Heat Rate Improvement Reference Manual

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REPORT SUMMARY

Performance optimization of fossil power plants is a high priority within the electric utilities in the new competitive environment. This manual can help utility engineers establish a heat rate improvement program.

Background

A 1983 utility survey covering 129 fossil generation units concluded that a mean heat-rate improvement of more than 400 Btu/kWh can be achieved. For a typical 500-MW fossil-fueled power plant, a 400 Btu/kWh reduction in heat rate translates into $4 million in annual fuel savings. In May 1986, EPRI published the Heat-Rate Improvement Guidelines for Existing Fossil Plants (EPRI report CS-4554). Following these guidelines enabled utilities to implement plant heat-rate improvement programs. In recent years the electric power generation community has focused on environmental control need, but the opportunity exists in the new competitive marketplace to optimize thermal efficiency while controling boiler emissions. However, many utilities have downsized and lack experienced staff in the area of performance technology. The goal for this project was to distill existing documents on heat rate improvement into a comprehensive manual for both training and application.

Objectives

To produce a manual that can serve as a training tool and reference book for utility heat rate engineers.

Approach

The project team first reviewed existing industry techniques and experiences relevant to heat rate reduction. They conducted a literature search and sought input from the Heat Rate Interest Group (HRIG). HRIG members reviewed and commented on the draft manual. The project team finalized the manual after incorporating all the comments.

Results

This manual is designed to be used by electric utilities as a training tool and reference book for heat rate engineers. The manual addresses the following topics:

- Heat rate basics

- Fossil steam station components
- Elements of a thermal performance monitoring program
- Instrumentation requirements for heat rate monitoring
- Cycle isolation
- Heat rate improvement programs

EPRI Perspective

This reference manual supplements the EPRI Heat Rate Improvement Guidelines published in May of 1986. It includes detailed heat rate monitoring, accounting, and calculation methodology not covered in the guidelines. In addition, the manual highlights the results of heat rate improvement guideline demonstration projects conducted at five member utility plants. The manual is an essential training tool and reference for utility heat rate engineers

Interest Categories

Fossil steam plant performance optimization
Fossil steam plant O&M cost reduction
Air emission control
Air toxics measurement and control

Keywords

Heat rate
Thermal efficiency
Optimization
Performance evaluation
Instrumentation
Boiler/turbine improvements

ABSTRACT

Performance optimization of fossil power plants is a high priority within the electric utilities in the new competitive environment. On the other hand, many utilities have downsized and lack experienced staff in the area of performance technology.

The objective of this project was to produce a manual to be used by electric utilities as a training tool and reference book for heat rate engineers. This document includes methodology for heat rate accounting, instrumentation requirements for heat rate monitoring, and key heat rate parameter measurement.

This manual used EPRI CS-4554 *Heat Rate Improvement Guideline* as a basis for development of the program. This manual includes:

- Illustration of thermal kits use
- Illustration of heat rate calculation using the I/O and O/L method
- Illustration of heat rate calculation for best heat rate, best achievable heat rate, and actual heat rate at various loads
- Illustration of the impact of component efficiency
- Illustration of the impact of controllable parameters such as LOI, excess air, spray flow, auxiliary power and cycle isolation
- Illustration of the impact of component modification to heat rate and cost benefit analysis
- Illustration of the impact of cycle modification to heat rate and cost benefit analysis
- Illustration of the impact of maintenance practice modification to heat rate and cost benefit analysis
- Performance test recommendations for key components
- Instrumentation requirements for heat rate monitoring purposes
- Justification for upgrading instrumentation for heat rate monitoring

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INTRODUCTION

The purpose of this manual is to define the performance standards necessary to successfully manage a heat rate improvement program. The information contained in this manual can enable management, staff and technical individuals to make their company more competitive and successful in the future production of electricity.

This report:

- Reveals the benefits of a performance program by explaining the philosophy, purpose and savings from a high quality performance monitoring program.
- Explains how to implement and integrate a successful performance monitoring program.
- Shows that improving efficiency of a units operation should be a continuing objective.
- Equips the performance engineer with a tool to reduce fuel costs and enhance the reliability and availability of their product.

Importance

A successful thermal performance program is essential to competitive success in a deregulated environment. Rising fuel costs and increased environmental regulations have directed many utilities to improve the performance of their generating stations. A more accurate knowledge of unit heat rates can improve economic dispatching costs and ensure that profits are maintained on a daily basis.

Complexity

A thermal performance program based on performance monitoring and heat rate improvement is dynamic and complex. To improve efficiency, the engineer must know the heat input, mass of fuel, the fuel analysis and the kW rating generation to determine actual heat rate. After the actual heat rate is calculated and understood, losses must be identified and understood. Good communication and teamwork between the engineer and staff within the station is essential to success. Dedication, initiative, and diligence at the station is also required.

Defined Actions

A thermal performance program is actually the development of performance parameters which characterize a unit's operation. Appropriate performance parameters can enable the performance engineer to either immediately correct performance or estimate when it would be cost effective to make corrections. Performance data and benefits have been defined by the action or decisions which the data can affect.

These actions are:

- Improve unit operation
- Predictive maintenance
- Comparison of actual to expected performance
- Baselining and tracking of units performance
- Improved economic dispatch of units
- Reduce uncertainty in actual costs for better MW sales.

Value

The performance parameters measure how well the unit is doing its job in producing electricity. Decisions should not necessarily be made only to improve thermodynamic efficiency, but rather to improve a company's overall performance.

Measurement

As with any program, a performance program should be measured. The following is a useful matrix to help measure the activities surrounding a successful program.

Table 1-1
Performance Program Matrix

Activity	Yes	No	Date to be Completed
1. Is the efficiency factor important?			
2. Does the efficiency factor show an improving trend?			
3. Is there an efficiency factor goal for the station or unit?			
4. Is fuel being measured/sampled accurately?			
5. Is generation measured accurately?			
6. Does management or station performance engineer review quarterly a performance report?			

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HEAT RATE PRIMER

Although the specific objectives of a heat rate improvement program will vary from utility-to-utility and from plant-to-plant, a necessary prerequisite to formalizing program activities is to obtain some understanding of the current conditions of a unit. This is essential to ensure the cost-effective expenditure of the limited resources available to improve unit heat rate. Although some problem areas may be known at a unit, many units have substantial heat rate degradations which are unknown. The purpose of the activities described below is to outline an approach which will begin to aid utility personnel in characterizing the performance of a unit in order to establish the amount of improvement which can be made in heat rate performance.

Heat Rate Definitions

Introduction

Heat rate is defined in units of Btu/kWh (kJ/kWh) and is simply the amount of heat input into a system divided by the amount of power generated by a system. While the definition seems simple, the number of ways in which heat rate can be computed are numerous. This section will describe the standard heat rate definitions currently being used by utilities and the purpose and usefulness of each. The definitions provided are those commonly used by utilities to report heat rate for management information purposes.

It is recognized that utilities use different fuels - coal, natural gas, oil, wood waste, refuse, bagasse, etc. - to produce electricity or steam for consumption. However, this discussion will center on the more traditional fuels - coal, oil, and natural gas. Special emphasis will be given to the calculation of the heat rate of coal-fired units due to the complexities of accurately measuring fuel flow and heating values. Regardless of the fuel burned, the principles involved in computing a heat rate are identical. However, the accuracy of measuring both fuel usage and the heating value of oil and natural gas are less uncertain than for other types of fuel.

As-Designed versus As-Built Heat Rate

To understand and determine the performance of a unit properly it is necessary to benchmark not only its design heat rate but also its best achievable heat rate. This can be obtained from the results of acceptance tests performed at initial unit start-up. A unit's design net heat rate curve can easily be generated using the information supplied by turbine and boiler vendors along with design auxiliary consumption.

The next step is to recognize that the “as-designed” heat rate is not usually equal to the “as-built” heat rate. The problem that exists is that the design curve is usually not representative of what the unit is actually capable of. Due to differences between "as-designed" and "as-built" renders this gap between its predicted performance and the unit's best achievable performance. Also this gap will continue to change if the design and best achievable performance curves are not maintained for any modification made to the unit.

Differences between "as-designed" and "as-built" conditions include:

- Extraction line pressure drops
- Reheater pressure drops
- Turbine design deficiencies
  - HP, IP, and LP efficiencies
- Boiler design deficiencies
  - Superheat and Reheat spray flow
  - Excess air requirement
  - Preheater efficiency
- Fuel quality

These factors are usually calculated during the design phase, but the actual installed or “as-built” valves often differ due to conservative assumption.

There are numerous other factors that exist which affect unit heat rate which are not accounted for in the design heat rate curve that produces the gap between “as-designed” and best achievable heat rate.

Factors that exist which are not usually included in the “as-designed” heat rate curve are:

- Heater vents
- Pump seal and leakoff flows
- Steam traps
- Plant auxiliary steam usage
- Cycle leakages
- Soot blowers steam usage
- Coal handling power consumption
- Steam coils
- Different fuel characteristics (grindability, HHV, moisture, ash)

These factors have to be quantified and a benchmark established in order for a performance program to have a firm foundation. Only by knowing how well a unit can perform will the performance person be able to make intelligent decisions regarding performance improvement.

Conditions which affect unit performance which are not controllable are air inlet temperature, cooling water temperature, and fuel quality. The expected design net heat rate and best achievable net heat rate have to be adjusted for these conditions. Once adjusted for these uncontrollable conditions then a comparison can be made between actual and expected (design and/or best achievable) heat rates.

Heat Rate Changes

The performance of a unit will begin to decline as the unit begins to age. A good performance program will be able to identify these losses as accounted-for losses. Unit overhauls can bring the unit back to the best achievable heat rate, but some of the lost performance may not be economically recoverable. It should be pointed out that the best achievable heat rate is a realistic value because it was once attained and can be achieved again if cost justified. To exceed the best achievable heat rate equipment modification and/or enhanced operating practices will have to occur. Many times during the course of a unit's operating life it is modified in a manner which directly affects heat rate performance.

Some typical modifications include: conversion from constant speed to variable speed fans, addition of cooling towers, changes in fuel used, and the retrofit of electrostatic precipitators or flue gas desulphurization. Other modifications could include: addition or removal of heat transfer surface or the replacement of heat transfer surfaces with more efficient design, replacement of turbine nozzles or blading with designs which may improve unit performance, replacement of feedwater heaters, or replacement of the condenser tubing with a more efficient design.

Whenever a modification is made on a unit a test should be performed to determine the actual performance of the equipment in question. Then the design heat rate curve should be adjusted based on design values and the best achievable heat rate adjusted based on test results. The adjustment to the design and best achievable curves can be made manually or through the use of a computer performance code. To perform these calculations manually, the performance person needs to evaluate each modification independently to precisely determine the effect it has on the operating parameters of the unit. The performance person then uses available information, such as the plant thermal kit or heat balances codes, to estimate the effect that the change will have on unit heat rate. Any adjustment to benchmark curves should be well documented for future reference.

Definitions for various levels of performance standards are shown in the Figure 2-1.

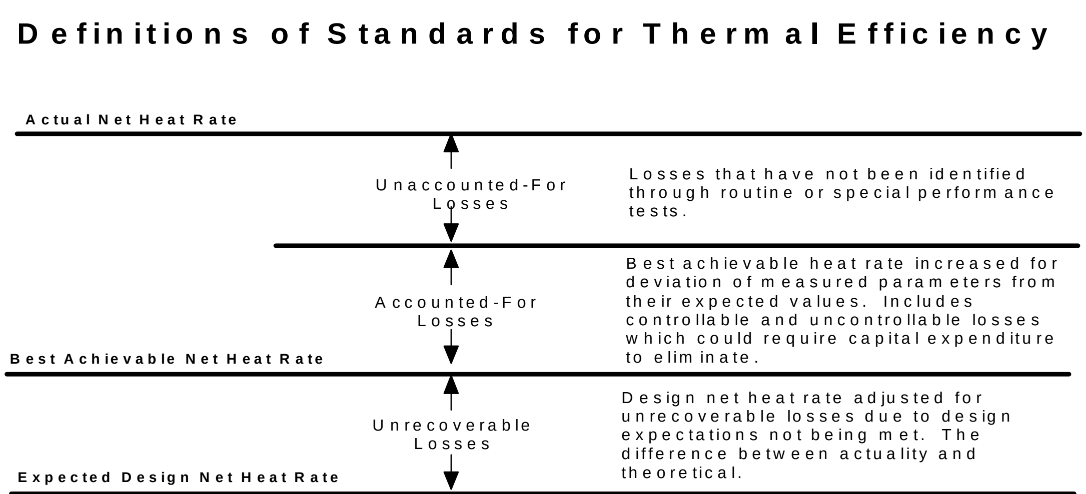


Figure 2-1
Definitions of Standards for Thermal Efficiency

As-Designed Heat Rate

The design heat rate of a unit is a tool that provides a definable benchmark for comparison and trending purposes. It is simply a curve generated from the following parameters:

- Turbine Heat Balance Curves
- Unit Expected Auxiliary Consumption
- Design Boiler Efficiency

The proper procedure for calculating design/expected heat rate corrects for expected absolute back pressure, coal quality, and ambient temperatures. This procedure is explained in detail in Appendix A.

Design heat rate should be based on a expected boiler efficiency that is calculated using an as-fired fuel analysis and ambient (inlet air after steam coils) temperature. Also a correction should be made for expected condenser performance. The expected condenser pressure given by the condenser manufacturer varies with condenser heat loading and inlet water temperature. Therefore, the expected pressure can vary considerably from the pressure used in the design heat rate curve. It is recommended that expected absolute back pressure be used to adjust the expected heat rate . The expected back pressure should be based upon the actual inlet water temperature, design CCW pump flow (number of pumps in service), and design steam flow to the condenser.

One might argue that the CCW pump flow might be less than design which is most likely a true statement, but the expected flow is still the design flow. The same applies to the steam flow to the condenser. The turbine efficiency may have degraded, thus the actual steam flow seen by the condenser increased, but the expected steam flow to the condenser is still the design flow. By properly calculating the expected/design heat rate any poor performance given by the condenser will not be overlooked in the efficiency factor deviation between actual and design.

Best Achievable Heat Rate

The best achievable net heat rate is the same as the net heat rate obtained from unit acceptance test when the equipment was new and the unit was operated at optimum. Therefore, this heat rate value is realistic and attainable for it has been achieved before.

It is a requirement that the expected design and best achievable net heat rates be adjusted for any equipment additions and/or modifications for any performance program to be of maximum benefit. 1) Actual net heat rate 2) Best achievable net heat rate 3) Expected design net heat rate must be accurate so that a performance person can make intelligent decisions regarding performance improvement.

Having determined the best achievable heat rate at one valve point, valves wide open for example, or at a load point, maximum guarantee for example, the performance person can estimate the shape of the best achievable heat rate performance over the remainder of the load range. This can be done by duplicating the shape of the design heat rate curve. Once the performance person has determined the best achievable performance of the unit, one can compare the actual and the best achievable heat rates to determine the amount of improvement that can be made in the unit's performance.

Operating Heat Rate

Operating heat rate is calculated from the heat energy consumed by a unit or station for a specified time period regardless of the operating status of the unit or station.

A common utility practice is the use of an accounting heat rate which is the ratio of the total fuel consumed by the unit or station divided by the gross electrical energy produced by the generator. The fuel input into the furnace or the unit, is simply the fuel consumed by the unit, multiplied by the fuels heating value. The total fuel consumed is considered in the heat rate calculation including fuel used during light-off and start-up.

Accounting heat rate may be referred to by different names at various utilities. Some of the more common synonyms are: generation cost heat rate, gross overall heat rate, and heat rate of record. Accounting heat rate is not a useful measure of unit performance to a performance engineer wishing to evaluate the efficiency of a particular unit. However, because this heat rate index has widespread exposure within a utility organization, the performance engineer should be familiar with how it is computed and with the manner in which he can influence its accuracy. This is normally the heat rate value which is reported for publication and comparison with the performance of other utilities.

Accounting heat rate is primarily influenced by fuel and power measurements. The performance engineer can influence the accuracy of this calculation by ensuring the proper maintenance and calibration of fuel weighing and sampling equipment and reviewing and auditing fuel sampling procedures, and by reconciling fuel inventory measurements with fuel as-received and as-burned records.

This latter point can be very important to an accurate accounting of heat rate because some utilities will adjust the heat rate calculation based upon fuel consumption which is adjusted for fuel inventories. The fuel consumed by a unit or a station is determined by the difference between the fuel received, as estimated from fuel supplier receipts or as-received measurements, and the fuel stored or stockpiled. The amount of stabilizing or start-up fuel must also be considered in determining the total amount of fuel consumed. For instance, oil-fired units, which burn No. 6 oil, will use either natural gas or No. 2 oil as a start-up fuel and coal-fired plants will use either oil or gas for flame stabilization at low loads and during start-up. The energy input into the unit is calculated by multiplying the amount of fuel consumed by the average heating value for that fuel. This total energy consumed is then divided by the gross electrical generation to determine the gross accounting heat rate. A net accounting heat rate can be computed by subtracting from the gross electrical generation the energy consumed by the unit or station during power generation and the energy consumed by the unit during non-operating periods during the specified time period.

The obstacles to calculating an accurate heat rate for a coal-fired unit using this method are:

- The difficulty of accurately assessing the quantity of coal which has been diverted to the stockpile.
- The inaccuracies of the as-received coal scales and the large uncertainty associated with measurements made over long periods of time.
- The use of an average heating value for a fuel whose heating value may vary widely with different fuel suppliers.

Incremental Heat Rate

Units within a utility system and within a power pool are dispatched, i.e., loaded upon the grid, based upon their incremental heat rate and resulting cost curve. In addition to its usefulness for economic dispatch, the incremental cost curve is also used in production simulation for maintenance planning and projecting fuel procurement needs and for pricing of power for sale or resale. The benefit of accurate input/output data for a unit (and the benefit of an accurate incremental cost curve) will depend upon the particular utility system which the unit is a part and on how the unit is operated.

The estimation of the unit incremental cost curve utilizes input/outpt (I/O) data for the unit. I/O data are obtained in two ways. One is by varying unit load and measuring the rate of energy into the boiler (i.e., fuel flow and heating value) and power produced by the generator (i.e., power at the generator bus bar). This approach is practical for oil and gas units where the technology for reasonable accurate fuel flow measurements and relatively consistent fuel heating values exist. However, for coal units this approach is less than optimum due to the difficulty in obtaining accurate coal flow measurements and due to the constantly changing heating value of the fuel. The possibility of introducing errors into the cost curves is high. For the most part a unit's incremental cost curve is based upon estimates of a unit's performance which are nontypical of that unit's actual performance. For example, a unit may be operating with some equipment problem which is affecting its heat rate yet no adjustments are made to that unit's incremental curves to account for the decreased (less efficient) performance.

A second approach to obtaining I/O data, which may be more suitable for pulverized coal-fired units, is to measure turbine heat rate vs. Load (corrected to standard conditions), boiler efficiency vs. load, and then calculate energy input rate for various values of power output according to:

$$Rate\ of\ Energy\ Input = \frac{Corrected\ Turbine\ Heat\ Rate}{Gross\ Generation / Boiler\ Efficiency} \qquad \text{(eq. 2-1)}$$

Once the I/O data are obtained, an I/O model must be developed which fits the data and also meets the constraints of operation. The main constraint is that the input/output curve, fitted through the unit valve points, must be monotonically increasing with a concave upward curvature. This curve is then differentiated to obtain the incremental heat rate curve which, together with fuel and maintenance costs, yields the incremental cost curve.

EPRI Reports EPRI CS/EL-4415 vols. 2 and 6 address the issue of incremental heat rate in more detail.

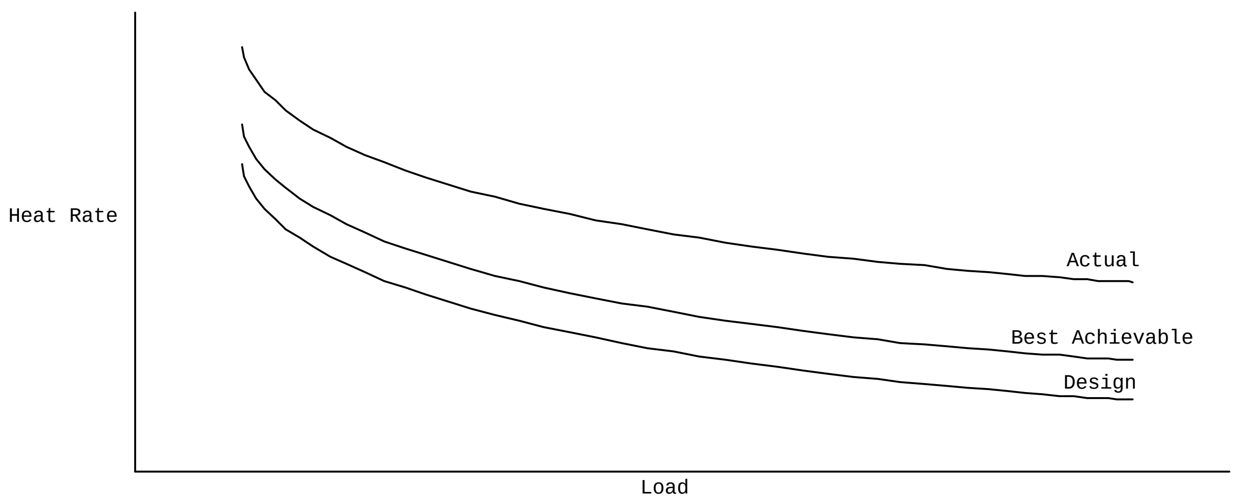


Figure 2-2
Typical Unit Heat Rate Curves

Heat Rate Measurement Methods

EPRI Report "Power Plant Performance Monitoring and Improvement - Boiler Optimization" (CS/EL 4415, Vol 4) contains substantial discussion concerning heat rate measurement methods. The following is a summary of some of the pertinent information taken from this report.

General

A new approach for the direct measurement of unit heat rate, referred to as the output/loss method, was developed; and the accuracies of the input/output technique and the output/input method were evaluated. Input/output utilizes measurements of the flow rate and heating value of the coal and of gross electrical generation and station service power. If used on-line, this technique would require measurements of the instantaneous heating value and coal flow rate of the coal as it flows from the storage bunkers to the mills. The output/loss method depends on measurements of quantities such as the flow rate, composition and temperature of the flue gas leaving the stack, feedwater flow rate, steam temperatures and pressures entering and leaving the boiler, and amount of unburned carbon. The flue gas measurements in the stack, used to determine the stack loss, provide an alternative to the absolute measurement of coal flow rate and heating value required by the input/output method.

Two sets of carefully controlled field tests were carried out in which heat rate and boiler efficiency were measured by both the input/output and the output/loss methods. The results show that both techniques are subject to bias errors, which in these particular tests ranged up to two percent. The bias errors associated with the input/output measurements were due to errors in the measurement of coal flow rate and/or higher heating value. Given current measurement techniques, it will be extremely difficult to make continuous and automated field measurements of heat rate using input/output which are any more accurate than the data that were obtained here.

The measurements by the output/loss approach were subject to systematic errors which varied with the level of O_2 and which ranged from a bit less than one percent at the high O_2 level to about two percent at the low O_2 level. These errors were due to systematic errors in the measurement of the flow rate of the stack gas.

The random error of measurement by the output/loss approach was found to be significantly lower than that obtained with input/output. Standard deviations in the range of 10 to 40 Btu/kWhr (11 to 42 kJ/kWhr) were obtained by output/loss, whereas the corresponding values for input/output were of the order of 100 Btu/kWhr ((106 kJ/kWhr). The smaller standard deviations associated with the output/loss measurements are particularly important in unit optimization where the emphasis is on detecting changes in heat rate as small as 10 to 20 Btu/kWhr (11 to 21 kJ/kWhr) due to changes in parameters such as coal grind size or 0_2 level.

Further work is now underway to improve the output/loss method. In its current state of development it offers substantial advantages over the input/output approach. The output/loss method gives smaller random errors than input/output. In addition, based on the data and analyses that have been carried out so far, it should be possible to reduce the systematic error in the output/loss measurement to levels significantly less than occur with input/output.

Actual Net Heat Rate

General

Two approaches for direct measurement of heat rate are described in EPRI CS/EL 4415. Vol 4. These are the "input/output" technique and the "output/loss" method. Input/output utilizes measurements of the flow rate and heating value of the coal and of gross electrical generation and station service power.

If used on-line, this technique would require accurate measurements of the instantaneous heating value and flow rate of the coal as it flows from the storage bunkers to the mills. An alternate approach, referred to here as the "output/loss" method depends on measurements of quantities such as the flow rate, composition and temperature of the flue gas leaving the stack, feedwater flowrate, steam temperatures and pressures entering and leaving the boiler, and the amount of unburned carbon. The flue gas measurements, used to determine the stack loss, provide an alternative to the absolute measurement of the coal flowrate and heating value.

A typical coal fired power plant has a boiler efficiency of approximately 90 percent. For each 100 Btu's (106 kJ's) of fuel into the plant, approximately 10 Btu's (11 kJ's) flow out as boiler and stack losses and the remaining 90 (95 kJ's) are transferred to steam flowing to the turbine cycle. Measurements of steam temperature and pressure and feedwater flowrate can be made relatively accurately. On the other hand, the ability to measure the boiler and stack losses with high accuracy is limited. Despite these larger measurement errors associated with the boiler losses, their contribution to the overall uncertainty in the heat rate calculation is relatively small. A heat rate calculation that involves these losses is only weakly dependent on the uncertainty in these terms.

Input/Output Method

In the input/output method all of the parameters contained in the definition of the heat rate are measured directly. The heat input to the boiler is obtained from measurements of fuel flow rate and from laboratory or on-line analyses of fuel heating value. The electrical output of the unit is obtained from measurements of the gross generation and station service power. Thus

$$ANHR = \frac{\dot{M}_{coal}\, HHV}{P_G - P_S} \qquad \textbf{(eq. 2-2)}$$

The measured value of coal flow rate required by this method can be obtained from gravimetric feeders, which, if they have just been properly calibrated, are capable of accuracies of 1 percent or better. Laboratory analyses of coal heating value have typical uncertainties of about 1 percent. The electrical quantities can be measured relatively accurately with uncertainties ranging from 0.1 to 0.5 percent. The uncertainty in the measured heat rate can be given as

$$\varepsilon\ (HR)^2 = \varepsilon\ (\dot{M}_{coal})^2 + \varepsilon\ (HHV)^2 + \frac{\delta\,(P_G)^2 + \delta\ (P_S)^2}{(P_G - P_S)^2} \qquad \textbf{(eq. 2-3)}$$

where the ε's are relative uncertainties and the δ's are absolute uncertainties of the quantities indicated.

Fuel Effects

To accurately calculate a unit's heat rate, only the fuel actually consumed during power production is factored into the heat rate calculation. With oil and natural gas this is not a deterrent since the fuel is measured as it being consumed. However, in coal-fired units, instantaneous fuel flow measurements cannot be made. The measurement of fuel usage is made either at the entrance to the coal bunkers or between the bunkers and the coal pulverizers.

The main advantage of the scales being located at the entrance to the bunker is that they can be periodically calibrated since they are not in continuous use and are easily accessible. This enables an accurate measurement of coal delivery to be made. The disadvantage of this method is that the fuel consumed cannot be easily measured over short time intervals with any degree of accuracy. However, determining the amount of fuel actually consumed, the coal delivery values must be adjusted to account for the coal which is stored in the bunkers. That relates this method of calculating operating heat rate to long periods of time.

Other uncertainties associated with this method are:

- The inaccuracies of the as-consumed scales and the large uncertainty interval associated with measurements made over long time periods.
- The inaccuracies associated with estimating the coal storage of the bunkers.
- The inaccuracies associated with using average fuel heating value measurements.

The method described in equation 2-2 can provide a gross estimate of a unit's actual net operating heat rate. However, these measurements do not provide adequate information for making sound decisions regarding heat rate improvement. It is used mainly for management information purposes and as an overall indicator of the performance of the unit. To obtain useful measurements of a unit's heat rate, the calculation must be performed over small time intervals with accurate measurements of fuel usage and heating value. The location of the fuel measurements in the above method make short time period tests nonfeasible.

To achieve the accuracy needed to perform short time period tests, the fuel measurement should be made as close to the combustion chamber as possible. Again, for oil and natural gas fired units this poses few problems since there is no fuel storage to take into consideration. In a pulverized coal-fired unit the optimum location to make the fuel flow measurement would be between the pulverizer and the furnace as this would eliminate errors introduced by coal storage within the pulverizers. However, this is not possible with present technology.

Presently, the most accurate fuel flow measurements can be made as the fuel is delivered from the bunker to the pulverizers. As mentioned, this introduces a storage error due to the coal which resides within the pulverizer. However, this error is small when compared to the total amount of fuel consumed during the period for which a heat rate is being calculated.

Fuel measurements made at the pulverizer inlet give an almost direct indication of the amount of fuel consumed by the unit over any period of time without the need to adjust the measurements for coal storage. The use of the pulverizer inlet coal scales to measure coal consumption can be used to characterize unit heat rate over the load range at which the unit operates. This is accomplished by measuring the coal consumed during short time periods when the unit is operating at a constant load point. By performing this test at several valve points, the information can be used to generate input/output curves for dispatching purposes and compare the unit's actual operating heat rate against its best achievable heat rate.

The uncertainties involved in making heat rate calculations from pulverizer inlet measurements are:

- The inaccuracies associated with the coal measuring device.

- The inaccuracies associated with the heating value of the fuel.

Output/Loss Method

With a control volume which includes the boiler, air preheaters, steam air heater, fans and mills, the principle of conservation of energy can be used to develop an expression which relates the rate at which energy flows into the power plant with the coal to the rate of heat transfer to the steam turbine cycle, the various boiler losses and other quantities such as mill and fan power. This relationship is given in approximate form by the following expression.

$$\dot{M}_{coal} \bullet HHV \approx Q_T - Q_A + L_B - P_{fm} \quad \text{(eq. 2-4)}$$

where Q_T = rate of heat transfer to the steam turbine cycle

Q_A = rate of heat transfer in the air preheater

P_{fm} = fan and mill power

L_B = boiler losses including terms such as stack gas, unburned carbon and sensible heat in the ash.

The unit heat rate can thus be evaluated independently of coal flow rate and higher heating value measurements, if measurements of Q_T, Q_A, L_B and P_{fm} are made instead. A rigorous derivation for the energy balance is given in Appendix B of EPRI CS/EL - 4415 Vol. 4.

An error analysis similar in concept to that given by Equation 2-2 was performed for the output/loss method.

These results indicate approximately a 10 to 1 ratio between the uncertainties in stack loss and heat rate. While the uncertainties in quantities such as unburned carbon and heat transfer in the air preheater were assumed to be relatively large, the energy flows associated with these quantities are small enough so that their contributions to the overall uncertainty in heat rate are small. With the output/loss technique, uncertainties in heat rate of approximately 1 percent are feasible even with uncertainties in the order of 10 percent in the stack loss and 50 percent in unburned carbon.

Efficiency Factor

Efficiency Factor (EF) is an important element in a strong performance monitoring program. The indicator is a quick reference of unit performance in relation to what it was designed to be. EF can also indicate the magnitude of effort needed to correct performance deficiencies. When the EF = 0, the unit is performing exactly as it was designed to. When EF is > 0, the unit is performing poorer (at a higher heat rate) than design and when EF is < 0, the unit is performing better than design. Once a performance monitoring program is implemented, a decreasing trend in EF indicates success in lowering and maintaining a good heat rate. As with most performance indicators it is not so important what the value of EF for a unit is, but rather which direction it is going over time.

$$EF = \frac{(Actual\ Online\ Heat\ Rate - Corr.\ Design\ Heat\ Rate)}{(Corr.\ Design\ Heat\ Rate)}\ x\, 100 \qquad \textbf{(eq. 2-5)}$$

EF accurately judges performance whether a unit is used for peaking, base load, or intermittent operation. Actual on-line heat rate achieved is always compared to the design heat rate curve at the point which best represents actual operation. The design heat rate to use in the calculation is at the average load for the unit during the measurement period. EF should be measured monthly for good data averaging, especially considering the importance of accurate coal weighing in determining actual heat rate.

Heat Rate Logic Trees

Introduction

Engineers at a fossil station responsible for monitoring Heat Rate need a systematic approach to aid them in identifying the root cause(s) of declining unit performance. The Heat Rate Logic Trees described in "Heat Rate Improvement Guidelines for Existing Fossil Plants", EPRI report CS-4554 provide an example of one such approach. The Logic Trees described in this report were modified and used during five demonstration projects at fossil stations in the United States. These demonstration projects are listed with a brief summary of how they modified the original guidelines at the end of this section.

Logic Tree Descriptions

Logic Trees are visual tools or diagrams of a Root Cause Analysis. Logic symbols are used to construct a diagram that defines a problem and various paths to a root cause(s).

Figure 2-3 describes the logic symbols used in the diagrams. This information was obtained from another EPRI report: "Coal-Handling System Problems at Gulf Power Company's Plant Crist: a Root Cause Analysis", EPRI CS-4743. All of the gates (blocks) in the Heat Rate logic trees developed here are "or" gates, meaning that any one of the lower level items can contribute to the higher level item.

Transfers are used to switch from one block or page of the logic tree to another. The transfer point is identified by a letter or number inside the triangle. Logic Trees can be constructed using the process described in CS-4743, but it will be more efficient to use the diagrams provided in CS-4554 or as modified in one of the demonstration project reports. Figures 2-4 through 2-21 show logic trees compiled from four different fossil plants.

Heat Rate Losses Tree

The heat rate logic tree is used to identify areas in the plant where heat rate degradation may be occurring without conducting expensive tests. The logic tree is structured to provide a process by which decisions can be determined that narrow down the cause of the problem based on available information. A statement of the overall problem (heat rate losses) starts the logic tree. This statement is placed in an "or" gate because any or all of the lower levels could contribute to heat rate losses. See Figure 2-4.

Major Cycle Component Trees

The next level of the logic tree identifies major areas in the plant cycle that have the potential for contributing to the overall problem. These include components such as the boiler and turbine, systems such as condensate/feedwater, cooling water, auxiliary systems (both electrical and mechanical), and fuel handling, and processes such as heat losses, and cycle isolation (leaks). These are plant dependent, (an oil fired unit would not have fuel handling components such as a coal pulverizer mill), and should be evaluated for applicability at each station. Each block shown on the main diagram is a part of a "transfer in" gate or a continuation from another page. The triangle below the block has an identifier showing where the logic tree is continued or "transferred from".

Figures 2-5 & 2-6 show how a logic tree is continued or "transferred from" boiler losses. This is also an "or" gate with several potential causes. These potential causes can also be "or" gates and/or transfer in gates. The process is repeated until the root cause(s) of the problem is identified (problems with the superheater due to fouling), or it is decided not to continue to any lower levels. Economic and time considerations determine how far the process is to continue.

Decision Criteria

Associated with the problem or potential cause is a decision criteria. These are listed below each block or gate. Decision criteria are conditions that need to be evaluated to determine if the potential cause (area or system shown in the block) is the actual cause of the heat rate loss. Decision criteria can be based on the value of a single performance parameter or the values of multiple performance parameters. In some cases the parameters may need to be trended, or calculations may need to be performed. Equipment checklist, special or periodic tests and comparisons between actual and expected values can also be appropriate decision criteria.

Performance Parameters

EPRI CS-4554 provides several tables and appendices that list performance parameters which can be used as decision criteria. Table 2-1 lists Performance Parameters and their Impact on Heat Rate. Table 2-2 is a list of performance parameters most often monitored by utilities. Appendix B of the EPRI CS-4554 is a Performance Parameter Accounting Manual. This is the basis for constructing, modifying or using a logic tree. It gives the effect on heat rate as a Utility Average, possible causes of the performance parameters deviation, and possible corrections that can be made to operations or maintenance.

Logic Tree Application

How can the station engineers modify and use the logic trees and decision criteria described above to evaluate the cause(s) of heat rate losses? First obtain data from various sources at the station including routine monitoring of selected plant performance parameters, special tests, outage reports, initial design documents and interviews with plant personnel. Then convert the data into decision criteria and associate these with the appropriate areas of the plant. Several options exist for displaying the performance parameters as decision criteria that relate possible causes for heat rate deviation.

1. The Logic Trees modified to suit each station can be used. Figures 2-4 through 2-20 are compiled from four different reports. All areas of the plant and both oil/gas fired and pulverized coal fired units are included. The engineer should select and modify whichever logic trees that best suites the needs of their station.

2. The Demonstration Project for Salem Harbor Station Unit 4 (an oil fired station), EPRI Report GS-7329, uses a tabular form to present this information. See Appendix Tables C-1 through C-5. These tables give the performance parameter(s), causes for degradation of the parameter(s), further testing for confirmation, and a list of

corrective actions. Major components or systems are separated into different tables. Not all areas or systems of a fossil station are included in the tables from this report.

3. In addition to the logic trees, the heat rate improvement guidelines describes a decision tree as a tool for identifying causes of heat rate degradation. See Figure 2-22. This technique relates the performance parameter with the possible causes of deviation depending on whether it is high or low compared to normal. A series of branching paths continue until some final or root cause is reached. Appendix B of the guidelines is used as the basis for constructing this representation.

Example

As an example, determine the potential causes for deviation of main steam temperature:

Tables 2-1 and 2-2 quantifies the effect its deviation has on heat rate.

Appendix B of the guidelines (B.4) also quantifies the effect on heat rate and lists possible causes and possible corrections.

The Heat Rate Logic Trees (Dry Gas Losses) show that the potential cause is fouling of heat transfer surfaces. See Figure 2-6.

The Diagnostic Tables (Boiler Losses) shows additional testing that can be conducted to confirm possible causes. See Appendix Table C-1.

The Decision Tree, provides a branching path to investigate operational or maintenance corrections such as sootblowing or damper adjustment. See Figure 2-22.

It can be seen that information from different sources is used to narrow the search for the cause of steam temperature deviation. The engineer can take the logic trees and diagnostic tables provided and modify them to suit the station's needs. Later sections of this document describe how to establish heat rate monitoring programs progressing from initial steps to performance testing of major components.

EPRI Reports referenced in this section:

Coal Handling System Problems at Gulf Power Company's Plant Crist: a Root Cause Analysis, Report CS-4743. Describes the symbols used and how Logic Trees are developed.

Heat Rate Improvement Guidelines for Existing Fossil Plants, Report CS-4554. The initial guidelines and logic trees describes the decision tree and gives Appendix B, Performance Parameter Accounting Manual. The logic trees for electrical and steam auxiliary losses, fuel handling losses and heat losses are used.

Demonstration of Heat Rate Improvement Guidelines (Gas Fired Unit), Report GS-7295 provides recommendations for improvement of the guidelines and modified logic trees. The logic trees for cycle isolation are used.

Heat Rate Demonstration Project (Oil Fired Unit), Report GS-7329 gives a cross reference between the report and the heat rate guidelines. The Performance parameter diagnostic table is used.

Heat Rate Demonstration Project, North Omaha Unit 5 (Coal Fired), Report TR-102122. Describes the logic trees and how to modify for this station. The logic tree main diagram and the top four areas are used.

Heat Rate Demonstration Project, Mt. Storm (Coal Fired) Unit 1, Report TR-102127. Describes use of logic trees. The logic tree on steam conditions is used. This area was added to the main diagram.

Heat Rate Demonstration - Ormond Beach (Gas/Oil Fired)Unit 2, Report TR-101249. Provides a discussion on cycle isolation and a walkdown isolation chart.

Table 2-1
Impact of Performance Parameter Deviation on Heat Rate

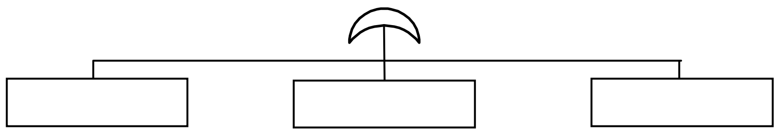
Performance Parameter	Engineering Units (E. U.)	Engineering Units (S. I.)	Variation Btu/kWhr	Variation Kj/kWhr	Range Btu/kWhr	Range Kj/kWhr
Condenser Backpressure	InHg	mmHg	204.0	8.5	42.0 - 269.0	1.75 - 11.2
Auxiliary Power	%		86.0	91.2	64.0 - 97.0	67.4 - 102.8
L. P. Efficiency	%		57.	60.4		
Excess Oxygen	%		29.4	31.2	18.0 -36.0	19.1 - 38.2
Makeup	%		24.0	25.4	4.0 - 88.0	4.2 - 93.3
H. P. Efficiency	%		18.8	19.9	17.9 - 21.5	19.0 - 22.8
I. P. Efficiency	%		14.5	15.4	11.0 - 19.1	11.7 - 20.2
Unburned Carbon	%		11.7	12.4	6.0 - 12.8	6.4 - 13.6
Feedwater Inlet Temperature	°F	°C	8.7	5.1		
Coal Moisture	%		7.8	8.3	6.0 -10.0	6.4 - 10.6
Flue Gas Temperature	°F	°C	2.7	1.6	2.1 -4.2	1.24 - 2.47
Main Steam Temperature	°F	°C	1.4	0.8	.7 -1.7	0.41 - 1.00
Hot Reheat Temperature	°F	°C	1.3	0.77	.9 - 1.9	0.53 - 1.12
Main Steam Pressure	psia	kPa	0.4	0.06	.03 - .65	0.0044 - 0.099

Table 2-2
Parameters Most Often Monitored by Utilities with Controllable Loss Programs and their Typical Deviations

Parameter	Heat Rate Deviation Per Percent Change From Expected Valve (Btu/kWh/%)	Typical Deviation From Expected Value	Heat Rate Deviation (Btu/WkHr
Backpressure	2.3	50.7	117.1
Auxiliary Power	5.1	11.6	59.2
Excess Oxygen	1.1	36.9	40.9
Makeup	2.3	13.0	30.8
Unburned	0.3	27.7	20.3
Coal Moisture	0.6	16.6	9.6
Throttle Temp.	13.4	-0.7	8.8
Hot Reheat	14.2	-0.6	8.6
Feed. Inlet Temp	44.2	-0.2	7.5
H.P Efficiency	16.9	-0.3	4.9
Main Steam Press.	6.7	0.7	-4.5
I. P. Efficiency	12.8	0.9	-11.2
Flue Gas Temp.	10.0	-2.0	-20.1

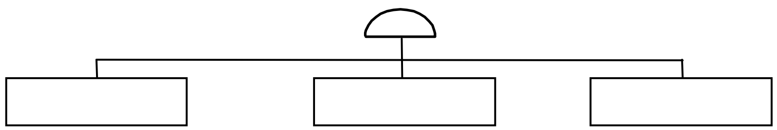
LOGIC GATE SYMBOLS

1. OR GATE



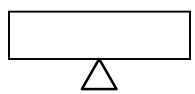
OUTPUT OCCURS IF ONE OR MORE OF THE INPUT EVENTS OCCUR.

2. AND GATE



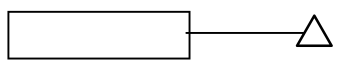
OUTPUT OCCURS IF ALL OF THE INPUT EVENTS OCCUR.

3. TRANSFER IN



INDICATES THAT THE LOGIC MODEL IS DEVELOPED FURTHER AT THE OCCURRENCE OF THE CORRESPONDING TRANSFER OUT (USUALLY ON ANOTHER PAGE). TRANSFERS CAN BE USED TO SIMPLIFY LOGIC MODEL CONSTRUCTION BY ELIMINATING THE NEED TO DEVELOP DUPLICATE BRANCHES.

4. TRANSFER OUT



DENOTES A TRANSFER OF A PORTION OF THE LOGIC MODEL TO CORRESPONDING TRANSFER IN A TRANSFER OUT SYMBOL CAN CORRESPOND WITH MULTIPLE TRANSFER IN SYMBOLS.

