have to be (or are very helpful if they are) 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, even including the impaired and in need of major work. Of course under very impaired equipment conditions, a unit will not be able to produce very good performance even with the best of operator attention, but performance improvements can be made nonetheless.

Operational optimization also serves as a valuable source of information to the mechanical optimization efforts. Through the operational work, mechanical problems can often be detected and brought to attention which otherwise may have been too subtle to have been recognized. 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 the operational work to implement and maintain. It requires (or benefits from) some analytical tools which may be fairly complex. Clearly it 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 requiring 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 with major or recurrent problems.

In the ongoing optimization effort, it is best to not attempt to separate the operational and mechanical areas from each other. Rather, they can be viewed as highly integrated aspects of unit performance, both of which 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 needs to be both to the results of performance optimization as well as to the team approach to achieving them.

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APPENDIX A-9 OPERATIONAL OPTIMIZATION — GENERAL METHODOLOGY

The following information describes general approaches and methods for operational optimization. There are many different ways that it can be done, with each having its own advantages and disadvantages relative to the other methods. This Appendix 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.

For further general information on operational optimization, refer to this Section's reference numbers 1-A,C,D,E,F; 4-E,I,J,L,M,N,O,U,V,W; 5-A,B,C,D,E,H,K,L,M,O,P,Q; 6-A,B,C,G,H,I,M; 24; 25; 26; 27; and 28.

A-9.1 OPERATIONAL APPROACHES

A general approach to operational optimization is described here for the document's users. 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 include the following.

A-9.1.1 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 it 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.

A-9.1.2 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 may be built the optimization efforts. It is most important that these people 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 people is not applied in the process, it is likely that results will be limited at best.

A-9.1.3 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 in order 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 imply that operators must be relegated to the role of robots; to the contrary, their judgment and active involvement in optimized operation is of paramount importance to its success. However, the reasonable uniformity or standardization of operating practices is needed to achieve the fullest benefits of optimization.

A-9.1.4 Optimal Operating Modes. This is another fundamental aspect of operating optimization, which involves the setting of operating values at those levels producing the "best" results. In optimization, this equates to the highest performance levels which can be cost-effectively and feasibly achieved, within the constraints of safety, reliability, and environmental soundness. Normally the objec-

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tive is to maximize operating efficiency; however, there are also considerations of equipment preservation and reliability which 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.

A-9.1.5 Operator Controllable Parameters Programs. Of extremely high value in operational optimization work are computer-based operator controllable parameters programs. Typically these programs continually receive various unit data, perform calculations, provide information displays to operators, and produce summary reports for operator and management use.

A-9.1.6 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 to the application stages of optimization work.

A-9.2 Methods of Determining Optimal Operating Modes

There are several methods which 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.

A-9.2.1 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 which 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. All in all, the empirical testing approach may generally be the most fruitful of all methods in revealing the truly "best" combinations of operating techniques and values.

A-9.2.2 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 "comfort factor" 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 which are ten percent 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.

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A-9.2.3 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.

A-9.2.4 Manual Engineering Methods. Manual calculation methods are also useful in some aspects of determining optimal operating modes. These include calculator 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 may be of value to include them as a part of the overall methodology.

A-9.3 Empirical Approach to Operational Optimization

The empirical approach for this work, as stated above, is recommended as being potentially 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 of 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, on operating experience, on judgment, and common sense. They are intended to protect the equipment and people during the optimization process which may enter previously unexplored oper-

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ating 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.

A-9.3.1 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. Measures are then taken of the results of these controlled changes, in order to establish cause-and-effect relationships. Iterative changes are continued for the parameter until the optimal operating levels and modes are determined for it.

In concept the process is no more complex than stated above. In practice it is not complex, but it 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 Section.

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 which 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 is needed of the effects measured to determine where is 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 possibly even 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 from it 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.

A-9.3.2 Sequence of Equipment Evaluation. There are many operating parameters which could potentially be evaluated for optimization. These range in magnitude of impact from almost negligible (such as only a few Btu/kWhr effect on heat rate) to major (conceivably hundreds of Btu/kWhr 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 be first handling those parameters which 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 have to

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repeat work already done which 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 which 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.