Figure 2-3
Logic Model Symbols

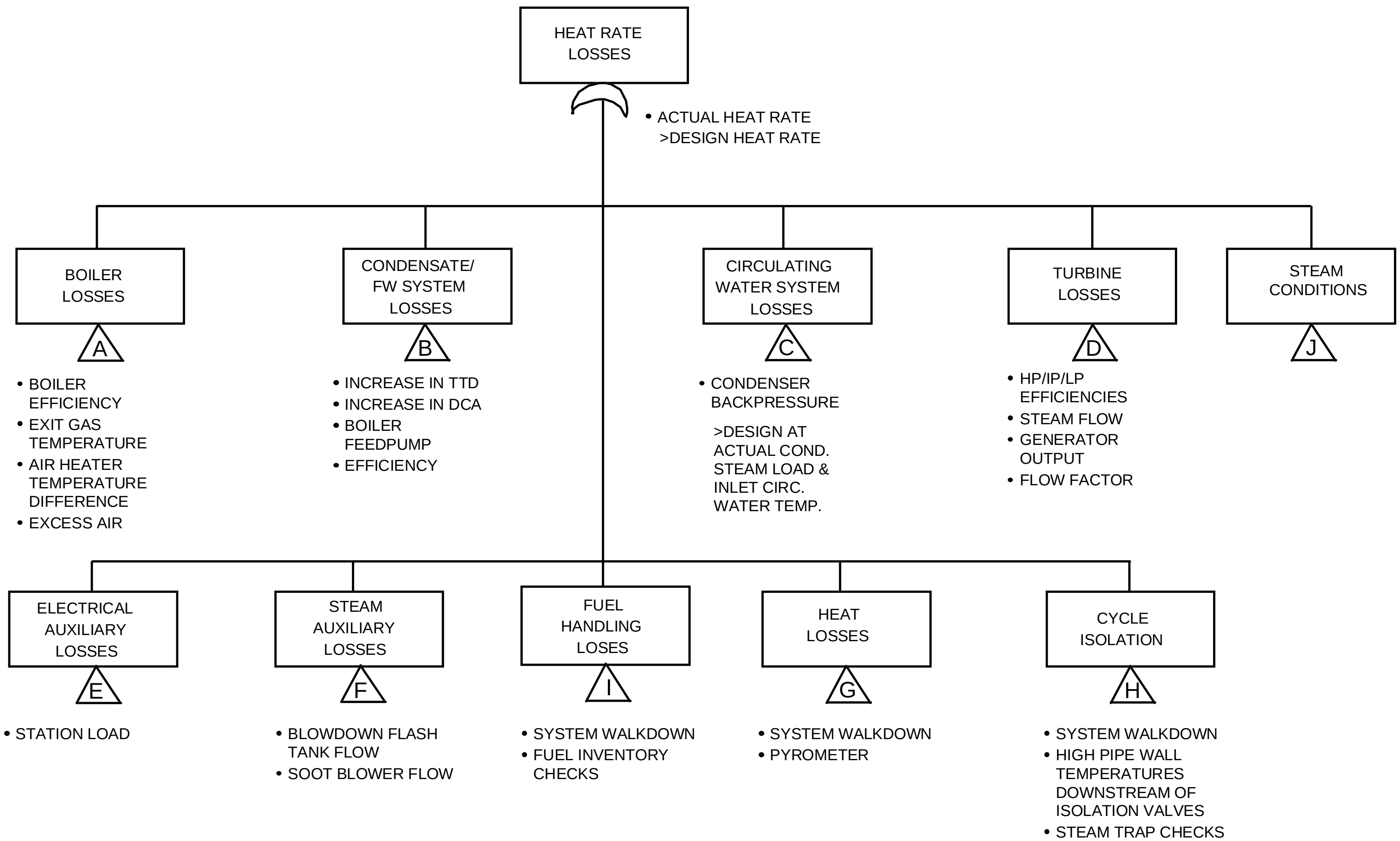


Figure 2-4
Heat Rate Logic Tree - Main Diagram

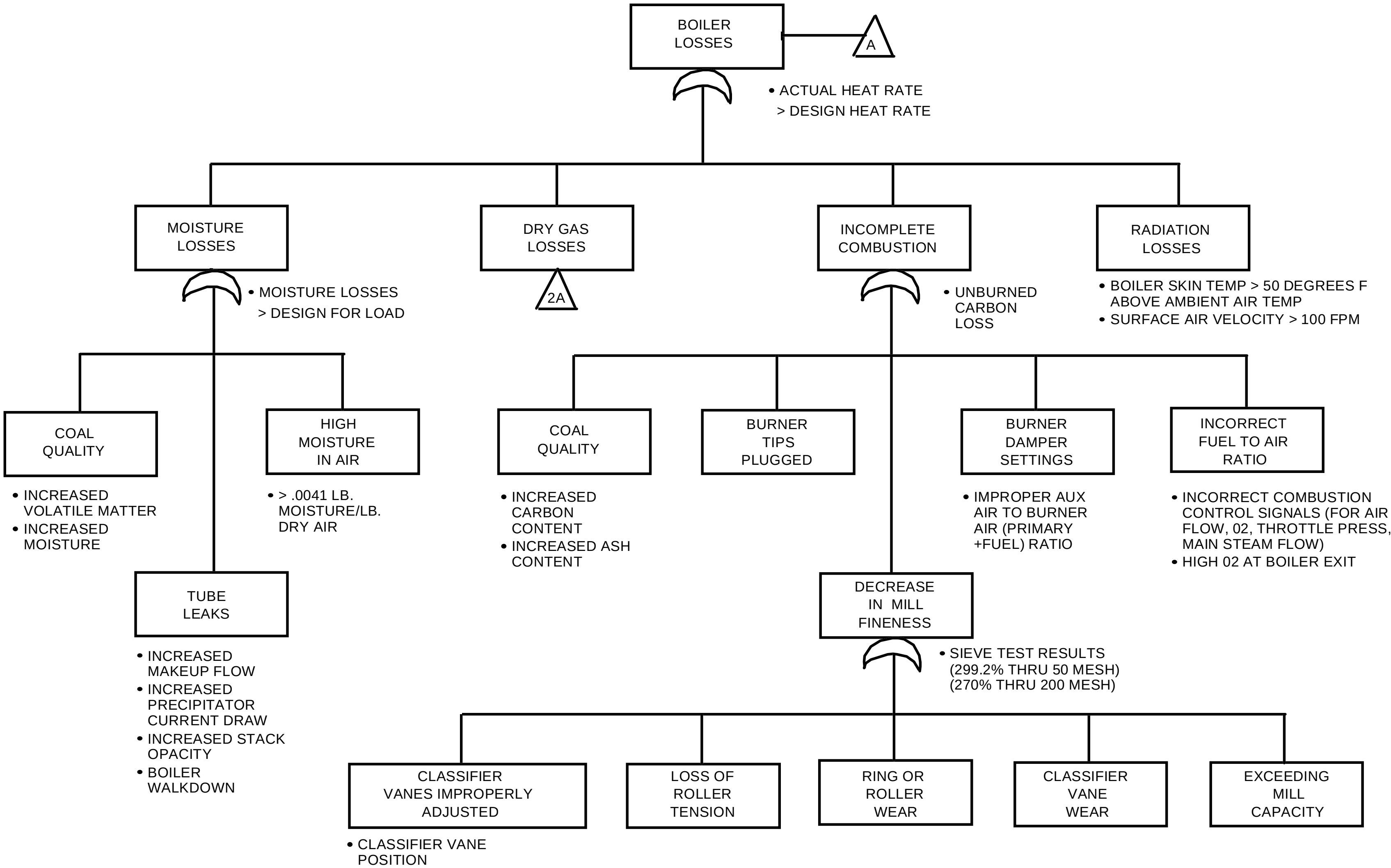


Figure 2-5
Heat Rate Logic Tree - Boiler Losses

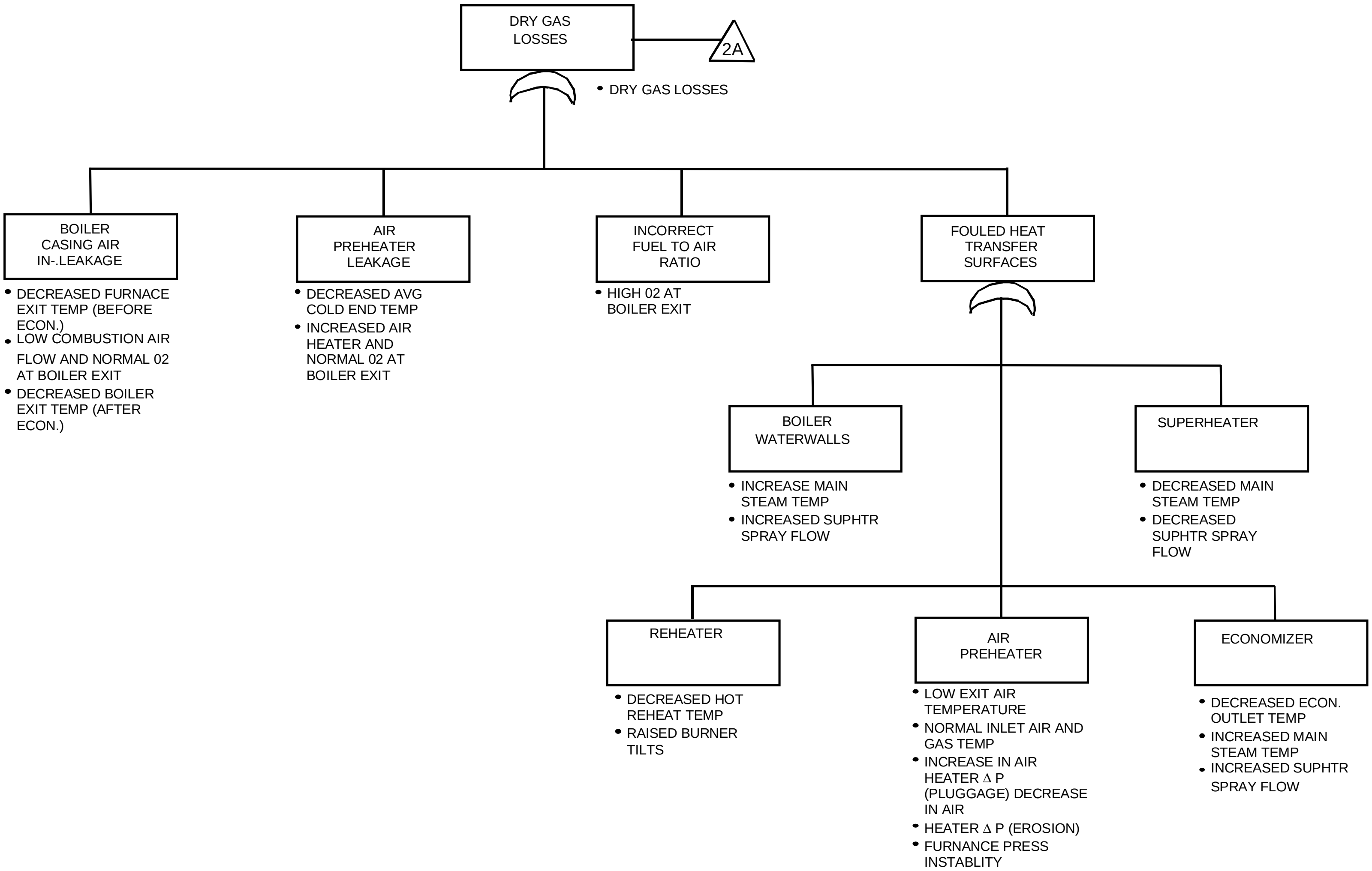


Figure 2-6
Heat Rate Logic Tree - Dry Gas Losses (Boiler Losses Continued)

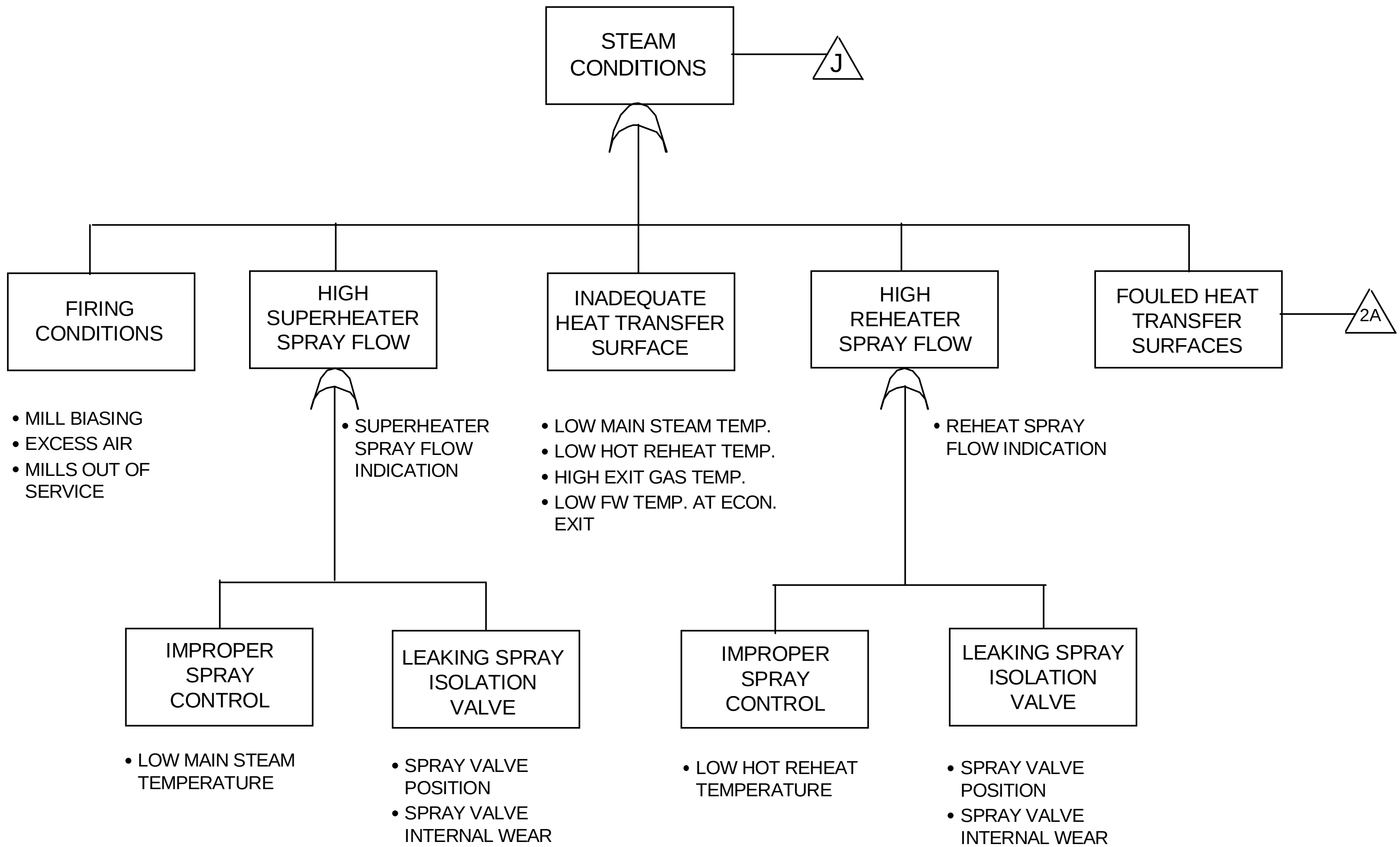


Figure 2-7
Heat Rate Logic Tree - Steam Conditions

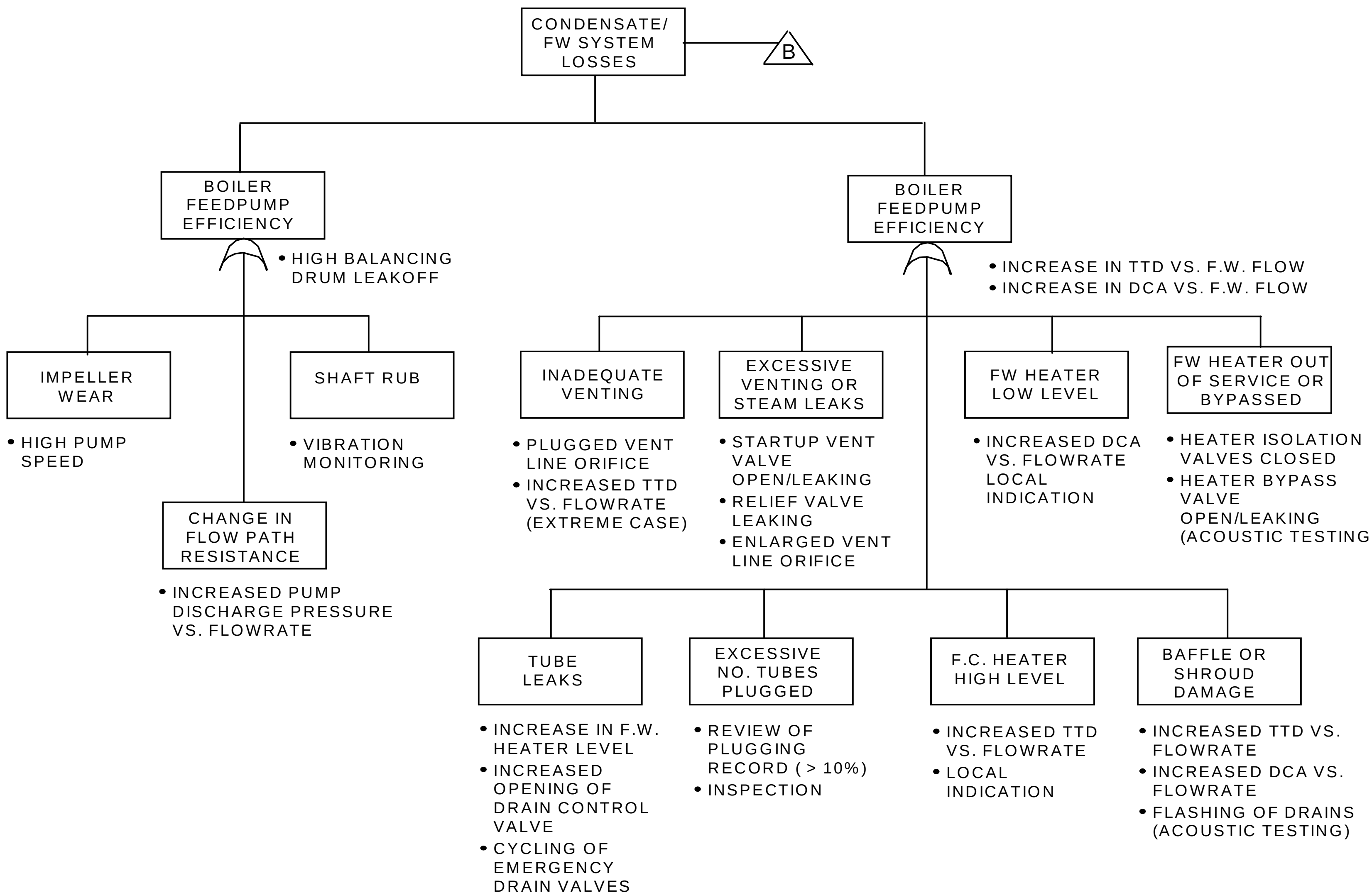


Figure 2-8
Heat Rate Logic Tree - Condensate/Feedwater System Losses

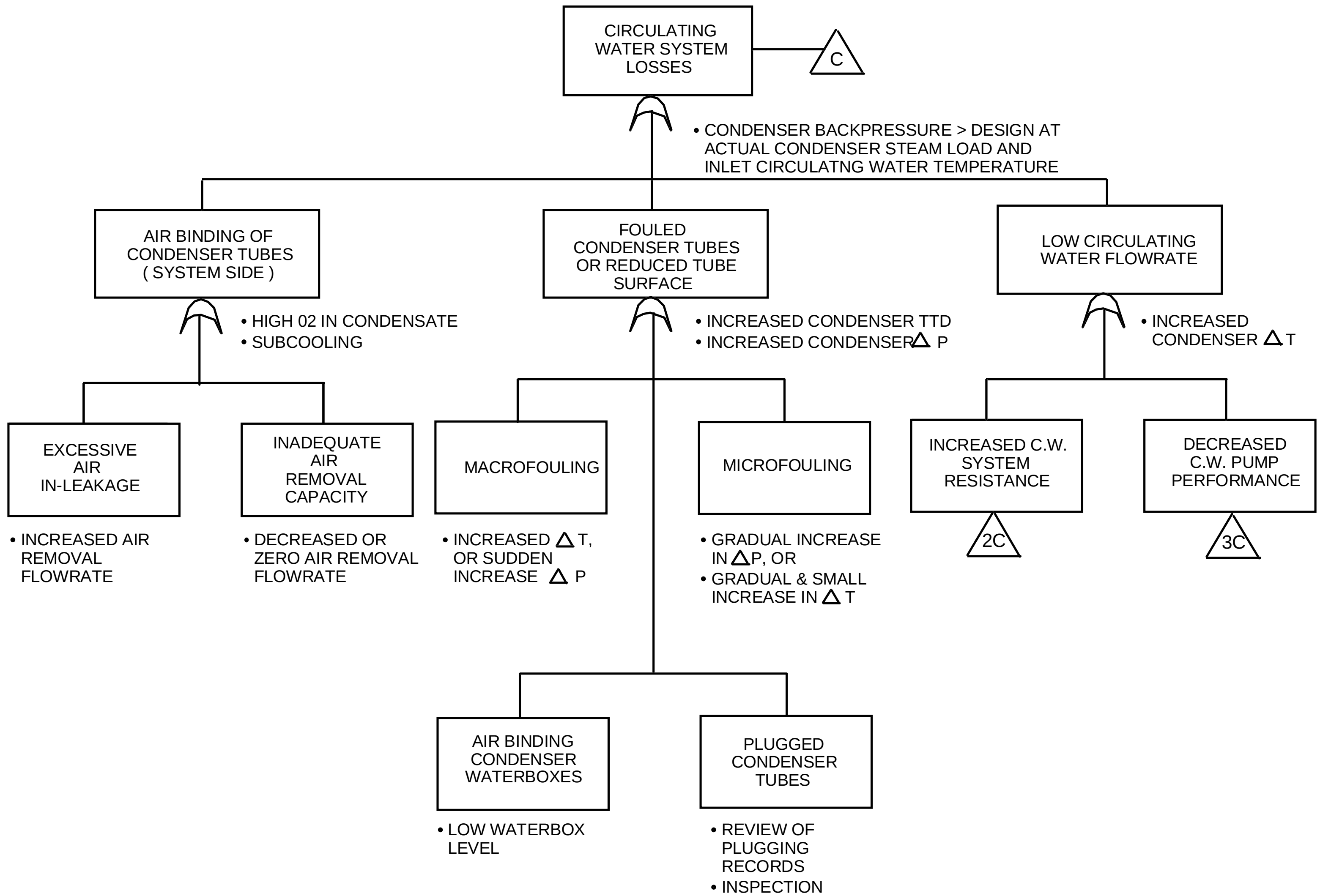


Figure 2-9
Heat Rate Logic Tree - Circulating Water System Losses

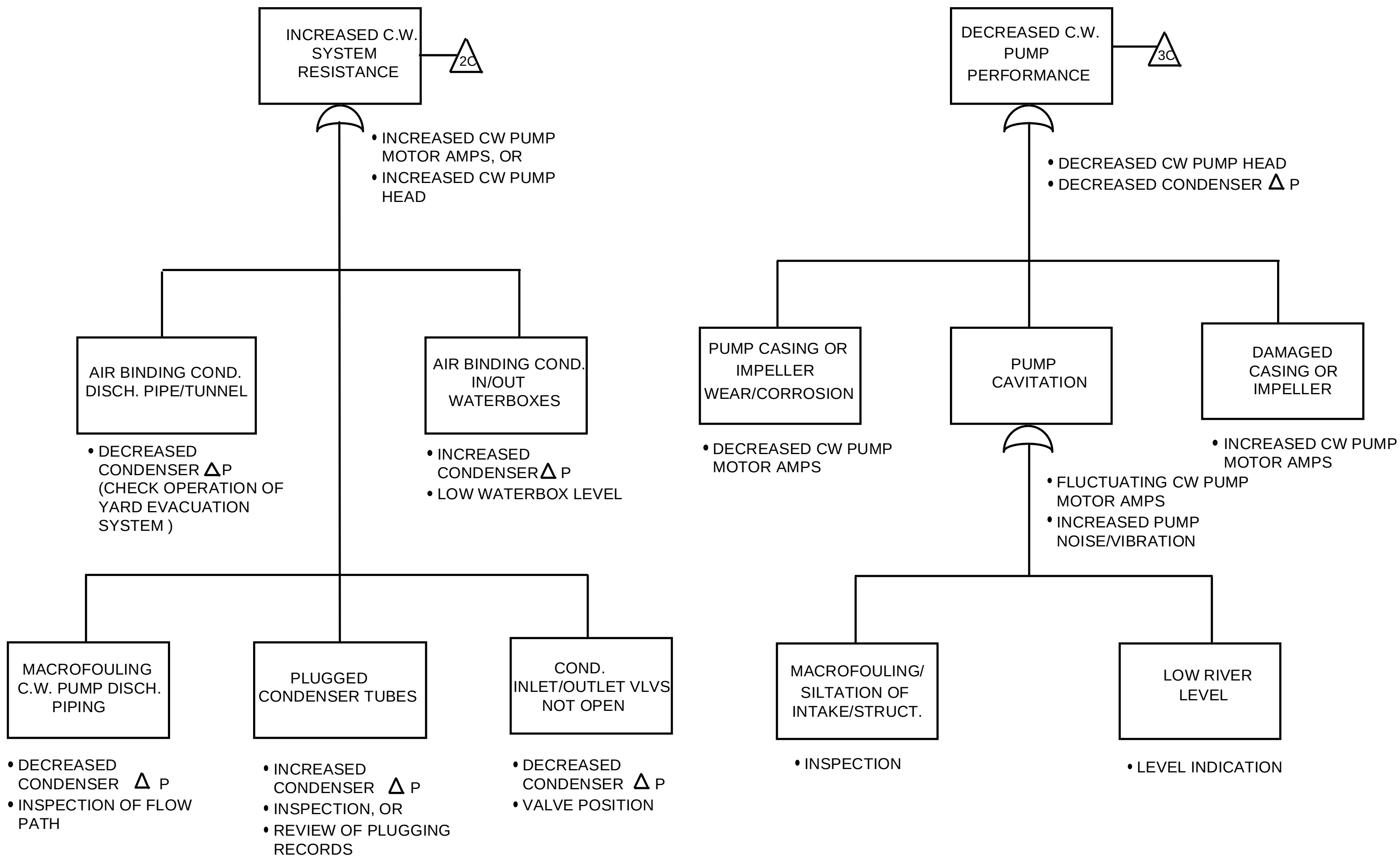


Figure 2-10
Heat Rate Logic Tree - Circulating Water System Losses (Continued - 2C and 3C)

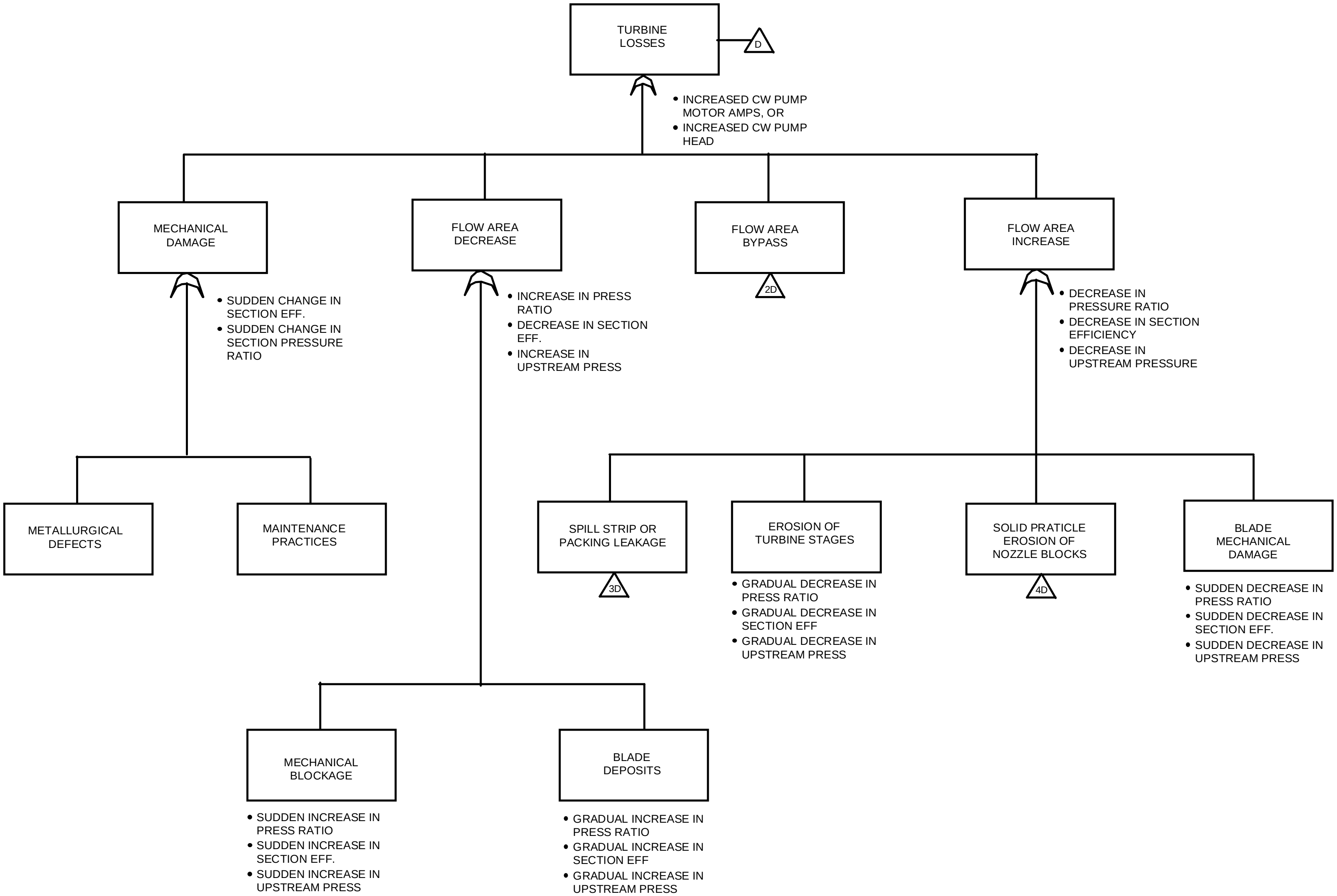


Figure 2-11
Heat Rate Logic Tree - Turbine Losses

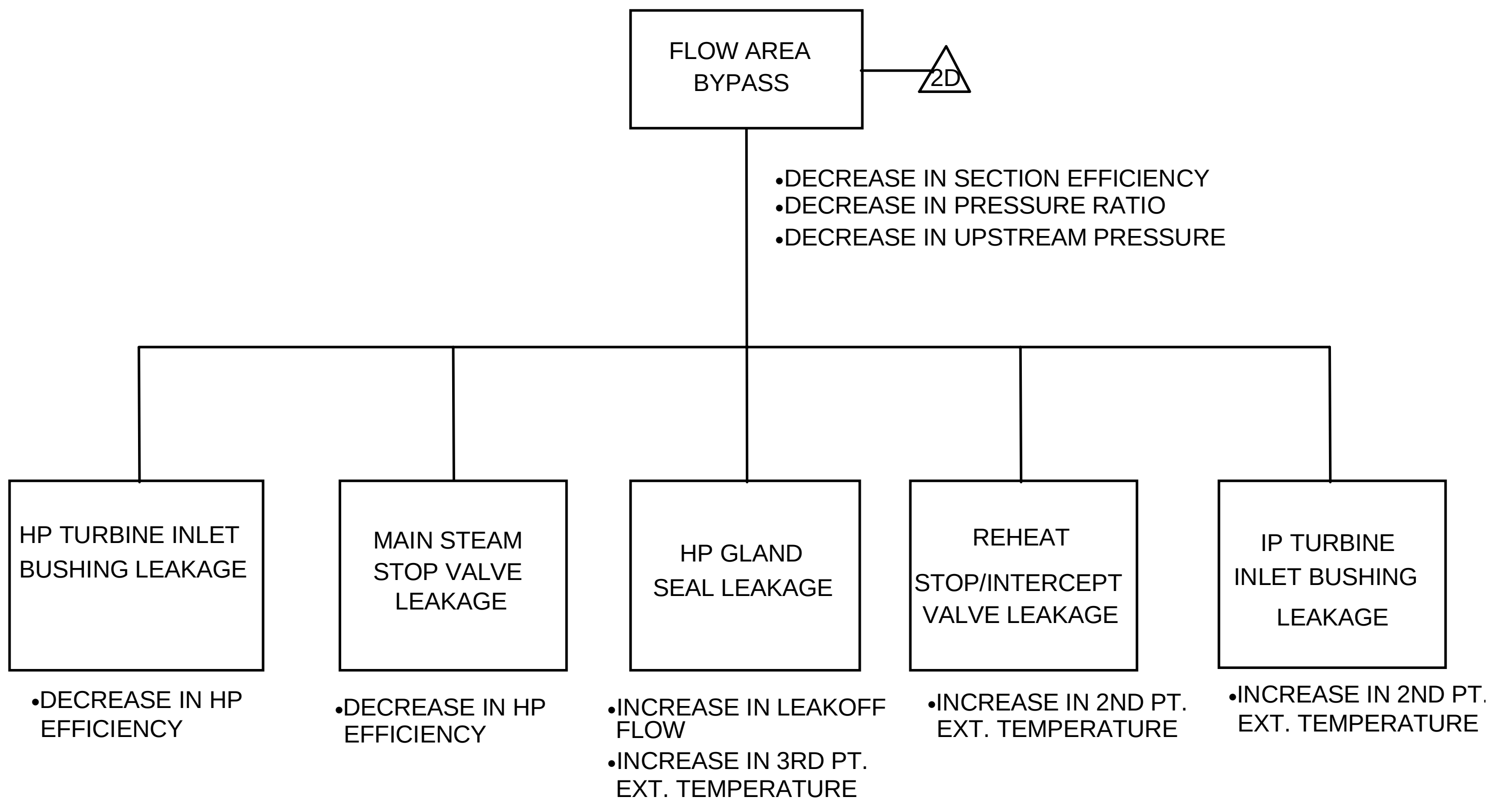


Figure 2-12
Heat Rate Logic Tree -Flow Area Bypass (Turbine losses continued)

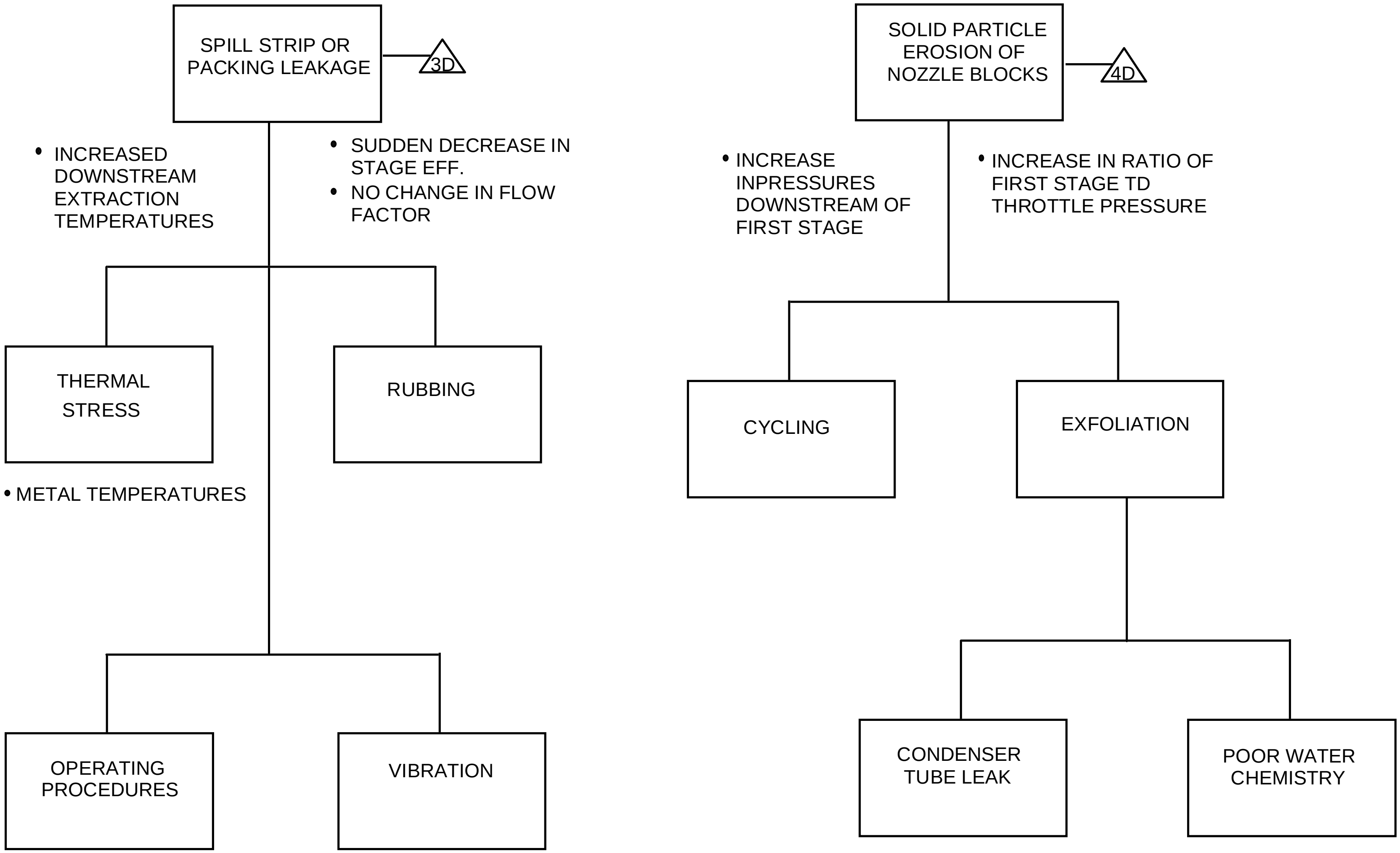


Figure 2-13
Heat Rate Logic Tree - Flow Area Increase (Turbine losses continued)

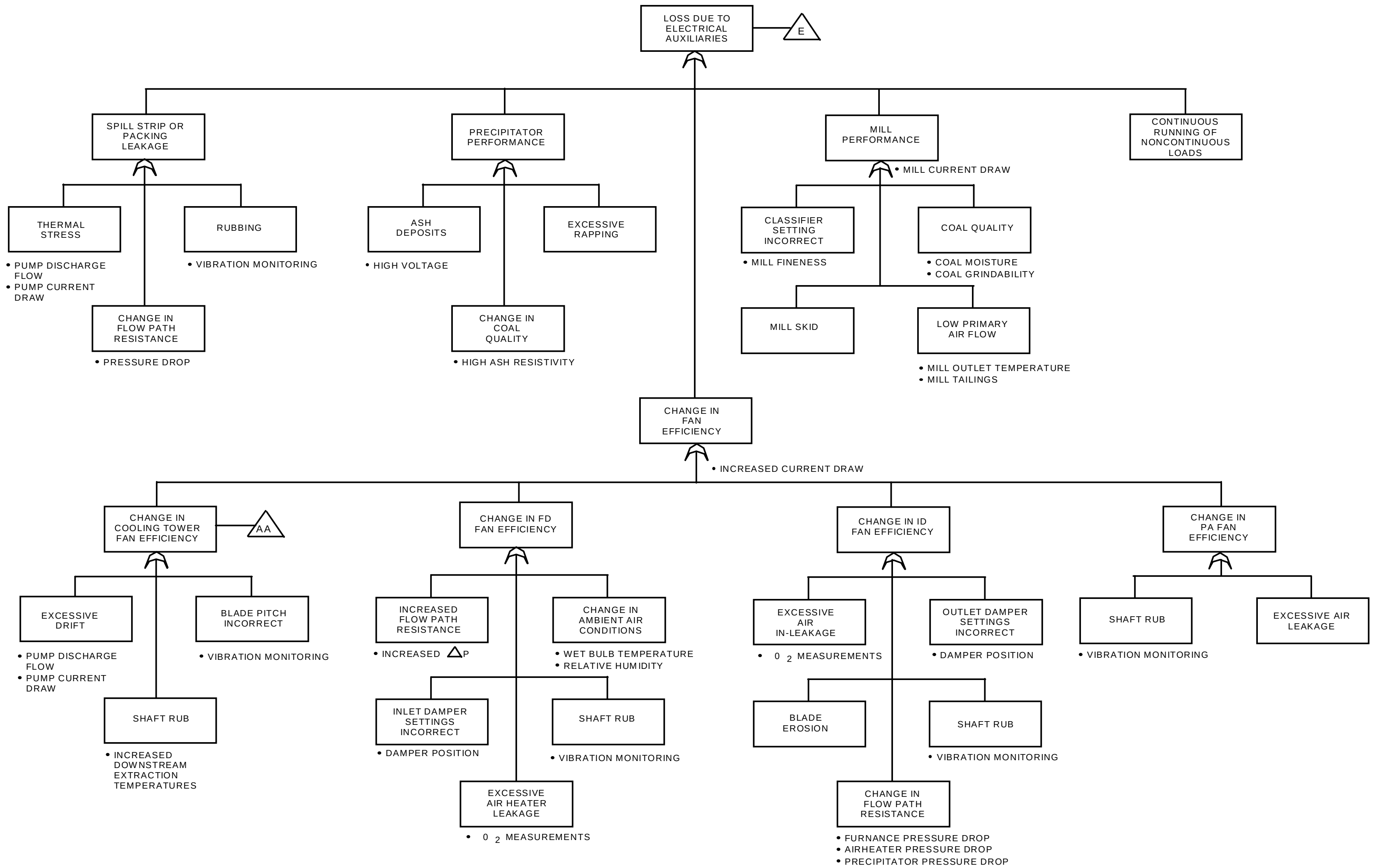


Figure 2-14
Heat Rate Logic Tree - Losses Due to Electric Auxiliaries

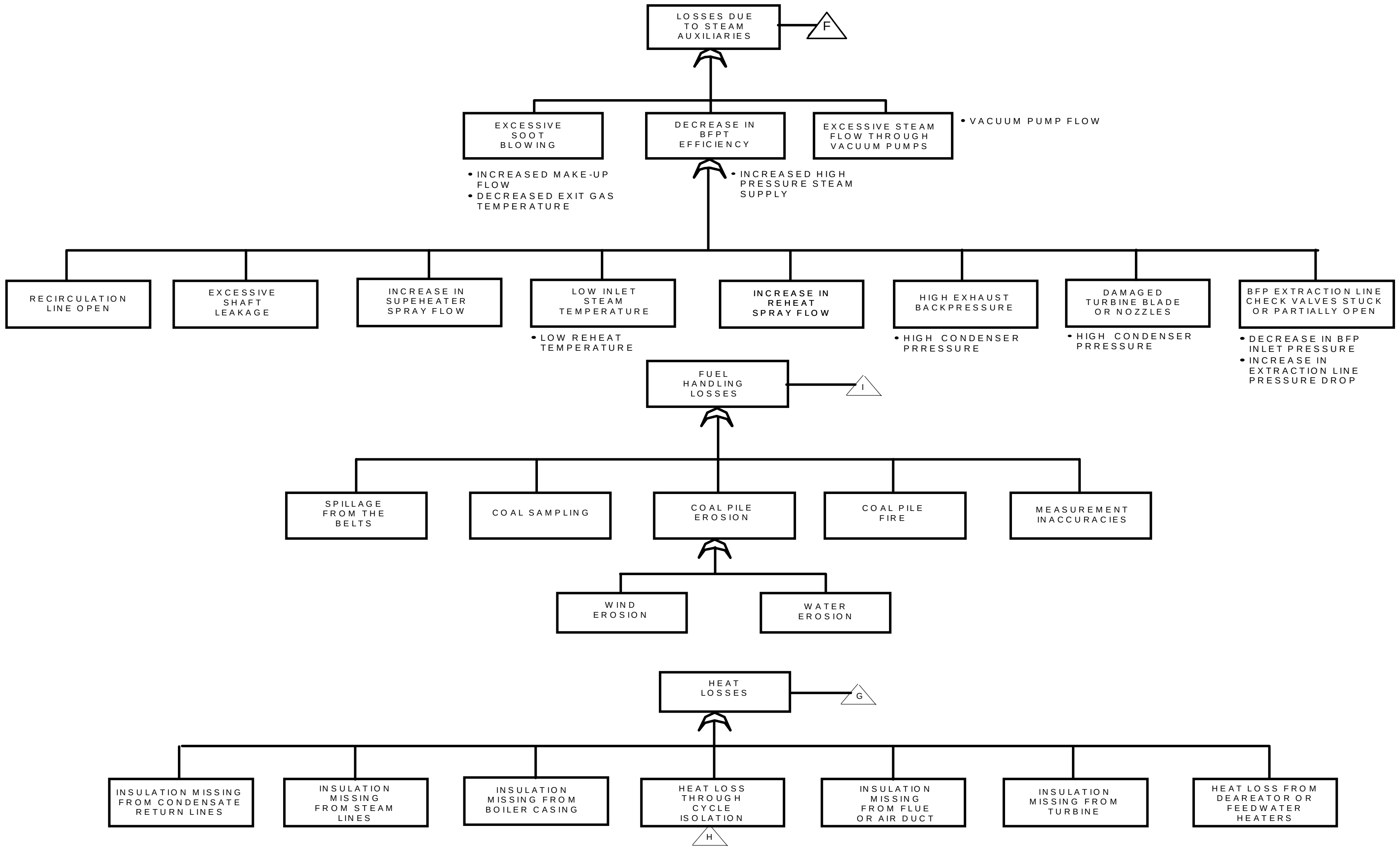


Figure 2-15
Heat Rate Logic Tree - Losses Due to Steam Auxiliaries

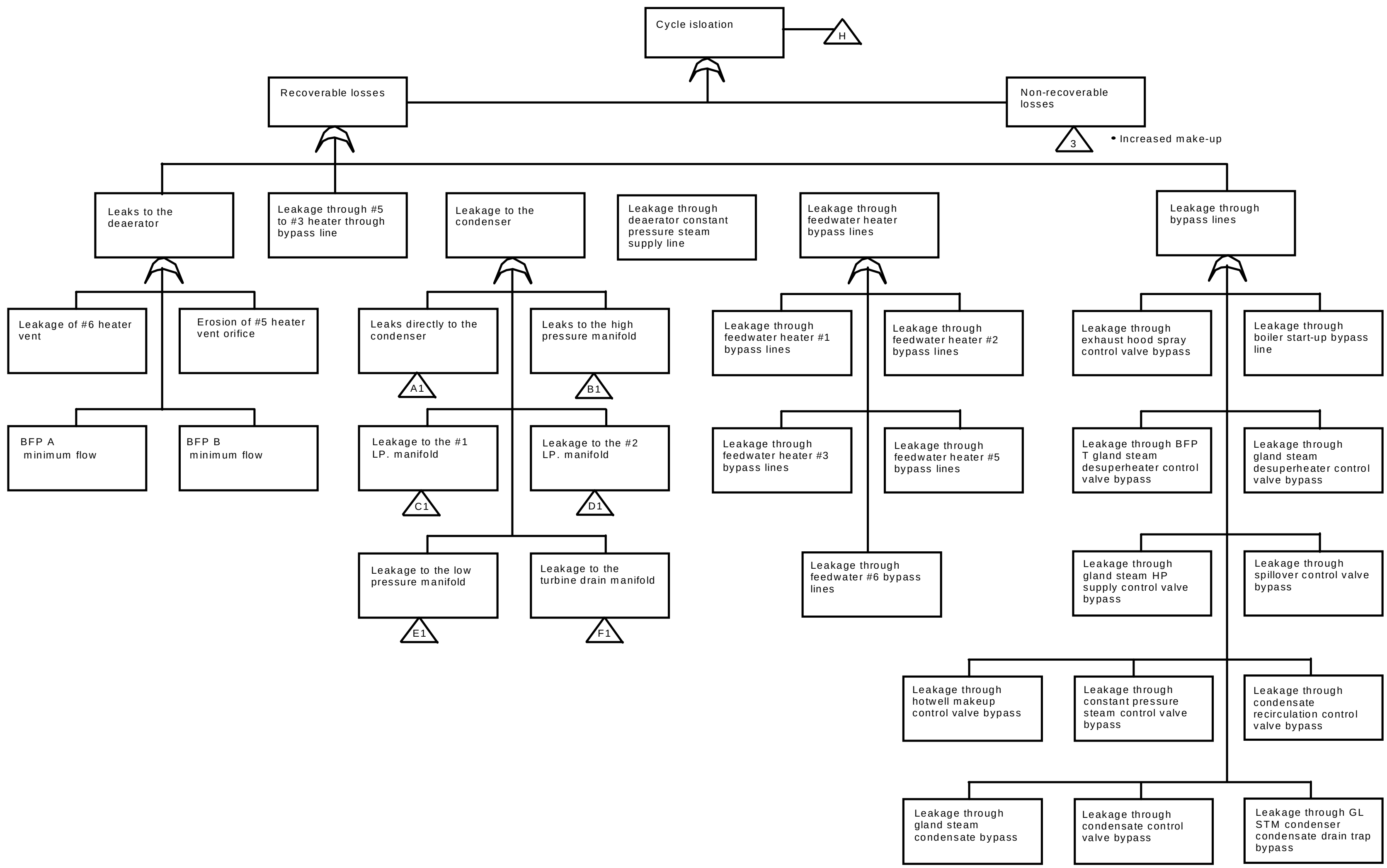


Figure 2-16
Heat Rate Logic Tree - Cycle Isolation

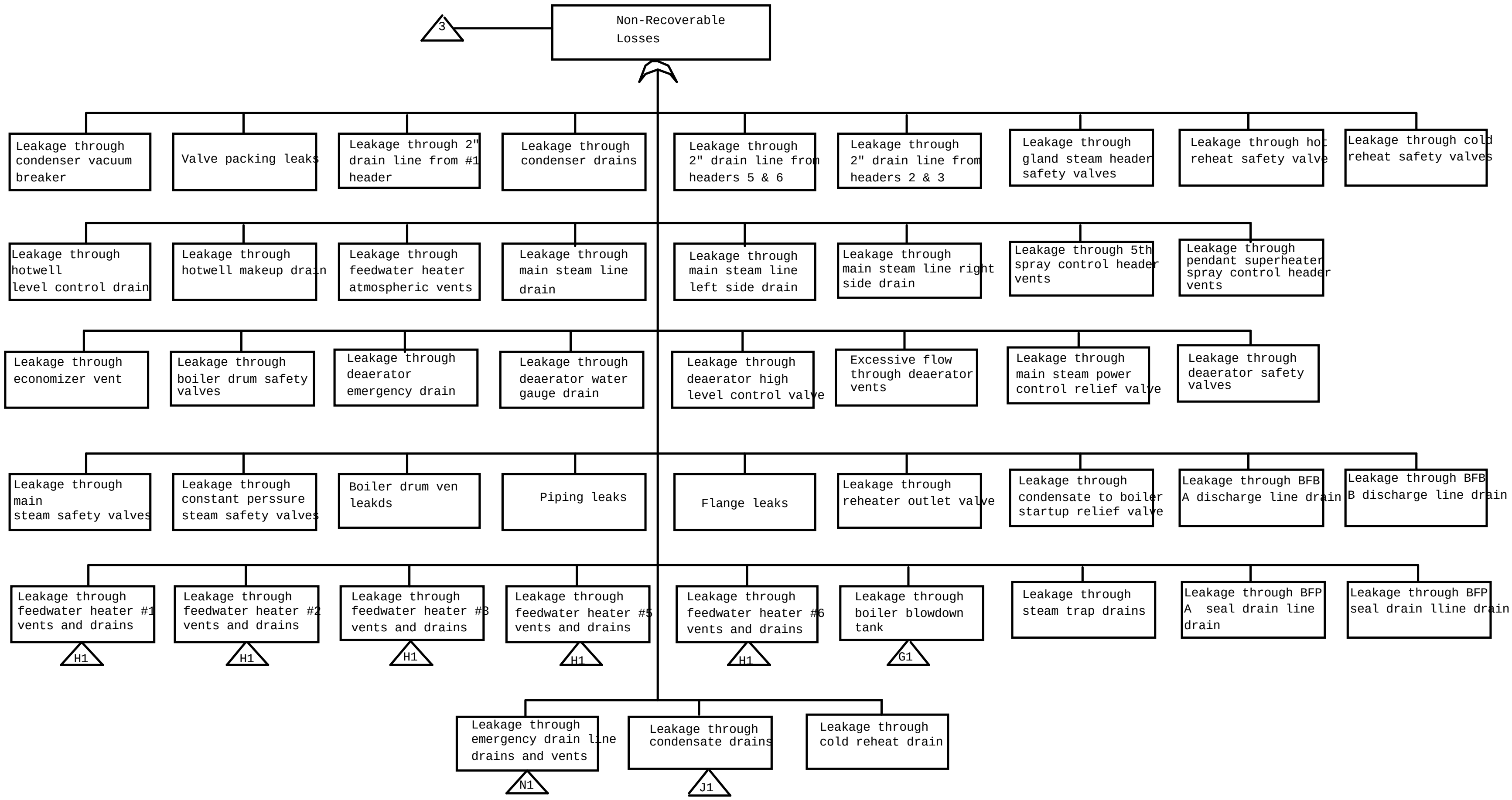


Figure 2-17
Heat Rate Logic Tree - Non-Recoverable Losses (Cycle Isolation Continued)

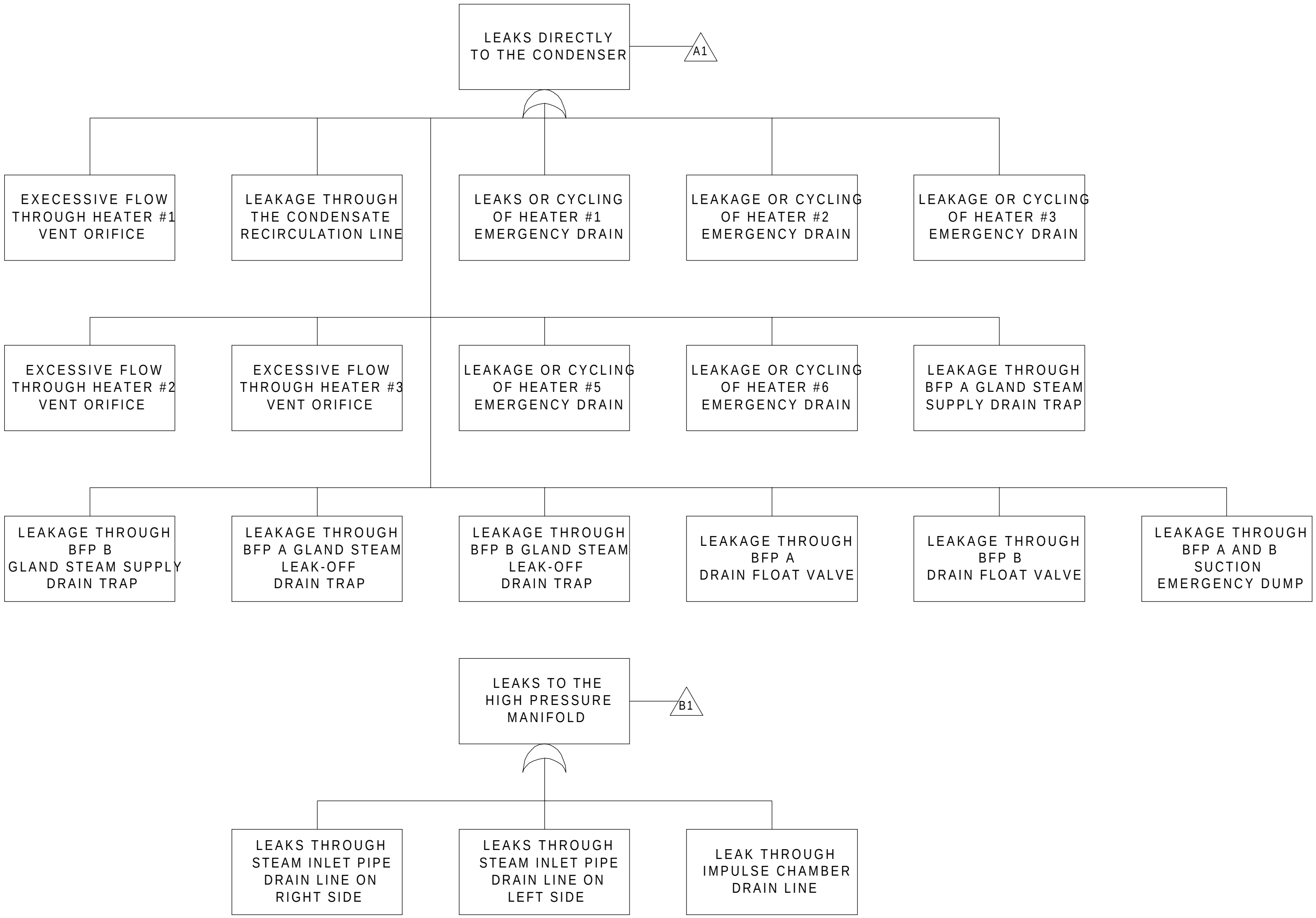


Figure 2-18
Heat Rate Logic Tree - (Cycle Isolation Continued)