A-9.3.3 Test Conditions. It is important to establish test conditions which 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.

A-9.3.4 Assurance of (Confidence in) Results. The empirical testing process is used to arrive at information on operating modes and values which 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 repeating 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.

A-9.3.5 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, 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, for each parameter studied, the patterns to be followed in normal operation through the full load range. 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 com-

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bustion 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, from careful evaluation, and from consideration of other important factors such as design data, and safety, reliability, and environmental aspects.

A-9.3.6 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 which must be followed under any conditions. The need for operator understanding and judgment as to 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.

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SECTION 10 — INCREMENTAL HEAT RATE

10.1 INTRODUCTION

An economic choice of which steam boiler or which turbine-generator should provide the next increment of power to a change in load demand should be based on incremental costs. How much of a change in fuel input will be required for an incremental change in power output, and what is the cost of that quantity of fuel? A performance monitoring program can provide the answers to these questions as well as optimizing costs for independent power producers and electrical interconnection sales by improving heat rate.

10.2 INPUT/OUTPUT RELATIONSHIPS

Input/output relationships are the basics for building incremental cost data. Input generally expressed in millions of Btu's per hour, and output, expressed typically in megawatts for power plants, do not vary uniformly over the entire load range. The relationship may be determined from design or guarantee data supplied by equipment manufacturers or from test data. Figure 10.1 illustrates input/output relationships for two generating units.

10.3 INCREMENTAL COSTS

Incremental costs are calculated from the input/output relationships developed by factoring in fuel costs. Table 10.1 illustrates incremental rates for the two generating units shown previously in Fig. 10.1. It 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.

10.4 INCREMENTAL HEAT RATE BY TEST

(*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 determine economically which unit or units to adjust as system load varies.

(*b*) The development of incremental heat rate is essentially a procedure which utilizes the same performance monitoring data that is collected for the various other purposes described elsewhere in this document.

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(*b*) The development of incremental heat rate is essentially a procedure which utilizes the same performance monitoring data that is collected for the various other purposes described elsewhere in this document.

In establishing test loads and procedures (for all uses) it is necessary to understand the unique nature of incremental heat rates, characteristically:

(*1*) It is the rate of change or derivative of the firing rate that determines the incremental value and therefore the determination of shape or curvature is more important than is absolute level.

(*2*) Most dispatch systems require monotonically increasing (positive) rates which may require compromise of the true physical shape determined by test such as caused by a partial arc turbine control valve loop (which is negative). This is illustrated in Fig. 10.2.

(*3*) Many dispatch systems can accommodate multiple curve segments which can be situated so as to coincide with control valve intercepts or "valve points" and auxiliaries such as pulverizers.

(*4*) The possibility of multiple curve segments emphasizes a generic characteristic of steam cycles that input is more firmly related to steam flow than to generation. Thus, turbine control valve position (which establishes system flow) should be measured and serve as an intermediate reference (rather than generation).

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TABLE 10.1 PROCEDURE FOR DETERMINING LOAD ALLOCATION FOR TWO UNITS WHOSE CURVES ARE SHOWN IN FIGURE 10.1.

Load MW	Million Btu/hr	Dollars/hr[1]	Charge for load increment dollars/hr	Incremental rate, dollars/MWhr
		UNIT 1		
0	400	1,000.00	—	—
50	700	1,750.00	750.00	15.00
100	1,200	3,000.00	1,250.00	25.00
150	1,800	4,500.00	1,500.00	30.00
200	2,600	6,500.00	2,000.00	40.00
250	4,000	10,000.00	3,500.00	70.00
		UNIT 2		
0	800	2,000.00	—	—
50	1,000	2,500.00	500.00	10.00
100	1,200	3,000.00	500.00	10.00
150	1,600	4,000.00	1,000.00	20.00
200	2,000	5,000.00	1,000.00	20.00
250	2,600	6,500.00	1,500.00	30.00

NOTE:
(1) At $2.50 per MBtu/hr.

(Based on Table 1, p. 42, of "Power System Operation," by Miller, R. H., McGraw-Hill, 1983, Published by permission of McGraw-Hill.)

(*5*) The curvature of the heat input or firing rate relative to flow or load determines the slope (and therefore the unit loading rate) of the incremental curve.

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.

(*c*) Once validated test data has been acquired, the mechanics of incremental heat rate development are:

(*1*) Plot the firing rate versus flow or load relationship as determined by test and illustrated in Fig. 10.2.

(*2*) Curve fit the data to one or more polynomials acceptable to the applicable dispatch algorithm.

(*3*) Take the derivative(s) as a function of the polynomial coefficients.

(*d*) In the absence of accurate heat input measurement capability, which is often the case for coal-fired units, the task of unit characterization requires a more complex procedure.

The steps required as illustrated in Fig. 10.3 are:

(*1*) Develop a net turbine cycle heat rate which subtracts from gross generation all auxiliary power consumption, even that used to support boiler operation.

(*2*) Develop a boiler efficiency as defined in Section 6.

(*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.

10.5 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 which 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

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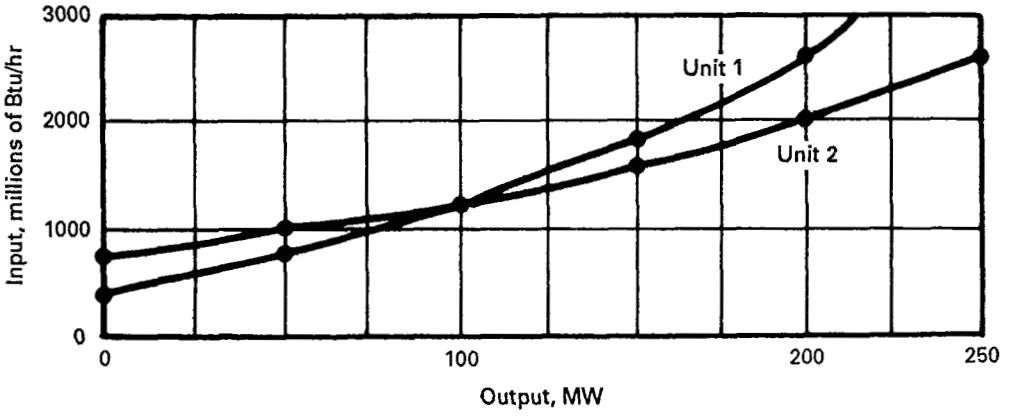

FIG. 10-1 INPUT/OUTPUT CURVES FOR THE TWO TYPICAL THERMAL UNITS

(Based on Figure 20, page 40, of "Power System Operation," by R. H. Miller; McGraw-Hill, 1983, Published by permission of McGraw-Hill)

GENERAL NOTE: Though 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.

(*4*) excess air
(*5*) exit gas temperature
(*6*) moisture in the fuel

Continuous or periodic updating of the incremental costs are desirable. A report CS/EL 4415 Vols. 1-8, "Power Plant Performance Instrumentation Project," available from the Electric Power Research Institute (EPRI Project 1681/2153) more fully describes the methodology for accomplishing these updates.