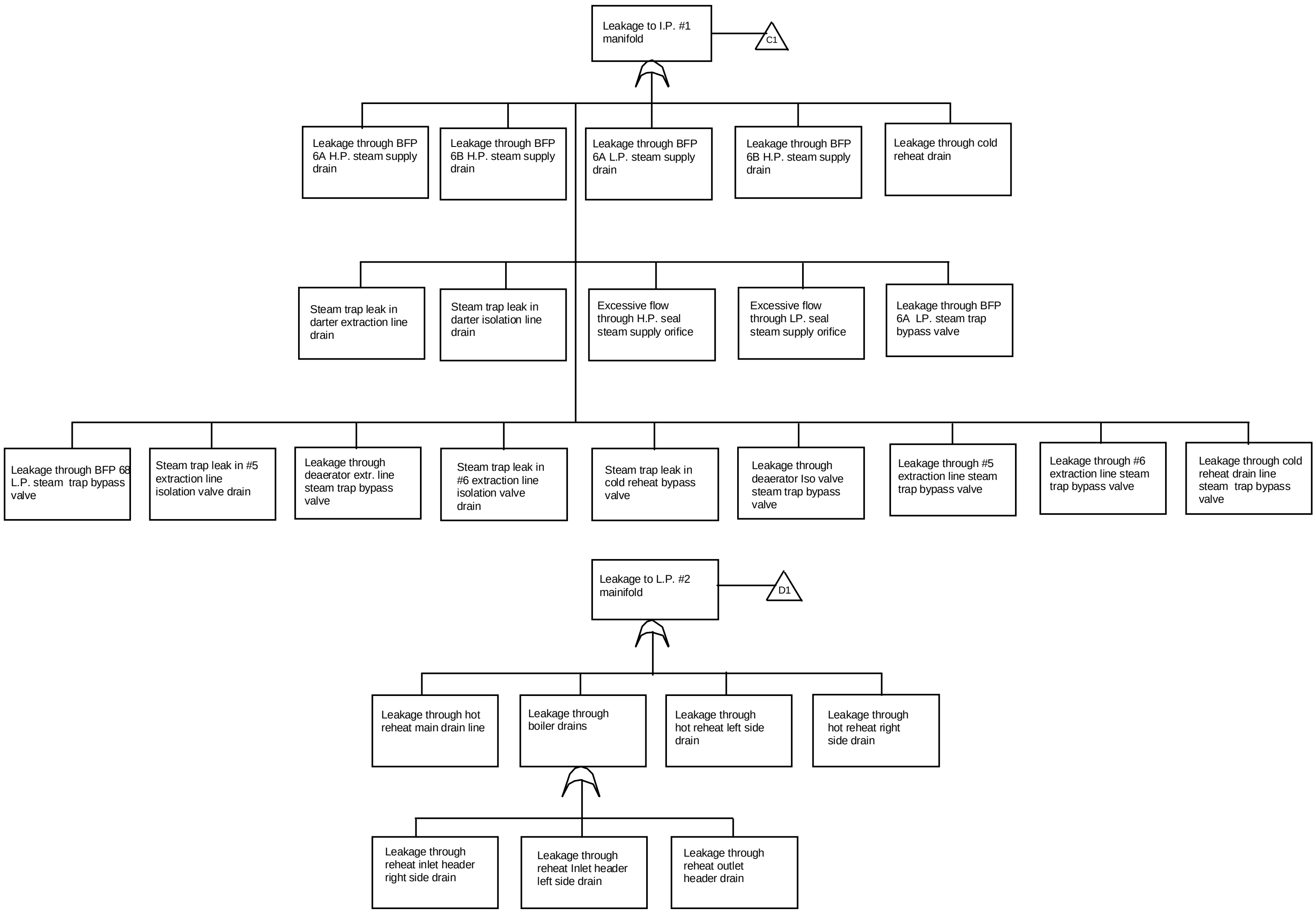


Figure 2-19
Heat Rate Logic Tree - (Cycle Isolation Continued)

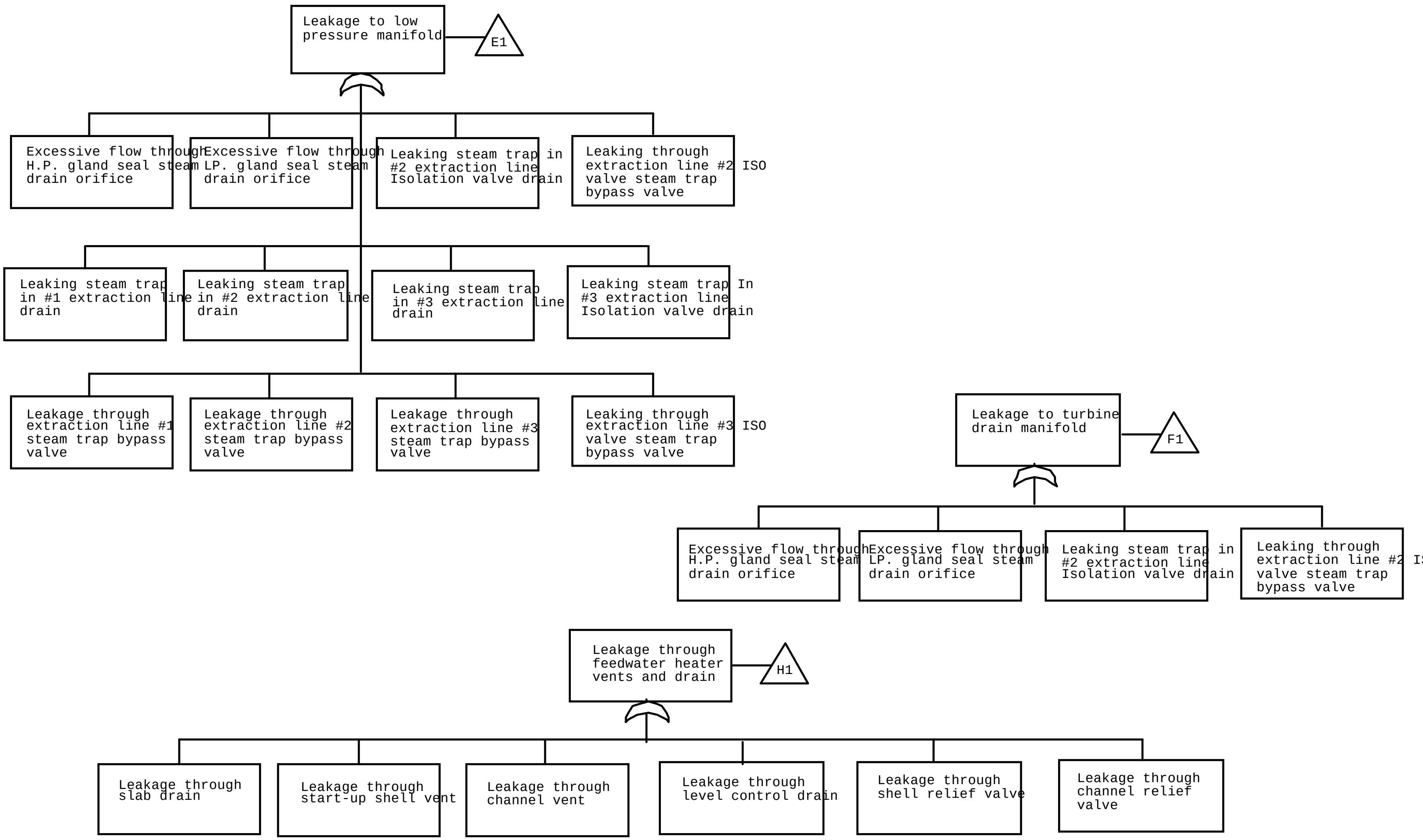


Figure 2-20
Modified Heat Rate Logic Tree - (Cycle Isolation Continued)

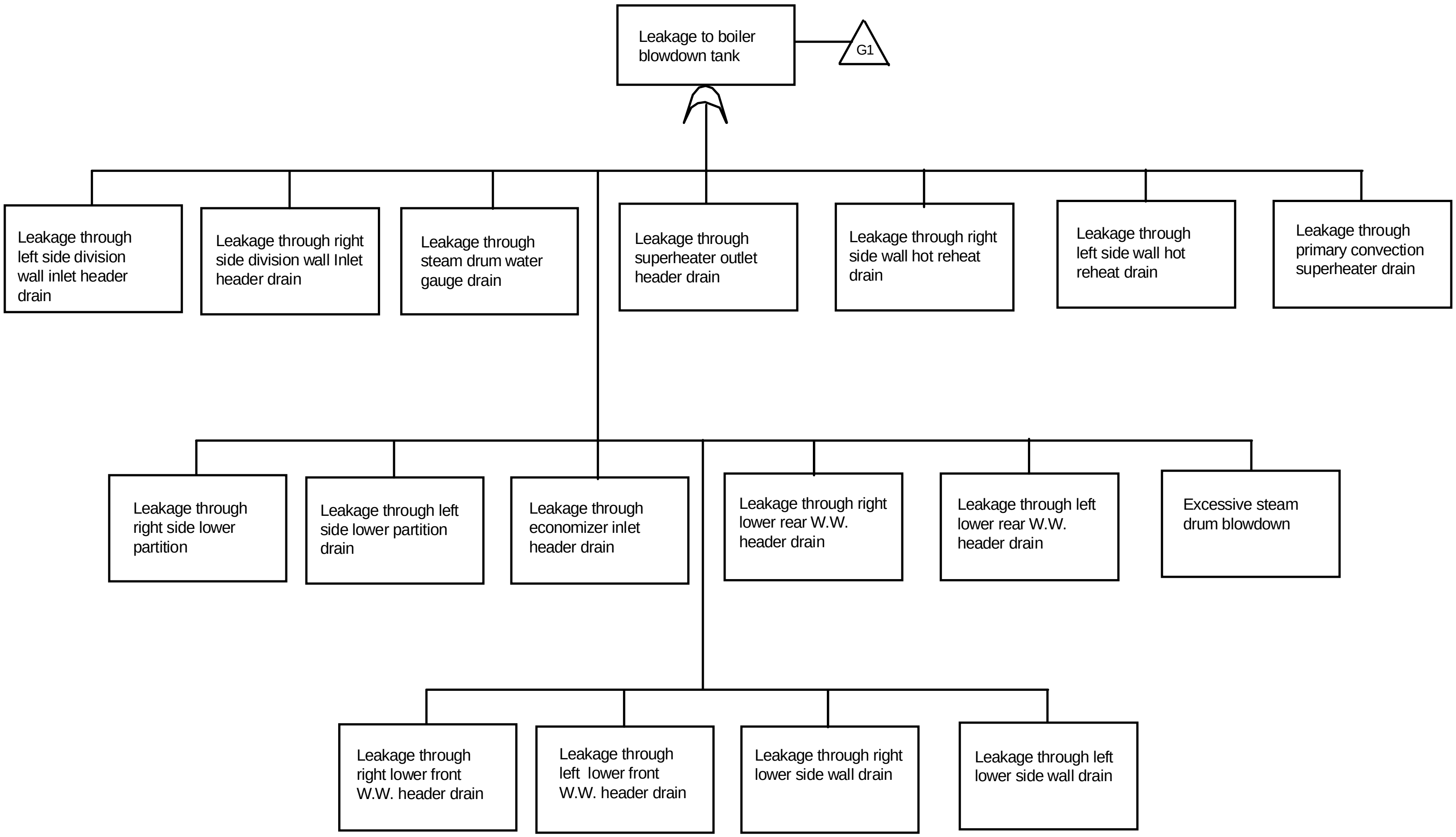


Figure 2-21
Modified Heat Rate Logic Tree - (Cycle Isolation Continued)

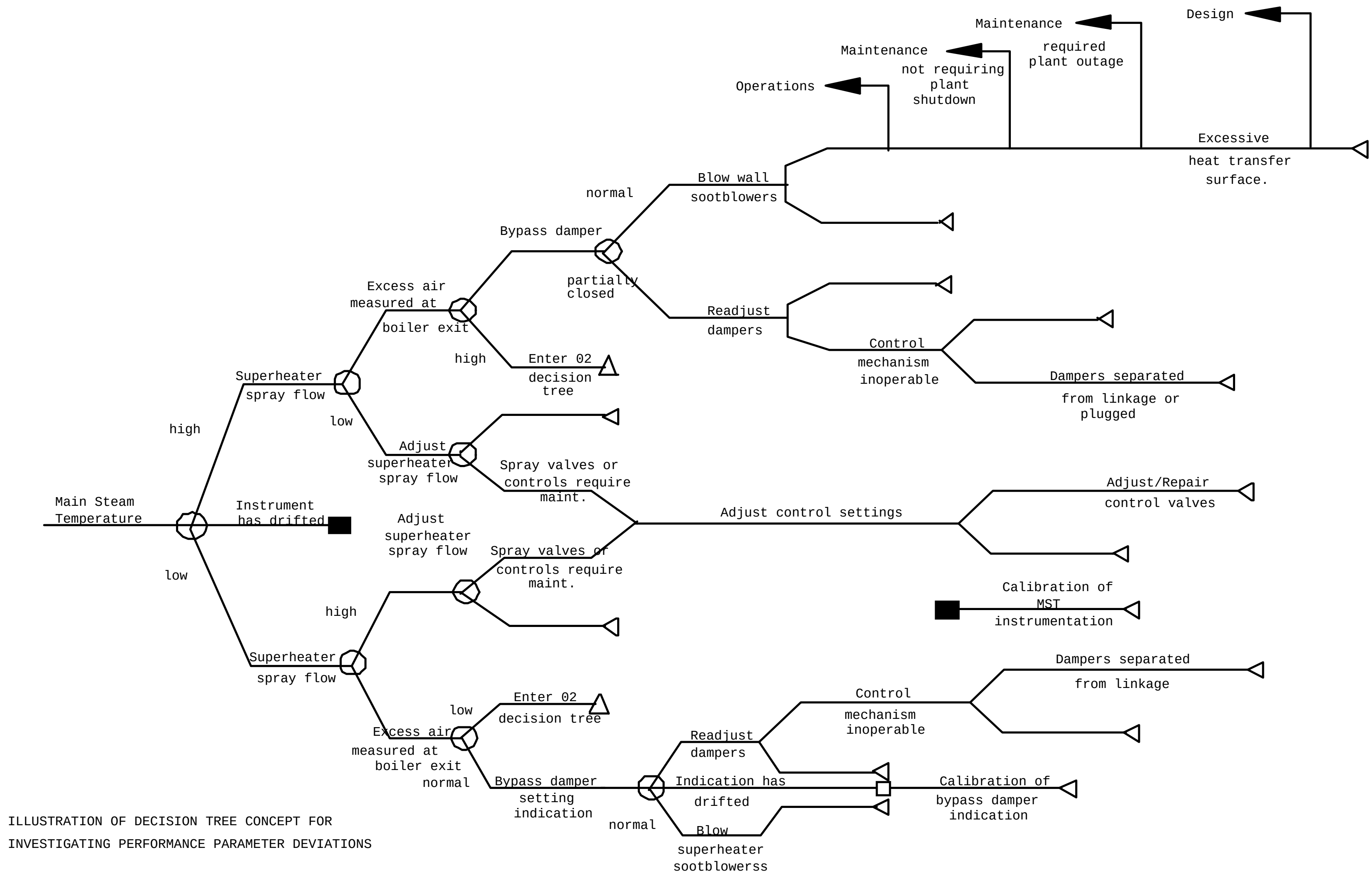


Figure 2-22
Main Steam Temperature Decision Tree

3

FOSSIL STEAM STATION COMPONENTS

Thermal Kits

Thermal kits are a collection of manufacturer turbine generator data in the form of secondary cycle diagrams, curves, equations, and constants. This data is supplied to the purchaser in order to best describe the expected (and/or guaranteed) performance characteristics of the turbine generator.

Purpose

The thermal kits are used primarily for the following functions:

1. Standards for monitoring
2. Data for cycle model verifications and studies
3. Unit net capability calculations
4. Turbine testing calculations
5. Corrected unit cycle heat rate or output computation

Thermal Kit Details

The following discussion covers the definition, explanation, and application(s) of the various items in thermal kits.

Heat Balance

A heat balance is a diagram of the unit's secondary cycle describing the expected conditions at a specific unit power level (% full power) or at a valve point (2, 3, or 4 valves). Turbine-generator vendors supply utilities with guaranteed and expected heat balances. Guarantee heat balances are provided for one load and calculated at a pseudo valve point i.e., calculated as if the guarantee load point coincides with a valve point.

The cycle conditions described on a heat balance typically include:

1. Enthalpies
2. Absolute pressures
3. Fluid temperatures
4. Main steam quality (where applicable)
5. Simplified normal cycle flow paths and flow rates
6. Feedwater and condensate booster pumps enthalpy rises
7. Gross generation
8. Turbine heat rate
9. Cycle expansion end points
10. Fixed mechanical and electrical turbine generator losses
11. Electrical generator conditions
12. Calculation assumptions, units, steam tables used.

There are two enthalpies listed on the turbine at each wet extraction zone which have moisture removal capabilities. The lower value is the enthalpy prior to extraction and moisture removal. The higher value is the enthalpy after moisture removal.

There are also two sets of flow and enthalpy data listed on the extraction lines for extractions with moisture removal. The higher enthalpy and corresponding flow is that of the wet extraction steam component of the total extraction flow. The lower enthalpy and corresponding flow is that of the saturated liquid removed at that extraction zone.

Turbine Heat Rate Curve

The heat rates and loads from the vendor heat balance are used to form the turbine heat rate curve. This expected heatrate curve has been drawn through the locus of valve points. A valve point is an easily repeatable condition at which throttling of inlet steam is minimized. During normal operation the turbine is not controlled at a valve point and an increased efficiency (heatrate) loss occurs because of the increased control valve throttling. The non-valve point heat balances provided by the vendor do not reflect this inefficiency, as the calculated heatrates and heat balances were developed at pseudo valve points with minimum control valve throttling and associated losses.

Expansion Lines/Mollier Diagrams

The steam conditions during expansion through the turbines are illustrated by a plot on a Mollier diagram. A Mollier diagram is a chart that describes Steam table properties (pressure, temperature, quality, enthalpy, and entropy) and their interrelationships. These plots are called expansion lines and their shapes have been defined by vendors to follow certain standard curves. Each wet extraction zone removes excess moisture from the Steam path resulting in an increase in enthalpy causing a step in the plotted expansion line. As unit load decreases, the expansion line shifts to the right due to increased control valve throttling.

The thermal kit expansion lines may be used as a basis for comparison for test results and cycle model verification. If a test data expansion line shifts to the right, there has been a decrease in efficiency of that specific section.

Extraction Pressure vs. Flow

The extraction zone pressure is determined by the turbine flow to the downstream (following) stage(s). Either a graph or equation is supplied that defines the expected Extraction pressure as a function of turbine downstream flow for each extraction and for the first stage shell pressure. There is a constant “Flow Factor" for each extraction point. Flow factors are used in the analysis of turbine test results. An inconsistent flow factor from one run as compared to others, denotes a problem with the test data.

Enthalpies and End Points

Expected stage, extraction, and expansion line end point enthalpies are plotted against flow by some vendors. These reflect the design at the turbine and its accessories. This information is used in benchmarking turbine cycle models.

Leakoffs

Expected flow rates and enthalpies are supplied for the following leakages:

1. Control, Stop, or throttle valve stems and packings

2. Turbine gland and steam seals

This data may be used for Cycle models when no test data is available. These vendor expected values may also be used as the baseline for comparison to monitoring/trend results.

Exhaust Loss Curve

Unrecoverable energy losses occur in the exhaust hood of each LP turbine. The total exhaust loss is plotted as a function of exhaust volumetric flow. The components of the exhaust loss are:

1. Leaving loss
2. Hood loss
3. Turnup loss
4. Shock wave pressure drop loss

The leaving loss is the "wasted" kinetic energy of the steam leaving the last stage. The hood loss results from the pressure drop of the steam passing through the exhaust hood. This loss may be positive or negative depending upon the design of the diffuser in the LP turbine exhaust hood. The turnup loss occurs due to flow instabilities and recirculation found at very low exhaust flows. The shock wave pressure drop loss is due to the shock waves formed at the turbine exhaust when the pressure drop across the last stage is greater than that required for sonic (critical) flow.

The exhaust loss information is used when calculating the used energy end point in determining the gross generation from cycle parameters.

Choked (Limited) Condenser Pressure

Each turbine-generator unit has a specific LP turbine exhaust pressure below which unit performance starts to deteriorate, assuming steady flow and heat cycle conditions remain constant. This pressure is choked (limited) condenser pressure. Choke limited pressure, for any given unit, is nearly a linear function of turbine exhaust end steam flow.

A low pressure turbine derives useful work from the steam passing through it by expanding the steam down to the pressure existing at the exit of the last row of blades. The blade exit pressure is not necessari1y the same as the condenser pressure. The pressure in the condenser is a function of the circulating water temperature for a given heat load. If the blade exit pressure is higher than the condenser pressure, a Positive hood loss exists.

Modern LP turbine hoods are designed to act as diffusers over a wide range of flow, in which the blade exit pressure will be lower than the condenser pressure. In this case the hood loss is negative.

When the condenser pressure decreases from a higher initial value, the turbine blade exit pressure will also decrease. It will continue to do so and as the hood becomes choked it cannot pass any additional volumetric flow as a result of the decreasing blade exit pressure. Up to this point a lower blade exit pressure has meant more work output and consequently a better heat rate.

Further lowering of the exhaust pressure below the point at which the hood becomes choked does not result in an additional lowering of the blade exit pressure. An increase in electrical auxiliary consumption for the operation of additional CCW pumps occurs. At the same time the condensate is cooled to a lower temperature corresponding to the lower exhaust pressure. Additional quantities of steam must be extracted to the lowest pressure heater to heat the condensate. Larger extraction quantities mean less steam flow left in the blade path and consequently less work output. The heat rate therefore increases. The unit output is at a maximum and the heat rate is at a minimum when the hood is just choked. The condenser pressure corresponding to this condition becomes the optimum exhaust pressure. Thus, operation at a condenser pressure lower than the choke limited pressure results in loss of net output and a higher heat rate, assuming that all other cycle parameters remain unchanged.

Turbine Section Efficiency and Effectiveness

Some vendors supply expected turbine section efficiency or effectiveness curves as a function of flow. These are typically not guaranteed but reflect expected performance.

Turbine test results are compared to these curves to determine which sections or components have performance levels that deviate from the norm.

Corrections

In an ideal situation in which the actual cycle steam conditions duplicate the contract heat balance steam conditions, the actual turbine-generator output and heat rate may be compared to guaranteed values. The vendor has supplied information to take into account off-design situations which routinely occur that may mask the true turbine-generator performance. Corrections to heat rate and output may be made for ASME PTC-6 group 1 and 2 corrections such as:

1. Absolute condenser back pressure
2. Main steam or throttle pressure
3. Main steam or throttle temperature
4. Reheat steam temperature
5. Reheater pressure drop
6. Feedwater heater performance
7. Make-up flow rate
8. Generator conditions
9. Condenser - Condensate temperature depression

Refer to ASME PTC-6 for a complete discussion.

Electrical Generator Losses

The total generator loss is a function of power factor, load, hydrogen purity, and hydrogen pressure. The vendor develops the loss relationships through shop tests and computer models.

This loss information is used for calculating the turbine used energy endpoint in order to calculate gross generation from cycle parameters.

Other Information

Other correction curves are available for plant performance calculations, including boiler efficiency curves, condenser performance curves and cooling tower correction curves.

Responsibilities

All engineers working with the performance monitoring program will need access to up-to-date station therma1 kits. The engineer assigned to support that station is responsible to obtain updates or corrections from the vendor.

Boilers

A typical Boiler system consists of the Steam Generator, Coal Pulverizer Mills or other fuel delivery system, Forced Draft Fan, Induced Draft Fan, and Air Preheater. Referring to Figure 3-1 combustion air is drawn from the hottest area of the facility to the Forced Draft Fan. Dampers or speed control on the Forced Draft Fan control air flow for proper combustion. The percent oxygen at the furnace outlet is measured and the air flow controls are trimmed to maintain a percentage excess air. The Induced Draft Fans control the furnace pressure.

The air travels through the Air Preheater to recover heat transferred from the boiler exit gas. The inlet air side of the preheater should be equipped with Steam Coils or a Preheater Bypass Duct arrangement to prevent moisture from forming in the Preheater. Vapor condenses on surfaces below its dewpoint of 250 to 300 °F(121 to 149 °C), but most Preheaters are designed to operate at minimum metal temperatures below the acid dewpoint , where the efficiency gained more than balances the additional maintenance. Figure 3-2 shows the recommended minimum temperatures. Preheated air is supplied to the Pulverizer Mills as primary combustion air and to the Boiler Windbox Dampers as secondary air.

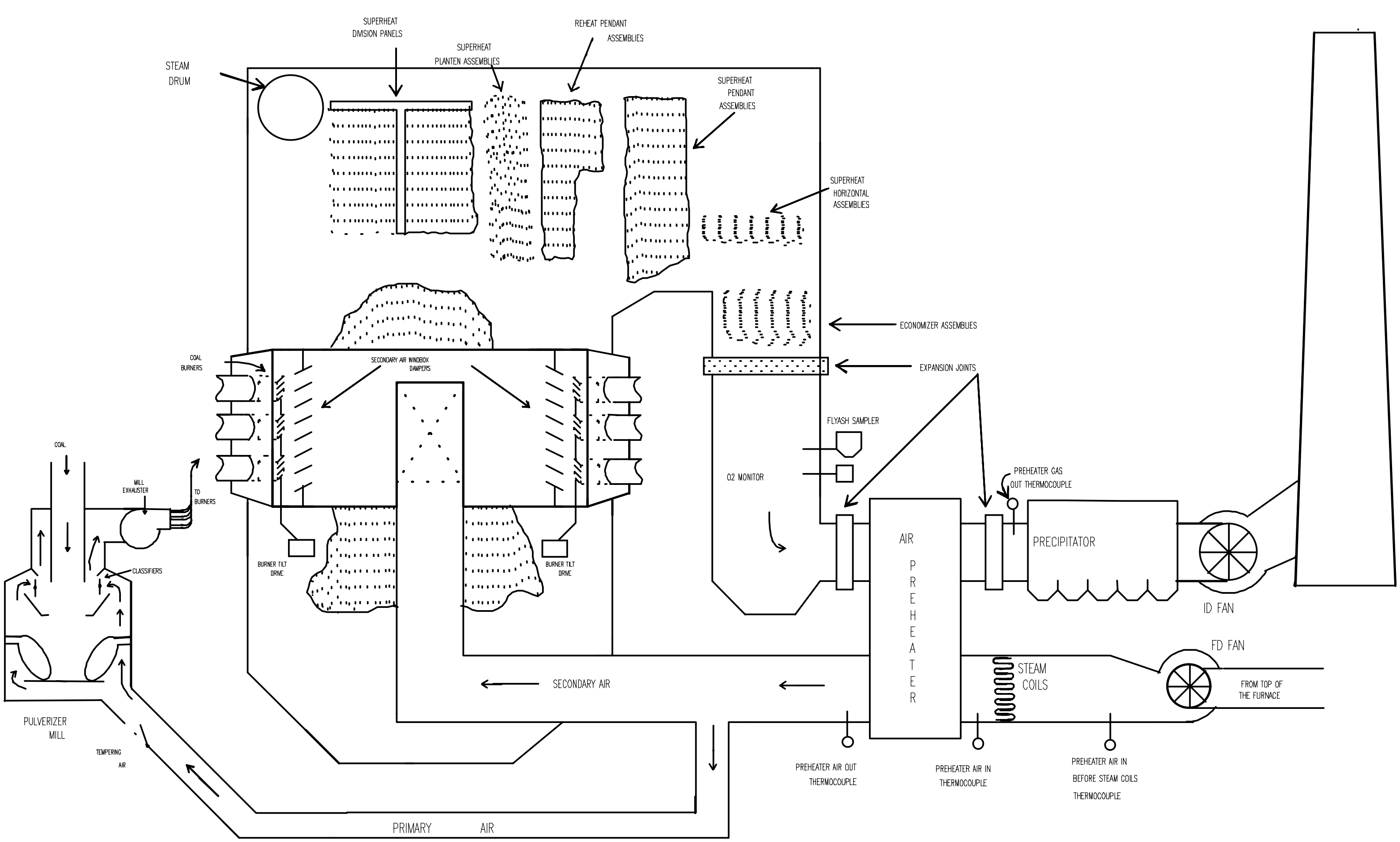


Figure 3-1
Typical Boiler System

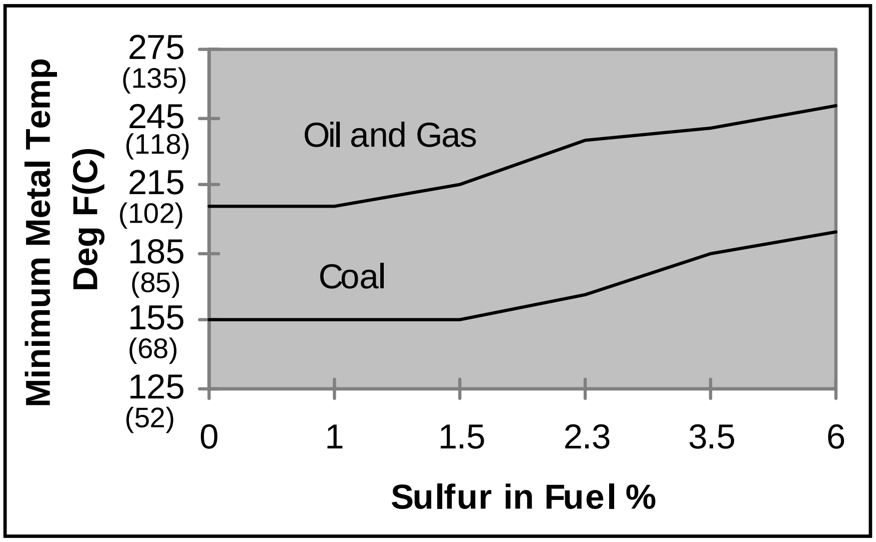


Figure 3-2
Regeneration Air Heater Cold End Minimum Metal Temperature Limits When Burning Sulfur Bearing Fuels

The primary air is mixed with tempering air at the Pulverizer to control the Mill temperature above a point to assure stable combustion and to thoroughly dry the coal, but below the point where there will be a risk of Pulverizer fires. The temperature range will be from 150 to 210 °F(65.5 to 99 °C) depending on the volatility of the coal. The coal is exhausted from the Pulverizer Mills through classifiers which are adjusted to control the fineness of the pulverized coal and transported to the burner elevations on the boiler where it is mixed with the secondary air for combustion.

If these processes are not monitored and controlled properly performance losses can occur such as Dry Gas Loss, Unburned Carbon Loss, Combustion Air Moisture Loss, and Radiation Loss. Below are listed causes with possible solutions for these performance losses.

Dry Gas Loss

High excess air and lower heat absorption in the boiler system can cause exit gas temperatures higher than expected resulting in Dry Gas Loss. A 40 °F(22.2 °C) rise in exit gas temperature can raise the Heat Rate by one percent. High exit gas temperatures can be caused by:

- Plugging or fouling of preheaters can occur on the hot side but is more common on the cold side where moisture has formed due to reaching the dewpoint. In addition to raising the exit gas temperatures, plugging can lead to load reductions if fan capacity is exceeded.

Ensure proper soot blowing for preheater. Dry superheated steam is normally used as a cleaning medium. The piping system should include automatic drain valves to insure that the steam temperature is adequate before admitting to the preheater. Compressed air can be used but its cleaning capabilities are not as good as dry steam.

Periodic high pressure washing may be necessary if the pressure drop across the preheater starts to limit fan capability.

- Corroded or eroded Preheater

  During the process of combustion, sulfur in the fuel is converted to SO_2 and depending on the excess air available, part of the SO_2 is converted to SO_3. The SO_3 reacts with any water vapor present in the preheater to form sulfuric acid (H_2SO_4). Ensure proper operation of steam coils or preheater bypass damper to keep the preheater above the temperature shown in fig 3-2. The average of the preheater inlet air and the preheater exit gas temperatures is used to determine the preheater metal temperature. If steam coils are used, perform periodic inspections for leaks which would increase water vapor to the preheater. Open or close plant windows and doors to circulate outside air through plant and around boiler to the Forced Draft Fan suction to minimize steam coil usage. Incoming oxygen-rich air, as a result of air inleakage, tends to increase the rate of acid deposition, increasing the corrosion potential.

- Inadequate boiler soot blowing can cause slag to build on heat-absorbing surfaces and proper heat transfer cannot occur. Severely plugged sections can restrict gas flow and tube erosion due to high local velocities can occur.

  Ensure proper operation of boiler soot blowing

- Air Preheater average air in-gas out temperature too high above dewpoint

  Excessive use of steam coils or bypass dampers

- Incorrect number of pulverizer mills in service at a given load causing an increase in tempering air. Increasing the tempering air decreases the percentage of total air flow which goes through the preheater, thus raising exit gas temperatures.

  Remove pulverizer mills from service as soon as possible when reducing unit load

- Excess pulverizer mill tempering air causing low mill temperature

Raise mill temperature controls setpoint

- Fuel/Air control system

  Maintain O_2 as low as possible without adversely affecting combustion. An increase in gas weight to the preheater will result in a higher exit gas temperature.

- Improper O_2 monitoring system

  Calibrate or repair O_2 monitors

  Ensure location and quantity of O_2 monitors gives a representative indication

- Air Inleakage in boiler, preheater, or ducts

  Run O_2 rise test on boiler to locate air inleakage and make repairs. Air inleakage in the furnace, boiler ducts, expansion joints, or preheater can adversely affect the heat transfer, give false indications of percent O_2, and increase fan auxiliaries. Incoming oxygen rich air also tends to increase the rate of acid diposition increasing the corrosion potential. O_2 readings should be taken at several locations simultaneously to isolate cause of air inleakage.

Dry Gas Loss Calculations

The following formulas are used in determining the effect of Dry Gas Loss on the unit Heat Rate.

To calculate dry gas loss the amount of dry gas per pound of carbon burned is needed. First calculate the amount of unburned carbon which is a function of the percent ash in the coal and the percent combustible in the flyash.

$$UBC = \frac{A}{100 - CR} - \frac{A}{100} \qquad \text{(eq. 3-1)}$$

where: UBC = unburned carbon per lb(kg) of coal burned (mass/unit mass)

A = percent ash in coal as fired (%)

CR = percent combustible in refuse (%)

$$UBC_x = \frac{A}{100 - CR_x} - \frac{A}{100} \qquad \text{(eq. 3-2)}$$

where: UBC_x = expected unburned carbon per lb(kg) of coal burned (mass/unit mass)

A = percent ash in coal as fired (%)

CR_x = expected percent combustible in refuse (%)

Carbon burned per pound of coal burned is calculated by subtracting the percent unburned carbon from the percent carbon in the coal.

$$CB = \frac{C}{100} - UBC \qquad \text{(eq. 3-3)}$$

where: CB = carbon burned per lb(kg) of coal burned (mass/unit mass)

C = percent carbon in coal as fired (%)

UBC = unburned carbon per lb(kg) of coal burned (mass/unit mass)

$$CB_x = \frac{C}{100} - UBC_x \qquad \text{(eq. 3-4)}$$

where: CB_x = expected carbon burned per lb(kg) of coal burned (mass/unit mass)

C = percent carbon in coal as fired (%)

UBC_x = expected unburned carbon per lb(kg) of coal burned(mass/unit mass)

The preheater exit gas temperature used in the dry gas loss calculation is corrected for air inleakage, that is, what would the exit gas temperature be if it had not been cooled by air inleakage.

Preheater outlet O_2 is calculated by adding the last O_2 rise obtained to the measured O_2.

$$O_2 = O_{2f} + O_{2r} \qquad \text{(eq. 3-5)}$$

where: O_2 = oxygen at preheater outlet (% O_2)

O_{2f} = oxygen at economizer outlet (%O_2)

O_{2r} = preheater O_2 rise

The air leakage across the preheater is calculated from the preheater O_2 rise divided by the CO_2+O_2 value minus the O_2 at the preheater outlet. The multiplier of 90 is from ASME PTC 4.3 and is used to correlate theoretical value with empirical data.

$$AL = \frac{O_{2r}}{K_p - O_2} \times 90 \quad \text{(eq. 3-6)}$$

where: AL = air leakage across preheater (%)

O_{2r} = preheater O_2 rise (% O_2)

K_p = CO_2 +O_2 at preheater outlet (%)

O_2 = oxygen at preheater outlet (% O_2)

$$T_{goiac} = \frac{((T_{aix} \times T_{fg}) - T_{go}) + (T_{fg} \times (T_{go} - T_{ai}))}{(T_{fg} - T_{ai})} \quad \text{(eq. 3-7)}$$

where: T_{goiac} = exit gas temperature corrected for contract air temperature

T_{aix} = guarantee inlet air temp

T_{fg} = flue gas temperature

T_{go} = uncorrected exit gas temperature

T_{ai} = actual preheater inlet air temperature after steam coils

$$T_{gonlc} = \frac{AL(T_{goiac} - T_{aix})}{100} + T_{goiac} \quad \text{(eq. 3-8)}$$

where: T_{gonlc} = exit gas temperature corrected for no leakage

AL = air leakage across preheater (%)

T_{goiac} = exit gas temperature corrected for contract air temperature

T_{aix} = guarantee inlet air temp

The expected exit gas temperature is calculated from curve fits of vendor information and is based on steam flow.

$$T_{gox} = A + B(F_s) + C(F_s)^2 \qquad \text{(eq. 3-9)}$$

where: T_{gox} = expected exit gas temperature with no leakage

F_s = corrected throttle steam flow

A, *B*, *C* are curve fit coefficients

Total dry gas per pound of fuel is calculated from *Boiler Performance Analysis*, by Babcock & Wilcox, Page 17, equation 27.

$$DG = \frac{1.33(k_e) - O_{2f} + 233.3}{K_e - O_{2f}} \times (CB + \frac{S}{267}) \qquad \text{(eq. 3-10)}$$

where: DG = dry gas per lb(kg) of fuel burned (mass/unit mass)

K_e = $CO_2 + O_2$ at economizer outlet (%)

O_{2f} = oxygen at economizer outlet (%O_2)

CB = carbon burned per lb(kg) of coal burned (mass/unit mass)

S = percent sulfur in coal as fired (%)

$$DG_x = \frac{1.33(k_e) - O_{2x} + 233.3}{K_e - O_{2f}} \times (CB_x + \frac{S}{267}) \qquad \text{(eq. 3-11)}$$

where: DG_x = expected dry gas per lb(kg) of coal burned (mass/unit mass)

K_e = $CO_2 + O_2$ at economizer outlet (%)

O_{2x} = expected percent O_2 at economizer outlet

O_{2f} = oxygen at economizer outlet (%O_2)

CB = carbon burned per lb(kg) of coal burned (mass/unit mass)

S = percent sulfur in coal as fired (%)

$$DGL = \frac{DG(24)(T_{gonlc} - T_{aix})}{HHV} \qquad \text{(eq. 3-12)}$$

where: DGL = dry gas loss (%)

DG = dry gas per lb(kg) of coal burned (mass/unit mass)

T_{gonlc} = exit gas temperature corrected for no leakage

T_{aix} = guarantee inlet air temp

HHV = higher heating value of coal as fired

$$DGL_x = \frac{DG_x(24)(T_{gox} - T_{aix})}{HHV} \qquad \text{(eq. 3-13)}$$

where: DGL_x = expected dry gas loss (%)

DG_x = expected dry gas per lb(kg) of coal burned (mass/unit mass)

$\mathrm{T}_{\mathrm{gox}}$ = expected exit gas temperature corrected for no leakage

T_{aix} = guarantee inlet air temp

HHV = higher heating value of coal as fired

$$DGL_d = \frac{DGL - DGL_x}{BE_x} \qquad \text{(eq. 3-14)}$$

where: DGL_d = dry gas loss deviation (%)

DGL = dry gas loss (%)

DGL_x = expected dry gas loss (%)

BE_x = expected boiler efficiency

$$DGL_e = \frac{DGL_d}{100 - DGL_d} \times 100 \qquad \text{(eq. 3-15)}$$

where: DGL_e = percent effect on heat rate due to Dry Gas Loss (%)

DGL_d = dry gas loss deviation (%)

Unburned Carbon Loss

A flyash sample should be collected from the furnace outlet and analyzed in the laboratory for unburned carbon. There are products on the market which can give an online measurement proportional to the amount of carbon in the ash so that operators can see the results of their adjustments. The percent unburned carbon is also called Loss On Ignition or LOI. High Carbon Loss can usually be traced to:

- Improper excess air in furnace

  Ensure O_2 setpoint is high enough for proper combustion. Care should be taken to not raise O_2 too far such that the Dry Gas Loss increases more than the Carbon Loss decreases.

- Poor mixing process of the fuel and air in the furnace

  Check operation of secondary air dampers and confirm the unit is operating with proper windbox to furnace differential pressure. Check condition of burners and confirm that range of burner tilt operation is not adversely affecting performance.

- Pulverized coal fineness is incorrect

  Collect coal samples from Pulverizer Mills and analyze for fineness. Adjust Mill classifiers for proper fineness. Table 3-1 lists desired fineness results while table 3-2 shows coal properties for various coal ranks.

- High surface moisture in the coal can lead to agglomeration and have the same effect as coarse coal during the combustion process because the surface area is reduced.

  Ensure that the proper amount of hot air is supplied to the pulverizer to remove the surface moisture.

Table 3-1
Desired Fineness for Various Coals

Coal Rank	Passing 200 Mesh (wt %)	Retained on 50 Mesh (wt %)
Subbituminous C coal and Lignite	60 - 70	2.0
High Volatility Bituminous C, and Subbituminous A and B	65 - 72	2.0
Low and Medium Volatility Bituminous. High Volatility Bituminous A and B	70 - 75	2.0

Table 3-2
Coal Properties

	Bituminous					Subbituminous			Lignite
	Low Volatile	Med Volatile	High Volatile A	High Volatile B	High Volatile C	A	B	C	A
Agglomerating character	Agg	Agg	Agg	Agg	Agg but not caking	Non-Agg	Non-Agg	Non-Agg	Non-Agg
Moisture (seam)	2.0	2.0	4.0	7.0	10.0	14.0	19.0	25.0	40.0
Volatile matter	21.1	32.3	38.4	33.8	35.9	35.3	34.5	25.8	25.9
Fixed carbon	68.6	55.8	51.5	47.3	43.3	41.2	37.5	40.9	27.4
Ash	8.3	9.3	6.1	11.9	10.8	9.5	9.9	8.3	6.7
HHV, Btu/lb, As-Fired	13,150	13,210	13,410	11,610	10,590	9,840	8,560	7,500	5,940

Carbon Loss Calculations

Carbon Loss percent effect on the unit heat rate is calculated using the following formulas.

$$UBC = \frac{A}{100 - CR} - \frac{A}{100} \qquad \text{(eq. 3-16)}$$

where: UBC = unburned carbon per lb(kg) of coal burned (mass/unit mass)

A = percent ash in coal as fired (%)

CR = percent combustible in refuse (%)

$$UBC_x = \frac{A}{100 - CR_x} - \frac{A}{100} \qquad \text{(eq. 3-17)}$$

where: UBC_x = expected unburned carbon per lb(kg) of coal burned (mass/unit mass)

A = percent ash in coal as fired (%)

CR_x = expected percent combustible in refuse (%)

Carbon loss is calculated from *Boiler Performance Analysis* by Babcock & Wilcox, Page 19, equation 36.

$$CL = \frac{(UBC)(1{,}450{,}000)}{HHV} \qquad \text{(eq. 3-18)}$$

where: CL = carbon loss (%)

UBC = unburned carbon per lb(kg) of coal burned (mass/unit mass)

HHV = higher heating value of coal as fired, Btu/lb., (kJ/kg)

$$CL_x = \frac{(UBC_x)(1{,}450{,}000)}{HHV} \qquad \text{(eq. 3-19)}$$

where: CL_x = expected carbon loss (%)

UBC_x = expected unburned carbon per lb(kg) of coal burned (mass/unit mass)

HHV = higher heating value of coal as fired, Btu/lb., (kJ/kg)

$$CL_d = \left(\frac{CL - CL_x}{BE_x}\right) \times 100 \qquad \text{(eq. 3-20)}$$

where: Cl_d = carbon loss deviation (%)

CL = carbon loss (%)

CL_x = expected carbon loss (%)

BE_x = expected boiler efficiency

$$CL_e = \frac{CL_d}{100 - CL_d} \times 100 \quad \text{(eq. 3-21)}$$

where: Cl_e = percent effect on heat rate due to carbon loss

Cl_d = carbon loss deviation (%)

Moisture Loss

Boiler efficiency calculations use the Higher Heating Value or HHV as the amount of heat generated in complete combustion of the fuel. The HHV of the fuel is determined by assuming heat is absorbed by the boiler system to return the combustion products back to their initial temperature, thus any moisture in the fuel or moisture formed by hydrogen in the fuel during the combustion process would be in the condensed state. Since the combustion products are not returned to their initial temperature the latent heat of water vapor from moisture in the fuel is not available for making steam. These losses are accounted for in boiler efficiency calculations as Moisture Loss and Hydrogen Loss.

Since these losses are related to the percent moisture and hydrogen in the fuel, improvements can only be made by minimizing the exit gas temperatures, as discussed under the Dry Gas Loss section.

Moisture Loss Calculations

$$MFL = \frac{M\left[0.458(T_{gonlc}) + 990.483\right]}{HHV} \quad \text{(eq. 3-22)}$$

where: MFL = moisture in fuel loss (%)

M = percent moisture in coal as fired (%)

T_{gonlc} = exit gas temperature corrected for no leakage, °F

HHV = higher heating value of coal as fired, Btu/lbm

$$MFL_x = \frac{M\left[0.458(T_{gox}) + 990.483\right]}{HHV} \quad \text{(eq. 3-23)}$$

where: MFL_x = expected moisture in fuel loss (%)

M = percent moisture in coal as fired (%)

T_{gox} = expected exit gas temperature, °F

HHV = higher heating value of coal as fired, Btu/lbm

$$HL = \frac{9(H)\left[0.458(T_{gonlc}) + 990.483\right]}{HHV} \quad \text{(eq. 3-24)}$$

where: HL = hydrogen loss (%)

H = percent hydrogen in coal as fired (%)

T_{gonlc} = exit gas temperature corrected for no leakage. °F

HHV = higher heating value of coal as fired, Btu/lbm

$$HL_x = \frac{9(H)\left[0.458(T_{gox}) + 990.483\right]}{HHV} \quad \text{(eq. 3-25)}$$

where: HL_x = expected hydrogen loss (%)

H = percent hydrogen in coal as fired (%)

T_{gox} = expected exit gas temperature, °F

HHV = higher heating value of coal as fired, Btu/lbm

$$MFL_d = \left(\frac{MFL - MFL_x}{BE_x}\right) \times 100 \quad \text{(eq. 3-26)}$$

where: MFL_d = moisture in fuel loss deviation (%)

MFL = moisture in fuel loss (%)

MFL_x = expected moisture in fuel loss (%)

BE_x = expected boiler efficiency

$$HL_d = (\frac{HL - HL_x}{BE_x}) \times 100 \qquad \text{(eq. 3-27)}$$

where: Hl_d = hydrogen loss deviation

HL = hydrogen loss (%)

HL_x = expected hydrogen loss (%)

BE_x = expected boiler efficiency

$$MFL_e = (\frac{MFL_d}{100 - MFL_d}) \times 100 \qquad \text{(eq. 3-28)}$$

where: MFL_e = percent effect on heat rate due to moisture in fuel loss deviation

MFL_d = moisture in fuel loss deviation (%)

$$HL_e = (\frac{HL_d}{100 - HL_d}) \times 100 \qquad \text{(eq. 3-29)}$$

where: HL_e = percent effect on heat rate due to moisture in fuel loss deviation

HL_d = moisture in fuel loss deviation (%)

Radiation and Unaccounted for Losses

There are a number of Boiler System losses which are not normally based on test measurements. These can be grouped under a heading called Radiation , Unaccounted for, and Moisture in Air Losses or RUMA.

Radiation losses account for heat losses to the air through conduction, radiation, and convection. The heat emanates from the boiler, ductwork, and pulverizers. If the unit is equipped with hot side precipitators, they can be a source of significant gas temperature drop. Partial recovery from radiation losses can be accomplished by opening and closing doors and windows to circulate Forced Draft Fan suction air flow across the plant equipment as discussed under Dry Gas Loss.

Unaccounted for losses include difficult to measure losses that are included in a heat balance to arrive at a guaranteed efficiency. These include heat lost in the ash leaving the furnace through the bottom ash hoppers and economizer hoppers and any apparent losses due to instrumentation errors.

Moisture in air losses are due to not returning the combustion air to its inlet temperature thus losing the latent heat of water vapor as discussed under Moisture Losses.

RUMA Loss Calculations

RUMA losses are calculated from curve fit data in the boiler contract and is based on steam flow. This is an expected value since it is not possible to accurately measure these losses.

$$RUMA = A + B(F_s) + C(F_s)^2 \qquad \text{(eq. 3-30)}$$

where: $RUMA$ = radiation, unaccounted for, and moisture in air losses (%)

F_s = corrected throttle steam flow

A, B, and C are curve fit coefficients for the particular Steam Generator

Boiler efficiency is the sum of all the actual losses subtracted from 100%

$$BE = 100 - DGL - MFL - HL - CL - RUMA \qquad \text{(eq. 3-31)}$$

Note: Note that this equation (3-28) for boiler efficiency is not the same as ASME PTC 4, but is practical and reasonable for routine testing.

where: BE = boiler efficiency (%)

DGL = dry gas loss (%)

MFL = moisture in fuel loss (%)

HL = hydrogen loss (%)

CL = carbon loss (%)

$RUMA$ = radiation, unaccounted for, and moisture in air losses (%)

$$BE_x = 100 - DGL_x - MFL_x - HL_x - CL_x - RUMA \quad \text{(eq. 3-32)}$$

where: BE_x = expected boiler efficiency (%)

DGL_x = expected dry gas loss (%)

MFL_x = expected moisture in fuel loss (%)

HL_x = expected hydrogen loss (%)

CL_x = expected carbon loss (%)

$RUMA$ = radiation, unaccounted for, and moisture in air losses (%)

$$BL_d = \left(\frac{BE_x - BE}{BE_x}\right) \times 100 \quad \text{(eq. 3-33)}$$

where: BL_d = total boiler loss deviation (%)

BE_x = expected boiler efficiency (%)

BE = boiler efficiency (%)

$$BE_e = \left(\frac{BE_d}{100 - BE_d}\right) \times 100 \quad \text{(eq. 3-34)}$$

where: BE_e = percent effect on heat rate due to total of boiler loss deviation (%)

BE_d = total boiler loss deviation (%)

Turbine

Steam is admitted through control valves to the turbine where the thermal energy is converted to kinetic energy and then to mechanical energy by expansion through the turbine sections. There will normally be a high pressure section, a low pressure section, and possibly an intermediate pressure or reheat section. Incorporating a reheat section in the turbine design can improve the thermal performance by 5 percent. As the steam enters the turbine it travels through stationary nozzles and rotating turbine blades or buckets alternately. The stationary nozzles organize the steam flow into high speed jets which are directed to the buckets.

For maximum efficiency the turbine stages contain a combination of impulse bucket and reaction blade designs. These designs are characterized by how the energy is extracted from the steam. Impulse nozzles orient the steam so it flows in well formed high speed jets containing kinetic energy which the moving buckets convert into shaft rotation, or mechanical energy, as the steam changes direction. The pressure drop occurs across the stationary nozzles only.

In reaction stages the stationary nozzles and rotating blades are similar in design, that is the blades are not bucket shaped. Steam pressure drops in passing through both moving and stationary blading. Steam velocity rises through the stationary nozzles and falls in the moving blades.

The method used to control the steam flow to the turbine at various loads affects the plant performance. Partial arc admission can be used where the control valves are throttled successively which adjusts the active nozzle area and the throttle pressure remains constant through the load range. In full arc admission the control valves remain fully open and the load is changed by varying the boiler pressure or the boiler pressure can remain constant and all the valves are operated together until the desired load is reached.

Each of these methods have their advantages and disadvantages. Full arc admission while maintaining throttle pressure throughout the load range wastes pumping energy at the lower loads. Efficiency gains can result from full arc admission with variable boiler pressure operation due to decreased throttle enthalpy losses in the turbine first stage and higher inlet steam temperatures, but it tends to result in a less responsive system and increased boiler fatigue since the boiler would be responsible for building pressure and increasing load simultaneously. Figure 3-3 shows an example of how throttle enthalpy losses can be minimized by using variable pressure operation. Usually the best operation is a combination of fixed and variable pressure operation where the control valves are throttled to a valve point and reduced pressure operation is used in a particular load range. For example, start reducing load with partial arc admission and constant throttle pressure until a valve point is reached, then maintain this valve position and reduce load by reducing throttle pressure. Maintain this mode of operation until the minimum throttle pressure is reached and then continue throttling the control valves to reduce load further.

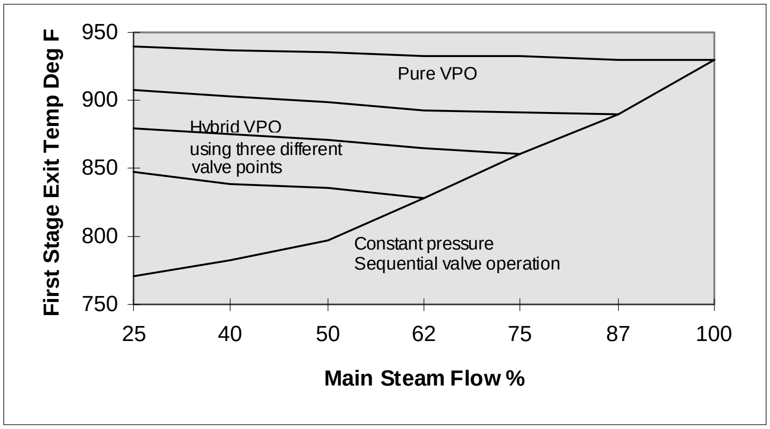


Figure 3-3
Variable pressure operation (VPO) can keep first stage exit temperature nearly constant during load changes reducing thermal stress and improving efficiency. Normal operation with constant throttle pressure and sequential valve operation can cause a drop from 930 to 770°F across the turbine first stage. Maintaining high HP exhaust temperature raises hot reheat temperature at reduced loads.

The most important factors to consider when evaluating variable pressure operation are:

1. Reduced available energy due to lower throttle pressure
2. Improved HP turbine efficiency since throttle temperature losses are minimized
3. Higher hot reheat temperatures at low loads
4. Higher throttle temperatures at low loads
5. Reduced power requirements for boiler feedpumps
6. Solid particle erosion is reduced

There are a number of factors dealing with the turbine which can affect the unit heat rate. These include process values such as steam temperature and pressure, design features such as pressure drops throughout the turbine, and maintenance items such as blade erosion and steam leakage. Figure 2-11 is the Heat Rate Logic Tree for Turbine Losses.

Main Steam Temperature

A throttle temperature change can affect the turbine load and heat rate.

1. For a fixed throttle value position, throttle flow is inversely proportional to throttle temperature as shown in this equation:

$$W = KAP\sqrt{\frac{1}{T}} \quad \text{(eq. 3-35)}$$

where: W = steam flow, lb/hr (kg/hr)

K = constant

A = area, ft^2, (m^2)

P = steam pressure, psi, (kPa)

T = steam temperature, °F, (°C)

Therefore, if the throttle temperature rises the flow will decrease.

2. An increase in throttle temperature will increase the available energy in the main steam.

3. A decrease in throttle flow decreases the total exhaust loss in the low pressure turbine which results in an increase in low pressure turbine efficiency and an increase in unit load.

The effect of a temperature change on unit load in each section of the turbine can be calculated by multiplying the ratio of the new and old steam flow by the ratio of the available energy in the steam under the new and old conditions.

$$G_{new} = \frac{W_{new}}{W_{old}} \times \frac{H_{avail(new)}}{H_{avail(old)}} \times G_{section} \quad \text{(eq. 3-36)}$$

where: G_{new} = turbine section output at new conditions

W_{new} = steam flow under new conditions

W_{old} = steam flow under old conditions

$H_{avail(new)}$ = available enthalpy under new conditions

$H_{avail(old)}$ = available enthalpy under old conditions

$G_{section}$ = design power generation for section

An increase in main steam temperature can have a negative affect on load from the above equations, but the increase will improve the overall turbine heat rate. Turbine heat rate is the ratio of boiler heat duty to turbine mechanical work. The following equation is used to determine the new turbine heat rate.

$$THR_{new} = \left[W_1 (H_1 - H_2) + W_2 (H_3 - H_4)\right] / G_{new} \quad \text{(eq. 3-37)}$$

where: W_1 = throttle flow

W_2 = reheat flow

H_1 = enthalpy at throttle conditions

H_2 = enthalpy at final feedwater conditions

H_3 = enthalpy at hot reheat conditions

H_4 = enthalpy at cold reheat conditions

G_{new} = unit load under new conditions

THR_{new} = new turbine heat rate

The above equations are supplied to give some insight into how main steam temperature can affect unit load and unit heat rate. In practice, curves supplied with the unit thermal kit are used to estimate the effects of temperature deviations on the unit heat rate. Figure 3-4 is a sample curve showing typical heat rate penalties. The following factors can cause deviations in main steam temperature.

- Slagging and fouling. Slagging is usually considered the formation of molten deposits on the furnace walls and other surfaces exposed to radiant heat. Fouling is defined as the formation of high temperature bonded deposits on convection heat absorbing surfaces. Fouling is caused by vaporization of elements in the coal as the heat is absorbed and temperatures are lowered in the convection sections of the boiler. Slagging on the radiant walls causes more of the heat transfer to occur in the convection sections thus raising steam temperatures.

If slagging is a problem furnace deposits can be reduced by increasing excess air. If steam temperatures are low and furnace walls stay clean then conditioners can be sprayed into the furnace which cause slag buildup. It is important to develop a good soot blowing program such that waterwall surfaces are not blown excessively which would cause more of the heat transfer to occur in the waterwall thus lower steam temperatures. Soot blowers in the superheat and reheat sections should be operated as necessary to maintain cleanliness.

- Pulverizer mill biasing in tangential fired boilers can be used to raise the fireball at reduced loads and thus increase the heat transfer in the superheat and reheat sections if temperatures are low.
- Burner tilt operation has the same effect an mill biasing. Elevated burner tilts reduce the total water wall surface exposed to direct radiant heat and the proportion of total heat absorbed by the water walls is reduced. Raising the burner tilts too much will lower the furnace's effective retention time and allow less residence time for complete combustion.

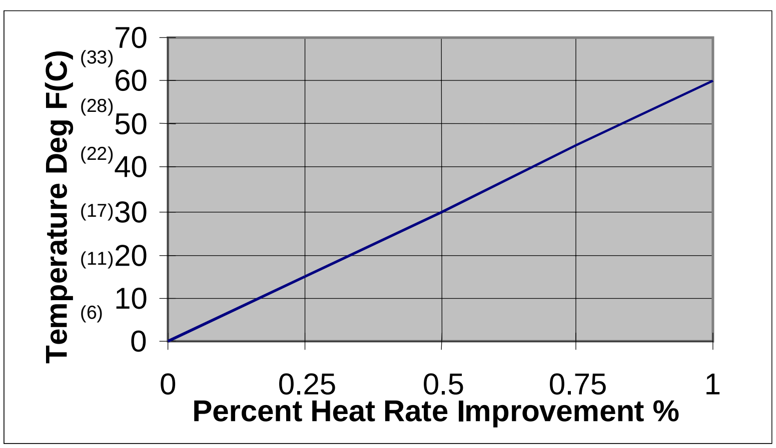


Figure 3-4
Typical heat rate improvement for an increase in main steam temperature. The curve is based on a 60% load factor.

Main Steam Pressure

As with main steam temperature a change in main steam pressure can affect the unit load in three ways.

1. From equation 3-35 a 5% increase in initial pressure will result in a 5% increase in steam flow which will in turn cause a 5% unit load increase.

2. The increase in flow will cause an increase in steam velocity leaving the last stage, increasing the total exhaust loss. An increase in exhaust loss results in poorer low pressure turbine efficiency.

3. The throttle available energy increases as the pressure increases.

The net result is a load increase of 4.9% at rated load. The increase in output would be greater at lower loads because turbine exhaust losses improve with increased pressure at loads less than 50%.

With increased energy in the steam it can be seen from equation 3-36 that the turbine heat rate will improve. The exception would be under variable pressure operation where we have already shown that decreased throttle temperature losses, improved reheat steam temperatures, and reduced BFPT power requirements override the energy losses. The curve for calculating heat rate improvements due to increased throttle pressure should be included in the unit thermal kit. Figure 3-5 is a typical example.

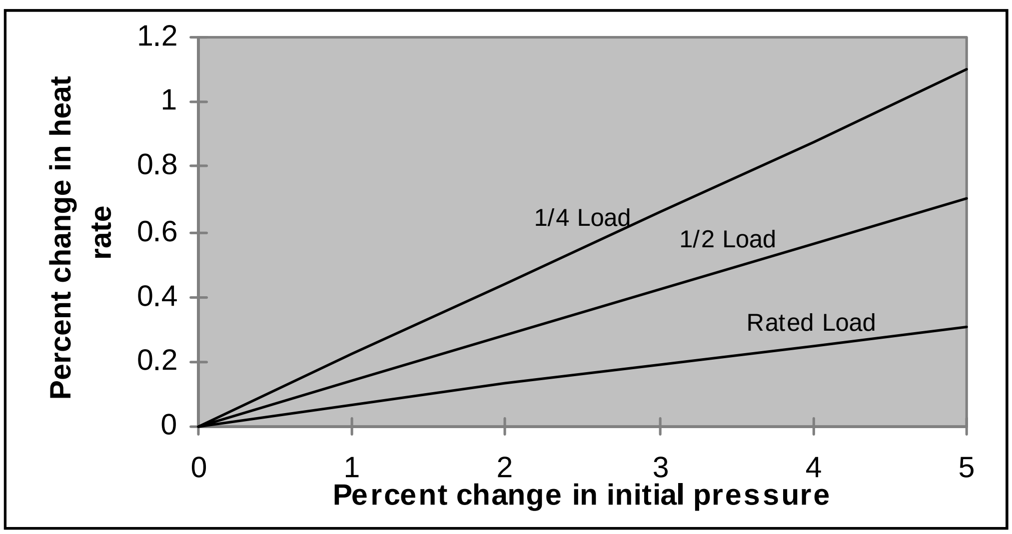


Figure 3-5
Turbine main steam pressure influence on unit heat rate. Most units are designed to run 5% overpressure.

Design Features

Design considerations affecting turbine efficiency include component design to minimize pressure drops, stationary and rotating blade design to obtain optimum steam velocity, and section design to minimize friction and leakage losses.

Pressure drops occur in various components without doing any work. Examples are listed below with approximate pressure drops in percent.

Component	Pressure Drop
1. Turbine stop valves	2%
2. Control valves (VWO)	2%
3. Boiler reheater	7-10%
4. Reheat stop and intercept valves	2%
Turbine crossover	3%

As a general rule a 1% pressure drop can cause about .1% effect on the unit heat rate. The pressure losses listed above could affect the heat rate as much as 1.9%.

The relationship between the speed of the rotating blades and the steam velocity affects the turbine efficiency. For a pure impulse stage maximum efficiency is obtained if the ratio of blade speed to steam velocity is 0.5 and for a stage with 50% reaction, maximum efficiency occurs when the ratio is 0.707. These ratios were derived assuming a stationary blade angle of zero degrees, therefore the stage design must consider optimum blade angle and the proper combination of impulse and reaction stages to obtain the most efficient velocity ratio.

The turbine efficiency can be affected by minimizing packing leakage within the turbine sections. Figure 3-6 shows that this leakage can occur as tip leakage bypassing the rotating blades, interstage leakage bypassing the stationary nozzle blades, and interstage leakage re-entering the steam path through dovetail holes. These leaks can be minimized by various packing designs. The dovetail holes are installed to minimize the pressure difference across the rotating blades in an impulse stage, but they provide an additional benefit of keeping the interstage leakage from re-entering the steam path where boundary layer flow losses would decrease the efficiency further. These holes cannot be used in reaction stages because large pressure drops across the rotating blades would cause steam to bypass the rotating blades.

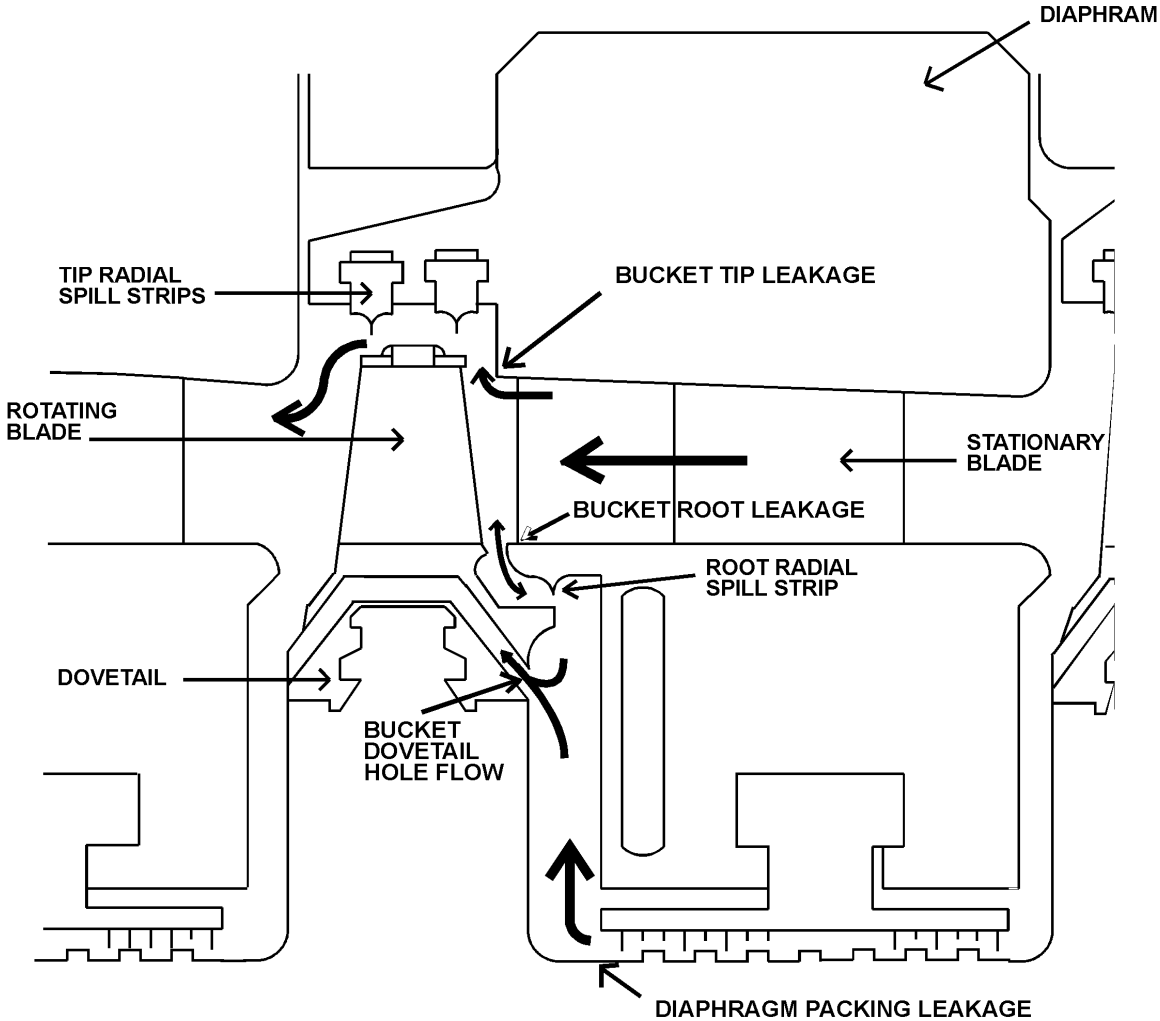


Figure 3-6
Cross-section of turbine impulse stage showing efficiency losses due to leakage.

Maintenance Items

During maintenance outages all packing seals should be inspected for wear and turbine blades and nozzles should be inspected for corrosion or erosion. Seal damage can occur from vibrations, thermal distortion, bearing failures, and solid particle erosion. Table 3-3 and 3-4 show kilowatt losses due to seal wear.

Table 3-3
Approximate Kilowatt loss per mil of excess clearance for radial tip spill strips.

*** These losses are for the first stage of the LP turbine**
**** These losses are for the second stage of the LP turbine**
***** These losses are for the last stage of the LP turbine.**

Use linear interpolation to determine losses between second and last stages.

Unit Size - MW	HP Stages	IP Stages	LP Stages
100	2.5	1.0	0.5*** to 1.5*
500	7.0	3.0	2.0*** to 6.0*

Table 3-4
Approximate Kilowatt loss per mil of excess clearance for interstage packings

These losses are for the first stage of the LP turbine
**** These losses are for the second stage of the LP turbine**
***** These losses are for the last stage of the LP turbine.**

Use linear interpolation to determine losses between second and last stages.

Unit Size - MW	HP Stages	IP Stages	LP Stages
100	4.0	1.0	0.5*** to 2.0**
500	8.0	2.0	1.0*** to 4.0**

Solid particle erosion (SPE) of steam turbine buckets or blades, nozzles and control valves has been a problem of concern to U. S. utilities for many years It is generally agreed that erosion damage is caused by oxide scale oxfolication from boiler tubes and/or steam leads which become entrained in steam flow to the turbine.

SPE damage can be minimized by water chemistry, thermal cycling, material changes and elimination of air inleakage. EPRI reports are available as background in all these areas, including the new TR-105040, “Selection and Optimization of Boiler Water and Feedwater Treatments for Fossil Plants”.

The high and intermediate pressure turbine sections are affected by solid particle erosion problems, but cycling and low load operation can cause corrosion problems in the low pressure turbine. Impurities such as sodium hydroxide, sodium chloride, and various organic and inorganic acids can deposit on the LP turbine blades especially near the wet/dry interface. Deposits can reduce the efficiency of turbines by changing the shape and smoothness of the nozzles and blades and can contribute to stress corrosion cracking and corrosion fatigue.

Damage to turbine seals, nozzles, or blades can normally be detected from performing enthalpy drop tests on the turbine. This tests should be done quarterly to determine if the deterioration is sudden or gradual. The following section details the test execution, calculations, and analysis.

Enthalpy Drop Test

Enthalpy Drop Test Introduction

The purpose of this type of testing is generally to determine the maximum unit capability and the performance characteristics of the superheated turbine sections, usually the HP turbine plus parts or all of the IP turbine.

The maximum unit capability is determined by normalizing the tested generator output for variation from design conditions of main steam pressure and temperature, reheat temperature, reheater pressure drop, and absolute back-pressure.

The performance characteristics of the superheated turbines which are of interest include efficiencies and normalized pressure. Since the main steam pressure and temperature measurements are taken prior to the control valves, the control valves and their effect are included in the HP turbine efficiency. For this reason, it is imperative that this testing be conducted at the VWO (Valves Wide Open) position, or a repeatable valve position. This valve position is repeatable and will minimize valve position effects on HP turbine performance characteristics. The repeatability of a verifiable valve position is critical. This ensures that valve position effects are equal between periodic tests.

Enthalpy Drop Test Preparations

Instrumentation

The parameters which should be measured include the following:

- Pressures:

Steam pressures should be measured with DWGs, suitable accurate pressure transducers, such as a Rosemount pressure transducer or calibrated gages. The number of suggested measurements is listed below:

One Measurement: First Stage Shell and Extractions (Bleeds)

Two Measurements: Main Steam (Throttle), HP Turbine Exhaust, Hot Reheat (IP Turbine Inlet), and Crossover (IP Turbine Exhaust)

- Temperatures:

The same number of temperature measurements should be taken at each point as listed under "pressures" above. Instrumentation should consist of calibrated continuous-lead thermocouples, ice-junction reference, and potentiometer.

- Generator Output:

The output is measured using the unit KWH meter located on the switch board. Output is measured using either a meter revolution counter or by timing meter revolutions using a stop watch. If both methods are available, they should be used to provide checks on each other. Newer units have high accuracy KWH meters which should be used to measure generator output at each phase of the generator.

- Steam Flow Measurements (Where Applicable):

This measurement should be taken using a high-pressure differential pressure transducer (manometer) for measuring the flow differential and a pressure transducer for the static pressure measurement. Other flow measuring instruments are available such as ultrasonic displacement types, but the accuracy of the instrumentation should be taken into account before application.

- Barometric Pressure and Absolute Back-pressure:

These are measured using mercury column instruments or calibrated pressure transducers. Impulse lines should be sloped to the tap to allow condensate drainage. Care should be taken to record the temperature of each instrument since the mercury columns are temperature corrected.

- Station Pressure and Temperatures:

Station gauges which indicate the measurement parameters should be trended for comparison with the "test" instrumentation. This is used to indicate erroneous station instrumentation which requires maintenance. Particular attention should be given to indications of main steam pressure and temperature and reheat temperature.

The instrumentation should be set up and checked the day before the test. Preferably, the person(s) who sets up and checks out the instrumentation should read it during the test unless a data acquisition system is used. Pressure lines should be blown down to steam prior to the test. Instrumentation where there are two measurements of the same parameter (such as two thermocouples in "A" main steam pipe) should be compared to each other for accuracy. A rule of thumb is that two pressure transducers from the same source should agree within 5 psi(34.5 kPa) and two thermocouples should be within 1° F(0.55 °C) of each other. Inoperable and broken thermocouples must be repaired or replaced. Unlike DWGs, thermocouples are not very durable and will require some maintenance or replacement. It is a good policy to remove all thermocouples prior to a turbine-inspection outage, which keeps them from being damaged and also allows for inspection and repair or replacement, as may be necessary.

Unit Configuration

Regarding the unit, all equipment such as heater drain pumps and feedwater heaters should be in service. Feedwater heater water levels should be correct. If some cycle component is out of service for a lengthy period, Turbine Output Check (TOC) type testing may continue during that period, but the cycle abnormality must be reported. Such cycle abnormalities do not affect turbine-section efficiencies, but will effect stage pressures. Much available information is still gained from the enthalpy drop tests.

The turbine steam pressure levels at a given valve position are affected by deviations in the feedwater-heating cycle or the quantity of reheat-desuperheat water. Such items as heaters-out-of-service, dumping of heater drains to the condenser, or variations in the use of systems, such as auxiliary steam system, should be avoided when conducting comparative turbine tests, since they do have an influence on the measured turbine pressure levels. Small deviations in feedwater-cycle performance such as those caused by changes in heater terminal temperature differences have a minimal effect on the turbine pressure levels and can normally be neglected in the comparative analysis. However, care should be taken to insure good water levels in the feedwater heaters to minimize any effects.

Test Execution

Unit Operating Set-up

In conducting the test, the test coordinator should remember that the basic objective is to measure maximum unit capability and turbine section performance at the VWO point, or some fixed valve position such as 3-VWO.

It is generally the responsibility of the operating shift to set the control valves at the VWO position with maximum unit generation. The test coordinator should assure that these conditions are met. Boiler equipment such as fans and pulverizers often operate at or near capacity during the test. This is expected. Whenever possible, measurements of valve stem lift may be taken as an indication of the VWO position.

When required, the test may be conducted at a reduced throttle pressure in order to obtain the VWO position. However, the test should be conducted at as high a pressure as possible in order to minimize the magnitude of correction required. Steam temperatures should be maintained at contract full load values and maintained as steady as possible. The VWO position should ideally be reached by opening the control valves until VWO or a boiler operating parameter (air, fuel or water) is reached. The boiler master pressure setpoint can be reduced, allowing throttle pressure to fall as turbine valves are fully opened. For G.E. units, a check should be made to assure that the motor speed changer is against the high speed stop and that the load limit-hand wheel is fully opened. A visual check of the cam and roller positions should also be made to further insure that the unit is at VWO.

Stabilization Period

As soon as the unit is prepared from an operations standpoint, a stabilization period should be observed. During this period, the test parameters are allowed to stabilize and steady-state conditions are established with boiler controls in automatic mode. However, on some units better results are obtained with the pressure control (or Boiler Master) in the manual position to fix the firing rate.

This is possible if the coal is of consistent and relatively dry quality. During this stabilization period and the test period, the test coordinator should emphasize to the control operator the importance of minimizing manual control changes. Such manual control adjustments introduce step changes, destroy steady-state conditions, and lengthen stabilization periods. Stabilization periods are typically fifteen to thirty minutes, depending on the control system and coal quality.

During the stabilization period, sample data should be taken and reviewed to assure that all the instruments are operating properly. The test coordinator should review each data sheet. For pressure measurements, the coordinator should look for such problems as malfunctioning deadweights or pressure transducers, improper summing of weights, and improper test methods. Comparison of redundant instrumentation should be checked for agreement. On temperature measurements, items such as open or shorted thermocouples, low-millivolt readings indicating an improperly installed thermocouple, improper ice-bath reference practices, and malfunctioning potentiometers should be checked.

Test Period

Test periods of one hour consisting of thirteen readings taken at five minute intervals are suggested, although some 30 minute tests are appropriate. Some stations utilize data acquisition systems which allows scanning the data as fast as 15 second intervals. The data sheet check by the test coordinator should be repeated early in the test period.

Calculations

Data Average

All data is averaged and corrected using standard corrections to pressure and temperature measurements. Data averaging should be done by the individuals collecting the data unless data acquisition systems are used to collect and average data. All pressures are converted to absolute by correcting for water legs and adding barometric pressure.

All thermocouples millivolt readings are converted to degrees and the appropriate thermocouple corrections applied.

Generator output is calculated using the two measurement methods. The agreement between the two methods should be *~0.25%*. Any discrepancy can usually be traced to a revolution counter malfunction. Therefore, any questionable tests should use the "timed" method generation calculation.

Corrections to Output

The tested output must be corrected to reference conditions in order to eliminate the effect of cycle conditions which are variable. These include corrections for main steam pressure and temperature, reheat temperature, reheater DP, and ABP. A sample correction is shown under "Normalizing Generator Output" below.

Enthalpy Drop Calculation

Using the inlet and outlet conditions on each turbine section, turbine section efficiencies are calculated. Turbine efficiency is defined as the actual enthalpy drop (or used energy) divided by the isentropic enthalpy drop (or available energy).

The actual enthalpy drop is the inlet enthalpy minus the exit enthalpy at test conditions.

The isentropic enthalpy drop is inlet enthalpy at test conditions (temperature and pressure) minus the exhaust enthalpy at test pressure with the change in entropy equaling zero.

Example:

	Throttle Conditions	HP Exhaust Conditions
Pressure	2345.1 psi (16169 kPa)	556.7 psi (3838.3 kPa)
Temperature	1049.1 °F (565 °C)	680.5 °F (360 °C)
Enthalpy	1495 btu/lb (3477 kJ/kg)	1343.1 btu/lb (3124 kJ/kg)

Enthalpy at 556.7 psia (3838.3 kPa) with no change in entropy = 1309 btu/lb (3044.7 kJ/kg)

$$EFF_{HP} = \frac{1495-1343}{1495-1309} \times 100 = 82\ \% \qquad \text{(eq. 3-38)}$$

Correcting Pressures

The first-stage shell pressure, HP exhaust pressure, hot-reheat pressure, crossover pressure, and bleed pressures should be corrected for deviations in the main steam pressure and temperature and the reheat temperature.

The pressure of steam expanding through the turbine blading is measured at numerous locations between the throttle and the condenser. Practically all pressure measurements are obtained from static pressure taps located on the steam piping or turbine shells. Since steam pressures are being measured, water columns will form in the connecting piping between the source and the pressure gauge (DWG). The effect of these water legs must be included with atmospheric pressure in obtaining absolute pressures. Sub atmospheric pressure connections should be properly vented to clear lines of accumulated moisture and be provided with purge air systems. Locations where pressure levels will be slightly above atmospheric pressure should also be purged, particularly when the manometer is located above the source connection.

Steam pressures on the turbine side of the throttle-stop valve are proportional to the main steam flow rate downstream of the point of measurement. The proportionality is affected, to some extent, by variations in the steam inlet temperatures (throttle, hot reheat) which cause small changes in the steam specific volume. Therefore, turbine pressures are normalized to design conditions using the following corrections factors.

The measured first-stage (impulse) pressure should be normalized to a reference condition by using the following relationship..

$$P_{fssn} = P_{fsst} \times \frac{P_{msd}}{P_{mst}} \qquad \text{(eq. 3-39)}$$

where:

P_{fssn} = Pressure of first stage shell normalized

P_{fsst} = Pressure of first stage shell during test

P_{msd} = Pressure of main steam design or base test

P_{mst} = Pressure of main steam during test

The remaining turbine pressures are normalized using equation 3-40 and the two correction factors below. Equation 3-41 corrects for variations in main steam pressure and specific volume. Equation 3-42 corrects for variations in the reheat steam specific volume.

$$P_n = P_t \times C1 \times C2 \qquad \text{(eq. 3-40)}$$

where: P_n = Normalized pressure

P_t = Test pressure

$$C1 = \sqrt{\frac{P_{msd} \times V_{mst}}{P_{mst} \times V_{msd}}} \qquad \text{(eq. 3-41)}$$

where: P_{msd} = Pressure of main steam design or base test

P_{mst} = Pressure of main steam during test

V_{mst} = Specific volume of main steam during test

V_{msd} = Specific volume of main steam design or during base test

$$C2 = \sqrt{\frac{V_{hrhdt}}{V_{hrht}}} \qquad \text{(eq. 3-42)}$$

where: V_{hrhdt} = Specific volume of hot reheat at design temperature and test pressure

V_{hrht} = Specific volume of hot reheat at test temperature and pressure

Steam flow is normalized using only equation 3-41.

$$Flow_n = Flow_t \times C1$$

Normalizing the pressures to design conditions allows for comparisons which otherwise could not be made.

Data Plots

Chronological plots of the deviation from base of calculated values should be plotted. These indicate unit performance trends which are indicative of unit abnormalities which require correction.

The individual turbine section efficiencies should also be plotted on a chronological basis.

Analysis

Test Validity

The first step on data analysis is test validity, i.e. if the plotted data represents present unit performance. This judgment is often subjective and dependent largely on experience. Several generalizations apply:

- Turbine section efficiencies under normal operating and test conditions should decrease gradually during the period between turbine inspections.
- Generator output, first-stage shell pressure, and standard throttle flow deviations should all indicate the same trend. This is because output is dependent on flow which is indicated by first stage pressure. These parameters should all increase or decrease simultaneously.
- Low HP turbine section efficiency accompanied by a large negative deviation of the first stage shell pressure usually indicates that the control valves were not at the VWO position.
- IP turbine section efficiencies are normally constant, or changes are smaller than testing accuracy's.

If for some reason the test is determined to be invalid, a repeat test should be scheduled.

Data Analysis

After determining that the test is valid, i.e. the plotted data indicates present unit performance at the VWO position, an analysis must be conducted to determine if any abnormalities are indicated which require further investigation and/or resolution. Indicated abnormalities must be further investigated by additional tests conducted as soon as possible.

Conclusions as to the turbines conditions can be made using the following analysis of pressure trended over time:

- THROTTLE PRESSURE - Throttle pressure may be reduced below design in order to obtain VWO. This pressure level measured upstream of the stop valves should be essentially the same in each steam lead.
- FIRST-STAGE SHELL PRESSURE (FSSP) - This pressure, measured after the first stage buckets in the HP turbine, is the most useful pressure measurement available for analysis. This pressure level is proportional to the

main steam (throttle) flow. For a constant control valve position, the normalized first stage shell pressure will be a constant, provided that the feedwater heating cycle is consistent between the comparative tests. An increase in this pressure, at constant valve position, would indicate either increased steam flow capacity due to nozzle block erosion or a reduction in blade path flow area downstream of the first stage measurement location due either to damage to either buckets or nozzles or fouling of the flow path surfaces. Analysis of additional pressure data, discussed later, would confirm which of the above flow path deficiencies caused the change in FSSP. If the normalized FSSP decreased at a constant valve position, then one should suspect either increased flow area downstream of the FS (excessive clearances in the HP turbine blading) or a flow restriction in either the nozzle block or inlet features. In the case of combined HP/IP elements, there is a possibility of excessive leakage from the inlet features (steam inlet pipes, nozzle chambers) to some lower pressure zone in the turbine element. gain, analysis of additional turbine pressure data should clarify which of the flow path abnormalities has taken place.

- CONSTANT FIRST STAGE SHELL PRESSURE - If the normalized FSSP at a fixed valve position is constant, then other downstream pressures (hot reheat, crossover) can be compared with the first-stage pressure for consistency. Again, assuming no large deviations in the feedwater heating cycle, these normalized pressures should be constant if the blade path has not been damaged or fouled. Should the normalized pressures increase, one should suspect flow path restrictions downstream of the measured pressure (hot reheat, crossover) location. Conversely, a decrease in these normalized pressures would be indicative of increased flow path areas downstream of the measurement location. Note again, these interpretations are valid only if the normalized first-stage shell pressure remains constant.

- HOT REHEAT PRESSURE - The hot reheat pressure at a constant inlet temperature is proportional to the steam flow entering the IP turbine element. Variations in the feedwater heating cycle and the quantity of reheat desuperheat flow can affect the hot reheat pressure level at a constant control valve position. Therefore, every effort should be made to conduct the test with no desuperheat flow. If the throttle flow remains constant at the fixed valve position (normalized first stage shell pressure level is constant), but the normalized hot reheat pressure level increases, the interpretation would be that a flow restriction existed in the flow path downstream of the IP inlet pressure location. Conversely, a decrease in the normalized hot reheat pressure level, at constant control valve position and steam flow, would infer that an increase in downstream flow area has taken place.

- CROSSOVER (LP BOWL) - The measured pressure is proportional to the steam flow entering the LP turbine. As noted for the hot reheat pressure, the level can be affected by deviations in the feedwater-heating cycle as well as the reheat spray flow rate. Should the normalized crossover pressure increase at a constant throttle flow (first stage pressure constant at fixed valve point), the proper interpretation would be to suspect a flow restriction or surface fouling of the LP turbine blading. Should the normalized crossover pressure decrease at constant throttle flow, increased LP flow path areas are indicated.

- EXTRACTON PRESSURES - The measured extraction pressures downstream of the IP inlet location behave in a manner similar to the hot reheat and crossover pressures. The extraction pressure levels are proportional to the flow downstream of the extraction point, and the levels can be influenced by feedwater cycle variations and quantities of reheat spray water. In order to minimize the effect of the extraction piping pressure losses on the level of the extraction pressure the measurement point should be located close to the turbine connection. If the throttle flow remains constant at the fixed control valve point (first stage pressure level constant), then increases in the normalized extraction pressure levels are indicative of downstream flow path restrictions, and decreases in pressure levels could result from greater flow path areas in the downstream blading.

- HP EXHAUST - The HP exhaust pressure level is determined by the hot reheat pressure level plus the pressure drop through the reheater section of the steam generator. The reheater pressure drop is essentially constant across the turbine steam flow range. The "normal" level of reheater pressure drop should be established and any significant deviation from the "normal" should be investigated. The level of HP exhaust pressure can be affected, at a constant throttle flow rate, by variations in the feedwater cycle or quantities of reheat spray flow. In actuality, the hot reheat pressure is affected by these variables and since the reheat pressure drop is a fixed percentage, the level of HP exhaust pressure must also respond. If the HP exhaust and hot-reheat pressures do not show the same trend, then a measurement error should be suspected.

- VARYING FIRST STAGE SHELL PRESSURE - Now consider the case where, at constant control valve position, the normalized first-stage shell pressure increases. As previously discussed, the deviation would indicate that either the throttle flow rate at the valve position has increased, or there has been a decrease in flow path area downstream of the first-stage pressure location resulting in greater pressure drop through the blading from the same steam flow rate. The key point here is looking at the downstream pressure levels (hot-reheat, extractions, crossover). If these pressure levels have increased the same percentage as the first-stage shell pressure level, then the steam flow has increased at the constant control valve position. If, on the other hand, the downstream pressure levels have not increased proportional to the increase in the first-stage pressure, then the steam flow has not increased and the elevated first-stage shell pressure is most likely the result of flow path area reduction by fouling or damage. The same procedures are applicable should the first stage shell pressure level decrease at the fixed control valve position. Again, the downstream pressure levels will either substantiate a steam flow decrease or confirm that a change has occurred in the first-stage pressure-flow relationship, in this case most probably the result of flow path area increases.

One should keep in mind the following basic actions when reviewing turbine flow path pressures to determine possible changes in the condition of the unit:

- Make comparison at fixed control valve position.
- Normalize measured pressures to account for significant deviations in inlet steam conditions.
- Utilize the various downstream turbine pressures to substantiate any change in steam flow rate at the fixed-valve position.
- If the valve point steam flow rate has not changed, determine if downstream turbine pressures indicate problems in the flow path areas.

In general, higher flow path steam pressure levels at constant steam flow rates indicate blade path damage or surface fouling. Conversely, lower inlet pressures in the various element blade paths, again at constant steam flow, are indicative of damage to the sealing of the blade path with increased leakage flow area.

It is difficult to assign limits as to the magnitude of changes in either pressure levels or pressure ratios which should be considered significant in terms of reflecting changes in the internal condition of the turbine blade path. In general, a change between comparative values where the magnitude is ~ 1.0% or less would not indicate significant change in the internal condition of the turbine. Should the magnitude of the change between consecutive sets of values exceed ~3 .0%, action should be initiated to verify the cause of the change. In like manner, a gradual change in level or ratio which exceeds +3.0% over a period of several comparisons should also be investigated. Step changes in consecutive comparisons where pressure levels or ratios change in excess of +5.0% require immediate attention, since they are indicative of serious damage to the turbine blade path.

As might well be expected, the above listed turbine pressure characteristics do not apply under all circumstances for all steam turbine configurations. In particular, combined HP/IP reheat elements, where sizable internal leakages can bypass blading groups or the reheater section of the steam generator can develop blade path pressure deviations which do not conform with the conventional interpretations described in the preceding paragraphs.

Test Frequency

Turbine output checks and enthalpy drop testing should be conducted on each unit on a quarterly basis. It is imperative that this testing schedule be reasonably maintained. As indicated by the information on data analysis, the object is to determine relative change which more often occurs over a relatively brief period of time. Testing at one-or two-year intervals will only reveal the end result of abnormalities, whereas regular periodic testing allows a determination to be made of possible causes and solutions.

Technical References

The following references were used in preparing the procedure.

- ASME PTC-1964: Steam Turbines
- ASME PTC-19.2-1964: Pressure Measurement
- ASME PTC-19.3-1964: Temperature Measurement
- ASME PTC-6S Report-1970: Simplified Procedures for Routine
- Performance Tests of Steam Turbines
- General Electric Paper Ger-2661: Analysis of Change in the Performance Characteristics of Steam Turbines
- ASME Fluid Meters, Sixth Edition
- Westinghouse Paper: Guidelines for Analysis of Steam Turbine Generator Operational Data

Normalizing Generator Output

The actual generator output is corrected to design or base conditions to eliminate the effect of cycle conditions. The following is the typical routine:

	Test	Design
Generator Output (kW)	215,400	208,134
Main Steam Press	1823.7 psi (12574 kPa)	1812.1 psi (12494 kPa)
Main Steam Temp	998.4 °F (537 °C)	1000.0 °F (538 °C)
Reheat Temp	986.9 °F (531 °C)	1000.0 °F (538 °C)
Reheater Pressure Drop (%)	9.4	10.0
Absolute Back-Pressure	1.69 inHg (5.72 kPa)	1.5 inHg (5.08 kPa)

Using the correction curves found in the unit thermal kit, the effects these variations have on the generator output are found. Using the equation for each curve minimizes error and inconsistencies:

	% Change In Kilowatt Load From Curves
Main Steam Pressure	+ .55%
Main Steam Temperature	+ .013%
Hot-Reheat Temperature	-.603%
Reheater Pressure Drop	+ .156%
Absolute Back-Pressure	-.350%

The kilowatt load normalized to design conditions is found by dividing the test output by the following for each correction:

$$1+\frac{\%KW}{100} \qquad \text{(eq. 3-43)}$$

where: $\%KW$ = % change in kilowatt load from each correction curve

$$L_n = \frac{215{,}400}{\left[1+\frac{0.55}{100}\right]\left[1+\frac{0.13}{100}\right]\left[1+\frac{(-0.603)}{100}\right]\left[1+\frac{0.156}{100}\right]\left[1+\frac{(-0.35)}{100}\right]} = 215{,}661 \text{ kW} \quad \text{(eq. 3-44)}$$

where: L_n = generator load normalized for test conditions

Plant Auxiliaries

Overall unit heat rate is calculated by dividing total btu input by total net generation. Since gross generation is not used the electrical auxiliaries used to operate the plant can affect the heat rate significantly. On the average excess electrical auxiliaries affect utility heat rates by 86 btu/kwh (90.8 kJ/kwh). Each unit should have a curve of expected auxiliaries for a given load and it should be updated whenever additional equipment is added or removed.

The effect on heat rate of a deviation from expected auxiliaries is calculated by using equation 3-45.

$$AUX_e = \frac{AUX - AUX_x}{LOAD - AUX} \times 100 \quad \text{(eq. 3-45)}$$

where: AUX_e = percent effect on heat rate for total auxiliaries

AUX = actual total auxiliaries

AUX_x = expected total auxiliaries

LOAD = unit gross load

An effect on the heat rate due to high plant electrical auxiliaries can be caused by the following.

- Operating plant equipment when it is not needed for the units present status

Develop a plan outlining when equipment, such as pulverizers, condensate booster pumps, and hotwell pumps, is needed for a specific unit load. Running the proper number of pulverizer mills for a given load can help reduce dry gas loss as well as auxiliary losses.

Cooling tower fans or circulating pumps depend on unit load and also ambient conditions. Develop calculations which compare condenser and auxiliary effects to determine the optimum cooling requirements.

- Operate equipment such as service water pumps and air compressors only as needed.
- Maintain equipment whose power usage increases with deteriorating performance such as pulverizers and pumps.
- Maintain boiler ducts and expansion joints to prevent air leakage to conserve FD and ID fan power.
- Investigate possible installation of variable speed drives for fans instead of using dampers for air flow control.
- Outdoor lighting should be controlled by automatic sensors.
- Maintain heating and air conditioning controls for proper operation.
- Turn off personal computers when not in use, especially overnight.

Condenser

The power plant condenser receives exhaust steam from the low pressure turbine and condenses it to liquid for reuse. Three types of condensers are direct contact condensers which spray water directly into the steam flow, air-cooled condensers where steam condenses on the inside of tubes while air flows over the tubes, and water cooled surface condensers where steam flows over tubes containing cooling water.

The water-cooled surface condenser is the most common type of condenser. The steam rejects heat which is transferred to the cooling water which in turn transfers the heat to the environment. This heat rejection process is very important to the Rankine Cycle efficiency. Efficiency increases as condenser absolute pressure decreases. With condenser pressure as low as possible the amount of heat rejected is lower and the amount of work of the turbine increases, since the enthalpy drop across the turbine becomes greater. This is accomplished by optimizing the heat transfer rate between the condensing steam and the cooling water, effectively allowing no leakage of air into the condensing space, and minimizing any cycle leakage to the condenser which would add heat load. Air ejectors are supplied to remove any noncondensable gases that enter the condenser.

Some of the things that affect the heat transfer are tube fouling, tube restriction, cooling water flow rate, unit load and cooling water inlet temperature.

- Tube fouling or restriction in the condenser can be detected by monitoring the absolute back pressure, tube side pressure drop and the terminal temperature terminal difference between the turbine exhaust and cooling water outlet temperatures. Typical elements found in the collected material after cleaning a condenser include silicon, calcium, manganese, and iron. The manganese and iron are of primary concern as they are responsible for scale buildup on the tube ID, adversely affecting heat transfer and corrosion rates. Possible corrective or preventative measures include:

  — Backwashing arrangements may be provided.

  — Sponge balls or brushes may be automatically recirculated through the condenser

  — Periodic condenser tube cleaning by shooting rubber plugs or wire brushes through the condenser with high pressure water during an outage. Material collection of greater than 0.75 grams per square foot of tube surface constitutes a “dirty” tube.

- Chemical Treatment
- Install condenser pressure differential transmitters in inlet and outlet water boxes to monitor tube restrictions.

• Cooling water flow rate inadequate

- Place additional circulating cooling water pump in service. See Figure 3-7 for proper balance to minimize condenser and auxiliaries performance effects.
- Install pressure differential transmitters at outlet water box or elbow to monitor flow rate.
- Determine if cooling water blowdown is installed on CCW inlet. If possible blowdown should be used on condenser outlet.
- Test circulating cooling water pumps for proper flow and rebuild as necessary.
- Maintain clean racks or screens and waterbox.

• Cooling water temperature too high

- Place additional cooling tower cells in service
- Perform cooling tower maintenance

• Condenser absolute back pressure too high with proper cooling water.

- Perform helium leak test to check for condenser air inleakage. Monitor air ejector discharge with a helium detector while spraying helium in areas that have potential inleakage such as turbine shaft seals, manway covers, rupture disks, vacuum breaker valves, penetration welds, flanges, valve packing, expansion joints, and pipe flanges.
- Inspect steam jet air ejectors for proper operation.
- Check incoming drain lines, feedwater heater high level dumps, minimum flow valves, and steam traps for leakage or improper operation which could add unexpected heat load to the condenser.
- Isolate pressure sensing line at condenser and take readings to check for instrument line leaks. The maximum leak should be 0.0491 psi/min (0.3385 kPa).

— Check pressure sensing line inside condenser for damage which could result in incorrect static pressure readings.

— Basket tip sensing line drainage.

— Transmitter calibration error.