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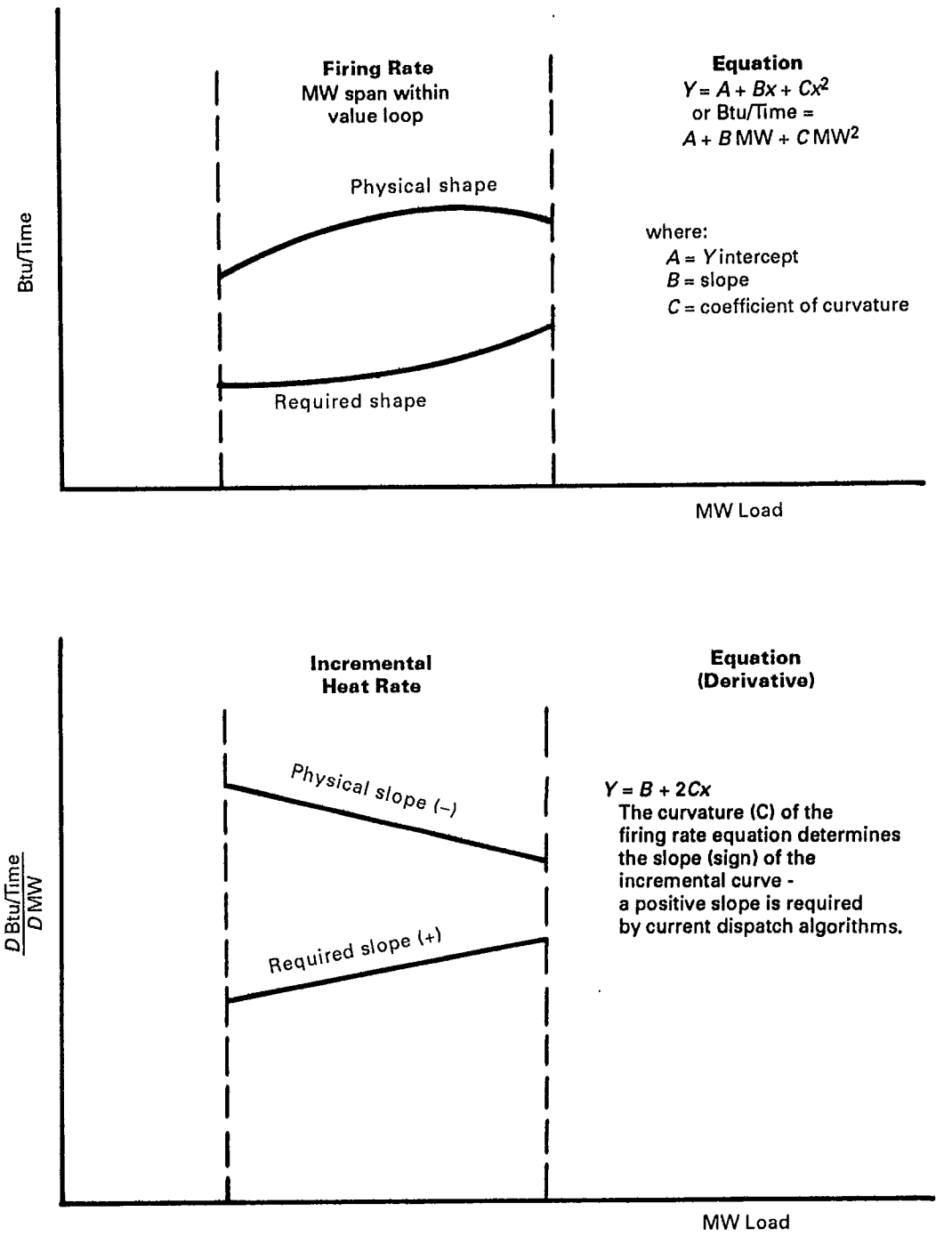