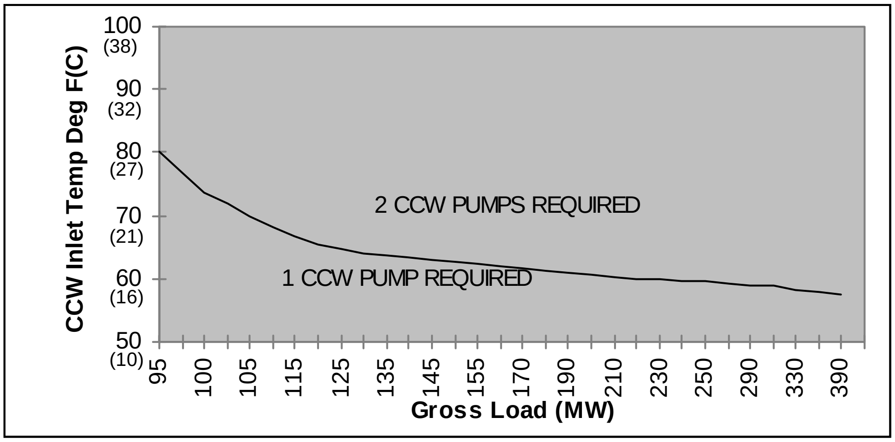


Figure 3-7
Typical steam station suggested economical CCW pump operation versus CCW inlet water temperature and gross load.

Cooling Towers

Condenser cooling water systems are classified as either once through (open) or closed systems. In once through systems water is pumped from a river or lake through the condenser and the warmer water is returned back to its source. In closed loop systems heat is rejected to the atmosphere through the use of either a cooling tower or a body of water such as a spray pond or cooling lake. Most newer systems use cooling towers because of increasing environmental constraints.

Cooling towers operate by evenly distributing the warm water from the condenser over a fill material in the tower and air is drawn or forced through the tower. Heat is rejected to the atmosphere by evaporation of some of the water and by heating the air. The cooling tower's performance is affected by the ambient wet bulb temperature, deterioration of the fill material, fill silt buildup, icing on tower structure, low water loading, and high water loading. During low load conditions some of the fans on multiple cell towers can be shut down to reduce plant auxiliary load. Condenser absolute back pressure should be monitored to determine the number of fans that can be removed.

Cooling tower performance tests can be very time consuming, but there is a simple method to determine if the tower is performing properly. Monitor the circulating water inlet temperature, circulating water outlet temperature, actual wet bulb temperature, and the circulating water flow. The cooling tower range is the difference between the water out and water inlet temperatures. Using the range and the tower outlet temperature, the corresponding calculated wet bulb temperature can be found from the vendor's design curves using either the design or measured value for circulating water flow. If the actual wet bulb temperature is higher than the calculated wet bulb temperature then the tower is performing at or better than expected. If the actual wet bulb temperature is consistently lower than the calculated wet bulb temperature then further testing or inspections are necessary to determine causes of the deficiency.

- Deterioration of fill material
  - Routine inspections are necessary to survey damaged or deteriorated fill material or to remove any debris.
- Fill silt or algae buildup
  - The purpose of the fill material is to increase the contact area between the air and water and to increase the water residence time. To maintain its maximum effectiveness buildup must be prevented by blowdown and proper water treatment.
- Low water loading can cause poor water distribution and high water loading can cause excessive air pressure losses.
  - Inspect distribution nozzles
  - Clogged distribution nozzles
  - Fan blade deterioration
  - Motor problems

Feedwater Heaters

Feedwater heaters serve three purposes in the power plant. They provide efficiency gains in the steam cycle by increasing the initial water temperature to the boiler, so there is less sensible heat addition which must occur in the boiler, they provide efficiency gains by reducing the heat rejected in the condenser, and they minimize thermal effects in the boiler. Steam is extracted from selected stages in the turbine to shell and tube heat exchangers or to open feedwater heaters where the steam and feedwater are in direct contact.

In shell and tube or closed type feedwater heaters the feedwater flows through the tubes and the extracted steam condenses on the shell side. The condensed steam from each feedwater heater drains successively to the next lower pressure heater and is returned to the feedwater through a heater drain pump or through the condenser. A drain cooling zone can be designed into the feedwater heater which cools the condensed steam to within a few degrees of the feedwater inlet. If the extracted steam is superheated additional improvement in performance can be obtained by designing a desuperheating zone in the heater. This allows the outlet feedwater to approach the saturation temperature of the shell side pressure.

Two variables are used to monitor a feedwater heaters efficiency. The heater Terminal Temperature Difference or TTD is a measure of how close the outlet feedwater temperature is to the feedwater heater saturation temperature. The heater Drain Cooler Approach or DCA is a measure of how close the heater drain outlet temperature is to the feedwater inlet temperature. There are a number of items that can affect these measures.

- Improper heater level can cause flashing in the drain cooler section and tube damage.
  - Check operation of automatic controls and level instrumentation
  - Check for possible tube leaks in feedwater heater. If a number of tube leaks have occurred in a feedwater heater tests should be run to verify proper level setpoint for optimum performance. See Figure 3-8 for effect of level on performance.
  - Vent valves may not be set up properly
- Improper extraction line pressure drops
  - Possible problem with extraction line check valve
- Tube fouling due to corrosion affects the heat transfer in the heater and also increases the problem of deposition of oxides on heat transfer surfaces of boilers. This can lead to boiler leaks and turbine damage
  - Reduce the level of dissolved gases such as oxygen and carbon dioxide in feedwater and adjust pH of the feedwater.
  - Clean tube bundles

- Continuous vent orifice plugging.
- Channel pass partition/gasket leak.

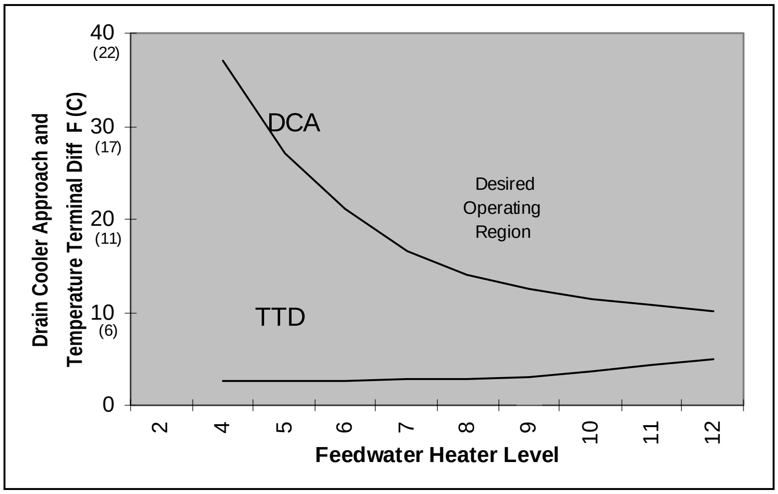


Figure 3-8
Feedwater heater level should be controlled at a point to optimize performance. If substantial tube plugging exists, a new optimum level should be selected by varying level while taking performance data.

Feedwater Heater Calculations

The following formulas are used in determining the effect feedwater heater deficiencies have on the plant heat rate.

$$TS = f\,(PHS) \qquad \text{(eq. 3-46)}$$

where: TS = saturation temperature of the steam at the extraction inlet pressure as found in the ASME steam tables

PHS = extraction inlet pressure

$$TTD = TS - TO \qquad \text{(eq. 3-47)}$$

where: TTD = feedwater heater Terminal Temperature Difference

TS = saturation temperature of the steam at the extraction inlet pressure as found in the ASME steam tables

TO = feedwater outlet temperature

$$TTD_x = C1 \tag{eq. 3-48}$$

where: TTD_x = expected feedwater heater Terminal Temperature Difference

$C1$ = constant as found in unit thermal kit

$$TTD_d = TTD - TTD_x \tag{eq. 3-49}$$

where: TTD_d = Terminal Temperature Difference deviation from expected

TTD = The feedwater heater Terminal Temperature Difference

TTD_x = expected feedwater heater Terminal Temperature Difference

$$TTD_e = A1 \times (TTD_d) + B1 \tag{eq. 3-50}$$

where: TTD_e = percent effect on heat rate due to the heater TTD being different than expected

$A1,B1$ = constants as found in unit thermal kit

$$DCA = TD - TI \tag{eq. 3-51}$$

where: DCA = drain cooler approach temperature difference

TD = heater drain outlet temperature

TI = feedwater inlet temperature

$$DCA_x = C2 \tag{eq. 3-52}$$

where: DCA_x = expected drain cooler approach temperature difference

C2 = constant as found in unit thermal kit

$$DCA_d = DCA - DCA_x \qquad \text{(eq. 3-53)}$$

where: DCA_d = drain cooler approach deviation from expected

DCA = drain cooler approach temperature difference

DCA_x = expected drain cooler approach temperature difference

$$DCA_e = A2 \times (DCA_d) + B2 \qquad \text{(eq. 3-54)}$$

where: DCA_e = percent effect on heat rate due to the heater DCA being different than expected

$A2, B2$ = constants as found in unit thermal kit

4

ELEMENTS OF A THERMAL PERFORMANCE MONITORING PROGRAM

Goals

The goal of a performance monitoring program is to improve unit efficiency. Efficiency improvements are needed because of increased fuel cost, increasing age of the units and their equipment, increased cost of capital improvements, and increased competition in the utility industry.

Performance improvement activities should initially concentrate on activities that can be accomplished with little capital investment in a relatively short time. These will include cycle isolation, evaluating selected parameters to improve operations control, and identify preventative maintenance that can easily be conducted on select components. After this work has shown to have improved the unit heat rate, then the engineer can begin evaluating losses that will require an expenditure of resources over a longer time period. This will require trending of performance parameters and conducting special performance test on major components or systems.

Initial Steps

The following steps can be used to establish a heat rate monitoring program. The intent is to start with work that can be accomplished quickly and will show results in a timely manner with little cost other than the engineers time.

1. Evaluate cycle isolation for leaks or improperly positioned valves. See Section 6 for details on cycle isolation. This evaluation can be accomplished at no cost to the unit, so it should have a high priority as an initial investment of time.

2. Determine which performance parameters are being monitored at the unit with existing instrumentation. See Table 4-1 for a list of parameters that can be monitored at a typical fossil station. This table came from EPRI Report TR-102127, Heat Rate Demonstration Project, Mt. Storm Unit 1. (Appendix K pp k10 and k11.)

3. Obtain readings of the selected parameters from Table 4-1 and compare the readings with expected values. If no historical readings are available, use Table 2-2 as a starting point. This table came from EPRI Report CS-4554, Heat-Rate Improvement Guidelines. (Table 3-3 on page 3-29 of CS-4554 is a more concise list of parameters.)

4. Determine the magnitude of the parameters deviation from expected. An aid in this is to review Table 2-1 for utility experienced deviations.

5. Determine the heat rate deviation as shown in Table 4-4.

6. List the parameters deviating from expected in descending order of effect on heat rate.

7. Select a parameter to be investigated from the list. Discuss the parameters and their deviation with unit operators and other plant personnel to insure its validity.

8. Review logic trees or parameter diagnostic tables to identify components effected and potential causes for the deviation. See the discussion on logic trees in Section 2. Modify the logic trees from one of the demonstration reports to suit the unit being evaluated.

9. Refer to a performance parameter accounting manual such as found in Appendix B of this manual (taken from EPRI GS-4554), or the Performance Tutorial in this section. Prepare a list of possible corrections to the deviation.

10. Review with operations and maintenance to determine the appropriate correction.

This completes the initial steps in evaluating heat rate degradation. Additional actions will require the expenditure of time, personnel and equipment resources.

Performance Parameters

Table 4-1
Performance Parameters *

Equipment	Parameter
TURBINE	
	Main Steam Temperature
	Main Steam Pressure
	Reheat Steam Temperatures (Cold and Hot)
	Reheater Pressure Drop
	HP Turbine Section Efficiency
	IP Turbine Section Efficiency
	LP Turbine Section Efficiency
	First Stage Pressure
	Turbine Shaft Seal Leak-Off Flow
	Throttle Flow
	IP Exhaust Steam Temperature and Pressure
	Vibration Levels
	Electrical Generation
TURBINE CYCLE	
	Cycle Isolation
	Feedwater Flow
	Condensate Flow
	Feedwater Heater Terminal Temperature Difference

Equipment	Parameter
	Feedwater Heater Drain Cooler Approach
	Feedwater Temperature Rise
	Extraction Line Pressure Drop
	Steam Attemperation Water Flow
	Unit Makeup Flow
	Air Preheater Steam Flow
	Pump Seal Leak-Off Flow
	Pump Seal In-Leakage
	Steam Driven Auxiliaries
	Generator Hydrogen Pressure and Purity
	Generator Power Factor
	Feedwater Heater Level
	Feedwater Heater Drain Valve Position
CONDENSER	
	Condensate Oxygen Level
	Terminal Temperature Difference
	Circulating Water Inlet Temperature
	Circulating Water Outlet Temperature
	Condenser Backpressure
	Condensate Temperature
	Condenser Waterside Pressure Drop
	Air In-leakage

Equipment	Parameter
BOILER	
	Air Preheater Leakage
	Exit Gas Temperature
	Exit Gas 02
	Exit Gas CO
	Exit Gas $C0_2$
	Combustibles in Ash
	Coal Moisture
	Mill Rejects
	Draft Losses
	Proximate or Ultimate Fuel Analysis, HHV
	Mill Fineness
AUXILIARIES	
	Auxiliary Electrical Power Consumption
	Auxiliary Steam Consumption
MISCELLANEOUS	
	Makeup
	Wet Bulb Temperature
	Wind Speed and Direction
	Dry Bulb Temperature

The intent of this list is to ensure that all major contributors to heat rate degradation will impact at least one of the performance parameters. Not only will monitoring these parameters characterize a unit's performance, these parameters will be useful in trending equipment performance for preventive maintenance.

Performance Tutorial

Introduction

The purpose of this tutorial is to be of some assistance in identifying and minimizing performance losses while operating a unit. Typical heat rate effect and dollar cost values are given to show the magnitude of individual performance parameters, for an actual US utility. Also, possible causes of deviations and possible corrections are listed to assist the operator in what action/s to be taken to eliminate or reduce the loss in the most cost effective manner. This information is similar to that found in Appendix B of EPRI CS-4554, and available in Appendix B of this report.

Throttle Temperature

Utility Average: Percent change in heat rate= .18%/10^{o}F (.32%/10°C)

Dollars cost per day = $369.60 /HR

Plant-Specific Value: The effect on heat rate of deviation in main steam temperature is provided with the unit thermal kit.

Possible Causes of Deviation

- Superheater spray control problems
- Superheater spray valve leakage
- Fouling of the superheater (low temperature)
- Fouling of the boiler waterwall (high temperature)
- High excess air
- Burner tilts mispositioned/broken - if applicable
- Bypass dampers mispositioned/if applicable

- Temperature control setting calibration drift
- Incorrect amount of heat transfer surface (superheat/waterwalls)
- Mill out of service/mill biasing
- Improper biasing of secondary air

Possible Corrections

Operator Controllable:

- Blow soot selectively
- Adjust burner tilts
- Adjust bypass damper settings/if applicable
- Adjust auxiliary air dampers
- Control excess air
- Manually control superheater spray flow
- Run proper number of mills/proper biasing

Maintenance Correctable:

- Calibrate temperature control setpoint
- Repair superheater spray control valve
- Clean boiler waterwalls
- Clean superheater platens
- Repair superheater tube leaks
- Add or remove superheater heat transfer surface
- Repair burner tilts
- Repair auxiliary air dampers

Throttle Pressure

Utility Average:	Percent change in heat rate	= .025%/10 psi (0.036%/100 Kpa)
	Dollars cost per day	= $49.68

Plant-Specific Value: The effect on heat rate of deviation in main steam pressure is provided with the unit thermal kit.

Possible Causes of Deviation

- Feedwater flow too low (once-through units)
- Firing rate inadequate
- Instrument
- Start-up/Silica
- Inadequate BFP/BFPT problems
- Recirculation valves leaking
- Pump problems

Possible Corrections

Operator controllable:

- Increase feedwater flow
- Increase firing rate (manual control only)
- Increase blowdown rate
- Instrument calibration

Maintenance Correctable:

- Pumps
- Valves

Reheat Temperature

Utility Average: Percent change in heat rate = 0.15%/10°F low (0.27%/10°C)

Dollars cost per day = $307.80

Plant-Specific Value: The effect on heat rate of deviation in reheat temperature is provided with the unit thermal kit.

Possible Causes of Deviation

- Reheat attemperation control problems
- Reheat attemperation control valve leakage
- Fouling of the reheater (low temperature)
- Fouling of the boiler waterwall (high temperature)
- Fouling of the superheater
- High excess air
- Burner tilts mispositioned/broken
- Bypass dampers mispositioned/if applicable
- Reheater tube leaks
- Incorrect amount of reheater heat transfer surface
- Mill out of service/mill biasing
- Improper biasing of secondary air

Possible Corrections

Operator Controllable:

- Blow soot/selective
- Adjust burner tilts
- Adjust bypass damper settings

- Adjust auxiliary air damper
- Control excess air
- Manually control reheat spray flow
- Run proper no. of mills/proper biasing

Maintenance Correctable:

- Repair reheater spray control valves
- Clean boiler waterwalls
- Clean superheater platens
- Clean reheater platen
- Repair reheater tube leaks
- Add or remove reheater heat transfer surface
- Repair burner tilts, binding, linkage, etc.
- Repair Auxiliary air dampers

Condenser Back-Pressure

Utility Average: Percent change in heat rate= +.25%/.1 Absolute Back Pressure (ABP)

Dollars cost per day = $950.00

Plant-Specific Value: The effect on heat rate of deviation in condenser backpressure is provided with the unit thermal kit.

Possible Causes of Deviation

- Air inleakage
- Excess condenser load (boiler feedwater pump & heater drain pump recir., flash t dump, steam traps)
- Tube fouling

- Tube bundle
- Design problem caused air binding and excessive pressure drop
- Steam by-pass into air cooling section
- Low circulating water flow
  - Continuous priming system
  - Vacuum breakers
  - Correct number of CW pumps
  - Clogged traveling water screen
- Increases in circulating water inlet temperature caused by:
  - Changes in ambient conditions
  - Problems with cooling tower performance
- Proper condenser setup (LP and HP condensers) and bundle design
- Poor performance from steam-jet air ejector (SJAE) and vacuum pumps
- Fouled water boxes and tube sheet

Possible Corrections

Operator Controllable:

- Increase circulating water flow/additional CW pump
- Add an additional vacuum pump/ SJAE
- Check cycle isolation
- Place additional cooling tower cells in service
- Maintain proper LP and HP service water
- Proper operation of Amertap tap systems
- Clean out water boxes

- Check operation of continuous priming systems/if applicable
- Shoot condenser
- Maintain proper rack and screen differential
- Check vacuum breakers seal

Maintenance Correctable:

- Repair condenser air leaks
- Repair cycle isolation valves
- Clean condenser
- Repair circulating water discharge control valve
- Repair cooling tower
- Repair valves on continuous priming system

Make-up Water

Utility Average:	Percent change in heat rate	= .12%/0.5% deviation
	Dollars cost per day	= $456.00

Plant-Specific Value: The effect on heat rate of deviation in make-up water is provided with the unit thermal kit.

Possible Cause of Deviation

- Boiler tube leaks
- Excess deaerator venting to atmosphere
- Excess continuous blowdown/ if applicable
- Excess steam lost through condenser venting (SJAE)
- Valve packing leaks
- Pump seal leaks

- Steam leaks to atmosphere
- Overflow of CST, RCW, BFP seal leakoff sump tank
- Lose RCW through vents on exciter cooler
- Soot blowers (boiler/preheater)
- Feed rate on condenser neck, vacuum breakers (if applicable)

Possible Corrections

Operator Controllable:

- Overflow of storage tanks
- Overflow of exciter cooler vents
- Overflow of condenser neck seal and vacuum breakers seal

Maintenance Correctable:

- Check deaerator vent orifices or valve settings
- Repair valve and pump packings and seals
- Repair boiler tube leaks
- Optimize continuous blowdown
- Isolate cycle losses

Feedwater Heaters

Utility Average:	Percent change in heat rate for 'A' heater	= .12%/5.0°F (0.22/5°C)
	Dollars cost per day	= $456.20
	Percent change in heat rate for 'B' heater	= .01%/5.0°F 0.018/5°C)
	Dollars cost per day	= $38.00
	Percent change in heat rate for 'C' heater	= .05%/5.0°F (.09/5°C)

Dollars cost per day	= $190.00
Percent change in heat rate for 'D' heater	= .03%/5.0°F (.054/5°C)
Dollars cost per day	= $114.00
Percent change in heat rate for 'E' heater	= .04%/5.0°F (.07/5°C)
Dollars cost per day	= $152.00
Percent change in heat rate for 'F' heater	= .04%/5.0°F (.07/5°C)
Dollars cost per day	= $152.00
Percent change in heat rate for 'G' heater	= .025%/5.0°F (.045/5°C)
Dollars cost per day	= $95.00

Plant-Specific Value: The effect on heat rate for changes in feedwater heater TTDs and DCAs are provided with the unit thermal kit. Terminal temperature differences (TTD) have a much more significant effect on heat rate than do drain cooler approach temperatures(DCA). Therefore, efforts should be made to obtain TTDs instead of DCAs.

Possible Causes of Deviation

- Improper heater level
- Improper extraction line pressure drop
- Reduced condensate flow through the heater(condensate bypass valve leakage)
- Heater partition plate leaks
- Failure to vent noncondensable gases
- Tube Fouling

Possible Corrections

Operator Controllable:

- Set feedwater heater levels

- Check vent system set-up

Maintenance Correctable:

- Optimize feedwater heater level
- Maintain heater vent valves and line orifices
- Repair partition plate leaks
- Clean tube bundles

Desuperheater Spray

Superheat Spray

Utility Average: Percent change in heat rate= .025%/1% of steam flow

Dollars cost per day = $51.30

Plant-Specific Value: The effect on heat rate of deviation in superheat spray flow is provided with the unit thermal kit. Usually the effect is negligible and an effect on heat rate penalty curve is not supplied in the unit thermal kit.

Possible Causes of Deviation

- Improperly adjusted control setpoint
- Leaking spray control valve
- Broken spray nozzle
- Fouling of boiler waterwalls
- High levels of excess air
- Burner tilts position
- Auxiliary coal-air dampers

Possible Corrections

Operator Controllable:

- Blow waterwall soot
- Reduce excess air to proper levels
- Adjust coal-air dampers

Maintenance Controllable:

- Repair spray valves
- Calibrate temperature controls
- Replace spray nozzle

Reheat Spray

Percent change in heat rate= .185%/1% of steam flow

Dollars cost per day = $379.62

Plant-Specific Value: The effect on heat rate of deviation in reheat spray flow is provided with the unit thermal kit. Normally a unit is designed to operate with zero reheat spray flow because it has a substantial heat rate penalty associated with it.

Possible Causes of Deviation

- Fouled waterwalls
- High levels of excess air
- Fouled superheater sections
- Improperly adjusted temperature setpoint
- Leaking spray control valve
- Broken spray nozzle
- Burner tilts position

- Auxiliary coal-air dampers

Possible Corrections

Operator Controllable:

- Adjust excess air to proper levels
- Soot blow waterwalls
- Soot blow superheater sections

Maintenance Controllable:

- Repair spray control valve
- Replace spray nozzle
- Calibrate temperature control setpoint

Auxiliary Steam Effect

Utility Average: Percent change in heat rate= .20%/.25% of steam flow

Dollars cost per day = $410.00

Plant-Specific Value: The effect on heat rate of deviation in auxiliary steam usage has to be determined from a heat balance code program or calculated by hand.

Possible Causes of Deviation

- Excessive soot blowing
- Excessive plant heating
- Steam traps leaking-blowing through
- Excessive steam coil usage and possible leaks

Possible Corrections

Operator Controllable:

- Optimize soot blowing
- Proper setup of plant heating equipment
- Check for proper steam trap operation
- Optimize usage of steam coils

Maintenance Correctable:

- Repair soot blower leaks
- Repair/Replace steam traps
- Repair/Replace leaks in steam coils

Boiler Losses

Dry Gas Loss

Utility Average: Percent change in heat rate= .25%/10°F (5.5°C) increase in exit gas

Dollars cost per day = $513.00

Plant-Specific Value: The effect on heat rate of deviation in dry gas loss is provided with the boiler contract performance data.

Possible Causes of Deviation

- Excess air
- In-leakage from bottom-ash hopper to preheater inlet
  - Hopper doors leaking/open
  - Hot precip. doors leaking/open
- Excess mill tempering air due to low mill temperature

- Incorrect operation of prewarming air preheater inlet air
  - FD fan room doors open
  - Preheater room windows open
- Preheater bypass damper open
- Poor mill performance
- Poor preheater efficiency
  - Pluggage/fouled
  - Corroded/eroded
- Mill exhauster damper position
- O_2 measurement calibration
- Correct no. of mills in service for a given load
- Excess furnace draft (higher draft is worst for in-leakage)

Possible Corrections

Operator Controllable:

- Reduce excess air
- Ensure hoppers, slag ports, etc., are closed tightly
- Adjust mill primary air flow to minimize tempering air flow
- Adjust mill exhauster dampers to maintain proper air flow
- Achieve design mill outlet temperature and exceed if possible
- Ensure proper setup of windows and doors throughout the plant for maximum inlet air temperature to preheater.
- Check setup of preheater bypass damper/if applicable
- Proper soot blowing for preheater

- Proper operation of steam coils to prevent fouling and corrosion.
- Proper soot blowing for boiler
- Proper no. of mills in service

Maintenance Correctable:

- Repair expansion joints, door gaskets on hoppers, and preheater seals
- Repair/Replace preheater baskets
- Mill maintenance

Auxiliary Power

Utility Average: Percent change in heat rate= .21%/1.0 MW

Dollars cost per day = $430.92

Plant-Specific Value: The effect on heat rate of deviation in auxiliary power usage can be obtained by using the following equation:

AUXTEF = ((AUXT - AUXTX) / (LOAD - AUXT)) * 100

AUXTEF = % Effect on Heat Rate for Total Auxiliaries

AUXT = Actual Total Auxiliaries, MW

AUXTX = Expected Total Auxiliaries, MW

LOAD = Unit Gross Load, MW

The expected total auxiliaries for a given load has been developed for each unit. This curve should be updated whenever additional equipment is added (i.e., precip., air compressors, scanner fans, etc.).

Possible Causes of Deviation

- Idling equipment
  - Transfer pumps
  - Air Compressors

- Outdoor Lighting
- Coal Handling equipment
- Air Conditioners

- Decline in efficiency of operating equipment
- Operation of redundant equipment at low load operation
- Operation of additional equipment for an excessive operating cushion (Heater Drain P, Condensate Booster Pump, Service Water Pump, Pulverizers).

Possible Corrections

Operator Controllable:

- Stop Idling equipment
- Reduce equipment operation at low loads
- Reduce excessive operating cushion
- ntenance Correctable:
- Repair or replace inefficient equipment
- Maintain equipment whose power usage increases with deteriorating performance, e.g., electrostatic precipitators, pulverizers, pumps, etc.

High Pressure Turbine Efficiency

Utility Average: Percent change in heat rate= .18%/1.0% decline in efficiency

Dollars cost per day = $381.77

Plant-Specific Value: The effect on heat rate of deviation in high pressure turbine efficiency can be determined from a heat balance code program or calculated by hand.

Possible Causes of Deviation

- Erosion of nozzle blocks
- Erosion of turbine blades

- Damaged turbine blades
- N2 packing seal (HP and IP turbine are in same shell)
- Excess shaft packing leaks
- Excess spill strips and diaphragm packing leaks
- Malfunctioning control valves
- Plugged nozzle blocks
- Fine mesh strainers not removed
- Deposits on nozzles and/or blades
- Improper stroke on stop, intercept, and control valves

Possible Corrections

Operator Controllable: The operator can minimize the reduction in the high pressure turbine efficiency by preventing these conditions from occurring by how well he starts and operates the unit.

Maintenance Correctable:

- Repair or replace nozzle block
- Repair or replace turbine blades
- Clean turbine blades
- Replace shaft packings
- Replace turbine spill strips and diaphragm packings
- Ensure proper setup of stop valve, intercept, and control valves.

Intermediate Turbine Efficiency

<u>Utility Average</u>:	Percent change in heat rate	= .13%/1.0% decline in efficiency
	Dollars cost per day	= $266.76

Plant-Specific Value: The effect on heat rate of deviation in intermediate pressure turbine efficiency can be determined from a heat balance code program or calculated by hand.

Possible Causes of Deviation

- Erosion of turbine blades
- Deposits on turbine blades
- Excess shaft packing leakage
- Excess spill strips and diaphragm packing leakage

Possible Corrections

Operator Controllable: The operator can minimize the reduction in the intermediate pressure turbine efficiency by preventing these conditions from occurring by how well he starts and operates the unit.

Maintenance Correctable:

- Repair or replace turbine blades
- Repair shaft packings
- Repair spill strips and diaphragm packings

Low Pressure Turbine Efficiency

Utility Average: Percent change in heat rate= .11%/1.0% decline in efficiency

Dollars cost per day = $225.72

Plant-Specific Value: The effect on heat rate of deviation in low pressure turbine efficiency can be determined from a heat balance code program or calculated by hand.

Possible causes of deviation and possible corrections are applicable to the low pressure turbine same as the IP turbine. A problem arises with the LP turbine because the steam conditions exiting the LP turbine cannot be easily measured. Hence, the amount of energy removed from the steam by the LP turbine cannot be determined from a typical turbine output and enthalpy drop test. The LP turbine efficiency effect on heat rate resides in the reported unaccounted-for loss value.

Generator Efficiency

Utility Average: Percent change in heat rate= .06%/15 psig H_2 above expected (.087/150 Kpa)

Dollars cost per day = $123.12

Plant-Specific Value: The effect on heat rate of deviation in generator efficiency is provided with the unit thermal kit.

Possible Causes of Deviation

- Improper hydrogen pressure for a given field and/or armature temperature
- Power factor setting
- Hydrogen purity
- Poor hydrogen cooling flow and/or temperature conditions

Possible Corrections

Operator Controllable:

- Set hydrogen pressure at the lowest pressure allowable
- Operate at nearest unity power factor permitted by dispatch
- Operate purification system properly to ensure highest hydrogen purity/if available
- Maintain optimum hydrogen cooling flow and temperature conditions

Maintenance Correctable:

- Repair or replace hydrogen coolers to ensure hydrogen purity
- Reinsulate generator to lower armature and field temperatures which allows lower hydrogen pressure operation.

Miscellaneous Items

Flashtank Dump Valve 30% Open (subcritical)

Utility Average: Percent change in heat rate= .13%/4.0% of steam flow

Dollars cost per day = $266.78

Plant-Specific Value: The effect on heat rate of deviation in miscellaneous items will have to be determined from a heat balance code program or calculated by hand.

Possible Causes of Deviation

- Excessive Boiler Circulating Pump seal leakoff flow which the HDP system cannot handle
- Controllers for flashtank level operating improperly
- HDP in service cannot develop enough head (weak pump)
- Excessive CBPs in service
- An heater upstream of the HDP has a tube leak

Possible Corrections

Operator Controllable:

- Start/Operate BCP properly to minimize seal damage
- Operate proper number of CBP (reduce BFP suction pressure to allowable limits
- Operate standby HDP until root cause for flashtank dump valve opening has been resolved

Maintenance Correctable:

- Repair/Replace BCP seals
- Calibrate controllers for flashtank level
- Rebuild HDP

- Rebuild CBP
- Repair heater with tube leaks

Mill Coal Spillage

Utility Average:	Percent change in heat rate	= 0.10%/0.10% of coal flow
	Dollars cost per day	= $205.00

Plant-Specific Value: The effect on heat rate of deviation in mill coal spillage is proportional to the heat rate. Expected mill coal spillage is zero percent.

Possible Causes of Deviation

- Inadequate air flow to mill
- Poor mill condition
- Improper journal and spring tension
- Foreign objects lodged in mill
- Coal grindability change
- Coal size (crusher working improper/excessive moisture (>10%)

Possible Corrections:

Operator Controllable:

- Set proper air flow to mill
- Remove foreign objects
- Set proper classifier settings/if applicable

Maintenance Correctable:

- Repair/Replace worn-out mill parts
- Set proper mill journal and spring tensions

- Set proper classifier settings (B&W)
- Repair/Replace crusher

Excess Drains to Condenser

Possible Causes of Deviation

- Valve stems broken and/or valve leaking
- Steam traps leaking
- Start-up valves operating improperly
- Recirculation valves leaking and/or setup improperly
- Vent orifices worn and/or bypass vent valves open/leaking
- Steam seal diverting valve setup improperly
- FWH dump valve leaking and/or setup improperly
- Miscellaneous drain and turbine drain valves leaking
- Flashtank dump valve on subcritical units operating improperly

Possible Corrections:

Operator Controllable:

- Check systems setup
- Maintain steam trap survey
- Ensure all startup and regulating valve are operating properly and are not leaking

Maintenance Correctable:

- Repair/Replace if needed

Excess Steam-Jet Air-Ejector in Service

A steam-jet air-ejector is a piece of equipment design to remove non-condensable gases from the condenser so steam exiting the turbine can be condensed. If non-condensables are not removed they act as a blanket on condenser tubes and prevent heat transfer from taking place. When the non-condensables present in the condenser exceeds the removing capability of the steam-jet air-ejector in service then blanketing could possibly take place. A simple check of placing the backup steam-jet air-ejector in service along with the other jet and monitoring the condenser vacuum will determine whether or not the existing jet's capability is exceeded. If any increase in condenser vacuum is experienced then the gains in efficiency easily offsets the operation of an additional steam-jet air-ejector. Therefore the operation of an additional steam-jet air-ejector with no gains in condenser vacuum causes an efficiency loss due to steam used for jet operation plus unnecessary wear on equipment.

Excess Recirculation

Possible Causes of Deviation

- Boiler feed pump to deaerator/condenser
- Heater drain pump to condenser
- Condensate recirculation to lower/upper storage tanks

Possible Corrections

Operator Controllable:

- Proper setup of feedwater/condensate cycle(no. of pumps, valves, etc.)

Maintenance Correctable:

- Repair/Replace broken valves
- Repair/Replace weak pumps

Baseline Evaluation

The performance engineer now needs to establish the present level of unit performance and to then compare the results with some baseline in order to evaluate conditions contributing to heat rate degradation. This baseline can be vendor design values (thermal kits), results from performance or ASME code testing, industry standards or any combination of these.

Take key parameter measurements around the boiler, turbine, feedwater, cooling water cycles, fuel handling systems, auxiliary systems and look for increased heat and inventory losses.

Table 4-1 will not apply to every type of unit but must be adjusted for the type of fuel consumed and any design differences which affect heat rate performance.

In formulating this list, the intent is to ensure that all major contributors to heat rate degradation will impact at least one of the performance parameters. In addition to helping to characterize a unit's performance, the performance parameters will be useful in trending equipment performance. The information presented in a logic tree which has been constructed for a specific unit can aid in identifying important performance parameters.

The intent is to define a subset of the total set of performance parameters which best characterizes the performance of the unit. This subset can be the performance parameters which have the greatest effect on heat rate, such as is shown in Table 2-2, or as a result of the information obtained by performing a limited cycle evaluation which identifies those areas where degradation is occurring. Other sources of information which can be used to help identify important performance parameters are: the experience of the plant operations and maintenance staff, information supplied by equipment vendors, the plant thermal kit, heat balance codes, baseline and acceptance tests, and the experience of other utilities.

Baseline Example

To illustrate this concept, let's look at an actual utility experience. This utility had determined that the deviation between a unit's actual and best achievable heat rate was 281 Btu/kWh (296 kJ/Kwhr). At first this utility was examining the effects on heat rate of the five parameters which are listed in the table below.

Table 4-2
Baseline Shortlist of Heat Rate Losses

Parameter	Loss	Loss
Condenser Back Pressure	31 Btu/kWh	32.7 KJ/kWh
Reheat Steam Temperature	11 Btu/kWh	11.6 KJ/kWh
Excess Air	2 Btu/kWh	2.1 KJ/kWh
Main Steam Temperature	3 Btu/kWh	3.2 KJ/kWh
Exit Gas Temperature	0 Btu/kWh	
Total Heat Rate Losses	47 Btu/kWh	49.6 KJ/kWh

The total of the heat rate losses which could be attributed to these five parameters was 17 percent of the known deviation. This indicated that the proper set of performance parameters was not being monitored. The list was expanded to include additional parameters.

Table 4-3
Expanded Heat Rate Losses

Parameter	Loss	Loss
Station Load	77 Btu/kWh	81.2 kJ/kWh
Throttle Pressure	13 Btu/kWh	13.7 kJ/kWh
Makeup	1 Btu/kWh	1 kJ/kWh
Reheat Attemperation	12 Btu/kWh	12.7 kJ/kWh
HP Efficiency	67 Btu/kWh	70.7 kJ/kWh
IP Efficiency	7 Btu/kWh	7.4 kJ/kWh
HP Htr TTD	1 Btu/kWh	1 kJ/kWh
Coal Moisture	11 Btu/kWh	11.6 kJ/kWh

The total heat rate losses which could be accounted for from all of the monitored parameters was now 210 Btu/kWh (221 KJ/kWh) or 75 percent of the calculated deviation.

Monitoring Recommendations

Once it has been decided which performance parameters should be monitored, and the affect of each of these parameters on unit heat rate has been determined, the type of instrumentation to be used and the frequency of monitoring each parameter must be decided. Measurements can be made using either normal station instrumentation, calibrated station instrumentation, or test instrumentation.

Calibrated test instrumentation would provide the most accurate indication, however, the cost of purchasing, installing, and maintaining this instrumentation for monitoring performance parameters may be beyond the budgetary constraints of the utility. The next preferable method would be to have all of the required station instrumentation calibrated prior to collecting the performance parameter data. There is normally no problem with calibrating plant pressure gauges but plant thermocouples normally cannot be calibrated in-situ. To resolve the latter problem, it may be necessary to install redundant thermocouples in spare thermowells to be read either locally or by an automated data collecting device. See Section 5 of this report for more detailed instrumentation recommendations.

The data should be collected for a period of time which will give a representative sample of the unit's performance. The data should be collected at all of the load points at which the unit is normally operated for extended periods of time. The data should be averaged to smooth out any fluctuations.

Example Calculations

Once an estimate of the deviation in each of the performance parameters has been made the next step is to determine the impact of these deviations on heat rate. This involves translating the deviations in each performance parameter into a heat rate loss. This is done by multiplying the deviation of the performance parameter from its expected value by the estimated worth of that parameter on heat rate. An example calculation of parameters having a major effect on heat rate is shown in Table 4-4. The expected value can be either the design value of that parameter, as determined by the boiler or turbine vendor, or it can be some value which is less than the design value determined by any limiting conditions of operation or design which affect that parameter.

Example Calculation of Heat Rate Loss

Table 4-4
Example Calculation of Heat Rate Loss by Parameter Deviation Method

Performance Parameters	Actual English	Actual S. I.	Target English	Target S. I.	Deviation English	Deviation S. I.
Throttle Pressure	1819.0 psig	12.6 mPa	1820.0 psig	12.65 mPa	0.4 Btu/kWhr	0.42 kJ/kWhr
Throttle Temperature	993.0 °F	534 °C	1000.0	538 °C	9.8 Btu/kWhr	10.3 kJ/kWhr
Hot Reheat Temperature	976.0 °F	524 °C	1000.0	538 °C	31.2 Btu/kWhr	32.9 kJ/kWhr
Station Service	7.63 %		6.39		106.6 Btu/kWhr	112.5 kJ/kWhr
Condenser Backpressure	2.14" Hg abs	.072469 Bar	1.84	.062309 Bar	61.2 Btu/kWhr	64.6 kJ/kWhr
Makeup	.99		.80		2.6 Btu/kWhr	2.7 kJ/kWhr
Exit Gas Temperature	348.0 °F	176 °C	293.0	145 °C	148.5 Btu/kWhr	156.7 kJ/kWhr
Excess Oxygen	5.04 %		4.26		22.9 Btu/kWhr	24.2 kJ/kWhr
Unburned Carbon	9.3 %		5.0		50.3 Btu/kWhr	53.1 kJ/kWhr
Coal Moisture	6.9 %		7.0		-0.8 Btu/kWhr	-.84 kJ/kWhr
H.P. Efficiency	82.0 %		89.5		141.0 Btu/kWhr	148.8 kJ/kWhr
I.P. Efficiency	89.5 %		84.1		-78.3 Btu/kWhr	-82.6 kJ/kWhr
TOTAL DEVIATION					495.4 Btu/kWhr	522.7 kJ/kWhr

Performance Loss Monitoring and Trending of Key Parameters

The following section describes how the units performance can be surveyed for losses and trended to follow the effects of operation.

Loss Monitoring

Performance loss monitoring is required for determining how well a unit is being maintained and operated, and in establishing its actual performance level. A successful loss monitoring program consists of gathering accurate operating data from adequate sources to provide a complete status of unit operating parameters. This data must then be incorporated into the proper calculations for determining actual performance losses for cost/benefits analysis. Once a source of degradation has been identified through loss monitoring the plant staff can then pursue determining the root cause for the degradation. From the root causes of heat rate degradation plant staff can optimize their preventive maintenance program, their instrumentation requirements, operating practices, and their performance parameter monitoring needs.

Classification

Many performance losses can be reduced or eliminated through careful unit operation, while others are correctable only by equipment maintenance. Other performance losses remain unknown as to cause or effect on unit efficiency. For these reasons, it is helpful to classify performance losses so responsibilities for loss reduction can be more effectively delegated. The suggested categories for classifying performance losses are controllable losses, accounted-for losses, and unaccounted-for losses. A description and listing of these losses is contained in the following subsections.

Controllable Losses

Controllable losses are performance losses that can be minimized by the plant operating personnel. A list to typical controllable losses is as follows:

- Throttle temperature
- Throttle pressure
- Hot reheat temperature
- Condenser pressure
- Make-up flow

- Feedwater heater terminal temperature difference
- Feedwater heater drain cooler approach temperature
- Main steam desuperheater spray flow
- Reheat desuperheater spray flow
- Auxiliary electrical loads
- Dry gas loss
- Carbon loss
- Coal weighing error

Accounted-for Losses

Accounted-for losses are those remaining performance losses for which an effect on heat rate can be determined. Accounted-for losses may consist of both controllable and uncontrollable performance losses and are usually correctable by maintenance. A list of accounted-for losses may include the following:

- Reheater pressure drop
- Extraction line pressure drop
- Hydrogen loss (boiler)
- Moisture in fuel loss
- RUMA (radiation, unaccounted, moisture in air) loss (boiler)
- Turbine efficiency
- Miscellaneous
- Light-off fuel

Unaccounted-for Losses

Unaccounted-for losses are performance losses for which an effect on heat rate cannot easily be established. These losses may or may not be known to exist. A list of unaccounted-for losses that may exist is as follows:

- Heat loss to condenser (i.e., steam traps)
- Soot blowing
- Steam coils usage
- Plant auxiliary steam heating
- Condensate/Feedwater recirculation
- Improper valve alignment (i.e., improper routing of feedwater heater drain flow)
- Excessive turbine shaft seal leakages
- LP turbine efficiency
- Miscellaneous
  - Ignitor and scanner air effect on dry gas loss
  - SO_3 system steam usage
  - Mill coal spillage
  - Increase in boiler radiation loss due to insulation degradation

When an unaccounted-for loss can be quantified through calculations and/or testing, the loss may then be placed in the accounted-for loss category.

Performance parameters are a collection of plant parameters that are directly related to heat rate. These parameters may be measured directly or are calculated from other parameters. By using the list below it is possible to obtain a reasonable complete characterization of all important contributors to heat rate.

Quantification

To achieve total benefit from monitoring performance losses, the effect on unit efficiency must be correctly quantified. Pertinent data must be logged from operating instrumentation for use in accurate calculation algorithms. To emphasize the importance of monitoring sufficient performance parameters to guarantee a quality performance program an example is given below showing the deviation between a unit's actual and best achievable heat rate. The difference was 250 Btu/kWh (263 KJ/kWh) between actual and best achievable heat rates.

The effect on heat rate of these parameters was:

- Main steam temperature 0.10%
- Main steam pressure 0.05%
- Reheat steam temperature 0.08%
- Exit gas temperature 0.35%
- ABP 0.15%
- Total Accounted-for losses 0.73%

Actual On-Line Efficiency Factor = (9500 - 9250)/9250 * 100

= 2.70%

Only 27.0% of the unit's performance degradation has been identified. This indicates that the proper set of parameters was not being monitored. The list was expanded to include additional parameters.

These additional parameters and their heat rate deviations were:

- Reheat spray flow 0.07%
- Auxiliaries 0.45%
- Combined heaters TTD 0.09%
- HP turbine efficiency 0.60%
- IP turbine efficiency -.25%

• Coal weighing error	0.10%
• Carbon loss	0.17%
• BFP turbine efficiency	0.13%
• Total Accounted-for losses	1.36 + .73 = 2.09%

Now a total of 77.4% of the unit's performance degradation has been identified by the additional parameters being monitored. To properly gauge the importance of any one parameter in identifying heat rate degradation, it is necessary to know not only what effect a deviation of that parameter will have on heat rate but also a dollar value associated with each deviation percent effect on heat rate. Examples of cost per day have been given in the performance tutorial section of this report for multiple performance parameters.

Loss Quantification Procedure

The examples which follow will illustrate the percent effect on heat rate and the dollars associated with deviations in key performance parameters. Examples from a typical unit were used, with the assumptions noted below:

• Capacity Factor	= 100%	
• Heat Rate	= 9500 Btu/kwh	10,022 kJ/kWh
• Coal Cost	= 1.8000 $/mBtu	1.90 $/mkJ
• Coal Cost	= 45.0 $/Ton	40.82 $/Ton (metric)
• Coal HHV	= 12,500 Btu/lb	(5,982 kJ/kg)
• Average Gross Load	= 500 Mw	

To adjust any fuel cost for the actual capacity factor, simply multiply the fuel cost by that capacity factor divide by 100.

Using the above assumptions, the fuel cost increase due to a 1.0 percent increase in heat rate is calculated as shown below:

(.01)(9,500 Btu/kWh)(500,000 Kw) = 47,500,000 Btu/hr (50,120,000 kJ/kWh)

(47,500,000 Btu/hr)/(12,500 Btu/lb) = 3,800 lb/hr (1724 lb/hr)

The use of 3,800 lb/hr additional coal for an entire month would yield the following monthly fuel cost increase at an capacity factor of 100 percent.

(3,800 lb/hr)(24 hr/day)(30 day/month)(45 $/ton)(ton/2000 lbs)(1.0) = $61,560/month

To adjust the fuel cost for actual capacity factor compute as follows:

($61,560/month)(.63 Capacity Factor) = $38,783/month

The dollars cost per day reported in the Performance Tutorial section of this document were calculated using the assumptions above and the percent change in heat rate given for each performance parameter. To obtain precise values for a given unit simply substitute correct inputs in the equations above.

Unit Performance Survey

Introduction

Not since the initial acceptance tests on many fossil units has an overall level of performance been established. Since that time unit operating modes have changed (i.e., from baseload to cycling to peaking), modifications have been made (i.e., addition of precipitators), coal quality has changed, and equipment has aged. With this in mind it is the primary goal of the Unit Performance Survey program to determine the present level of overall performance and to improve it by identifying and minimizing losses. Data gathered during the survey should be compared where possible to contract data, expected data or previous data. The main objectives of the Unit Performance Survey program are:

To stress the importance of performance and to provide a means for "on-the-job" performance training for station and results personnel.

To establish a present level of performance for each fossil unit.

To identify cycle and equipment problems and obtain information for use in problem resolution.

To improve the present level of performance of each fossil unit by identifying and minimizing losses.

The program as described herein would be implemented on one unit at a station. With the Results Group working closely with station personnel, the subject unit would be examined from "top-to-bottom" as outlined. As losses are uncovered, many will lend themselves to immediate correction, others will require long-range planning and budgeting.

If significant improvements are realized on the surveyed unit then the station would apply the knowledge gained to their other units.

Preparation

Preliminary work by Results Groups

Cycle Review and Familiarization

The Results Group (RG) working with the Station Survey Coordinator (SSC) would assemble one-line drawings, heat balance diagrams, valve set-up drawings, and other available information on the subject unit for review and familiarization.

Manufacturer's Contract Reviews

RG (with the SSC) would assemble all boiler, condenser and heater contracts for the subject unit for review. These contracts along with items from Cycle Review and Familiarization would be put into a working notebook to be used during the survey for reference.

Review Past Test Data

Past test data will be reviewed by the RG (with SSC). Such data to be reviewed would be turbine acceptance tests, boiler tests, turbine output checks (TOCs), heater TDs, and 0 rise tests, possibility of plotting some of this information to reveal trends, such as heater TDs.) Actual data or summaries of this data should be included in the working notebook for the survey.

Review Known Problems

Known problems on the subject unit should be reviewed to identify the need for special tests or investigations during the Unit Survey.

Formulation of a Tentative Plan and Schedule

Having established a background on the subject unit, then a tentative plan and schedule for the survey should be prepared by RG (with SSC) to present to the station management.

Meeting with Station

Station manager, all superintendents, supervisors, plant engineers and the station survey coordinator (if different from preceding) would attend this Unit Survey meeting.

Review of Preliminary Work

A review of preliminary work by the RG (and SSC) for accuracy in regard to systems set-up, changes to systems, etc., should be done.

Known Problems

A review of known problems for the subject unit should be done to allow additional feedback on these problems as well as a chance to add any items overlooked.

Review of Tentative Plan and Schedule

Layout of the tentative plan should allow for comments and changes as necessary from the various departments of the station. A review of the schedule would show the impact on workforce for the different groups.

Formation of Final Plan and Schedule

After the plant meeting, a formal plan and schedule would be prepared and the survey period would begin. Schedule and plan both should have to be flexible due to scope of work.

Unit Survey

The unit survey should address each major system of the unit as a separate item. These systems should be covered as follows:

Turbine-Cycle Survey

Heat Loss to Condenser

Using a pyrometer or a sonic tester, all drain lines to the condenser should be checked. If a line is found hot, it should be traced to the source of the heat and noted. If steam traps are found leaking, then they should be identified for replacement with appropriate design. Water-induction system should be checked for leaks.

Unit Isolation, Water Balance, TOC and Enthalpy Drop

The subject unit should be isolated as is done for a turbine acceptance test. This will include a normal flow path verification for the Condensate-Feedwater System. This should find extraneous flows, such as condensate to the ditch or to the storage tanks. Irregularities in flow path will be noted and corrected where possible.

A water balance with the unit isolated should then be performed at full load. This is an attempt to account for water losses from the system by measuring the drop in the hotwell level, storage tank levels, and drum level over a period of time, and comparing this to the known leakage from the system. Determining leakages will require actual physical measuring of leaks at pumps, gland-steam exhausters, valve packing, condenser neck seal, boiler blowdown, etc. Drains to condensate storage tanks should be noted.

At this point a routine TOC, enthalpy drop, and seal leakoff test should be performed on the unit. A comparison to base for percent deviation would allow turbine losses to be accounted for to a degree (plot section efficiencies and pressure ratios). Where possible, turbine clearances in shaft packings, diaphragm packings, and spill strips would be evaluated for effect on heat rate. This clearance data would come from the last turbine outage reports. Included in seal leakoff tests will be a check of the Steam-Seal Regulator System. (Main steam leaking through, etc.)

Feedwater Cycle

All elements of the Feedwater Heater System should be checked. These items are:

- Heater terminal differences and drain cooler approach as applicable.
- Heater level controls over the load range.
- Heater drain system (proper routing and valve alignment).
- Heater vent system and shell-pressure relief valves.
- Heater bypasses checked for feedwater bypassing heater. Bleed pressure drops.
- Heater drain pumps over the load range (covered in more detail under pump performance).
- Heater hydraulic pressure drops.

Boiler Survey

Mills

The pulverizer mills should be looked at in detail for this is an area that can contribute significant gains in regard to performance. The areas to be addressed during the survey of the mills are provided in the following sections.

Mill Inspection

A full mill inspection should be performed on each mill. Items to be inspected include:

- Roll adjustment
- Roll wear
- Ring wear
- Cone condition
- Classifier conditions
- Liner wear
- Pyrite scraper wear
- Air gap (restriction angle) between bowl and mill housing
- Damper conditions (hot air, exhauster and tempering). Hot-air duct inspection for flyash buildup

Mill Clean-Air Flow Test

The clean-air flow as measured in each transport pipe should be compared to mill design air flow. Also, velocities in individual transport pipes should all be within +5% of each other for a particular mill. Problems here could be the result of plugged transport piping, worn orifices, worn exhauster blades or improperly adjusted exhauster dampers.

Mill Capacity Check

The ability of a mill to grind to its design capacity will be tested. A Hargrove grindability number will be obtained as this is the key to a mill's ability to pulverize.

Included in this test will be a mill tailings test to determine the amount of tailings a mill is spilling. A coal fineness sample will be taken during this test and if found necessary the mill fineness can be set. Also the feeder speed and primary air damper controls will be checked for proper operation. A carbon loss sample is collected during this test, but a closer look at this will be made during the boiler tests.

Boiler Test

Boiler tests should be run. A boiler manufacturer's field engineer should probably be available to assist during the boiler tests. Results of the boiler tests should reveal problem areas such as carbon loss, air in-leakage, dry gas loss and air heater problems. Existing instrumentation on the boiler such as temperature indicators and 0_2 analyzers should be checked for accuracy by temperature and 0_2-traverses where possible. Damper operation effect on fuelball, ignition point, mill temperature, capacity, carbon loss, and temperature balance in the boiler should be examined. A comparison of actual boiler draft losses to predicted draft losses should be made.

Steam Temperature Study

A steam temperature study should be conducted over the load range. The effects of the following items will be examined:

- Desuperheating spray
- Soot and slag blowing practices
- Damper operation
- Excess air

The above characteristics will reveal the ability of the boiler to maintain the manufacturer's expected steam temperatures over the load range. If a problem exists and the temperatures cannot be maintained, steps should be taken to resolve the problem.

Boiler Heat Losses

Steam loss from the boiler from boiler blowdown, drain valves, relief valves will be noted and eliminated where possible.

Condenser Survey

The condenser has a major impact on unit performance. Therefore, a detailed study should be conducted in regard to condenser performance. The following parameters will be examined:

- Absolute back-pressure
- Condenser terminal difference
- Cooling-water temperature rise
- Air inleakage
- Condenser cleanliness factor
- Condenser tube pluggage
- CW pump operating criteria

Miscellaneous Steam Surveys

Other items that should be covered in the Unit Survey are:

Performance of Boiler Feed Pumps, Circulating Water Pumps, Hotwell Pump, Condensate Booster Pumps, and heater drain pumps be investigated. The performance of these pumps will be evaluated by one of several methods depending on the individual situation.

The electrical auxiliaries will be reviewed in two main categories unit and non-unit auxiliaries. This will not be a kilowatt search, such as measuring the power consumption of each component, but a look at the economics of the operating practices (i.e., if the component is operated when it is not needed). A comparison between the actual running auxiliaries and the expected curve will be made over the load range. Any significant deviations would warrant a closer look with an attempt to determine the reasons for these deviations.

Unit auxiliaries - Example of unit auxiliaries would be pumps (including sump pumps), fans, and precipitators.

Non-unit auxiliaries - Examples of non-unit auxiliaries would be lighting, coal handling, air conditioning, and air compressors.

Fuel Weighing and Sampling

A general review of the fuel weighing (coal) and measuring (oil) and sampling arrangement should be conducted.

- Material and chain test
- Trackscale and belt scale maintenance
- Coal accounting and distribution
  - Fuel oil accounting (start-up and firing)
  - Coal sampling methods and procedures

Auxiliary Steam System

The auxiliary steam system should be examined in detail. Traps, drains, relief valves, heating units, temperature and pressure controls will be checked for proper operation. Steam-to-air heater coils, demineralizer systems and others should be accounted for in the survey.

Instrument and Control Systems

The ability of the units instrument and control systems to maintain good operating parameters should be examined.

Unit Capacity Test

A unit maximum capacity test should be run to determine the parameters that limit load on the unit. A look at these limiting parameters will be made to determine if any can be eliminated.

Compressed Air System

This system should be examined to see if air leakage is causing compressors to be overworked or the air supply to be borderline. A "walk through" of the air system should reveal any major problems.

Unit Performance Survey Conclusions

When the unit survey proper has been completed, then a report of findings will be compiled. The following steps will be taken:

- Compare all data to contract, predicted or start-up data, where possible.
- Identify losses and problems on the following:
  - Turbine cycle and Boiler cycle (i.e., Condenser and Other Cycles)
- Assign heat rate and unit capacity effects to losses (show dollar figures).
- Plan for follow-up and correction of losses (responsible persons).
- Evaluate the overall benefit of unit performance survey program, both by station and general office.

Performance Testing of Specific Components

Performance testing is valuable in determining equipment performance degradation when the methods defined above do not provide sufficient direction. Plant performance testing can be greatly simplified through the use of automated data collection and data processing devices, however, it is possible, but at times less attractive, to perform these tests manually. This section will describe the procedure for performing those tests which can greatly aid the performance engineer in maintaining or optimizing unit heat rate and which are typical of tests normally performed by utilities who routinely test. In addition, some of the described tests, while not directly improving heat rate, can provide the performance engineer with the necessary information to influence plant maintenance decisions which result in improved performance.

Boiler Testing

The primary purpose of conducting boiler tests is to trend the performance of the boiler over time and to compare that performance against accepted performance. The parameter which is normally used to trend boiler performance is boiler efficiency which is the ratio of the energy absorbed by the working fluid to the energy input into the boiler.

Boiler Efficiency

Boiler efficiency can be calculated by one of two methods – the input/output method or the method of heat losses. In the input/output method, the energy absorbed by the working fluid is calculated from the enthalpy rise of the fluid entering the economizer and the enthalpy rise of the fluid entering the reheater. The equation for boiler efficiency is:

Efficiency = Energy out/Energy in

= w1* (hl-h2)+w2* (h3-h4)/Fuel Flow* Heating Value

where:

w1 = throttle flow

w2 = reheater flow

h1 = enthalpy at throttle conditions

h2 = enthalpy at economizer inlet

h3 = enthalpy at hot reheat

h4 = enthalpy at cold reheat

The above form of the boiler efficiency equation assumes that there is no flow diversion from the system in the form of boiler tube leaks or boiler mass blowdown. It also does not take into account any desuperheating spray which may exist. If either superheater or reheater spray is functional during the test, the following terms should be added to the numerator of the above equation:

Energy added to superheater spray = w3*(h1 - h5)

Energy added to reheater spray = w4*(h3 - h5)

where:

w3 = superheater spray flow

w4 = reheater spray flow

h5 = enthalpy of the spray water

Boiler continuous blowdown is normally isolated during a boiler test and does not need to be factored into the calculation.

The energy input to the boiler is the product of the measured fuel input into the boiler for a specified time period and the fuel's higher heating value (HHV). From this information it can be seen that the following measurements must be made to calculate boiler efficiency by the input/output method:

- Throttle Pressure and Temperature
- Cold Reheat Pressure and Temperature
- Hot Reheat Pressure and Temperature
- Economizer Inlet Pressure and Temperature
- Feedwater Flow
- Reheater Flow
- Fuel Flow
- Higher Heating Value of Fuel

The prescribed routine for the performance of a boiler efficiency test is to isolate boiler continuous blowdown and combustion air heating equipment during the period of the test. Rather than measure the reheater flow, it can be estimated using the measured value of the feedwater flow and the heat balance estimates of high pressure heater extraction flow and seal flows. If a more accurate estimate of the reheater flow is desired, additional measurements must be made to allow the extraction flow to be calculated from a mass and energy balance around the high pressure heater. These measurements include:

- Extraction Pressure and Temperature
- Heater Inlet Pressure and Temperature
- Heater Drain Cooler Pressure or Temperature

Most of the measurements needed for performing an input/output efficiency calculation are part of the plant's normal complement of instrumentation. It would be desirable to have all pressure instrumentation calibrated prior to the test and to use test quality thermocouples for the required temperature measurements. Due to the criticality of accurate fuel flow measurements, it is essential to have fuel flow measuring equipment, particularly coal scales for coal-fired plants, calibrated prior to the test.

Calculating boiler efficiency by the input-output method is desirable because of the simplicity of the method but is subject to the inaccuracies of all of the measurements required. The method of heat losses increases the accuracy of the calculation but, while the number of measurements is decreased, the difficulty of obtaining accurate measurements is increased.

The abbreviated heat loss method, as described in ASME PTC 4.1, determines the boiler efficiency by defining the amount of heat lost from:

1. Dry Gas Losses
2. Fuel Moisture Losses
3. Moisture Losses From Hydrogen Combination
4. Losses Due to Moisture in Combustion Air
5. Unburned Combustibles in Refuse
6. Radiation Losses, and
7. Unmeasured Losses.

Unmeasured Losses

Unmeasured losses are normally supplied by the steam generator manufacturer and can be found in design boiler efficiency calculations. If these are unavailable, a typical accepted value is .5 to 1.0 percent. Radiation losses can be estimated using information provided in Figure 8 of PTC 4.1.

Dry Gas Loss

Dry gas loss is calculated from the measured levels of oxygen, carbon dioxide, and carbon monoxide in the flue gas and the calculated level of nitrogen. The sensible heat of these gases is computed from the difference between the exit gas temperature and the reference air temperature. In the case of units with a cold precipitator, the exit gas temperature should be measured at the air heater exit. If the temperatures are measured at the air heater exit, oxygen (0_2) and carbon dioxide ($C0_2$) level measurements must be taken on both sides of the air heater to allow the temperature measurements to be corrected for air heater in-leakage. If the unit has a hot precipitator, the 0_2 and $C0_2$ level measurements should be taken at the precipitator inlet and outlet and at the air heater outlet. This is to allow separate air in-leakage calculations to be performed for the air heater and the precipitator.

Fuel Moisture Loss

Losses due to moisture in the fuel are calculated from the measured value moisture determined from either the fuel proximate or ultimate analysis difference between the enthalpy of water vapor at 1 psia (6.89 kPa) and the exit gas temperature and the enthalpy at the reference air temperature.

Hydrogen Loss

Moisture losses due to hydrogen combination are calculated from the measured hydrogen levels taken from the ultimate analysis. In this calculation it is assumed that all of the hydrogen combines with oxygen to form water vapor. The weight of water formed is nine times the weight of hydrogen contained in the fuel. The heat loss is determined using the enthalpy difference as described above.

Heat Rate Using Condenser Heat Rejection

An alternate method of determining turbine heat rate has been proposed and is presently being used to calculate "as-run" turbine heat rate in operating units in Britain. This method reduces the total number of measurements that are needed to calculate a heat rate to four. The method determines the heat input to the turbine by calculating the heat rejected to the condenser and the power output of the generator. An added advantage of this method is that the heat rate being calculated is an operating heat rate which will include the effects of any cycle isolation problems which may be occurring.

The heat rejected to the condenser is calculated by measuring the circulating water flow rate and multiplying it by the temperature rise across the condenser and the specific heat of the fluid. The power output of the generator is converted into its energy equivalent. The four measurements needed are:

- Circulating water flow into the condenser
- Circulating water inlet temperature
- Circulating water outlet temperature
- Generator output

The difficulties of measuring circulating water flow rates and accurate inlet and outlet temperatures have already been discussed previously. This system also allows the thermal performance of the condenser to be determined with the addition of pressure measurements. This has the advantage of measuring the two parameters considered to be of most value in identifying performance problems in a survey of utility performance engineers, condenser efficiency and turbine heat rate.

Improving Unit Performance

The previous discussions have concentrated on the methods used to characterize unit performance. Methods have been presented for determining the amount of improvement which can be made in a unit's heat rate. This section deals with methods for improving a unit's performance. Performance improvement activities are divided into those which can be realized with little capital investment, short term achievable, and those which would require an expenditure of resources, long term achievable.

- Actual Unit Heat Rate
- Controllable losses through improved operation and maintenance
- Short-Term Achievable Unit Heat Rate
- Recoverable losses through cost-effective capital expenditures
- Long-Term Achievable Unit Heat Rate
- Recoverable losses through non-cost-effective capital expenditures

Attached is a performance tutorial that quantifies the change in heat rate with a dollar cost of operation.

Monitoring and Trending

The attached pages describe how a unit’s performance can be surveyed for losses and trended to follow the effects of operation.

Input parameters required for determining the monthly performance losses are listed below under the source from which they were obtained. On the next page are notes with additional comments on particular items on the list.

Monthly Generating Station Report

- hours on-line gross generation auxiliary power usage coal usage (tons)
- fuel oil usage

- fuel oil heating value

Daily Operating Log

- condenser back pressure
- throttle steam flow
- LP condenser CW inlet temperature
- CW crossover temperature
- throttle temperature
- hot reheat temperature
- HP turbine exhaust pressure
- hot reheat pressure
- make-up water usage throttle pressure HP spray flow
- air preheater average cold end temperature
- air preheater exit gas temperature
- economizer outlet oxygen
- feedwater heater TTD’s and DCA’s

Performance Test Values

- HP turbine efficiency
- combustibles in ash

Design Values

- air preheater inlet gas temperature
- air preheater oxygen rise

Miscellaneous

- fuel analysis
- HP turbine efficiency
- number of CW pumps in service
- feedwater heater TTDs and DCAs

Key Parameters Audit Work Plan

The following audit work plan provide an audit work plan for monitoring key parameters.

Table 4-5
Audit Work Plan

1.0 BOILER

1.1 Review boiler contract data and/or expected boiler performance data for:

- expected boiler efficiency
- expected flue gas temperature
- expected exit gas temperature
- expected inlet air temperature
- expected boiler exit 0_2/$C0_2$
- expected preheater 0_2 rise or $C0_2$ drop
- expected desuperheating spray flow rates

1.2 Review boiler operating instrumentation for:

- steam temperature measurement
- flue gas temperature measurement

- preheater gas outlet temperature measurement
- inlet air temperature measurement
- boiler exit $0_2/C0_2$ measurement
- preheater gas outlet $0_2/C0_2$ measurement
- combustibles in refuse sampling (flyash, bottom ash, and economizer hoppers)
- type of instrumentation, frequency of calibration, and maintenance

1.3 Review boiler test instrumentation for:

- location of sampling grids/ports available for temperature and $0_2/C0_2$ measurements
- type of instrumentation, frequency of calibration, and maintenance

1.4 Review boiler performance data collection methods for:

- required input data and scan frequency for operator displays
- accumulated data used for performance records
- manual input data required for computer calculations

1.5 Review boiler performance testing program for:

- types of boiler tests performed
- testing procedures
- frequency of testing

1.6 Review boiler performance data analysis methods for:

- data verification, reduction, correction and/or conversion procedures
- boiler efficiency calculations

- air preheater calculations
- calculation of expected boiler performance parameters
- interpretation of results (comparison of actual vs. expected performance parameters and troubleshooting techniques

1.7 Review reporting of boiler performance results for:

- records reports
- test reports

1.8 Review pulverizer performance monitoring

2.0 TURBINE

2.1 Review turbine design data and/or expected performance data for:

- expected turbine cycle heat rates
- expected turbine efficiencies
- expected turbine shaft seal-leakage flow rates
- expected heat rate and kilowatt output effects for deviations from design for:
  - main steam temperature
  - main steam pressure
  - reheat steam temperature
  - reheater pressure drop
  - turbine back-pressure
  - generator hydrogen pressure and power factor

2.2 Review turbine instrumentation (operating and testing) for:

- throttle pressure and temperature measurements

- throttle flow (or first-stage shell pressure) measurement
- control valve position measurement
- HP turbine exhaust pressure and temperature measurements
- IP turbine inlet pressure and temperature measurements
- IP turbine exhaust pressure and temperature measurements
- extraction pressure measurements
- generator output measurement
- shaft seal-leakage flow measurements
- location of instrumentation, type of instrumentation, frequency of calibration, and maintenance

2.3 Review turbine performance data collection methods for:

- performance displays for operators
- accumulated data for performance records

2.4 Review turbine performance data analysis for:

- data verification, reduction, correction and/or conversion methods
- turbine efficiency calculations
- calculations for correcting generator output
- interpretation of results (troubleshooting performance degradations)

2.5 Review turbine performance testing program for:

- types of turbine tests performed
- testing procedures
- frequency of testing

2.6 Review reporting of turbine performance results for:

- records reporting
- test reporting

3.0 CONDENSER/COOLING TOWER

3.1 Review condenser/cooling tower design data and/or expected performance data for:

- expected condenser pressure for all CW pump schemes
- expected cooling tower performance

3.2 Review condenser/cooling tower instrumentation (operating and testing) for:

- condenser pressure measurement
- inlet cooling-water temperature measurement
- outlet cooling-water temperature measurement
- index to CW flow rate
- ambient temperature measurements at cooling towers
- hotwell temperature measurement
- condenser air leakage measurement
- index to condenser heat load
- location of instrumentation, type of instrumentation, frequency of calibration, and maintenance

3.3 Review condenser performance data collection methods for:

- input data and scan frequency for operator displays
- accumulated data for performance records (expected and actual)

3.4 Review condenser performance data analysis methods for:

- data verification, reduction, correction and/or conversion methods
- expected condenser pressure calculation
- condenser cleanliness calculation
- determination of economical CW pump operation
- interpretation of results (comparison of actual vs. expected parameters and troubleshooting techniques)

3.5 Review condenser testing program for:

- type of condenser tests performed
- testing procedures
- frequency of testing

3.6 Review reporting of condenser performance results for:

- records reporting
- test reporting

4.0 CONDENSATE/FEEDWATER CYCLE

4.1 Review design and/or expected performance data for:

- expected feedwater heater TTDs and DCAs
- expected heat rate and kilowatt output effects for TTD and DCA deviations (including feedwater heaters out-of-service)
- expected extraction line pressure drops
- expected feedwater heater hydraulic pressure drops
- expected pump performance
- expected make-up water flow rates
- expected heat rate and kilowatt output effects for deviations in make-up flow

rates

4.2 Review condensate/feedwater cycle instrumentation (operating and testing) for:

- feedwater heater inlet and outlet water temperatures
- feedwater heater drain outlet temperatures
- feedwater heater shell pressure
- feedwater heater extraction pressures at turbine (for extraction line pressure drops)
- condensate and feedwater flow rates
- pump differential pressures
- location of instrumentation, type of instrumentation, frequency of calibration, and maintenance

4.3 Review condensate/feedwater cycle data collection methods for:

- input data and scan frequency for operator displays
- accumulated data for performance records

4.4 Review condensate/feedwater cycle performance testing program for:

- types of tests performed
- testing procedures
- frequency of testing

4.5 Review condensate/feedwater cycle data analysis methods for:

- data verification, reduction, correction and/or conversion methods
- interpretation of results (comparison of actual vs. expected parameters)

4.6 Review reporting of condensate/feedwater cycle performance results for:

- records reports

- test reports

4.7 Review monitoring procedures for:

- steam traps
- drain valves
- relief valves

5.0 COAL QUANTITY AND QUALITY

5.1 Review coal analysis methods for:

- sample location/type
- type of analysis
- frequency of analysis

5.2 Review coal quantity measurement methods for:

- type of measurement device(s)
- location of measurement device(s)
- calibration frequency of measurement device(s) and maintenance

6.0 UNIT HEAT RATE

6.1 Review expected unit heat rate determination for:

- inputs to expected unit heat rate
- corrections to expected unit heat rate

6.2 Review actual heat rate determination for:

- method of generator output and electrical auxiliary measurement
- frequency of heat rate determination
- reporting of heat rate information

7.0 ORGANIZATION AND WORKFORCE

7.1 Review station organization structure

7.2 Review functional activities of station departments

7.3 Review departmental workforce allocations

8.0 AWARENESS AND INCENTIVES

8.1 Review present awareness and incentive programs

5

INSTRUMENTATION AND TESTING REQUIREMENTS FOR HEAT RATE MONITORING

Instruments and Performance

A problem with the objective of efficient unit operation is one which provides the operators with the tools (instruments, data analysis, and reporting) they require to maintain all important controllable parameters at an optimal value. The program should also motivate the operators to utilize this information and strive for efficient operation. A program which collects, displays, and reports data, but which the plant does not pay careful attention to is not achieving the objective.

Performance data which is collected but which is not effectively used to influence maintenance, operations, and chemistry planning, due to lack of confidence in the results or due to inadequate decision making mechanisms, should be resolved. The instruments used to operate the plant should be high quality. The instruments must be accurate and reliable to obtain efficient operation of the plant by the operators. Any inaccuracies in operating instrumentation causes inefficiencies in plant operation.

The Thermal Performance Program is jeopardized severely when instrument error exists. The errors in instrumentation is an unaccounted-for loss undetectable until a calibration is performed. Performance is essential to ensure a cost efficient operation, but one does not have performance at any cost, but on the other hand, the neglect of performance is costly.

ASME Performance Test Codes

The American Society of Mechanical Engineers has produced Performance Test Codes for power plant major equipment and large components. Testing performed in accordance with the ASME Codes will produce test results of the high accuracy and low uncertainty, but may involve off-line testing, special or additional instrumentation, and rigorous calibrations. If a plant component performance is suspect, based on on-line testing with normal instrumentation, an ASME Code test can be performed to verify, identify and quantify the problems.

Additionally, ASME has issued a valuable new Performance Test Code covering the testing of an entire plant, yielding output and heat rate. This code is titled:

- ASME PTC 46 - 1996 Performance Test Code on Overall Plant Performance

ASME PTC 46 contains extensive detail relative to the location, selection, installation guidelines, and uncertainty requirements of instrumentation for an overall power plant heat rate test. Although this will produce a Code grade test result of the lowest uncertainty, if applied correctly, many of the recommendations also can be applied to the routine and online determination of plant heat rate.

Some of the additional Performance Test Codes of interest for plant performance are:

- ASME PTC 1 General Instructions
- ASME PTC 2 Definitions and Values
- ASME PTC 3.2 Coal and Coke Generating Units
- ASME PTC 3.3 Gaseous Fuels
- ASME PTC 4 Steam Generators
- ASME PTC 4.2 Coal Pulverizers
- ASME PTC 4.3 Air Heaters
- ASME PTC 6 Steam Turbines
- ASME PTC 6S Procedures for Routine Performance Tests of Steam Turbines
- ASME PTC 6.1 Interim Test Code for an Alternative Procedure for Testing Steam Turbines
- ASME PTC 7.1 Displacement Pumps
- ASME PTC 8.2 Centrifugal Pumps
- ASME PTC 11 Fans
- ASME PTC 12.1 Closed Feedwater Heaters
- ASME PTC 12.2 Steam Condensing Apparatus
- ASME PTC 12.3 Deaerators

- ASME PTC 21 Particulate Matter Collection Equipment
- ASME PTC 23 Atmospheric Water Cooling Equipment
- ASME PTC 39.1 Condensate Removal Devices for Steam Systems

Instrumentation

Once it has been decided which performance parameters should be monitored, and the effect of each of these parameters on unit heat rate has been determined, the type of instrumentation to be used and the frequency of monitoring each parameter must be decided. Measurements can be made using either normal station instrumentation, calibrated station instrumentation, or test instrumentation.

Operator Aid

It is extremely important in thermal performance monitoring to provide the operators and performance personnel with accurate, reliable, and calibrated operating instrumentation. Efficient unit operation depends on their decisions and actions. Inputs to the performance program must be as accurate as practical, but absolutely must be repeatable and results from the program must be fed back to station personnel, especially the operators. The most important feedback available to the control room operator comes from operating instrumentation. The instrumentation must provide real time information with a high degree of accuracy. The most critical parameters (main steam temperature and pressure, reheat temperature, condenser back pressure, boiler O_2) are very sensitive and must be routinely calibrated to ensure a high degree of confidence. The calibration period is determined by whatever it takes to establish a high degree of confidence which will vary per instrument.

Accuracy

Calibrated test instrumentation would provide the most accurate indication however, the cost of purchasing, installing, and maintaining this instrumentation for monitoring performance parameters may not be cost justified. The accuracy of performance monitoring instrumentation can be compromised, but repeatability is a requirement. This philosophy gives the accurate direction of a performance parameter along with a reasonable magnitude. Whenever precise numbers are required then high accuracy instrumentation can be utilized. For example, when determining the best achievable heat rate curve, equipment acceptance test, and actual heat rate high accuracy is needed. This approach will not only lessen the cost of establishing and maintaining a performance program but also reduces the labor requirement without sacrificing the quality of the performance program. Units with computer data collection capabilities should be able to perform the majority of routine testing with a push of a button, hence freeing up test technicians for instrument calibration work, which would therefore increase the quality of data being collected.

Periodic Testing

Periodic tests must be performed to support a thermal performance program. This provides continual data under controlled conditions that may be trended over time. Equipment degradation may be monitored and maintenance requirements accessed. Along with periodic testing is the accountable loss performance program. This is the process of identifying performance deviations (from design and/or best achievable) and accounting for the heat rate effect of the deviation.

Testing Program

Since many parameters required for performance monitoring cannot be obtained from operating data, a structured performance testing program must exist. To support the performance testing program, it is desirable to have instrumentation that is reliable and repeatable. Whenever operating instrumentation is available it should be utilized since it should be the most accurate and most recently calibrated on a unit.

It is valuable to collect performance parameter data from as many sources as possible, this is necessary when only station instrumentation is being used to cross check the validity of the measurements. Many times additional redundant measurements can be obtained in most power plants. Potential sources include control room indicators, local gauges, special test gauges, and automated data collecting devices which might be available in the control room.

The data should be collected for a period of time which will give a representative sample of the unit's performance. The data should be collected at all of the load points at which the unit is normally operated for extended periods of time. The data should be averaged to smooth out any fluctuations. A suggested test schedule is shown in table 5-1.

Periodic Test

Periodic performance tests should include the following:

- Turbine output check and enthalpy drop tests
- Boiler oxygen rise tests
- Coal fineness

- Coal lost of ignition
- Performance belt scale calibration test
- Calibration of performance belt scale to as-received coal scales (mat'l test)
- Feedwater heater (TTD & DCA)
- Flow path verification
- Air preheater performance test
- Boiler feedpump test
- Unit maximum capability test
- Condenser Cleanliness

Special Test

Special tests are usually performed on an as-needed basis. Experience should dictate the need and frequency of tests.

- Soot blowing optimization tests
- Mill clean air tests
- Mill dirty air tests
- Coal flow balancing tests (rotoprobe)
- Heater drain pump test
- Condensate pump test
- Isolation/Water balance
- Turbine steam seal system
- Cooling tower

The time frames given below should be used as guidelines when beginning the testing program. A performance person should establish the required time frame for testing for each unit. The frequency should vary due to unit operation and service hours, and various types of equipment being used.

Table 5-1
Test Schedule

Test	Continuous	Weekly	Bi-weekly	Quarterly	Annual	As-needed
1. Fineness			X			X
2. LOI/Combustibles			X			X
3. Weightometer calibration		X				
4. Turbine output check				X		
5. Enthalpy drop tests				X		
6. Preheater O_2 rise				X		
7. Material test				X		
8. Feedwater heater				X		
9. Flow path verification				X		
10. Air preheater efficiency				X		
11. BFP efficiency					X	
12. Max. capability test					X	
13. Soot blowing optimization						X
14. Mill clean air test						X
15. Mill dirty air test						X
16. Fuel balancing test						X
17. Heater drain pump efficiency						X
18. Condensate pump efficiency						X
19. Isolation/Water balances						X
20. Turbine steam seal system						X
21. Cooling tower efficiency						X
22. Steam trap survey						X
23. Condenser				X		
24. Furnace O_2 rise						X

Test	Continuous	Weekly	Bi-weekly	Quarterly	Annual	As-needed

This test frequency is based on a minimum of 2190 hours of operation a year.

Instrumentation

General

Section 4 of this report should be reviewed as a aide in establishing the parameters to be part of either an outline, periodic or single event monitoring program. In particular Tables 4-1 and 4-2 provide a comprehensive list.

It cannot be over emphasized the importance of ensuring that the operators have the most accurate, reliable, and calibrated operating instrumentation. Any error in this instrumentation results in a real time efficiency loss; not an error in identifying where a performance loss maybe occurring. The most critical parameters are:

- Main steam temperature
- Main steam pressure
- Reheat steam temperature
- Condenser back pressure
- Boiler O_2
- Feedwater flow
- Electric Generation

These parameters are very sensitive and must be routinely calibrated to ensure a high degree of accuracy. The calibration period is determined by whatever it takes to establish a high degree of confidence which will vary per instrument. The calibration period for the parameters stated above should not exceed six months if the unit operates over 2190 hours a year. If the unit operates less hours than this the calibration period will need to be established to give the best schedule based on unit operation and quality of the instrument being used.

Of equal importance is the availability and location of test instrument connections (pressure taps, thermocouple wells, sample ports, etc.). The use of test instrumentation also provides a check against operating instrumentation. Many times faulty performance and/or operating instruments may be located by comparing them to test instruments. This should be common practice when running all test which involves using special test instrumentation. If performance and/or operating instrumentation is being utilized for gathering data for the performance monitoring program (which is highly recommended) most likely a situation will occur where special testing instrumentation will have to be utilized to resolve a problem.

Devices

General

Good references are available to aide in the selection of instrumentation for testing and monitoring. Among these are ISA Standards and Manufacturer's Literature. Additionally, The American Society of Mechanical Engineers has produced a set of Performance Test Codes for measurements. Some of these applicable codes are:

- ASME PTC 19.1 Measurement Uncertainty
- ASME PTC 19.2 Pressure Measurement
- ASME PTC 19.3 Temperature Measurement
- ASME PTC 19.5 Application, Part II of Fluid Meters: Interim Supplement on Instruments and Apparatus
- ASME PTC 19.6 Electrical Measurements in Power Circuits
- ASME PTC 19.12 Measurement of Time

Fuel Measuring and Sampling

Introduction

To calculate heat rate using input/output methods it is necessary to have accurate measurements of both the amount of fuel consumed and its heating value. These two important aspects of heat rate calculation using the input/output method will be discussed in this section.

Fuel Flow Measuring Methods and Devices

The accuracy with which the amount of fuel delivered to a unit can be measured is highly dependent upon the type of fuel being consumed. Homogeneous, single phase fuels such as oil and natural gas can be measured with greater accuracy than can non-homogeneous fuels such as coal. When using the input/output method of heat rate determination, every one percent error in the measurement of the fuel flow is reflected as a one percent error in the heat rate calculation.

Natural Gas Flow Measurement

Natural gas can be measured using orifice flow meters, densitometers, or turbine flow meters. Orifice flow meters have a generally accepted accuracy of ± 2 percent, while the accuracy of turbine flow meters is ± 1 percent. Commercially available on-line gas densitometers are available which have accuracy's of ± .25 percent however, they have not gained wide spread acceptance in the utility industry. A discussion of current utility practices in the measurement of gas flow, as well as that of oil and coal, can be found in Reference EPRI CS-4554.

Oil Flow Measurement

Oil flow measurements can be made with an accuracy of ± 1 percent to ± .25 percent with rotary type positive displacement flow meters with the higher accuracy being associated with calibrated flow meters. It is further stated in Reference EPRI CS-4554, that orifice flow meters are more widely used by U.S. utilities than any other type and that these meters are capable of accurate measurements when calibration is maintained. Unlike the positive displacement flow meter, the orifice meter is not accurate over wide range of flow rates.

The Central Electricity Generating Board reports that the turbine flow meter is replacing the positive displacement flow meter as a means of oil flow measurement in many industries, yet have not achieved widespread use by utilities. The CEGB cites the main advantage of turbine flow meters to be the repeatability of measurements taken over wide load ranges at varying viscosity's. The stated accuracy of these flow meters given in EPRI CS-4554 is ± 0.1 percent. More information on the measurement of natural gas and oil can be found in the following reports:

- ASME PTC 4.1 - 1964
- American Gas Association: Orifice Metering of Natural Gas - Gas, Measurement Committee Report No. 3, April 1955

Coal Flow Measurement

Coal flow is normally measured by either coal scales or belt feeders. The latter can be either volumetric or gravimetric feeders. Both coal scales and gravimetric feeders directly measure the weight of the coal being consumed. Volumetric belt feeders deliver a constant volume of coal but do not compensate for density changes. Gravimetric feeders are normally in-line devices through which all coal consumed by the unit must be delivered. Coal scales and volumetric feeders are often designed with provisions for bypassing the scale if a malfunction should occur. The accuracy of gravimetric feeders and coal scales are considered by many utility personnel to be no better than ± 4 percent, although the manufacturers claim much higher accuracy's.

Some utilities whose units contain scales or feeders with bypass features have an alternate method for estimating the amount of coal consumed during those periods when the fuel measuring devices are bypassed. During periods when the scales or feeders are operational, the amount of coal weighed is correlated with the power usage of the pulverizer mills. From this correlation, the fuel used during periods when the scales are not available can be estimated.

The accuracy of coal scales or gravimetric feeders would be much higher if proper maintenance and calibration could be provided. However, because these scales are in-line, they are in continuous use during plant operation requiring maintenance and calibration to wait until the plant is operating at a reduced load or during an outage. The drift in calibration during this time span can, in some cases, be several percent. This problem has been further intensified due to declining coal quality which necessitates units being run with all pulverizers operating to maintain the unit near its rated load. The accuracy of fuel measurements made at either the entrance to the bunker or at the coal unloading site is much higher. Utility estimates place the accuracy of "as-received" belt scales to be on the order of ±.25 percent and that of the "as-burned" scales to be ± 1 percent. These scales have an advantage in that they are not in continuous use and are positioned where it is easy to maintain and calibrate them.

Further information on coal weighing can be found in the following sources:

- ASME PTC 4.1 - 1964
- ASME PTC 3.2 - Solid Fuels

Fuel Sampling Methods and Devices

Introduction

The accurate determination of heat input into the combustion chamber of a fossil unit is dependent upon the accurate measurement of the energy contained in that fuel. The energy content of the fuel is referred to as its heating value. There are two heating values for any fuel -high and low. The difference between the two is the latent heat of evaporation of the water vapor formed during combustion is recovered in HHV. In calculating the energy input to the boiler it is normally the high heating value (HHV) which is used.

In homogenous fluids and gases the heating value will not change significantly within a batch and changes between batches provided by different suppliers will normally be within 5 percent. One oil burning utility reported that the weekly variance in the heating value of daily samples was less than 1 percent. With coal, the variability of the heating value is much greater. The heating value of fuel from the same supplier may vary as much as 15 percent while between suppliers the variance may be as much as 25 percent.

Coal Sampling

Therefore, to accurately determine the energy input into a boiler, it is necessary to take frequent and voluminous samples of coal. To meet the guidelines of the American Society for Testing and Materials (ASTM) and the ASME Performance Test Code (PTC) for coal sampling, it is almost a necessity that the utility have a mechanical sampling system. The shear volume of the coal removed for sampling can be overwhelming. For example for a 500 MW unit using 2" x 0 sized coal, the number of samples required during an assumed eight hour shift of coal loading would be 175 and the total sample weight would be 1050 pounds if the guidelines of ASTM D-2234 were followed.

With a mechanical sampling system continuous sample of coal is taken from the coal transfer chutes. Even with mechanical samplers, it is necessary to insure that a representative sample is being taken. This requires that the sampling system be installed so that the sample taken is not biased, but rather is a composite of the coal being transported by the coal conveyors. The cost of installation, the need for continuous maintenance, and the lack of justification for sophisticated sampling devices have precluded the wide acceptance of mechanical coal samplers by utilities.

An accepted method of coal sampling is to take a manual sample either at the end of the last coal transfer chute or from the feeder at the pulverizer inlet. When the sample is taken at the pulverizer inlet the general procedure is to take one sample per shift from each feeder to form a composite sample of the fuel burned in a 24 hour period. The sampling frequency at the coal transfer chute will depend upon the frequency with which the bunkers are filled. This period usually does not exceed 24 hours under normal operating conditions.

The procedure for taking a manual sample is no different than for the collection of a sample by an automated device. The operator should collect a uniformly distributed sample from either the transfer chute or feeder discharge. Once the sample has been taken, it should be tightly sealed to prevent moisture evaporation.

Industry estimates of the accuracy of hand sampling is +4 percent when performed according to ASTM procedures and +8 percent when procedures are not followed. The industry standard for the accuracy of mechanical sampling devices is ii percent.

Further information on sampling and the determination of heating values can be found in the following sources:

- ASME PTC 3.1 - Diesel and Burner Fuels
- ASME PTC 3.2 - Solid Fuels
- ASME PTC 3.3 - Gaseous Fuels
- ASTM D-1322 - Sampling Petroleum and Petroleum Products
- ASTM D-240 - Heat of Combustion of Liquid Hydrocarbon Fuels
- ASTM D-2234 - Collection of a Gross Sample of Coal
- ASTM D-2013 - Preparing a Coal Sample for Analysis
- ASTM D-2015 - Determining the Heating Value of a Coal Sample
- ASTM D-900 - Test for Calorific Value of Gaseous Fuels

On Line Gas Sampling and Oil Sampling

Technology presently exists to perform on-line caloric value analysis of gas on a continuous basis. It is stated that the accuracy of on-line gas sampling is +.5 percent. The caloric value of oil does not vary significantly between batches and therefore, sampling need only be performed periodically. On line sampling of coal is still in the developmental stage and may be too costly and complex for general utility application.

Temperature

Temperature measurement technology offers several selections. In general, critical temperature measurements should be made with devices with an inaccuracy of no more than 1°F for measurements of 200°F and greater and 0.5°F for measurements less than 200°F. Non -critical measurements should be made with instruments having an inaccuracy of no more than 3°F.

Calibrated continuous lead thermocouples. This type of thermocouple, when used properly with an ice-bath and calibration correction data, is highly accurate and reliable.

The resistance temperature device (RTD) may be used in testing. Care should be taken to ensure that the required accuracy is achieved and the max. temperature allowed by the RTD is not exceeded.

Pressure

Calibrated precision pressure transmitters or deadweight gauges. The portability, accuracy, and large range (span) of electronic pressure transmitters make them versatile and reliable. Transmitters with an accuracy of 0.1% are available and should be used for critical pressure measurements. For non-critical measurements, 0.25% accuracy is available and should be used.

Dead weight gages, manometers and other measurement devices may be as accurate in some applications and could be used.

All pressures should be corrected for atmospheric pressure and water legs to obtain absolute pressure values.

Signal cables must have a grounded shield to remove any induces currents from nearby electrical equipment. Signal cables should be installed away from EMF producing devices such as motors, generators, conduit and electrical panels.

Turbine Exhaust

The exhaust pressure of a steam turbine is not controllable in the same sense that the inlet steam conditions are. Units using a mechanical draft cooling tower can have controlled exhaust pressure by changing the number of fans running. Units using cooling water directly from a cooling water source (river, lake, cooling pond) have no real time control over exhaust pressure. The output of the stream turbine is affected by exhaust pressure which, in turn, influences the input-output characteristics used for economic dispatch. The accurate measurement of this pressure is very necessary for accurate and reliable monitoring.

Modern steam turbine exhaust hood designs have improved the steam pressure loss from the last rotating stage of the turbine to the condenser inlet. These hoods, however have peculiar flow distributions requiring pressure to be measured in four (4) places at each single flow exhaust flange. The positions for these measurements have been determined by extensive laboratory and field tests. Because of the chaotic and high velocity flow patterns at the turbine exhaust a device referred to as a basket tip, Figure 5-1, is used to obtain a measurement of static pressure.

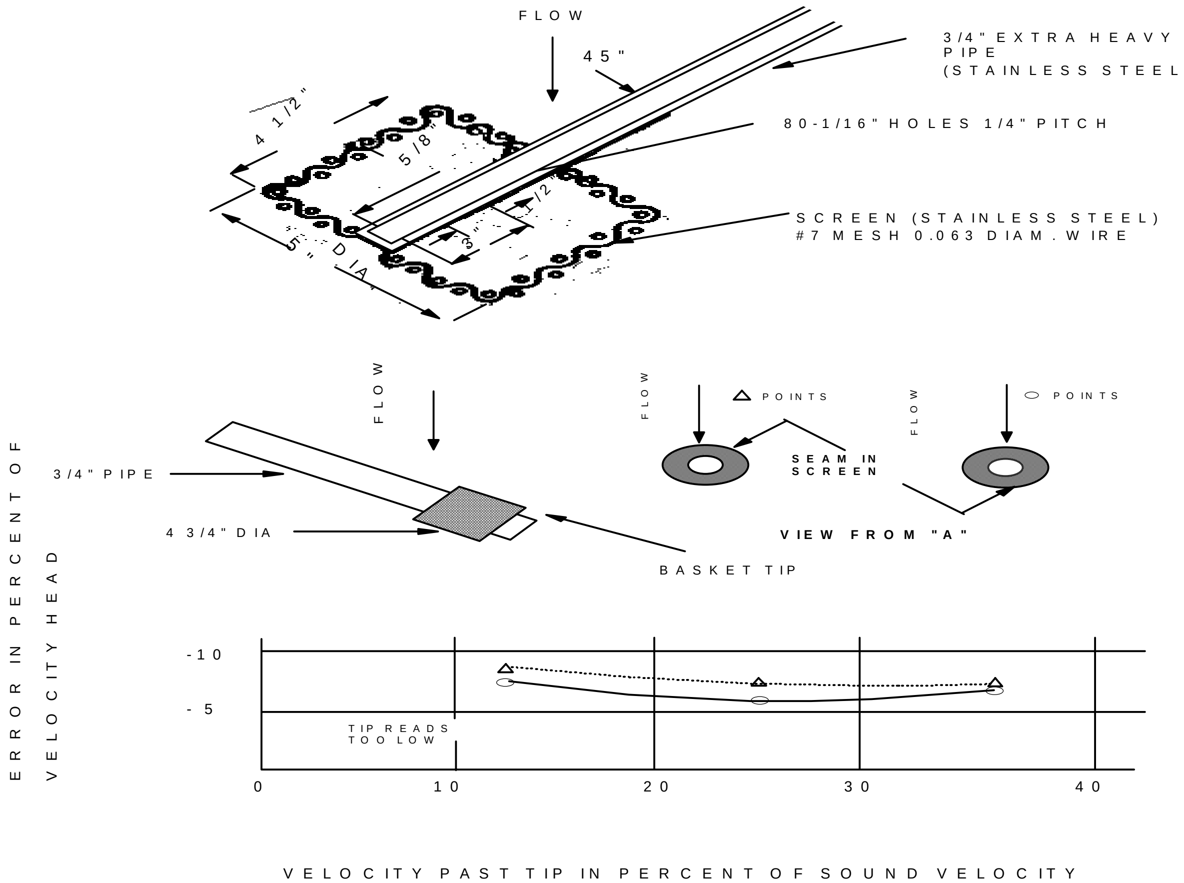


Figure 5-1
Basket Tip

A basket tip is formed by wrapping a wire mesh around the open end of a pipe. The pipe and mesh are then inserted into the LP turbine exhaust flow stream with the open end of the pipe facing downstream at an angle of 45° with respect to the flow path. The wire mesh serves to dissipate much of the local flow stream momentum resulting in a static pressure measurement which is almost independent of the flow stream velocity as illustrated in Figure 3-1. For this reason, a basket tip is one of only two devices recommended by PTC-6 for measuring steam turbine exhaust static pressure.

Basket tips are installed at the exhaust flange of the low pressure turbine. Pressure at the exhaust flange is not the same as the pressure in the condenser. There are Feedwater heaters, and perhaps other obstacles, located in the space between the exhaust flange and the condenser causing additional pressure drop.

The pressure at individual basket tips is not necessary for the calculation of performance; only the average is required. However the pressures must be measured individually. When basket tips are manifolded together flows are induced in the connecting piping due to pressure differences between pairs of basket tips. Pressure measured in a manifolded system will not be the average of the individual basket tip pressures. It will be influenced most by the two basket tips closest to the pressure transducer and may be completely insensitive to the pressure of the basket tips farthest from the transducer. In addition, since the pressure is sub atmospheric and usually less than five inches of mercury absolute (about 2.46 PSIA), the basket tips must be purged in order to prevent a water column buildup which would result in an erroneous (lower) turbine exhaust pressure reading.

Flow

General

Water flows can be measured more accurately than steam flows. When possible, water measurements should be made to calculate steam flows.

Code Grade Testing

ASME PTC 6 - (Steam Turbines) provides details for the high accuracy throat tap nozzle recommended for primary flow measurements to the Deareator or Boiler.

Water and Steam Flow Devices

ASME MFC-3M contains guidelines for construction, calibration and installation of other flow measuring devices.

Care should be take to insure that fluctuations have been eliminated that may be due to system conditions or instruments situations.

Liquid Fuel

Liquid fuel flows should be measured using flowmeters that are calibrated throughout the Reynolds number range expected. Volume flow meters must be corrected base on temperature of the fuel, so temperature must also be accurately measured.

Gas Fuel

Gas flows may be measured using orifices or turbine type flow meters. Measurements must also be made to determine the actual flow rate such as fuel analysis to determine density, static and differential pressures, temperature and frequency (if a turbine meter).

ASME MFC-3M details the uncertainties for an orifice metering run manufactured and installed correctly.

Electrical

The measurement of electrical energy is critical to the accurate and repeatable monitoring of the turbine-cycle heat rate, low pressure turbine efficiency, condenser performance and heat input to the steam generator calculated by the heat loss method. It also has a direct influence on the determination of incremental heat rate if it is used to determine net output. An error of 1% in the measurement of gross generation will result in an error of 1% in gross turbine heat rate. The only other measurement having a similar effect on gross turbine heat rate is that of the primary flow: condensate flow or feedwater flow.

The importance of power measurement is well recognized and ASME PTC-6 states that "A single-phase watthour meter or indicating wattmeter of the 0.25 percent accuracy class shall be used in each phase to determine the power output." A watthour meter per phase avoids errors introduced by unbalanced phase power or by operation at non-unity power factor since separate calibration of watthour meters, potential transformers and current transformers permits correction factors to be developed and applied per-phase.

Instrument Transformers

Errors in power measurement, in addition to those inherent in the watthour meter, are introduced by the potential transformers and current transformers through:

- Transformer ratio variations: calibration of the instrument transformers allows a ratio correction factor to be determined.
- Phase angle displacement between primary and secondary windings of the instrument transformer: calibration allows a phase angle correction factor to be determined.

Both these effects are caused by the magnitude of the current exciting the transformer, fraction of rated voltage or current, power factor, and impedance (or burden) of the instrumentation connected to the secondary winding. Potential transformers should be recalibrated at several burdens. It would be desirable to actually measure the burden but if this measurement is impractical, the potential transformers can be calibrated at zero burden so that the potential transformer ratio correction factor and phase angle shift can be calculated for any burden.

Ambient Condition Measurements

Inlet Dry Bulb Air Temperature

The dry bulb temperature is the static temperature at the inlet to plant equipment. The temperature sensor must be shielded from solar and other sources of radiation and must have a constant air flow across the sensing element.

Inlet Moisture Content

- The moisture content of the ambient air may be determined by the measurement of adiabatic wet-bulb, dew point temperature or relative humidity. Measurements to determine moisture content must be made in proximity with measurements of ambient dry bulb temperature to provide the basis for determination of air properties. The following devices can be used:

- Wet Bulb Temperature - can be inferred by a properly designed Mechanically Aspirated Psychrometer.

- Cooled Mirror Dew Point Hygrometer - uses a cooled mirror to detect dew point as air is drawn across a mirror which is cooled to the temperature at which vapor begins to form on the mirror. A temperature sensor mounted in the mirror measures the surface temperature.

- Relative Humidity Hygrometers - thin film capacitance and polymer resistance sensors provides a direct measurement of relative humidity. Measurement uncertainties vary with sensor type and design.

It may be necessary to conduct limited tests to completely collect all of the needed data. Depending upon the arrangement and location of oxygen measuring probes, it may be necessary to have 0_2 data manually collected. Oxygen measurements taken at the boiler exit are sufficient to provide the utility engineer with needed information on incomplete combustion, however, by also measuring 0_2 at the exit of the air heater, information can be obtained on the performance of the air preheater and the effect of air inleakage on exit gas temperatures.

Other special measurements which may be required are coal and ash samples, wet bulb temperature measurements, or pitot traverses of flue gas and circulating water flow to determine average flow rates and temperatures.