FIG. 10-2 INCREMENTAL CURVE SHAPE

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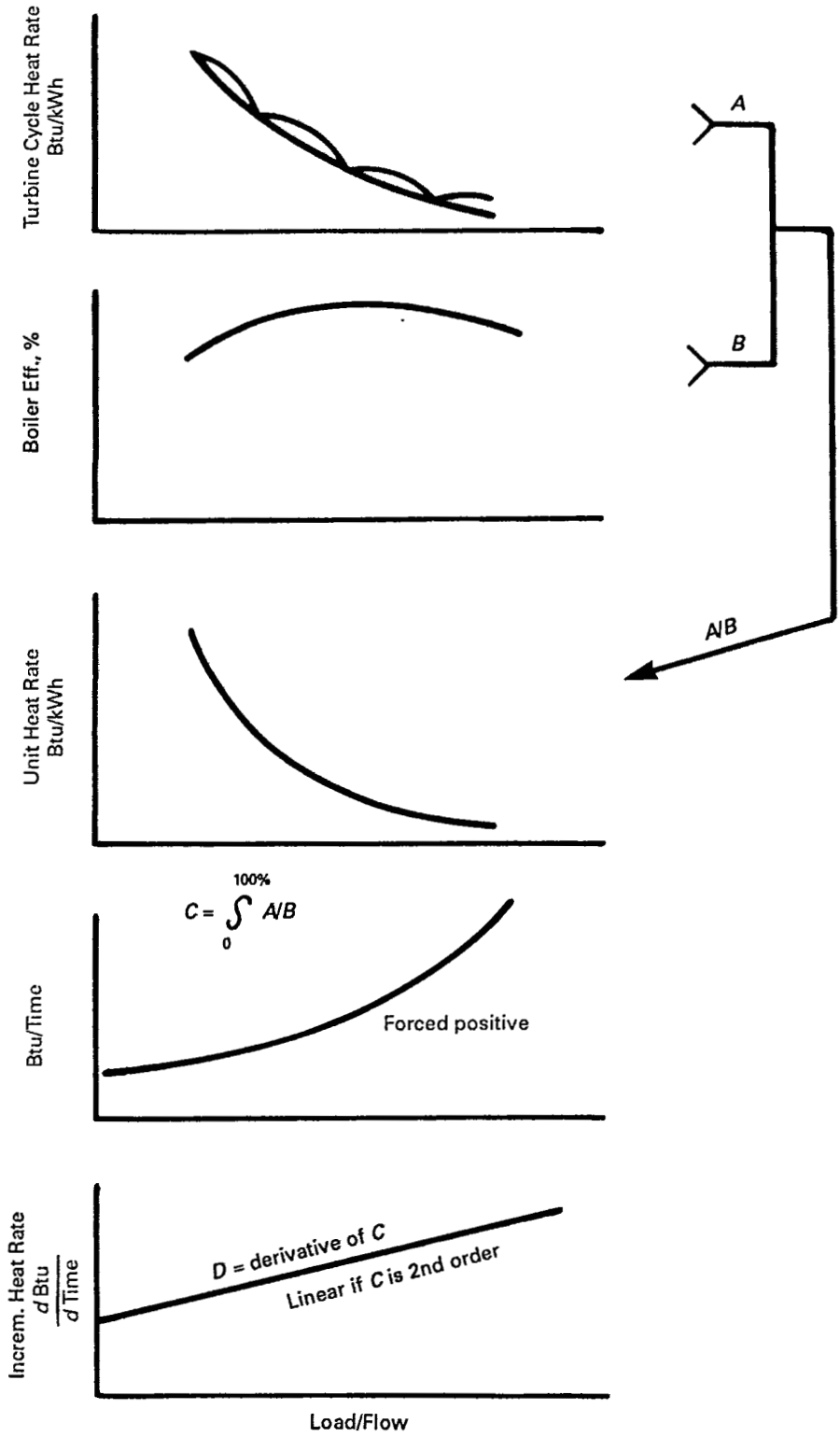

FIG. 10-3 ILLUSTRATION OF DEVELOPMENT OF INCREMENTAL HEAT RATE INFORMATION FROM BASIC PLANT MEASUREMENTS

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SI UNITS

This ASME PTC PM-1993 document is based on U.S. Customary (ft-lb) units of measurement which are to be regarded as the standard. This supplement is provided as a convenience to the user and contains SI conversion factors for units contained in this document.

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LIST OF SI UNITS FOR USE WITH ASME PTC PM-1993[1]