Data Analysis

Data and results from performance tests should be thoroughly analyzed to determine the existence or cause of a performance problem and to delegate the responsibility of problem resolution. The first step of effective data analysis is determining raw data validity (i.e., if the indicated pressure represents a realistic value). This kind of judgment is often subjective and dependent on experience. It is not necessary to wait until a test is complete before analysis begins. While a test is in progress, the performance person should review the data being collected for erroneous information. This can help ensure good test results and eliminate the need for a re-test. The importance of accurate data analysis cannot be over-emphasized due to possible consequences of an incorrect analysis (personnel safety, maintenance requirements, operating procedures, etc.). Skill in analyzing test data often must be obtained by analyzing a large number of test results and by studying the history of problems experienced by other utilities. General analysis methods and calculation procedures are covered within the applicable appendix. After the analysis of a performance test, the effective reporting of test results is of equal importance.

6

CYCLE ISOLATION

Every steam plant has a normal path for the flow of liquid or steam at every load point. These paths are usually well represented on the heat balance diagrams provided by the turbine and boiler vendors. What is not shown on these diagrams are the numerous paths which are available for either liquid or steam to escape from the normal flow paths.

When flow is diverted from the normal steam path it is either lost from the cycle completely or returned to a section of the cycle where energy is removed from the fluid without providing any useful work. This is true of fluid which is diverted from the condensate and steam cycles to the condenser or deaerator before it has performed work in the turbine.

Most utilities do not place a great deal of emphasis on ensuring that these potential steam or water diversion paths are isolated during normal operation. The result of a survey of typical low cost heat rate improvement activities indicated that few utilities routinely perform cycle isolation checks. There could be hundreds of flow paths through which energy can be lost from a cycle. These potential diversion paths range from instrument tapping lines which are .25 to .5 inch (.64 to 1.25 cm) in diameter to feedwater heater bypass lines which can be 14 inches (36 cm) or greater in diameter. Some pathways represent a greater potential for energy loss than do others. For example, it is obvious that a leak in a partially open 6 inch (15 cm) feedwater heater drain can divert more energy from the cycle than will a fully open .25 inch (.62 cm) instrument line.

The only way to appropriately deal with cycle isolation problems is to perform periodic cycle isolation walkdowns. Periodic walkdowns will allow a utility to identify the particular losses that are occurring in a unit and schedule maintenance activities to correct these problems as required.

The first step in preparing for a cycle isolation walkdown is to prepare a detailed cycle configuration checklist. This checklist contains all of the lines not used during normal operation. These lines should be isolated. The list should contain the following information:

- description of the line, including origination and termination points

- identification of the isolation valve (this is the valve which should be closed to prevent flow through a diversion flow path)
- location of the isolation valve, the valve size
- identification of the drawing from which the information was obtained
- identify when the valve should be open or closed.

It is also important that the isolation checklist contain all drain lines which have steam traps in order to perform a steam trap operability check. Once the checklist is compiled, the performance engineer should begin a cycle isolation walkdown using the isolation checklist.

The walkdown consists of determining if there is any leakage through the isolation valve by checking either the downstream pipe wall temperature or by listening for flow through the isolation valve. The downstream temperature can be checked by hand or by using a pyrometer and flow checks can be made using a stethoscope. The performance engineer should make note of any valve seals, pipe flanges, or pump seals which may be leaking and require repair. Caution should be exercised around high energy steam piping to avoid possible injury from high energy steam leaks. To make cycle isolation checks simpler, consideration should be given to installing contact thermometers on piping downstream of an isolation valve. This thermometer should be within the expected upstream temperature. The magnitude of a leak can be determined by the proximate of the temperature to upstream temperature.

Lines which terminate at the base slab drain or are vented to the atmosphere can be checked for leakage visually. Some lines which vent to atmosphere include main steam safety and relief valves and steam drum safety valves. Drains which discharge to the base slab include feedwater heater shell side drains, condenser hotwell drains, and boiler drains.

The performance engineer may wish to determine whether the normal cycle vent paths are functioning properly. Normal cycle vent paths include both the continuous blowdown deaerator vent flow, and the feedwater heater vent flows. The continuous blowdown from the steam drum is normally set to ensure that the unit will experience no imbalance of chemical impurities in the condensate. However, the amount of margin may be too great and the amount of blowdown may be excessive resulting in unnecessary energy losses.

Attention to the deaerator and feedwater vents can be important for two reasons. First, excessive venting will result in unnecessary energy losses from the cycle. This can occur if the deaerator and feedwater heater vent have eroded and are venting excessive steam. If, however, the feedwater heater vent are plugged or have mineral build-up, they may be venting inadequate amounts of noncondensables thereby impeding heat transfer and increasing the possibility of corrosion to heater tubes. Also, many units vent the deaerator back to the condenser. If the vent flow is greater than necessary, an additional load upon the condenser would occur, resulting in higher than expected condenser back pressures.

Having completed the inspection of cycle isolation, the performance engineer now has a list of those valves which require repair. Once a complete cycle isolation walkdown has been performed, the list can be reduced to a manageable number of flow paths to inspect on a routine basis. This list should include those flow paths which were found to be leaking, those which are known to be high maintenance items - identified from maintenance work orders, and those which are known from utility experience. The following flow paths are known from utility experience to be the most problematic:

- High Pressure and Low Pressure Emergency Drain Lines
- High Pressure Heater to Low Pressure Heater Low Load Bypass Line
- Boiler Feed Pump and Condensate Pump Minimum Recirculation Lines
- Start-up Valves on High Energy Steam Systems

The reduced list should be used on a frequent basis to check cycle condition; however, a walkdown using the complete checklist should be periodically performed, preferably prior to the scheduled annual maintenance. Inspection should be performed sufficiently in advance of maintenance activities in order to increase effectiveness.

If the performance engineer is unable to prepare a detailed cycle isolation checklist, he should perform the cycle walkdown, developing a list of leaking lines which require repair and a list of those paths which are known “offenders.” Included on the checklist should be all lines which have the potential to divert large amounts of energy away from the normal flow path.

It should be noted that for many plants which are experiencing large deviations between actual and best-achievable levels of heat rate, as much as 20 percent of the total deviation can be recovered from proper cycle isolation. Accurately determining the flow paths which are the biggest contributors to heat loss requires some data gathering and analysis using one of the heat balance codes. An analysis of this type has been performed by a utility and is detailed in EPRI CS-4554. The results are shown in Table 3-5, and the complete results are presented in Reference 10 of CS-4554.

Another method used to determine the overall effect of inadequate cycle isolation is to perform two limited cycle evaluations, one at the unisolated condition and the other with the major leaking lines isolated. The results can be compared to determine changes in the power output of the unit. These tests should be performed at a valve point to allow accurate comparisons.

7

HEAT RATE IMPROVEMENT PROGRAM

The financial success of electric utilities in an increasingly competitive environment depends largely on improving plant performance to maintain or lower the cost of producing electricity while meeting ever more stringent environmental regulations. As the average age of fossil plants increases, new approaches are needed to ensure their continuing productivity.

A heat rate improvement program can include cycle modifications, component modifications, improved maintenance practices, or increased use of microprocessor based controls and instrumentation.

Cycle Modifications

Examples of cycle modifications include retrofitting a unit for variable pressure operation, installing variable speed drives on plant equipment and modification or addition of equipment for improved heat recovery.

Variable Pressure Operation

EPRI Report GS-6772 presents a discussion of variable pressure operation for efficiency improvements, a portion of this report is highlighted here.

Major modifications to a unit may or may not be required to implement a variable pressure operation retrofit. Tests involving the boiler and turbine manufacturers must be performed to evaluate what revisions would be necessary. In most cases an increase in unit cycling initiates the desire to implement variable pressure operation and the testing would involve determining what revisions would be necessary to improve unit reliability and stability at reduced loads and pressure.

Advantages of variable pressure operation include:

- Lower plant heat rate at part load by reducing throttling losses.
- Ability to maintain higher main steam and reheat steam temperatures at low loads.

- Ability to extend the automatic control range from about 50% down to about 20% load.

- Shorter startups in the superheater and reheater at low loads because of the higher specific volume of the steam at lower pressures and higher temperatures.

- Improved availability and reliability by reduction of turbine stresses, particularly during startup and load changes.

- Lower life expenditure for the same duty cycle.

The use of variable pressure is often coupled with full arc admission turbines because it results in the best full load heat rate. However, the use of variable pressure in plants with partial arc admission turbines can also result in significant improvements in part load heat rates. The use of sequential valve operation followed by variable pressure operation - hybrid operation - generally yields the lowest part load turbine heat rate.

Background

Up until the oil crisis of the 1970s, the electric utility industry in the United States experienced a growth rate of electric power demand between six and ten percent per year. To satisfy the rising demand, large new generating units were built. The maximum unit size increased from about 200 MW in the early 1950s to over 1000 MW in the early 1970s. Most of the units installed during that period were designed solely for baseload operation. As the units grew in size, difficulties arose in manufacturing some of the larger components such as steam drums, turbine rotors, etc. Steam generators 600 MW and larger were equipped with two steam drums installed in parallel, since the required thickness exceeded the manufacturing capability of existing mills. To overcome the manufacturing difficulties, as well as to improve the heat rate, supercritical units were introduced. In 1957 the first U.S. commercial size supercritical unit was commissioned by Ohio Power Co. (Philo Unit 6, 120 MW, with steam parameters of 4496 psi (31 MPa), 1148 °F (620 °C), 1033 °F (556 °C), 1000 °F (538 °C). Shortly thereafter (1959) the Philadelphia Electric Company's.' 325 MW Eddystone Unit 1 was commissioned and to this day remains the unit with the highest design steam parameters (5076 psi (35 MPa), 1202 °F (650 °C), and double reheat to 1051 °F (566 °C).

About one-sixth of all large fossil fired units in operation in the United States use the supercritical steam cycle. Many first generation U.S. supercritical units employed a bypass system with an external low-pressure flash tank for start-up, rolling, synchronizing and initial loading of the turbine. At some load point - usually at about 7 MPa throttle pressure - steam flow to the turbine is switched from the external flash tank circuit to the mainline flow path. The valve coordination and system response required to prevent sudden transients in steam parameters is difficult to achieve and requires great operator skills. The inherent instability of this system during startup contributed to forced outages and operating problems.

The 1973/74 oil crisis was followed by a rapid increase in fuel prices, high interest rates and high inflation. In response to this situation, drastic energy conservation measures were taken by industrial, commercial and residential electric power users. The rate of increase in demand for power has since slowed to about 2-3 percent per year. The utilities had to adapt to the new situation, making the best use of the facilities already installed.

On most large utility systems, daily load variations require that only some of the installed capacity should run continuously on base-load. This operation is normally assigned to the units with the lowest fuel costs, thus favoring nuclear plants. The other units are relegated to intermittent and cycling duty, including both cycling between low and full loads and/or "daily start-stop" cycling. These units generally were not designed for operation for extended periods at low loads, or for rapid load changing, and for this reason, can suffer in both heat rate and availability. Further, when supercritical plants are considered for intermittent duty, the problems are intensified due to the more stringent operating requirements imposed on these plants.

The most common operating mode used in the United States is constant (steam) pressure operation. Main steam is supplied to the turbine at constant pressure over the entire load range and steam flow is controlled by the main steam control valves. For turbines using throttle controlled, full-arc admission, all valves are opened simultaneously over the entire load range. When the turbine is equipped with a control stage, the control valves, which feed separate sectors (or arcs) of the first stage nozzles, are opened sequentially, so that each valve is almost fully opened (closed) before the next valve in the sequence starts to open (close). This type of turbine control is referred to as partial arc admission. Variable (steam) pressure operation is achieved by holding the turbine control valves in a fixed position (usually fully open) and varying the main steam pressure from the steam generator over the load range. A combination of both constant and variable pressure operation, the hybrid mode, is often used as a compromise that provides convenient tradeoffs between the advantages and disadvantages of the two operating modes. It is characterized by variable pressure operation from a certain minimum load point up to a predetermined transition load point at which full main steam pressure is established and then kept constant up to full load. The transition point is normally selected in the range of 60% to 85% load. For partial arc admission turbines, this point corresponds to a certain valve point in this range.

Variable and hybrid pressure operation present the greatest benefits for units subjected to frequent peaking and cycling, including both cycling between low and full load and/or daily stop/start cycling. These benefits are also shared by units that are on intermittent duty, operating for long periods at low loads.

The use of variable pressure operation presents several advantages for power plants including improved heat rate, reduced stresses on turbine components, reduced time for ramping up in load, reduced pressure stresses in boilers at partial load, increased control range etc. For these reasons, variable pressure will be used for the next generation of advanced supercritical plants to be built in the U.S. This design approach is already common practice on European and Japanese supercritical plants.

Drawbacks Of VPO

Variable pressure operation has two potential drawbacks especially significant for drum units: sluggish boiler load response and increased boiler fatigue stress. Drum boilers rely on energy storage to permit rapid response to small load change requests. Operating at reduced pressure worsens response by diminishing this storage. Varying pressure causes energy storage to change with load. This requires overfiring on load increases and underfiring on load decreases. Steam and metal temperature excursions in the convective pass caused by firing/evaporation rate mismatches often limit rate of load change. Rapid changes in temperature and pressure lead to cyclical stresses which shorten the lives of boiler components.

Pure VPO (in which the furnace walls operate at part pressure) is difficult to implement on once-through units originally designed only for constant pressure operation. Tube-to-water heat transfer in furnace walls degrades at low pressures and can lead to insufficient cooling. Pressure and mass velocity are varied to approximate their relation in a once-through unit with vertical furnace wall tubes changing load under pure VPO.

At each load, there is a minimum fluid mass velocity needed to adequately cool furnace wall tubes. That minimum velocity is referred to as the “critical velocity”. U.S. supercritical once-through boilers use various construction features to achieve that minimum velocity at all loads. These features include superimposed recirculation (at loads below about 60%) or series arrangement of furnace wall panels. The critical velocity for a given heat flux is much lower at subcritical pressures than at supercritical pressures. Therefore, nearly all U.S. once-through supercritical boilers use throttle valves downstream of the furnace to keep all furnace circuitry at supercritical pressures at all loads. By original design, these throttle valves only have the capacity to provide variable pressure below about 30% load. These valves can be redesigned to provide variable turbine throttle pressure over the full load range while maintaining full furnace pressure.

Alternately, spiral furnace walls make it possible to choose the design mass velocity independently of gas-side constraints on furnace size and area. Running wall tubes on a slant rather than vertically allows the furnace walls to be covered by a smaller number of tubes of a given size and spacing. Reducing the number of tubes proportionally reduces the furnace water flow area and thereby increases mass velocity. Since any value can be chosen for the angle of slant, designs are possible which provide mass velocities high enough to support pure VPO while in a once-through mode. The spiral arrangement also tends to equalize heat absorption among the parallel circuits since each circuit flows through all walls.

There are several other less significant problems in implementation of VPO. Retrofit usually requires control system upgrade. It may also be advantageous to change turbine throttle valve actuation methods to provide the greatest benefit of VPO. Some modification to the boiler feed system may at times be necessary to avoid either critical speed operation of the boiler feed pump turbine or vibration problems in variable speed drives.

Worldwide Development of Variable Pressure Operation

Fossil plant designs in the U.S., Japan and Europe have evolved in somewhat different ways, due to the requirements of the different loading patterns. In Japan and Germany, most new fossil units are designed for "daily start-stop" cycling. The cycling capability affects the turbine, the boiler and their control systems. Variable pressure supercritical boilers, turbine bypass systems, automatic digital control systems and integrated boiler and turbine controls are the most common features of new power plants.

Until late in the 1950s natural circulation boilers, operated at constant pressure, were the standard for utility applications everywhere. Due to limited capacity for producing heavy pressure parts and the very high cost of austenitic steels, once-through steam generators without large steam drums became popular in Northern Europe, particularly in Germany and Switzerland. The relative small size of most European utilities at that time required that the units be operated in a cycling mode to follow the system load demand. Once-through steam generators with small water inventory and few thick walled pressure parts were most suitable for this service. Steam turbines were developed with full-arc admission, without control stages, with light casings and in some cases with rotors welded from smaller forgings to avoid the need for the large and costly forgings used by U.S. manufacturers. This provided an easier transition to cycling duty. To improve heat rate and allow fast load change, variable pressure operation was introduced during the early 1950’s and quickly gained acceptance in Germany and other European countries.

The need of cycling operation and quick response has led to the development in Europe of once-through steam generators with helical furnace tubing and single pass construction. They can be used for any pressure and are very flexible, being able to compensate for varying heat pick-up rates in the furnace due to varying coal properties or slagging. Thin, small diameter water wall tubes are used to limit the heat stored in the metal and fluid, thus making the steam generator more capable of following changes in load. The single pass design with a fully open furnace, (no division walls or platens) allows easy cleaning by wall blowers and fully drainable arrangement of heating surfaces. For boiler-turbine matching during startup and load changes, all new units in West Germany are equipped with high pressure and low pressure turbine bypass systems, designed to handle 100% of the steam flow.

Supercritical steam generators with waterwall recirculation and high capacity turbine bypass systems, operated in the variable pressure mode with full arc admission turbines, are now used by many European utilities. These units exhibit great operating flexibility. Load change rates of greater than 5%/minute firing coal and 8-10%/minute firing oil or gas have been demonstrated.

Until late into the 1970s, power plant equipment built in Japan was based on U.S. technology through licensing arrangements. Japan's very rapid load growth during the last years required the addition of large units. Japan adopted large supercritical units about the same time as the U.S., but unlike the U.S., they have not returned to the subcritical design. Their fossil fired units are often cycled, which required them to build these plants for maximum flexibility by using advanced control concepts, and design features that allow rapid start-up and high load change rates. Rate of change measurements for both steam generator and turbine critical components are usually incorporated in the control system. Operating requirements have led to designs that differ from both U.S. and European approaches. European-type spiral wound boiler furnace designs are used in conjunction with partial-arc turbine control and hybrid pressure operating mode. or improved efficiency, an additional feedwater heater is provided to cool the water collected from European-type separators or recirculation pumps are added to introduce the drains into the high pressure water flow. High capacity turbine bypass systems are also employed but only to about 50% rating, since the bypass is not used as a safety valve as allowed in European practice. Automatic digital controls are used to keep thermal stresses within allowable ranges during load changes and start-up. Load change rates of greater than 3%/minute have been demonstrated with large supercritical coal fired units.

U.S. Experience

Most U.S. steam generators and turbines are designed primarily for base load operation and high equipment reliability. Natural and controlled circulation subcritical steam generators with sequential arc admission turbines operated at constant steam pressure were, and continue to be the most common design used in U.S. power plants. Supercritical units introduced in the early 1960s, using U.S. turbine designs and steam generators of European concept, increased in size and steam conditions. The first generation units had complex start-up systems/procedures and exhibited relatively lower reliability than existing subcritical natural and controlled circulation units.

U.S. designs did not employ the high start-up firing rates (normally required in Europe) which essentially eliminated the requirement for a turbine bypass system. Many U.S. steam generator designs used valves between the waterwalls and the superheater to maintain the waterwall pressure at full load conditions. This practice is still the design basis for most U.S. once-through units.

The U.S. turbine manufacturers have retained the partial arc admission design concept, which was developed to provide improved high-load heat rates. With such turbines, load changes create large temperature changes in the first stage of the HP turbine. To control the stresses in the turbine, long startup periods and slower load change rates are required.

Variable pressure operation has made some inroads in U.S. utility applications. With the increasing need of cycling of fossil-fired units, some utilities built VPO features into their plant designs and others converted constant pressure units to VPO operation. Of the utility generating units employing variable pressure operation most use a hybrid mode, some use variable pressure over their entire operating range and a few units keep constant pressure in the steam generator water walls and evaporative sections and throttle the steam prior to entering the secondary superheaters to vary the pressure to the turbine.

Among the largest units converted to variable pressure operation are four 1300 MW supercritical units in the American Electric Power Co.'s System (Gavin 1 & 2, Rockport 1 and Mountaineer 1). These units are equipped with full-arc admission turbines. The conversion resulted in an overall annual gain in unit efficiency of 1.5 percent. At half load, the net plant heat rate improved by 3 percent. The conversion was made by adding four 50 cm diameter sliding pressure throttle valves between the primary and secondary superheater. The cost of the modifications, reportedly was recovered through fuel savings in about 1-1/2 years.

Although surveys indicate that U.S. utilities that operate their fossil units for extended periods at low loads or in a cycling mode are interested in VPO operation, there is not sufficient published information available on the technical and economic merits of retrofit to VPO for existing units. The technology basis exists for such conversions, but more detailed feasibility and demonstration work is needed before wider acceptance of VPO conversions can be expected.

U.S. and European steam generator manufacturers and their Japanese associates have now developed designs, to be first used in Japan, which will permit full variable pressure operation of supercritical boilers and operation under full automatic control down to 10-15% of full load rating. This, combined with automation of the auxiliary equipment and the use of bypass systems, will enable turbines installed in the future to utilize their full capability for cycling service.

Converting Existing Plants to Variable Pressure Operation

To retrofit variable pressure operation, generally some modification of equipment and the control system is needed together with boiler operation changes. This retrofit can be applied to any existing power station which anticipates frequent low load operation, and which can produce, or be modified to produce, low feedwater pump outlet pressures at low flow rates.

A power plant which is modified to operate efficiently over a wide range of load requirements must also be designed to achieve high reliability. The conversion requires consideration of the many aspects of variable pressure operation in conjunction with existing plant equipment. Areas of concern which must be addressed are briefly discussed below.

Steam Generator -- An important aspect to consider are the coal characteristics and their impact on the operation of the steam generator. Most reactive low rank coals need less Ignition/stabilizing energy input, but require higher air temperatures for drying and combustion. Low grade coals with high ash content could create ignition and flame instability problems. The chemical constituents of these coals can affect the heat transfer of the boiler tubes as a result of tube slagging and fouling. High inherent moisture affects boiler performance by increasing the flow rate since a higher boiler heat input is necessary for drying the coal. An increase in flue gas flow rate in-turn requires additional fan power to pull the gases through the convective sections. The increase in the flue gas flow rate increases the heat transferred to the steam, usually requiring higher desuperheating flow which reduces cycle efficiency. All these effects must be considered when analyzing boiler performance in a mode which differs from that for which it was originally designed.

The specific performance capabilities of a variety of components must be evaluated, e.g., steam/water drum (subcritical units); headers, including the nipples, links, lug attachments and riser tubes; furnace framing system; valves; duct expansion joints; furnace wall tubes and attachments; pressure part hanger rods and supports, etc.

Circulation in the waterwalls and evaporators must be carefully checked and the need to modify the waterwalls determined. Most U.S. once-through subcritical steam generators use waterwalls made of straight tubes. Some experienced problems with circulation at low loads. Jacksonville Electric Authority (JEA) modified their 275 MW Northside Unit 1 boiler by replacing the straight vertical waterwalls with a spiral wound design. The unit now operates satisfactorily at loads as low as 12% in a variable pressure operating mode. Prior to the furnace modification, bypass of steam to the condenser was necessary for operation at low loads.

The good results obtained by modification of Northside 1 led other utilities to consider converting their once-through steam generators to spiral wound furnaces. Recently

Ohio Edison completed the retrofit of their W. H. Sammis Unit 5 to spiral furnace and other utilities are now investigating the merits of such conversions for their units.

By using spiral wound furnace waterwalls, higher in-tube mass velocities are obtained, and pseudo film boiling (or heat transfer impairment phenomenon) inside the tubes is avoided. Higher mass velocities are achieved since by inclining the tubes fewer parallel circuits are needed to cover a given furnace depth or width. In this example, by inclining each tube at angle of 30° the number of tubes required is reduced to half. An alternative to the spiral wound water wall furnace is the vertical water wall furnace made of rifled tubes. The rifled tubes have been used successfully on subcritical controlled circulation and natural circulation boilers. Rifled water wall tubes yield improved heat transfer characteristics compared to smooth tubes over a wide range of heat flux rates, steam quality and operating pressures.

Conversion of once-through boilers to variable pressure operation requires a cost/benefit evaluation whether to modify the furnace water walls and operate the entire unit on variable pressure or to keep the boiler on constant pressure (without modification). and throttle the steam with sliding pressure control valves added between the primary and secondary superheater (mixed pressure operation).

Limiting the rate of temperature change as a result of changing load will increase the life span of the boiler drum and superheater without significantly affecting the system response to load change. However, design modifications may be necessary to increase the flexibility of the headers and terminal tube configurations.

Coal Preparation -- An area of concern are the coal pulverizers. Although most base loaded plants have satisfactory turndown capabilities, it will be necessary to verify that the pulverizing system will not unduly limit load change rate or minimum load capabilities. It must also be verified that the existing pulverizers, feeders and primary air fans have adequate margins to provide the energy input required when over-firing, to satisfy energy storage requirements of the evaporative sections of the boiler.

Major sources of stresses in the turbine are large temperature mismatches between the steam and the turbine. Large mismatches lead to high temperature gradients in the turbine which in turn result in thermal stresses. To avoid stresses which will lead to fatigue problems in the two shift mode plants with constant throttle pressure control, a slow ramp-up procedure is adapted. This is needed because in ramping down at constant throttle pressure the drop in steam temperature across the turbine inlet valves and particularly in the control stage of partial arc admission type turbines leads to lower steam temperatures throughout the high pressure turbine. This results in the turbine cooling below the temperature required for minimum mismatch on start-up.

In contrast, variable pressure operation maintains the steam in the high pressure turbine at close to the design temperature or much of the ramp-down leaving the turbine at a much higher temperature. This allows more rapid ramp-up of the plant for the same level of stresses allowed in the fixed pressure case.

The more rapid ramp-up leads to lower overall fuel consumption at the plant and makes possible more economical operation of the grid to which it is connected. Besides faster load changes with variable pressure operation, the availability of the plant is also improved.

It is estimated that a 0.3 percent improvement in availability may be possible with a variable pressure retrofit, with corresponding benefits in generation savings and turbine life expectancy.

Variable pressure operation also leads to substantial reduction in the power required by the boiler feed pump at part load if variable main steam pressure is achieved by varying the boiler pressure. Operating in this mode allows running the boiler feed pump at lower head and lower speed than in the constant pressure case. This results in the pump operating at close to its full load efficiency at part load. The improvement in heat rate compared to constant pressure can be approximately 0.5%. If the boiler feed pump is driven by a steam turbine, variable pressure and constant pressure operation differ in yet another way which can affect the net heat rate. In variable pressure operation it is generally possible to use extraction steam to drive the feed pump at close to 40% of full load, resulting in an increase in overall cycle efficiency.

As load is reduced to some point the main and reheat steam temperatures will start to drop. This is referred to as the boiler droop and the load at which the steam temperature decreases below the design value is called the control point. The droop characteristic is a function of boiler design, method of operation and the fuel fired. With constant pressure operation, the control point is at about 50 to 60% load. With variable and hybrid pressure operation, the control point occurs at about 20 to 30% load, thus the range of automatic control is broadened considerably.

Conclusions

From the survey work conducted to date, and from observations of plant design techniques throughout the world, it is apparent that variable pressure operation is attractive for plants relegated to cycling and part-load. Reduced time for ramping up in load, reduced stresses on turbine and boiler components (with resulting availability improvement), and increased control range are achieved with VPO.

Variable Speed Drives

An option increasingly being considered in both new unit plant designs and life extension projects is the variable speed motor. Increasing fuel costs and improvements in technology and reliability have caused a decisive shift toward the use of this equipment, particularly where unit cycling is experienced.

There are three basic ways motor speeds can be changed.

- Change the number of poles on the motor (two speed motors)
- Change the slip of the motor (wound rotor motors)
- Change the frequency of the energy supply to the motor.

Two speed motors have been used on induced draft fan service in generating stations on a limited basis for years. The two speed single winding motor is a good example of changing the number of poles to make a speed change. These types of machines do not have the flexibility to meet more than two conditions of speed and are not generally classified as variable speed motors.

Wound rotor motor drives are a form of variable speed drive that has been found in generating stations throughout the years. Increasing the rotor winding resistance limits the magnitude of the starting current, making it possible to operate the motor at very high slip values. A varying speed torque characteristic is obtained by varying the value of the resistance. As the number of resistance steps increases, the machine begins to take on the characteristics of a variable speed motor.

With the introduction of solid-state power inverters, it became possible to eliminate the power losses caused by the resistors used to vary the speed. As the device became energy efficient, interest increased in using it to replace existing wound rotor drives.

The most efficient and most commonly used type of variable speed equipment in power generation is the adjustable frequency synchronous motor (AFSM) or load committed inverter (LCI). The speed of a synchronous motor is independent of motor slip and depends only on the frequency of the supply. A feeder breaker feeds constant voltage, constant frequency ac power to a converter consisting of stacks of silicon controlled rectifier cells. The converter output is an adjustable dc voltage which is connected to an inverter which supplies variable frequency ac to the motor.

Benefits of the variable frequency drive include improving the unit heat rate by decreasing electrical auxiliary usage at reduced loads and giving the motor soft start capabilities. Heat rate improvement from a variable frequency drive retrofit would depend on the unit operation. As an example a 600 MW unit operating at 60% output factor could expect an estimated improvement of 0.5 % effect on unit heat rate.

Drawbacks are high initial cost of the equipment, a good deal of space is required for installation, the equipment is complex and requires trained maintenance personnel, and care must be taken in applying variable frequency drive equipment because harmonics can be distributed throughout the system, causing heating and possible failure of other components in the electrical system. Other than draft fans, circulating water pump motors could also be a candidate for AFSM.

Heat Recovery Modifications

As energy costs rise, energy conservation and plant efficiency become increasingly important. A power plant Rankine thermal cycle suffers tremendous energy losses, therefore it is important to recover even small heat losses. Examples include recovery of boiler blowdown by returning it to the condensate cycle and installing heat exchangers on steam jet air ejectors and steam packing exhausts using condensate as the cooling supply. Larger examples include recovery of heat lost to the condenser and air preheater modifications.

Circulating Water Heat Recovery

More than half the energy of the fuel burned is lost to the condenser cooling system. These losses are considered unavoidable, but it is possible to recover some of this waste energy for a secondary use such as air preheating. Plans to accomplish such recovery have been developed and used, but most have suffered from operational and material problems. Today's technology has the means to eliminate most of these problems and make it possible to turn old ideas into operational cost savings for both existing and new plants.

Preheating combustion air to protect regenerative and recuperative air heaters against cold end erosion is common practice in fossil-fired power plants. There are numerous preheating methods in use today, the most common is hot water or steam heat exchange coils located in the air ducts. These heaters are normally supplied with steam or water from the main plant cycle, resulting in significant plant heat rate penalties. The use of warm circulating water can be used to offset some of this penalty and thus provide significant reduction in plant heat rate.

Other uses of warm circulating water have included agriculture for greenhouses and fish farms. Using the water as a supply for direct heating or energy to water source heat pumps, with resultant coefficient of performance (COP) values of 5 to 8 is another application. This may be of significant value for space heating of power plant facilities that would otherwise require much more costly energy use by conventional heating systems.

It has been determined in previous installations that the cost of implementing this equipment and systems can be economically justified when it is part of the original plant design. The retrofit value is more difficult to determine, since the installation evaluation is very site sensitive with regard to the plant cycle, turbine and boiler back end design, existing equipment arrangement, remaining plant life, plant capacity factor, plant operating modes, annual ambient temperature range, plant cooling system design and fuel cost.

Introduction

In the 1960s, major power plants were designed to be located near their coal fuel sources rather than at large sources of cooling water. These “mine-mouth”plants used large cooling towers for a heat sink. One of these plants, the two-unit 1800 MW Keystone Station in western Pennsylvania, operated by Pennsylvania Electric, was the second U.S. power plant to use natural draft cooling towers. Unit 1 went into service in 1967. Unit 2, a year later.

The cold water from a cooling tower is always warmer than the ambient air and the temperature difference is called the "approach." The approach differential added to the condenser water temperature rise is the total temperature difference available between the circulating water and the ambient air. However, in sub-freezing ambient air conditions, a cooling tower is operated in a de-icing mode where cold water temperature is limited to a minimum of 50 to 55 °F (10 to 13° C). Thus, the differential between the hot water and air actually increases as the air temperature drops below the freezing point. These characteristics increase the potential of the hot circulating water as a viable heating source.

The implementation of circulating water heat recovery has been successful at several utility sites including the Keystone Station. After 20 years of operation, the installed systems have experienced problems and replacement, but have proven their economic value.

Case History

<u>Air Preheating Evaluation</u>. The characteristics described above were investigated for possible use in combustion air preheating at Keystone. At a worst ambient air condition of -20 °F (-29 °C), the hot circulating water temperature was 78 °F (26 °C), or a 98 °F (37 °C) differential. At 0 °F (-18 °C) the water was 85 °F (29 °C). To recover heat from water to air under these conditions requires a substantial water flow and a large heat transfer surface area. The design selected for evaluation was a once-through bank of water-to-air coils with four rows of 1-inch (25mm) diameter finned tubes. The coil bank for each of the three forced draft fans was to be 20 feet high and 40 feet wide (6.1 x 12.2m), and was arranged as an inlet wall to the fan room. Since the heat recovery was insufficient for the total air preheating requirement, these coils were combined with several other methods of Lungstrom cold end protection for evaluation (hot water coils, hot air recirculation, and cold air bypass). They were then compared with a

system using only hot water coils. The evaluation considered the operating conditions over the entire year and the percentage of time operated within 10 °F (5.5 °C) increments of ambient air. Fan power (draft losses), pumping power, and cycle losses were included in the evaluation. The results of the original evaluation (in 1964 dollars) are shown in Table 7-1.

Table 7-1
Keystone station air preheating evaluation

	Extraction Steam Heated Hot Water	50% Circ Water 50% Hot Water	50% Circ Water 50% Hot Air Recirc	50% Circ Water 50% Air Bypass
Installed Cost	636,000	923,000	722,000	595,000
Energy Penalty	544,000	208,000	82,400	64,000
Capacity Charge	366,000	94,000	50,600	47,000
Total Cost	**1,546,000**	**1,225,000**	**855,000**	**706,000**

The Keystone plant design used three forced draft fans per unit, with each fan enclosed in a concrete block room for sound attenuation. The circulating water preheating coils were placed as an internal wall on the inlet to the fans. The coil design operating conditions are shown in Table 7-2.

Table 7-2
Coil design operating temperatures

Ambient Air In deg F (deg C)	Air Out deg F (deg C)	Circ Water In deg F (deg C)	Circ Water Out deg F (deg C)
-20 (-29)	39.2 (4)	78 (25.6)	63.2 (17.3)
0 (-17.8)	50 (10)	85 (29.4)	72.5 (22.5)
20 (-6.7)	62.5 (17)	92 (33.3)	81.4 (27.5)
40 (4.4)	74.2 (23.5)	98 (36.7)	89.5 (32)
60 (15.6)	87 (30.6)	106 (41)	99.2 (37.3)
80 (26.7)	100 (37.8)	114 (45.4)	109 (42.5)
100 (37.8)	113 (45)	122 (50)	119 (48.4)

The system was arranged to pump water from the condenser discharge circulating water line via a 30-inch (762mm) pipe and to use two half-size booster pumps rated at 7500 gpm (1700m^3/hr) at 30 ft. (9.lm) TDH. After passing through the coils, the water is returned to the same main circulating water line. In addition, the piping was connected between units and a common steam heat exchanger provided anti-freeze protection during shutdown.

Operating Problems

During the first ten years of operation there were significant operating problems. Having the fan inlets located at the back end of the plant created the major operating problem for the coils. Fugitive flyash from the pressurized boiler and precipitator area clogged the narrow fin spacing on the coils, causing uneven air distribution, high fan draft losses, and poor performance. The coils were washed with water from fire hoses periodically, but the high pressure water damaged the thin aluminum fins on the coils. The flyash also caused corrosion damage to the fins. As the air pressure differential increased, cracks appeared on the wall and there was concern about the room wall collapsing. The uneven air flow, poor water flow distribution, slow coil drainage, and debris blockage on the water side froze the water in the tubes and caused tube failures. The result of these problems finally rendered the coil system unusable.

The Conemaugh Station, which is essentially a duplicate of Keystone and located about 25 miles (40km) away, included the same type air preheating system and has experienced the same operating problems.

Redesign, Test and Replacement

After ten years of operating problems it was obvious that in order to provide adequate air heater protection it was necessary to replace or restore the existing system. The existing type system still proved to be the most viable and economical alternative, so steps were taken to improve the design (Table 8-3). A new coil design with stainless steel tubes having only five aluminum fins per inch, 1/2 inch (13mm) high and having a much greater thickness, was installed in 1978 as a test bank on Unit 2. Two-inch (51mm) header drains and vents were also replaced with a 10-inch (254mm) drain and eight-inch (203mm) vent for quick drainage freeze protection. Successful results with this test bank resulted in a design to replace the remaining coils both at Keystone and Conemaugh.

Table 7-3
Air preheating design changes

	Original	New
Tube Fins per in (per cm)	9 (3.5)	5 (2)
Fin Height, in (mm)	5/8 (16)	1/2 (13)
Tube Material	Admiralty	316 SS
Pitch, in (mm)	2.875 (73)	2.125 (54)
Height, ft (m)	24 (7.3)	25.6 (7.8)
Draft Loss, in wg (mmHg)	0.35 (0.65)	0.29 (0.54)
Coil Outlet, (quantity) in (mm)	(1)-10 (254)	(2)-8 (203)
Header Drain, in (mm)	2 (51)	10 (254)
Header Vent, in (mm)	2 (51)	8 (203)
Coil Drain, in (mm)	3/4 (19)	3 (76)
Coil Vent, in (mm)	3/4 (19)	2 (51)

The original coils had only one water connection on the inlet and outlet headers. The test coils increased this to two each for better water flow distribution in the tubes. Also, check valves on the outlets were considered unnecessary and were removed. A simplex strainer was installed on each pump discharge to eliminate debris fouling.

Based on the positive results of these changes, replacement of all the coils at the Conemaugh Station was ordered with a design similar to the Keystone test coils. The remaining coils at Keystone are now being replaced along with the system changes noted. While the new coils do not provide the same amount of heat recovery as the original design, their use will still result in average annual savings of more than 50,000 tons (45,400 tonnes) of coal.

Heat Pipe Air Heater Technology

Heat pipes are high-performance heat transfer devices that are simple, inexpensive and reliable over a long service life. Heat-pipe-based air heaters offer a combination of benefits not possible with other air heaters. No moving parts and positive seal connections reduce leakage to effectively zero, while high heat transfer rates permit a low-weight, low-volume, low-cost package. Heat pipes also add the benefit of lower cold-end corrosion potential and improved overall boiler performance.

This section provides an introduction to heat pipe technology and an overview of how this technology is incorporated into an air heater. The heat-pipe-based air heater and its benefits are discussed, along with how this technology can improve reliability and availability in fossil plants.

Introduction

Heat pipes are self-contained devices for thermal energy transport. They are known for their ability to transport heat at high rates over considerable distances. Yet they require only small temperature differences to drive the heat flow. For example, a heat pipe generally can transport heat at a rate between 100 and 1000 times greater than the rate that is achievable in a solid conductor of the same container material, outside diameter, length, and temperature difference as the heat pipe. Heat pipes are passive devices; that is, they do not require external power to operate.

Hardware

Heat pipes have three functional areas: an evaporator, an adiabatic or transport section, and a condenser. The location of each section in the heat pipe is defined by the position of the heat source and sink to which the heat pipe is exposed The evaporator section is located at the heat source, and the condenser section is located at the heat sink. The region between the evaporator and condenser sections is the adiabatic or transport section, where there is no heat input or loss. Heat pipes consist of three physical components: working fluid, wick and container.

Selecting the working fluid for use in a specific application depends on a number of factors. The first consideration is to identify fluids that have suitable liquid-vapor saturation temperature ranges (between the triple and critical points of the fluids) for the particular application. Water is a suitable fluid for the operating temperature range of air heaters where the fluid temperature is typically 100° to 150°F (56 to 83 °C) cooler than the flue gas. Naphthalene and methanol may be used to cover the highest and lowest temperatures, respectively.

Wicks are porous or grooved structures on the inside surface of the tube and provide the structure necessary to transport the working fluid condensate by capillary action. Condensate must be transported from the condenser to the evaporator and distributed over the surface of the evaporator. With heat pipes used in air heaters, gravity provides the dominant force augmented by capillary action to transport condensate from the condenser to the evaporator, but a wick generally is still required to distribute the condensate over the entire surface of the evaporator. Wicks may also enhance heat transfer between the container and working fluid and protect the condensate from entrainment in the high-velocity counterflowing vapor.

The purpose of a container is to enclose the working fluid and wick. Heat pipe shape is not limited to a tube, despite the word “pipe”. A heat pipe may be constructed as a rigid or flexible flat plate, disk, annulus, etc. Container materials vary widely, but carbon steel is typically used in large heat exchangers.

Operation

The basic principles of heat pipe operation are relatively simple. A working fluid inside the heat pipe exists as liquid and vapor at nearly equilibrium thermodynamic conditions. During operation, two transport mechanisms occur - heat and mass transport.

At the evaporator, heat addition causes the liquid within the wick to evaporate and enter the vapor core. Heat removal at the condenser causes the vapor to condense from the vapor core and enter the wick. This simultaneous influx and efflux of vapor create the driving pressure difference for the flow of vapor from the evaporator to the condenser. To complete this thermodynamic and transport cycle, the liquid or condensate in the wick must be transported from the condenser to the evaporator against the vapor flow. Condensate return is accomplished by capillary forces created at the condensate-vapor interface in the wick, augmented by external forces such as gravity. The temperature and pressure of the working fluid approach saturated conditions and are nearly uniform throughout the heat pipe.

Performance Limits

Heat transport capability of heat pipes may be limited by five internal thermal-hydraulic phenomena:

- Viscous limit - Impaired flow of vapor due to viscous forces.
- Sonic limit - Vapor reaches and cannot exceed the sonic velocity.
- Entrainment limit - Shear forces at the vapor-condensate interface become large enough to entrain condensate droplets in the counterflowing vapor, thus interrupting the return of condensate flow to the evaporator.
- Wick or capillary limit - Capillary forces are not sufficient to overcome viscous pressure drops or any opposing external forces, thus interrupting condensate flow.
- Boiling limit - The evaporator wick dries out at high heat fluxes and boiling no longer takes place on the evaporator surface.

Some or all of these limits may be significant in a specific heat pipe application. The conditions of the application and characteristics of the particular working fluid-wick system determine which limits apply.

Service Life

The operational life of heat pipes depends on the chemical compatibility of all system components, assuming the containers are not mechanically ruptured. Inside a heat pipe, compatibility of the container and wick material(s) with the working fluid is important. Corrosion and generation of noncondensable gas are two significant results of material incompatibility. Corrosion products are likely to be deposited in the evaporator, resulting in local hot spots or a plugged wick. Noncondensable gases accumulate in the condenser, blocking the heat flow path to the heat sink.

Chemical compatibility of heat pipe internal components is very sensitive to several factors in addition to the materials themselves including material treatment processes, cleaning processes and assembly procedures. Consequently, compatibility and life tests may be necessary for each set of manufacturing procedures and final heat pipe operating conditions.

Outside a heat pipe, compatibility of the container material with the external environment is important. For example, chemical corrosion and mechanical erosion are two important concerns in applying heat pipe air heaters to utility boilers. Corrosion of the heat pipe container and extended surface may result if the surface temperature is less than the acid or water dew point temperature. Erosion of the extended surfaces may result if the flue gas dust loading is great or if extensive sootblowing is required. Fortunately, these concerns can be overcome by proper material selection, thermal-fluid design of the air heater and operation of the air heater.

Unique and Beneficial Characteristics

Aside from lacking any moving parts, heat pipes have demonstrated a number of features that make them unique among heat transfer devices. These unique, beneficial characteristics are covered in the following paragraphs.

<u>High Conductivity and Extended Surfaces</u>. The effective thermal conductivity of a heat pipe is 100 to 1000 times higher than a comparable copper rod. This feature, combined with the fact that heat transfer from the hot gas and to the cold air occurs on the outside of the heat pipe, permits using extended surfaces in both air and flue gas streams. Since the major thermal resistance in gas-to-gas air heaters are between the gas or air and the metal surface, an increase in the surface area per unit length of heat transfer tube substantially reduces the size, weight and cost of the heat exchanger.

<u>Isothermal Surface Behavior</u>. Because boiling and condensation take place in the heat pipe under constant pressure and saturated conditions, the heat pipe surface in the condenser and evaporator are each at a uniform temperature. Therefore, the gas stream will see only a constant temperature instead of a temperature gradient along the tube length found in other heat-exchanging devices. Thus, the "cold-corner" phenomenon

with its associated corrosion and fouling problems found in tubular air heaters can be avoided. Conversely, for a minimum-design metal temperature, the average flue gas temperature can be further reduced below that in a tubular or plate air heater for fuel savings through an improvement in overall plant heat rate.

Heat Flux Transformer. In the heat pipe, the length of the heat input area (evaporator) is independent of the heat rejection area (condenser) while still transferring the same amount of heat. This permits customizing the condenser and evaporator lengths and external fin spacing and type to easily accommodate different gas/air environments. For example, a longer evaporator and wider fin spacing permit easier cleaning and low erosion in a coal-ash-laden flue gas stream. But a shorter condenser with more aluminum fins allows a lower cost and a more compact surface in the benign clean air stream. In this case, low heat input per unit length of tube occurs in the flue gas side; higher heat output per unit length of tube occurs in the air side.

Thermal Diode and Temperature Control. Currently under development are heat pipes that will operate only at or above a given temperature. Below this temperature, the heat pipe would stop conducting heat and prevent tube metal temperatures from falling below the set temperature.

Current Technology

In 1985, Hudson Products signed a license agreement with Furukawa Electric Corp. of Tokyo to design, manufacture and sell heat pipes and heat pipe heat exchangers. Furukawa brings 15 years of heat pipe development and application to Hudson and the Babcock & Wilcox (B&W) Co.

Furukawa Electric is a Japanese wire and cable maker that began developing heat pipes in 1972 as an outgrowth of its materials technology research. Initial development activities focused on small, copper heat pipes for electronic cooling applications. The company's first semiconductor cooling device was introduced in 1973. Research and development activities then expanded to cover large-diameter steel and copper heat pipes for large applications. The first heat pipe heat exchanger was installed in 1975, and since that time, heat pipe heat exchangers have been installed in Japan for more than 760 heat recovery and process applications.

Performance Studies

Hudson Products conducted a series of experimental laboratory heat pipe tests at B&W's Alliance (Ohio) Research Center and at its own labs in Beasley, Texas, to supplement and extend the heat pipe technology data base licensed from Furukawa. These specific tests evaluated the thermal characteristics of individual heat pipes to determine power input and temperature performance and the effect of internal corrosion.

Power input and temperature performance: Tests showed that the outside metal temperature of the heat pipe is fairly uniform (isothermal) along the length of the evaporator and condenser sections. Also established were power limits for the heat pipe applicable to Hudson's air heater.

Internal corrosion: A proper match of fluid and tube results in no significant corrosion and degradation in performance.

Fouling and Cleaning Technology

The heat pipe air heater has incorporated the data base and operating experience of utility and industrial dirty-environment applications. Understanding the behavior of the flue gas environment and how the associated fouling affects the heat transfer and pressure drop characteristics is important to the design methodology to ensure uninterrupted operation. The type of fouling and its rate of change during operation are used to design the proper cleaning methods and cleaning cycles to prevent degradation in the air heater thermal load (higher flue gas outlet temperatures) or to minimize pressure drop increases that can limit plant capacity and increase fan power use.

Through several years of experimental work, fouling and cleaning performance characteristics of finned, staggered, tube bundles were obtained for coal-fired flue gas environments. These tests were conducted on B&W's basic-combustion test facility at its Alliance Research Center to duplicate the flue gas characteristics and flow rate approaching the test bundle. Bundle tests were conducted using Wyoming sub-bituminous and eastern bituminous fuels; test results:

- Demonstrated cleanability with sootblower equipment.
- Provided the functional performance characteristics (fouling factor and pressure loss coefficient) of the bundle before and after the cleaning cycle.
- Characterized the transient fouling behavior of the bundle over time.

The in-line, helical, finned, tube arrangement is currently accepted for coal-fired flue gas environments. However, the staggered, helical, finned, tube arrangement-offering the advantages of lower weight and volume-can also be used in a coal-fired flue gas environment. These test results show that the staggered tube bundle is cleanable via sootblowing. Combining this with the other design features incorporated in the system, air heater plugging can be avoided, making this tube arrangement a viable alternative to the in-line design.

Heat Pipes as Applied to Air Heaters

A large heat pipe air heater consists of 100 to several thousand finned heat pipes that have been assembled into the desired tube bundle arrangement. These heat pipes are sealed to the divider plate that supports the heat pipes at approximately their mid point and provides a gas-tight seal between the air and flue gas streams. The fin geometry for each gas stream can be customized for each environment to optimize the unit for low weight, volume, and cost. The heat pipe bundle, including intermediate support plates, is enclosed in a pressure-tight casing. This bundle is then slightly inclined from horizontal to ensure gravity-assisted heat pipe operation, with the hot flue gas flowing through the lower end of the bundle. The air heater can be made into a number of separate modules for ease of shipment and field assembly, depending upon the unit size and customer requirements. Cleaning equipment is then added to meet the specific flue gas application.

Ducts leading to the air heater are arranged to provide gas flow over the outside of the tubes in one direction and the airflow over the outside of the tubes in the opposite direction. This flow arrangement combines with the uniform temperature characteristic of individual heat pipes to provide close to ideal counterflow heat transfer operation and a high heat exchanger thermal efficiency.

When heat pipes are incorporated into an air heater, a number of benefits result, some of which were touched upon earlier. They are discussed in more detail in the following paragraphs.

Lightweight and compact design - The high thermal conductivity, extended surfaces, gas-stream-customized surfaces, and true counterflow heat exchanger behavior combine to provide an air heater package far smaller and lighter than a tubular unit. The weight can be comparable to a regenerative unit.

"Zero" leakage - The heat pipes are stationary and are sealed to the divider plate, providing no direct leak path between the air and gas streams. In addition, a heat pipe must fail twice, once in each of the air and flue gas streams, before appreciable leakage can occur.