Quantity	Unit	Symbol	Other Units or Limitations
Space and Time			
plane angle	radian	rad	degree (decimalized)
length	meter	m	
area	square meter	m^2	
volume	cubic meter	m^3	liter (L) for liquid only (use without prefix other than in milliliter, mL)
time	second	s	minute (min), hour (h), day (d), week, and year
Periodic and Related Phenomena			
frequency	hertz	Hz	revolutions per second (r/s)
rotational frequency	revolutions per second	s^{-1}	revolutions per minute (r/m)
Mechanics			
mass	kilogram	kg	
density	kilogram per cubic meter	kg/m^3	
moment of inertia	kilogram · $meter^2$	kg · m^2	
force	newton	N	
moment of force (torque)	newton-meter	N · m	
pressure and stress	pascal	Pa	(pascal=newton per square meter)
energy, work	joule	J	kilowatt-hour (kW · h)
power	watt	W	
impact strength	joule	J	
section modulus	$meter^3$	m^3	
moment of section (second moment of area)	$meter^4$	m^4	
fracture toughness	Pa · $\sqrt{m}$		
Heat			
temperature — thermodynamic [Note (2)]	kelvin	K	degree Celsius (°C)
temperature — other than thermodynamic	degree Celsius	°C	kelvin (K)
linear expansion coefficient	meter per meter-kelvin	K^{-1}	$°C^{-1}$
quantity of heat	joule	J	
heat flow rate	watt	W	
thermal conductivity	watt per meter-kelvin	W/(m · K)	W/(m · °C)
thermal diffusivity	square meter per second	m^2/s	
specific heat capacity	joule per kilogram-kelvin	J/(kg · K)	J/(kg · °C)
Electricity and Magnetism			
electric current	ampere	A	
electric potential	volt	V	
current density	ampere per $meter^2$	A/m^2	
magnetic field strength	ampere per meter	A/m	

NOTES:
(1) Conversion factors between SI units and U.S. customary are given in SI-1, "ASME Orientation and Guide for Use of SI (Metric) Units," and ASTM E 380.
(2) Preferred use for temperature and temperature interval is degrees Celsius (°C), except for thermodynamic and cryogenic work where kelvins may be more suitable. For temperature interval, 1 K = 1°C exactly.

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COMMONLY USED CONVERSION FACTORS
(For Others See ASTM E 380) (See [Note (1)])

Quantity	To Convert From	To	Multiply by [Note (2)]	
plane angle	degree	rad	1.745 329	E–02
length	in	m	2.54*	E–02
	ft	m	3.048*	E–01
	yd	m	9.144*	E–01
area	in^2	m^2	6.451 6*	E–04
	ft^2	m^2	9.290 304*	E–02
	yd^2	m^2	8.361 274	E–01
volume	in^3	m^3	1.638 706	E–05
	ft^3	m^3	2.831 685	E–02
	US gallon	m^3	3.785 412	E–03
	Imperial gallon	m^3	4.546 09	E–03
	liter	m^3	1.0*	E–03
mass	lbm	kg	4.535 924	E–01
	ton (metric) (mass)	kg	1.000 00*	E+03
	ton (short 2000 lbm)	kg	9.071 847	E+02
force	kgf	N	9.806 65*	E+00
	lbf	N	4.448 222	E+00
bending, torque	kgf · m	N · m	9.806 65*	E+00
	lbf · in	N · m	1.129 848	E–01
	lbf · ft	N · m	1.355 818	E+00
pressure, stress	kgf/m^2	Pa	9.806 65*	E+00
	lbf/ft^2	Pa	4.788 026	E+01
	lbf/in^2 (psi)	Pa	6.894 757	E+03
	$kips/in^2$	Pa	6.894 757	E+06
	bar	Pa.	1.0*	E+05
energy, work	Btu (IT) [Note (3)]	J	1.055 056	E+03
	ft · lbf	J	1.355 818	E+00
power	hp (550 ft · lbf/s)	W	7.456 999	E+02
fracture toughness	ksi $\sqrt{in}$	Pa · $\sqrt{m}$	1.098 843	E+06
temperature	°C	K	$t_K = t_C + 273.15$	
	°F	K	$t_K = (t_F + 459.67)/1.8$	
	°F	°C	$t_C = (t_F - 32)/1.8$	
temperature interval	°C	K	1.0*	E+00
	°F	K or °C	5.555 555	E–01

NOTES:
(1) Care should be taken when converting formulas or equations that contain constant terms or factors. The value of these terms must be understood and may also require conversion.
(2)(a) Relationships that are exact in terms of the base units are followed by a single asterisk.
(b) The factors are written as a number greater than 1 and less than 10 with 6 or less decimal places. The number is followed by the letter E (for exponent), a plus or minus symbol, and two digits which indicate the power of 10 by which the number must be multiplied to obtain the correct value. For example: 3.523 907 E–02 is $3.523\ 907 \times 10^{-2}$ or 0.035 239 07.
(3) International Table

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COMPLETE LISTING OF ASME PERFORMANCE TEST CODES

PTC 1	— General Instructions	1991
PTC 2	— Definitions and Values	1980 (R1985)
PTC 3.1	— Diesel and Burner Fuels	1958 (R1992)
PTC 3.2	— Coal and Coke	1990
PTC 3.3	— Gaseous Fuels	1974 (R1992)
PTC 4.1	— Steam-Generating Units (With 1968 and 1969 Addenda)	1964 (R1991)
	Diagram for Testing of a Steam Generator, Fig. 1 (Pad of 100)	
	Heat Balance of a Steam Generator, Fig. 2 (Pad of 100)	
PTC 4.1a	— ASME Test Form for Abbreviated Efficiency Test — Summary Sheet (Pad of 100)	1964
PTC 4.1b	— ASME Test for Abbreviated Efficiency Test — Calculation Sheet (Pad of 100)	1964
PTC 4.2	— Coal Pulverizers	1969 (R1991)
PTC 4.3	— Air Heaters	1968 (R1991)
PTC 4.4	— Gas Turbine Heat Recovery Steam Generators	1981 (R1992)
PTC 5	— Reciprocating Steam Engines	1949
PTC 6	— Steam Turbines	1976 (R1991)
PTC 6A	— Appendix A to Test Code for Steam Turbines (With 1958 Addenda)	1982
PTC 6 Report	— Guidance for Evaluation of Measurement Uncertainty in Performance Tests of Steam Turbines	1985 (R1991)
PTC 6S Report	— Procedures for Routine Performance Tests of Steam Turbines	1988
PTC 6.1	— Interim Test Code for an Alternative Procedure for Testing Steam Turbines	1984
	PTC 6 on Steam Turbines— Interpretations 1977—1983	
PTC 7	— Reciprocating Steam-Driven Displacement Pumps	1949 (R1969)
PTC 7.1	— Displacement Pumps	1962 (R1969)
PTC 8.2	— Centrifugal Pumps	1990

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PTC 9 — Displacement Compressors, Vacuum Pumps and Blowers (With 1972 Errata) 1970 (R1985)
PTC 10 — Compressors and Exhausters 1965 (R1986)
PTC 11 — Fans 1984 (R1990)
PTC 12.1 — Closed Feedwater Heaters 1978 (R1987)
PTC 12.2 — Steam-Condensing Apparatus 1983
PTC 12.3 — Deaerators 1977 (R1990)
PTC 12.4 — Moisture Separator Reheaters 1992
PTC 14 — Evaporating Apparatus 1970 (R1991)
PTC 16 — Gas Producers and Continuous Gas Generators 1958 (R1991)
PTC 17 — Reciprocating Internal-Combustion Engines 1973 (R1991)
PTC 18 — Hydraulic Turbines 1992
PTC 18.1 — Pumping Mode of Pump/Turbines 1978 (R1984)
PTC 19.1 — Measurement Uncertainty 1985
PTC 19.2 — Pressure Measurement 1987
PTC 19.3 — Temperature Measurement 1974 (R1986)
PTC 19.5 — Application, Part II of Fluid Meters: Interim Supplement on Instruments and Apparatus 1972
PTC 19.5.1 — Weighing Scales 1964
PTC 19.6 — Electrical Measurements in Power Circuits 1955
PTC 19.7 — Measurement of Shaft Power 1980
PTC 19 8 — Measurement of Indicated Horsepower 1970 (R1985)
PTC 19.10 — Flue and Exhaust Gas Analyses 1981
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The Philosophy of Power Test Codes and Their Development

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PERFORMANCE TEST CODES

A complete list of all Performance Test Codes appears at the end of this book.

While providing for exhaustive tests, these Codes are so drawn that selected parts may be used for tests of limited scope.

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