Low maintenance - With no moving parts, heat exchanger maintenance is minimal. Customizing the flue gas surface for the specific application environment also minimizes the likelihood of cleaning problems.

Reduced cold-end corrosion potential - Because of the isothermal heat pipe behavior and nearly ideal counterflow heat transfer, no cold corner exists in the unit. The averaging effect of the heat pipe fluid biases the tube metal surface temperatures closer to the average flue gas temperature. This feature plus the zero leakage combine to either: 1) keep minimum metal temperatures in the flue gas 10°to 40°F (5.5 to 22 °C) or more higher than tubular or regenerative units; or 2) permit a lower flue gas exit temperature for the same minimum metal temperatures.

Lower stresses - Heat pipes are securely connected at their midpoint divider plate locations only. The ends are permitted to grow axially, thus minimizing the build-up and cycling of thermal stresses.

Lower pressure drop - For a given application, design studies and measurements demonstrate that the heat pipe air heater has less total pressure loss than other heat transfer equipment for the same operating conditions.

Lower erosion potential - Finned heat pipes operate at a higher thermal efficiency with lower flue gas velocities, minimizing erosion potential.

The carbon steel-water heat pipe system permits the highest heat transfer rates of any fluid while using inexpensive carbon steel tubes. The result is a lower cost, lighter, lower volume heat pipe air heater for a specific application. Flue gas and air-side surfaces are specially designed to typically use:

Widely spaced carbon steel fins in the flue gas to minimize fouling and maximize cleanability.

High-density aluminum fins on the air side to optimize heat exchanger surface.

The heat pipe air heater offers significant economic and technical benefits for retrofitting existing plants and designing new plants. The primary savings are in efficiency, operation and maintenance. The heat pipe air heater can be arranged to suit the needs of the coal-fired air heater application in a compact, lightweight package that offers no leakage, low maintenance, minimized cold-end corrosion and low pressure drop.

High heat transfer rates result in an air heater that is more compact than a tubular design. The leak-free design results in a lower heat rate and reduced fan power requirements. The heat pipes themselves are passive devices, so the air heater has no moving parts, bearings, or seal wear-all of which reduce maintenance cost and extend time between outages. Finally, heat pipes are isothermal, which improves corrosion resistance and lowers cold-end temperatures for greater boiler efficiency.

Component Modifications

There are many areas in the power plant where small investments can improve unit heat rate and provide cost savings. Most of these problem areas can be found by following a good test plan.

1. Boiler duct expansion joint leaks can cause multiple problems from improper combustion due to false O_2 readings to increased corrosion in the air preheaters. If leaks are found investigation into improved expansion joint designs should be made.

2. The boiler O_2 measurement is extremely important to proper boiler operation. Investigation should be made to determine if the proper number of analyzers exist and if they are in the proper locations for accurate indications. Most units have O_2 analyzers installed in the economizer outlet ducts, but the possibility of installing them in the furnace could prove beneficial if air inleakage is a problem.

3. Windbox damper operation is critical for proper fuel and air mixing. The dampers should be operated automatically by reliable drive units.

4. It is important that accurate feedwater heater level controls are installed and maintained. Incorrect levels in the heaters can cause performance problems and tube damage.

Boiler Corrosion

Experience at the Pennsylvania Power and Light Co., Martins Creek station has shown that several components in the boiler can suffer corrosion from acid that condenses from the flue gas. The components most affected by dewpoint corrosions are air heaters, air preheaters, ID fans and ductwork.

Air Heaters

Several alternatives have been studied to try and minimize air heater corrosion:

- Varying air heater speed to increase average cold end temperature during startups and low load operation.

- Removing baskets or changing basket heights for winter operation.

- Installation of air side or gas side bypass.

- Change of basket materials.

Of the above alternatives the only one that was found practical and easy to implement was the change of the basket materials. Hot end basket material was changed from carbon steel to corten, intermediate baskets are now enameled iron instead of corten; no change was made on the material for the cold end baskets which are still made of enameled iron.

Air preheater coils located under the air heater in the air side ducts experienced severe dewpoint corrosion problems. The original coils corroded through, causing many coil leaks. They were manufactured using copper tubes and fins for high thermal conductivity. The acid-containing atmosphere affecting these coils during the unit shutdowns and during air heater washes caused these coils to corrode in the first two years of cycling operation. To solve this problem the coils were replaced with stainless steel tubes and copper fins which prevented the tubes from corroding while the copper fins still maintained the necessary heat transfer capability. As one would expect, the copper fins began to corrode away leaving only the stainless tubes, with a reduction in heat transfer capability.

It was evident that new ideas needed to be considered to resolve the corrosion problem. The following alternatives were studied:

Duplicate the present 4-row stainless steel tubes with copper fins but with thicker fins and coated with "Heresite", a thermosetting protective coating which was designed to resist attack from corrosive fumes.

Replace the present coils in the same location with stainless steel tubes and fins.

Replace the coils with the original copper tube and fins but using the "Heresite" coating.

The alternative using stainless steel for the tubes and fins presented a major problem concerning performance. To use stainless steel would require additional surface area so that the temperature rise would remain unaltered. This meant that due to space constraints the coils had to be relocated. At this time the first alternative was preferred since it demanded a lower investment. In addition, the coating proposed by the manufacturer was backed by years of experience at other plants under similar corrosion conditions. Within a year of service, the coating started to blister and peel off, leaving the copper fins exposed to the fumes. Two years later the fins were completely eaten away resulting in pluggage of the air passages and much reduced thermal performance.

By the third attempt to resolve the corrosion problem of the air preheat coils, the consensus was that the coils had to be moved away from the air heaters to eliminate the corrosive environment. This would allow us to convert the fins and tubes to stainless steel by virtue of the additional room gained at the new location. The coils are now located at ground level near the forced draft fans and all the corrosion problems are virtually eliminated.

Ductwork and ID Fan

The ductwork and ID fan housings have suffered severe damage due to the dewpoint corrosion. This area of the plant requires attention during the unit annual outage: repairs to expansion joints, patching of ductwork floor and walls, rebuilding of structural struts and replacing of insulation. Initially the repairs were done using the same materials as the original construction, carbon steel, but in the past several years new alloys have been tested with various degrees of success.

The first attempt to improve the quality of the ductwork repairs was utilizing stainless steel 304 and 316 of the same thickness as the original. To our surprise, the corrosion of the stainless was not better than the carbon steel and in some cases, worse, since deep craters of localized corrosion were observed particularly in the 316 stainless. In addition, both stainless steels showed early degradation of mechanical properties which were evident by brittle fractures found emanating from the areas under acid attack. The degree of metal loss of the 304 and 316 stainless was found to be about 160 mils/year which is equivalent to the metal loss sustained by the mild steel.

Corrosion rates observed in the ductwork leading from the air heaters to the ID fans is not uniform. It is most severe on the floor area directly below the air heaters and decreasing in severity from that point to the ID fans. However, some pockets of extreme corrosion have been found in the inlet housing of the ID fans. Walls and ceilings of the ductwork haven't suffered as much corrosion damage as the floor. This pattern of corrosion corroborates observations during the unit shutdowns when virtual "rain" of acid condensate is seen running down the walls and collecting in the low lying areas of the floor as the ductwork cools off below the acid dewpoint.

Taking into account the dollars spent on the repairs to the ductwork and the pattern of corrosion, it was concluded that for the floor areas below the air heater a high nickel alloy should be used since it would afford a good degree of protection against corrosion and also provide adequate resistance against mechanical abrasion and impact caused by the tools used during the cleaning of the debris following the air heater washes. For other areas where corrosion and mechanical abrasion are less severe the use of a protective coating should be investigated.

A list of alloys used in typical applications were evaluated for price and suitability. Table 7-4 shows these alloys indicating results of tests. Carpenter 20 (34% Ni, 20% Cr, 2.5% Mo) was selected based on price and performance. Its expected performance will increase the life of the ductwork by a factor of 8 as compared to the existing carbon steel.

Table 7-4
Alloy corrosion rate vs cost

Alloy	Corrosion Rate mils/yr	Cost $/lb
Hastelloy C-276	1.6	9.25
Hastelloy G	3.6	6.07
Cabot 625	7.2	8.53
Cabot 825	24.4	4.30
Carpenter 20	24.8	4.28
Cabot 904L	27.2	3.90
Carbon Steel	200.0	0.35

Digital Controllers

With the importance of maintaining performance parameters at expected values under all operating conditions combined with almost continuous cycle and component modifications, analog control systems which require extensive wiring revisions for control logic upgrades are becoming obsolete. As an example, the requirements associated with variable pressure operation place a large burden on the steam generator control system with the steam temperature controls the most severely affected.

The energy storage of a steam generator is greater at full load than at low loads. Therefore, an increase in load temporarily requires raising the firing ramp rate above the load ramp rate to increase the stored energy of the steam generator. With variable pressure operation, the slope of the stored energy curve is much greater than for constant pressure.

Over-firing to satisfy the energy storage requirements of the waterwalls, in most cases, results in higher heat transfer rates in the superheater and reheater which in turn have less energy storage capacity. Also, the distribution of heat between the waterwalls and the superheater changes as a function of pressure. The redistribution of heat is primarily attributed to changes in the heat of vaporization of water and the specific heat of superheated steam with changes in pressure. Some superheater temperature control systems employ cascade sprays to reset the final temperature errors for superheater outlet temperature. This type of control system is inadequate for load changes when employing variable pressure operation. A system with sophisticated anticipation signals to predict the reaction of the superheater and reheater as a function of overfiring to meet changes in load is needed.

Control of the feedwater circuit must also be considered. The boiler drum level is susceptible to shrink and swell due to changes in the steaming rate. With variable pressure operation, a change in mass inventory also results due to changes in pressure. To assist the drum level controller, instrumentation must be provided to distinguish between the effects of shrink and swell as well as mismatches in feedwater and steam flow rates, and more sophisticated controls will be required to maximize the rate of load change.

Maintenance Practices

It could go without saying that if proper unit maintenance is not carefully planned and executed then unit thermal performance will suffer greatly. A good performance program should help drive the maintenance plan. The maintenance plan should not drive the unit performance results.

Predictive maintenance is an area which can draw heavily on the performance data generated by a heat rate program to aid in outlining a comprehensive maintenance program and be used as input into the maintenance decision making process. There are several ways in which the use of performance data can enhance utility predictive maintenance activities. These include using performance data for preventive maintenance planning, predicting equipment maintenance or modification needs, and cost/benefit evaluations of maintenance projects.

Efficiency related preventive maintenance is defined as that which is expected to be regularly required to prevent or minimize the degradation of equipment which reduces the efficiency of the unit. Examples of efficiency related preventive maintenance include inspection and checks of fuel supply system components, control system components, fuel burning equipment, heat transfer surfaces, instrumentation, fan blades, fan and pump motors, and steam traps and drains. The need to periodically perform a particular task and the time interval between task performance is based on past experience, vendor recommendations or degradation of a performance parameter.

One way in which performance data can aid in preventive maintenance planning is through the use of heat rate deviation data. These data identify the parameters which are the major contributors to heat rate degradation. Each of these parameters has associated with it a set of equipment whose operation directly influences the performance of that parameter. This can be used as a simple guideline for the establishment of a preventive maintenance program.

Predicting the need for maintenance or cycle modification is another potentially valuable use of performance data. There are several examples of utilities which have identified efficiency degradation through their testing programs and then have justified an early unit outage based upon the savings associated with correcting the efficiency losses. Other utilities have used the results of testing or the trending of performance parameters to predict the condition of the turbine prior to a scheduled overhaul. This allows the utility to order critical parts in advance to minimize the length of the outage.

Examples of maintenance activities which can be influenced by performance data are: condenser tube cleaning, sootblowing, turbine nozzle block repair or replacement, turbine blade repair or replacement, control system turbine, repair or replacement, location and repair of cycle water losses, and location and repair of boiler casing leaks. On an industry wide basis, the use of performance data to predict maintenance needs is an action which could be used to a much greater benefit than it currently is.

A third use of performance data in predictive maintenance is as a factor in the cost/benefit evaluation of maintenance and modification projects. Some utilities have cultivated a disciplined capability to make decisions on improvement projects based on cost/benefit ratios. The costs and benefits relative to heat rate improvement include the project costs (labor and material), lost revenue costs for the outage, and fuel costs with and without the modification. Performance data are required to estimate the fuel savings and also to aid in identifying the root cause of the problem and designing a modification to resolve the problem.

One of the highest priorities of the plant staff is to keep the unit operating and capable of generating power. Therefore, many of the preventive maintenance routines which are important to maintaining good unit heat rate are not perceived as being essential to continued plant operation. However, many or most of these preventive maintenance activities are necessary to prevent partial deratings of the unit. For example, much has been said about the importance of condenser back pressure on unit heat rate, yet a high condenser back pressure can also limit the maximum output of the unit. The importance of coal fineness on boiler efficiency has been discussed several times. If pulverizer maintenance is neglected, increased coal size can result in limitations on the units capacity.

Cost Benefit Analysis

General

In 1992, Southern California Edison Company demonstrated the usefulness and applicability of EPRI's *Heat Rate Improvement Guidelines for Existing Fossil Plants* at the Ormond Beach Generating Station Unit 2. This study is described in EPRI Report TR-101249. This report applied EPRI developed guidelines and reports to:

- Install turbine cycle instrumentation to provide ASME Performance Test-quality data and an automated data acquisition system.

- Perform baseline and subsequent post-modification testing to evaluate cycle and component performance.
- Identify cycle and components that exhibited degraded efficiency.
- Establish heat rate improvement goals for the unit.
- Develop and evaluate operational, equipment, maintenance actions and cycle modifications for heat rate improvement.
- Implement selected unit modifications and evaluate resulting performance improvement.

As a result of this study, methods to increase efficiency of some of the degraded components were evaluated as part of the program. Major equipment modifications were:

- Mod 1 - New steam supply for main feedpump drive turbine.
- Mod 2 - Refurbished high pressure turbine.
- Mod 3 - Adjustable speed forced draft fan motors.
- Mod 4 - Low excess-air burners.

Additionally, three maintenance actions implemented to improve heat rate were:

- Steam-generator control valves repaired to reduce leakage.
- Improved feedwater level control to increase final feedwater temperature.
- Condenser tube cleaning to reduce condenser pressure.

One of the capital projects and one of the maintenance projects from the Ormond Beach Report is highlighted here for demonstration of the applied principles of cost/benefit for heat rate improvement.

Capital Expenditure Improvement of Heat Rate

The following is an example from the EPRI Report TR-101249 for Ormond Beach.

Item 1 - Main Feedpump and Drive Turbine

Cycle performance tests determined that the steam flow to the main feedpump drive turbines was about twice that expected from the As-design calculations. Moreover, at higher loads, this steam was taken from the HP turbine exhaust rather than the IP turbine exhaust, as designed. The effect of these deviations from design resulted in close to two percent increase in NUHR at higher loads.

Analysis and potential modifications of the main feedpump and turbine performance deficiency were extensively investigated as part of the SCE/EPRI program. This work included a heat balance model of Unit 2 current configuration that matched the tested turbine steam flows and equipment performance. It was found from using the model that the altered steam source caused about three quarters of the heat rate deviation (1.5 percent NUHR), while component inefficiency caused the remainder. These results are described in Appendix B and summarized in (8), of the original EPRI Report for Ormond Beach EPRI TR-101249.

The main feedpump drive turbines were designed to use three different steam sources, depending on load. From start-up to about twenty percent load, steam was taken from a source within the steam-generator, and as load increased, from the HP turbine exhaust (also called cold reheat or second point extraction). For improved cycle efficiency above forty percent load, the system was designed to automatically provide steam from the IP turbine exhaust (also called crossover, or fourth point extraction). However, actual operation at higher loads required cold reheat steam because of both poor controllability at high loads and unacceptably slow load response when the steam supply to the turbines was from the IP turbine exhaust. The slow response when operating with crossover steam was found due to insufficient BFP turbine control capability. With lower-than-design pump and turbine efficiency, a proportionate increase in steam flow was required to generate the necessary pump power. This was achieved by opening the turbine control valves to approximately ninety percent open during normal operation. Testing found that the turbine governor became increasingly nonlinear above eighty percent open, which resulted in the earlier mentioned poor controllability.

Component Efficiency

Performance measurements determined that the combined (the product of) pump and drive turbine efficiency was 57 percent at 670 MW, some 11 percentage points less than design prediction at this load. Heat balance calculations indicated this deviation in efficiency caused 30 to 40 Btu/kWh increase in Unit 2 heat rate.

Feedpump and drive turbine efficiency could not be determined separately. However, bounds of possible efficiencies were estimated from the definition of combined pump turbine efficiency as:

1. If the pump performed at its design efficiency (85 percent), the drive turbine efficiency would have been 67 percent.

2. If the drive turbine performed at its design efficiency (80 percent), then the pump efficiency would have been 71 percent.

One direct measurement of drive turbine efficiency at minimum load (50 MW) found that it was only 54 percent. At this load, the turbine exhaust steam was superheated and consequently the measured steam temperature and pressure defined leaving enthalpy (at four circumferential locations). Although such measurement had large uncertainty, it did appear to indicate that the drive turbine was the primary cause of the system deviation. More testing was deemed necessary to conclusively prove this deduction. No corresponding pump efficiency could be determined, however, as cycle isolation problems resulted in uncertain feedwater flow through the main feedpump.

Mod 1 - Main Feedpump Turbine Steam Supply

To show the heat rate improvement unequivocally, input/output tests were performed sequentially, first with the drive turbines supplied with second point extraction steam (for the Baseline test) and then with the new steam supply.

Results

Heat rate and turbine cycle testing determined that the new steam supply for the drive turbine provided a 1.2 percent improvement in NUHR at 500 MW and at 670 MW.

<u>Benefit</u>. Annual levelized savings in fuel cost = $1,600,000.

<u>Cost</u>. Total investment cost for the modifications = $1,000,000, for an annual levelized capital cost of $200,000.

<u>Benefit-to-Cost Ratio</u>. = $1,600,000/$200,000 = 8

<u>Background</u>. Life cycle costs for various pump and cycle arrangements were evaluated to improve Unit 2 thermal performance as part of the EPRI program. As discussed in Appendix B, candidate modifications included:

- Upgrade drive turbine efficiency

- Provide a third feedpump with either a motor or turbine drive
- Add a steam supply from the third point extraction.

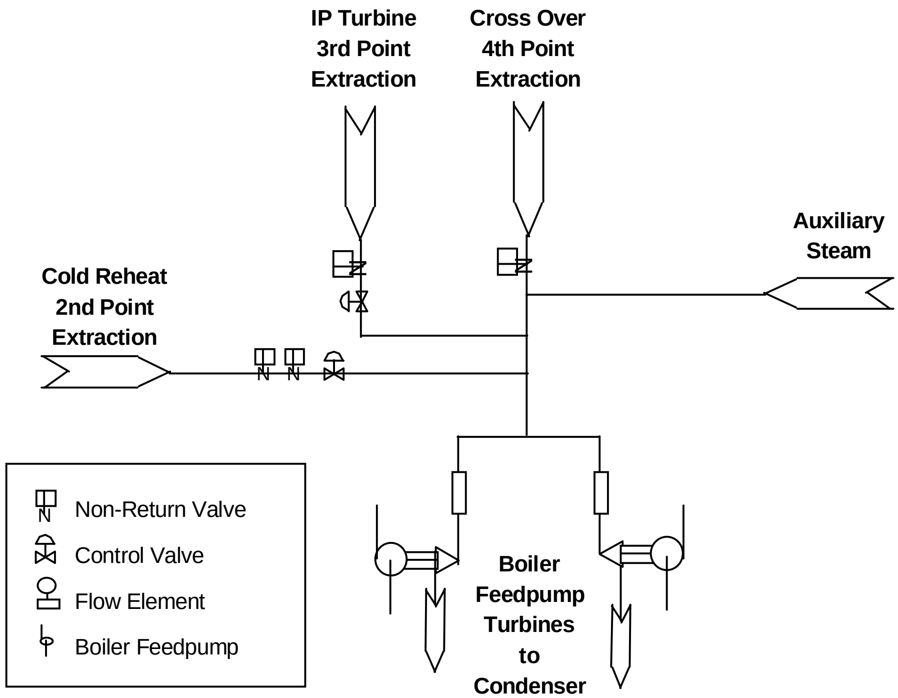


Figure 7-1
Schematic of boiler feedpump steam supply arrangement shows the additional extraction source taken from the Intermediate pressure turbine.

Results of this study showed that a new steam supply - from the third turbine extraction point - was the most cost effective (12). The new steam supply, along with a new digital governor for the drive turbines, were installed in late February 1989 during a four week maintenance outage.

Maintenance

The 1988 maintenance outage included overhaul of the steam-generator flow control valves, condenser cleaning and airheater resealed, in addition to the HP turbine refurbishment, previously discussed. This section briefly describes the first two actions and the corresponding heat rate improvement; instrumentation inaccuracy prevented the evaluation of the reduction in leakage provided by airheater seals.

Steam-Generator Control Valves

Control valves were installed as part of the low load modification for Unit 2 to bypass excess feedwater or steam during emergency and start-up conditions. Although not conclusively proven, it was deemed that the refurbished control valves reduced the leakage flow and reduced Unit 2 heat rate at lower loads.

Improved cycle isolation was deduced from analysis of the Comparison of the total component heat rate deviations to deviation, see Table 7.5 (EPRI Table 4-1), determined the NUHR improvement 10.3 percent at 50 MW and 2.8 percent at 250 MW. The same determine the loss due to cycle isolation, as was applied in Section 3.

Benefit. Annual fuel-cost savings = $190,000.

Cost. Assumed annual cost to refurbish control valves = $25,000.
Benefit-to-Cost Ratio. = $190,000/ $25,000 = 7.6

Table 7-5
EPRI Table 4-1 - Post Modification Heat Rate at Ormond Beach Unit 2 Compared to Baseline

	Nominal Load (MW)			
A. Component Improvement (Percent)	50	250	500	670
1. BFP Turbine	0	0	1.2	1.2

Table 7-6
EPRI Table 4-2 - Summary of Economic Evaluation of the Implemented Modifications and Maintenance Actions to Improve Heat

Modification	Benefit (1000 $/yr)	Cost (Cost $/yr)	Benefit/ Cost
Component:			
1. BFP Turbine	1600	200	8.0
Maintenance:			
a. Cycle Isolation	190	25[3]	7.6

[3] Estimated for annual valve refurbishment.

8

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REFERENCES

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A

PROCEDURE FOR CALCULATING EXPECTED UNIT NET HEAT RATE

1.0 Purpose

The purpose of determining design unit net heat rate is to provide a standard for comparison with actual unit net heat rate.

2.0 Data Required

- Generator Gross Load
- Throttle Steam Flow
- Condenser Pressure
- CCW Inlet Temperature
- Number of CCW Pumps in Operation
- Fuel Analysis (optional)
- Boiler Inlet Air Temperature (optional)

3.0 Calculation

$$EUNHR = \frac{DTGHR\,(1 + \frac{EPCF}{100})}{TBE\,(1 - \frac{ESSCF}{100})} \qquad \text{(eq. A-1)}$$

where:

EUNHR = Expected Unit Net Heat Rate

DTGHR	=	Design Turbine Gross Heat Rate (Design)
EPCF	=	Exhaust Pressure Correction Factor (%)
TBE	=	Theoretical Boiler Efficiency (%) (Design)
ESSCF	=	Expected Station Service Correction Factor (%)

4.0 Notes

Design Turbine Gross Heat Rate (DTGHR) is a function of generator gross load. A relationship fir this can be developed using a series of turbine-generator cycle design heat balances having the same operating parameters throughout the load range (i.e., same throttle pressure, turbine back-pressure, percentage make-up flow, etc.). See example.

Exhaust Pressure Correction Factor (EPCF) is a function of throttle steam flow and turbine absolute exhaust pressure. A computer modeling code can be used to obtain this relationship if it is unavailable from the turbine manufacturer. See example.

Expected Condenser Absolute Back-Pressure is a function of CCW inlet temperature, condenser heat load (steam flow), and number of CCW pumps in operation (see example). This relationship can be developed using Heat Exchanger Institute (HEI) condenser design calculations if it is unavailable from the condenser manufacturer. As an index to condenser heat load, a relationship between throttle steam flow and condenser steam flow can be developed from the heat balances mentioned above.

Design Boiler Efficiency is a function of fuel quality, inlet air temperature, and expected boiler operating parameters (i.e., furnace O_2, air heater leakage, flyash combustibles, exit-gas temperature, etc.). It can be calculated using the heat loss method of boiler efficiency determination. As an alternative, design boiler efficiency can be obtained from the boiler manufacturer's performance summary sheet as a function of throttle steam flow. See example.

Expected Station Service (Auxiliaries) Correction Factor is a function of generator gross load. It is a measure of the expected megawatt load required for station electrical service and is a percentage of generator gross load.

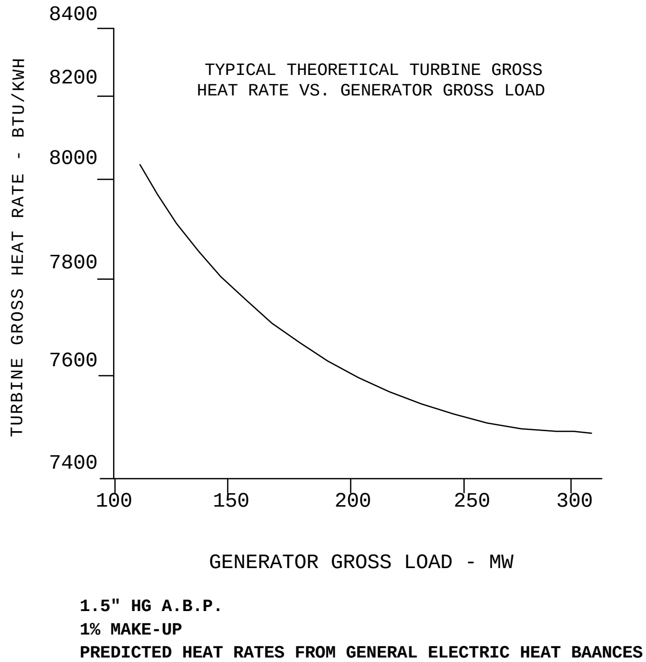

Figure A-1
Typical Theoretical Turbine Gross Heat Rate vs. Generator Gross Load

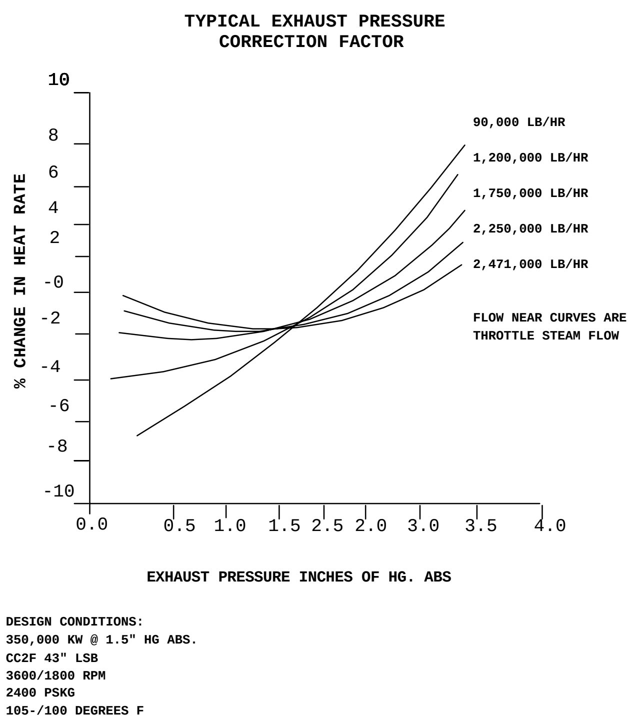

Figure A-2
Typical Exhaust Pressure Correction Factor

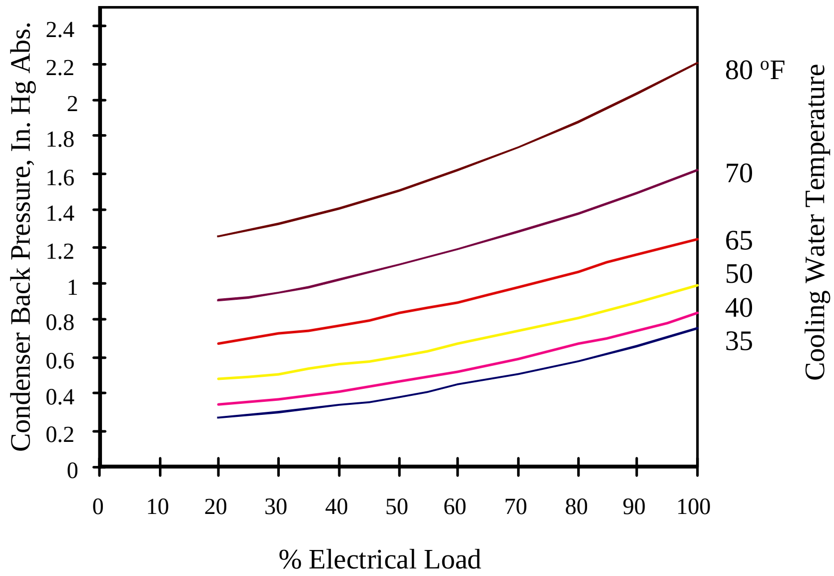

Figure A-3
Typical Expected Condenser Back Pressure v.s. Electrical Load

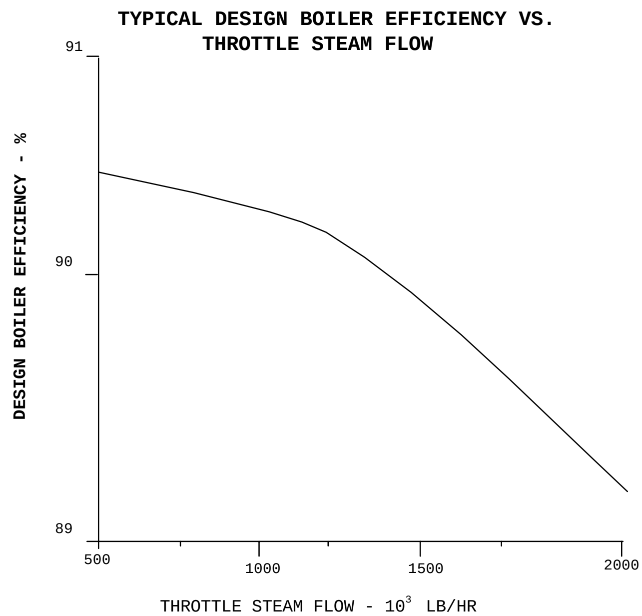

Figure A-4
Typical Design Boiler Efficiency vs. Throttle Steam Flow

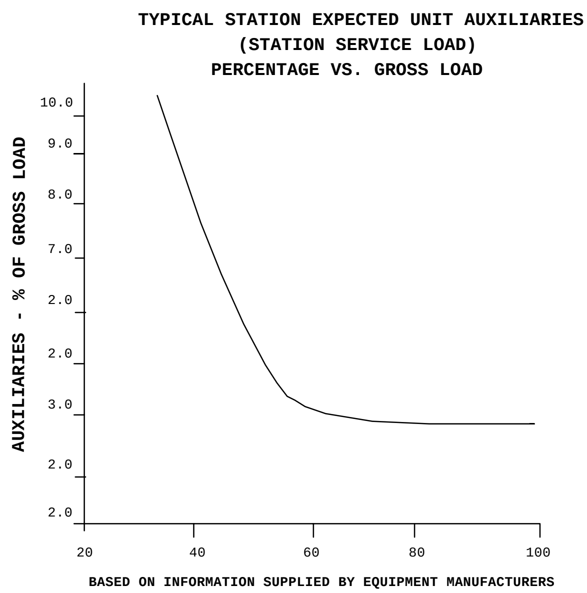

Figure A-5
Typical Station Expected Unit Auxiliaries (Station Service Load) Percentage vs. Gross Load

B

PERFORMANCE PARAMETER ACCOUNTING MANUAL

Note: **References in this section to Appendix D refer to the original manual CS-4554.**

B.1 High Pressure Turbine Efficiency

B.1.1 Effect on Heat Rate

Utility Average: 18.8 Btu/kWh/% (19.9 kJ/kWh/%)

Utility Range: 17.9-21.5 Btu/kWh/% (19.0-22.8 kJ/kWh/%)

Plant-Specific Value: Industry data can be used as an estimate. More specific value can be obtained from heat balance code analysis.

B.1.2 Possible Causes of Deviation

- Erosion of nozzle blocks
- Erosion of turbine blades
- Broken turbine blades
- N2 packing peak (HP and IP turbine are in same shell)
- Excess gland packing leaks
- Strip seal leakage
- Malfunctioning control valve

B.1.3 Possible Corrections

Operator Controllable: None

Maintenance Correctable:

- Repair or replace nozzle block
- Repair or replace turbine blades
- Clean turbine blades
- Replace gland packing
- Replace turbine seal strips

NOTE: The items noted by an asterisk have been extracted from the Heat Rate Accounting Manual furnished by Consumers Power Company.

B.2 Intermediate Turbine Efficiency

B.2.1 Effects on Heat Rate

Utility Average: 14.5 Btu/kWh/% (15.4 kJ/kWh/%)

Utility Range: 11-19.1 Btu/kWh/% (11.7-20.2 kJ/kWh/%)

Plant-Specific Value: Industry data can be used as an estimate. More unit-specific value can be obtained from a heat balance code analysis.

B.2.2 Possible Causes of Deviation

- Erosion of turbine blades
- Deposits on turbine blades
- Reheater bypass valve leakage
- Excess gland seal leakage
- Strip seal leaks

B.2.3 Possible Corrections

- Operator Controllable: None
- Maintenance Correctable:
  - Repair or replace turbine blades
  - Repair leaking reheater bypass valve
  - Repair strip seals
  - Repair gland seals

B.3 Main Steam (Throttle) Pressure

B.3.1 Effects on Heat Rate

Utility Average: .35 Btu/kWh/psi (3.4 kJ/kWh/kPa)

Utility Range: .03-.65 Btu/kWh/psi (3 - 6.3 kJ/kWh/kPa)

Plant-Specific Value: Effects on heat rate of deviation in throttle pressure is normally supplied with the unit thermal kit.

B.3.2 Possible Causes of Deviation

- Feedwater flow too low (once-through units)
- Firing rate inadequate

B.3.3 Possible Corrections

Operator Controllable:

- Increase feedwater flow
- Increase firing rate

B.4 Main Steam (Throttle) Temperature

B.4.1 Effects on Heat Rate

Utility Average: 1.4 Btu/kWh/^{0}F (2.7 kJ/kWh/^{0}C)

Utility Range: .7-1.7 Btu/kWh/^{0}F (1.3 - 3.2 kJ/kWh/^{0}C)

Plant-Specific Value: Effect on heat rate of deviation in main steam temperature is normally provided with the unit thermal kit.

B.4.2 Possible Causes of Deviation

- Superheater spray control problems
  - Superheater spray valve leakage
  - Fouling of the superheater (low temperature)
  - Fouling of the boiler waterwall (high temperature)
  - High excess air
  - Burner tilts mispositioned
  - Gas tempering flow inadequate
  - Bypass dampers mispositioned
  - Temperature control setting calibration drift
  - Superheater tube leaks
  - Incorrect amount of superheater heat transfer surface

B.4.3 Possible Corrections

Operator Controllable:

- Blow soot
- Adjust burner tilts
- Adjust bypass damper settings

- Adjust attemperating air flow damper
- Control excess air
- Manually control superheater spray flow

Maintenance Correctable:

- Calibrate temperature control setpoint
- Repair superheater spray control valve
- Clean boiler waterwalls
- Clean superheater platens
- Repair superheater tube leaks
- Add or remove superheater heat transfer surface

B.5 Reheat Temperature

B.5.1 Effects on Heat Rate

Utility Average: 1.3 Btu/kWh/^{0}F (2.5 kJ/kWh/^{0}C)

Utility Range: .9-1.9 Btu/kWh/^{0}F (1.7 - 3.6 kJ/kWh/^{0}C)

Plant-Specific Value: Effects on heat rate of deviations in reheat temperature can be obtained from the unit thermal kit or using the method of Appendix D.

B.5.2 Possible Causes of Deviation

- Reheat attemperation control problems
- Reheat attemperation control valve leakage
- Fouling of the reheater (low temperature)
- Fouling of the boiler waterwall (high temperature)
- Fouling of the superheater

- High excess air
- Burner tilts mispositioned
- Gas tempering flow inadequate
- Bypass dampers mispositioned
- Reheater tube leaks
- Incorrect amount of reheater heat transfer surface

B.5.3 Possible Corrections

Operator Controllable:

- Blow soot
- Adjust burner tilts
- Adjust bypass damper settings
- Adjust attemperating air flow damper
- Control excess air
- Manually control reheat spray flow

Maintenance Correctable:

- Repair superheater spray control valve
- Clean boiler waterwalls
- Clean superheater platens
- Clean reheater platens
- Repair reheater tube leaks
- Add or remove reheater heat transfer surface

B.6 Superheater Attemperation

B.6.1 Effects on Heat Rate

Utility Average: 2.46 Btu/kWh/10,000 lb./hr spray flow (1.18 kJ/kWh/10,000 kg/hr)

Utility Range: 1.3-3.1 Btu/kWh/10,000 lb./hr spray flow (.62 - 1.48 kJ/kWh/10,000 kg/hr)

Plant-Specific Value: Can be computed from the following formula or the methods of Appendix D.

$$HRDeviation = NUHR * (0.028) * \frac{(Wa - We)}{Wthr} \quad \text{(eq. B-1)}$$

where

NUHR = Net unit heat rate, Btu/kWhr (kJ/kWhr)

W_a = Actual spray flow, lb./hr. (kg/hr)

W_e = Expected spray flow, lb./hr. (kg/hr)

W_{thr} = Expected throttle flow, lb./hr. (kg/hr)

k = Constant, .028 (.0134 SI)

B.6.2 Possible Causes of Deviation

- Improperly adjusted control setpoint
- Leaking spray control valve
- Broken spray nozzle
- Fouling of boiler waterwalls
- High levels of excess air
- Improperly set gas attemperation
- Improperly set gas bypass dampers

B.6.3 Possible Corrections

Operator Controllable:

- Blow waterwall soot
- Reduce excess air to proper levels
- Adjust gas attemperation
- Adjust gas bypass dampers

Maintenance Controllable:

- Repair spray valves
- Calibrate temperature controls
- Replace spray nozzle

B.7 Reheat Attemperation

B.7.1 Effects on Heat Rate

Utility Average: 21.5 Btu/kwh/10,000 lb./hr spray flow (10.3 kJ/kWh/10,000 kg/hr)

Utility Range: 10-36.6 Btu/kWh/10,000 lb./hr spray flow (4.79 - 17.5 kJ/kWh/10,000 kg/hr)

Plant-Specific Value: Can be computed from the following formula.

$$HR\ Deviation = NUHR * (k) * \frac{(Wa - We)}{Wthr} \qquad \text{(eq. B-2)}$$

where

NUHR = Net unit heat rate

W_a = Actual spray flow, lb./hr. (kg/hr)

W_e = Expected spray flow, lb./hr. (kg/hr)

W_{thr} = Expected throttle flow, lb./hr. (kg/hr)

k = Constant 0.2 (.096 SI)

B.7.2 Possible Causes of Deviation

- Fouled waterwalls
- High levels of excess air
- Fouled superheater sections
- Improperly set gas bypass dampers
- Improperly adjusted temperature setpoint
- Leaking spray control valve
- Broken spray nozzle

B.7.3 Possible Corrections

Operator Controllable:

- Adjust gas bypass dampers
- Adjust excess air to proper levels
- Soot blow waterwalls
- Soot blow superheater sections

Maintenance Controllable:

- Repair spray control valve
- Replace spray nozzle
- Calibrate temperature control setpoint

B.8 Excess Air (0_2)

B.8.1 Effects on Heat Rate

Utility Average: 29.4 Btu/kWh/% (31kJ/kWhr)

Utility Range: 18-36 Btu/kWh/% (19. - 38 kJ/kW)

Plant-Specific Value: Can be computed from the following formula:

$$HR\ Deviation = NUHR * [\frac{BE}{(BE - .26 * O_2\ \ DEV)}] - NUHR \qquad \text{(eq. B-3)}$$

where

NUHR = Net unit heat rate

BE = Expected boiler efficiency

0_2 DEV = Actual expected 0_2 measurement.

B.8.2 Possible Causes of Deviation

- Fuel/air flow control problems
- Change in mill fineness
- Boiler casing leaks
- Air heater leaks
- Hot precipitator leaks
- Malfunctioning burner(s)
- FD fan inlet vanes mispositioned
- Burner registers mispositioned

B.8.3 Possible Corrections

Operator Controllable:

- Adjust FD fan inlet vanes
- Adjust FD fan speed (variable speed)

Maintenance Controllable:

- Adjust burner registers
- Clean or repair burners
- Repair air leaks
- Calibrate fuel/air flow controls
- Adjust pulverizer classifier vanes
- Replace pulverizer grinding wheels, balls, or rings

B.9 Exit Gas Temperature

B.9.1 Effects on Heat Rate

Utility Average: 2.7 Btu/kwh/°F (4.3 kJ/kWh/°C)

Utility Range: 2.1-4.2 Btu/kWh/°F (3.3 - 6.6 kJ/kWh/°C)

Plant-Specific Value: Exit gas temperature must be corrected for air heater leakage and for combustion air inlet temperature. The correct exit gas temperature can be calculated using the following equations:

$$AH\ \ leakage\ (\%) = 90 * \frac{\left(O_2 in - O_2 out\right)}{\left(O_2\ \ out - 21\right)} \qquad \text{(eq. B-4)}$$

or

$$AH\ \ leakage\ (\%) = 90 * \frac{\left(CO_2\ \ in - CO_2\ \ out\right)}{CO_2\ \ out} \qquad \text{(eq. B-5)}$$

where all gas measurements are made at the entrance and exit of the air preheater.

$$TG(NL) = [AH\,leak(\%) * 0.24 * \frac{(TG - TA)}{24.2}] + TG \quad \text{(eq. B-6)}$$

where

TG (NL) = Gas temperature corrected to zero leakage

TG = Measured exit gas temperature at the air heater exit

TA = Measured air temperature at the air heater inlet.

B.9.2 Possible Causes of Deviation

- Bypass dampers mispositioned
- Air heater baskets corroded/eroded
- Air heater baskets fouled
- Attemperating air flows misadjusted
- Combustion air heater in use

B.9.3 Possible Corrections

Operator Controllable:

- Reduce excess air
- Adjust bypass dampers
- Adjust tempering air flows
- Use air heater sootblowers
- Adjust steam flow to coils
- Adjust air recirculation dampers
- Remove combustion air heater from operation

Maintenance Correctable:

- Repair or replace air heater baskets

- Repair air heater leakage

B.10 Condenser Backpressure

B.10.1 Effects on Heat Rate

Utility Average: 204 Btu/kWh/In. HG (8.47 kJ/kWh/mmHG)

Utility Range: 42-269 Btu/kWh/In. HG (1.7 - 11.2 kJ/kWh/mmHG)

Plant-Specific Value: The effects on heat rate of a deviation in condenser backpressure can be obtained from the unit's thermal kit.

B.10.2 Possible Causes of Deviation

- Air leakage
- Excess condenser load
- Tube fouling
- Low circulating water flow
- Tube plugging
- Air binding (Design deficiency, not inleakage caused)
- Water box not full

- Increases in circulating water inlet temperature caused by:
  - Changes in ambient conditions
  - Problems with cooling tower performance

B.10.3 Possible Corrections

Operator Controllable:

- Increase circulating water flow
- Add an additional vacuum pump

- Check cycle isolation
- Place additional circulating water pumps in service
- Place additional cooling tower cells in service
- Vent water boxes

Maintenance Correctable:

- Repair condenser air leaks
- Repair cycle isolation valves
- Clean condenser
- Repair circulating water discharge control valve
- Repair cooling tower

B.11 Unburned Carbon

B.11.1 Effects on Heat Rate

Utility Average: 11.73 Btu/kWh/% (12.4 kJ/kWh/%)

Utility Range: 6-12.8 Btu/kWh/% (6.33 - 13.5 kJ/kWh/%)

Plant-Specific Value: The effect on heat rate due to unburned carbon in the ash refuse can be calculated by the following equations:

$$\begin{matrix} Heat\ Loss\ Due\ to \\ Unburned\ Carbon \end{matrix} (UBHL) = (UBE - UBA) * ASH * \frac{145}{HHV} \qquad \text{(eq. B-7)}$$

$$BE(COR) = BE(EXP) + (UBHL) \qquad \text{(eq. B-8)}$$

$$HR\ (DEV) = NUHR * [\ \frac{BE(EXP)}{BE(COR)}\ ] - NUHR \qquad \text{(eq. B-9)}$$

where

UBE	=	Expected level of unburned carbon expressed as percent by weight of carbon in ash
UBA	=	Measured level of unburned carbon
ASH	=	Percent ash content of coal from proximate analysis
HHV	=	Higher heating value of the as-fired fuel
BE (EXP)	=	Expected value of boiler efficiency
BE (COR)	=	Corrected value of boiler efficiency
NUHR	=	Net unit heat rate.

B.11.2Possible Causes of Deviation

- Incorrect fuel/air ratio change in mill fineness change in mill air flow

B.11.3Possible Corrections Operator Controllable:

- Adjust fuel/air ratio
- Adjust mill secondary air flow damper settings Maintenance Correctable:
- Adjust classifier vane settings
- Repair or replace grinding wheels, balls, or rings
  - Check secondary air heater for blockage
  - Calibrate fuel/air control

B.12 Coal Moisture

B.12.1Effect on Heat Rate

Utility Average:	7.8	Btu/kWh/% (8.2 kJ/kWh/%)
Utility Range:	6-10	Btu/kWh/% (6.3 - 10.6 kJ/kWh/%)

Plant-Specific values: The effect on heat rate of changes from design values of the coal moisture can be calculated from the following equations:

$$Heat\ Loss\ Due\ to\ Coal\ Moisture\ \ MHL = \frac{(ME - MA) * (H1 - H2)}{HHV} \quad \text{(eq. B-10)}$$

where

HR (DEV) = NUHR * [BE (EXP)/BE (COR)] - NUHR

BE(COR) = BE(EXP) + MHL

BE(COR) = Corrected boiler efficiency

BE(EXP) = Expected boiler efficiency

ME = Expected moisture content of coal

MA = Measured moisture content of coal (108.2 kPa)

H_1 = Enthalpy of vapor at 1 psia and corrected exit gas temperature TG (NL) (see B.9)

H_2 = Enthalpy of liquid at air heater inlet temperature TA (see B.9).

The remainder of the terms are the same as defined in B.11.

B.12.2 Possible Causes of Deviation

- Change in coal quality

B.12.3 Possible Corrections Operator Controllable: None Maintenance Correctable:

- Proper mill air temperature

B.13 Auxiliary Power

B.13.1 Effects on Heat Rate

Utility Average: 86 Btu/kWh/% (90.7 kJ/kW/%)

Utility Range: 64-97 Btu/kWh/% (68.0 - 102 kJ/kW/%)

Plant-Specific Value: The effect on heat rate of a deviation from power usage level can be obtained by using the expected auxiliary following equations:

$$HR(DEV) = NUHR * \frac{[100 - AP(EXP)]}{[100 - AP(ACT)]} - NUHR \quad \text{(eq. B-11)}$$

where

AP (EXP) = Expected power usage %

AP (ACT) = Measured power usage %

B.13.2 Possible Causes of Deviation

- Continuous running of noncontinuous loads
- Decline in efficiency of operating equipment
- Operation of redundant equipment during low-load operation

B.13.3 Possible Corrections

Operator Controllable:

- Stop noncontinuous loads
- Reduce equipment operation at low loads
- Maintenance Correctable:
- Repair or replace inefficient equipment
- Maintain equipment whose power usage increases with deteriorating performance, e.g., electrostatic precipitators, pulverizers, etc.

B.14 MAKEUP

B.14.1 Effect on Heat Rate

Utility Average: 24 Btu/kwh/% (25.3 kJ/kWh/%)

Utility Range: 4-88 Btu/kWh/% (4.2 - 92.8 kJ/kWhr/%)

Plant-Specific Value: The effect on heat rate of changes from the expected amount of makeup can be calculated from the following equations:

MU (DEV) = MU (ACT) - MU (EXP)

H (DEV) = $X (H_1 - H_2) + (1-X) * (H_3 - H_2)$

TOP TERM = [MU (DEV) * THRTFLOW * H (DEV)]/(GROSS * 100)

BOT TERM = [BE (EXP)]/1OO * [1 - AP (ACT)]

HR (DEV) = TOP TERM/BOT TERM

where

MU (ACT) = Actual makeup flow, %

MU (EXP) = Expected makeup flow, %

H_1 = Throttle enthalpy, Btu/lbm (kJ/kg)

H_2 = Makeup water enthalpy, Btu/lbm (kJ/kg)

H_3 = Feedwater enthalpy, Btu/lbm (kJ/kg)

THRTFLOW = Throttle flow, 1 bm/hr (kg/hr)

GROSS = Gross generation, kW

BE (EXP) = Expected boiler efficiency

AP (ACT) = Actual auxiliary power usage %

X = Estimated fraction of losses occurring after the boiler inlet.

B.14.2 Possible Cause of Deviation

- Boiler tube leaks
- Excess deaerator venting to atmosphere
- Excess continuous blowdown
- Excess steam lost through condenser venting
- Valve packing leaks
- Pump seal leaks
- Steam leaks to atmosphere

B.14.3 Possible Corrections Operator Controllable: None Maintenance Correctable:

- Check deaerator vent orifices or valve settings
- Repair valve and pump packings and seals
- Repair boiler tube leaks
- Optimize continuous blowdown
- Isolate cycle losses

B.15 Feedwater Heater Performance

B.15.1 Effect on Heat Rate

Top High Pressure Heater Performance:

Utility Average: 2.1 Btu/kWhr/°F (TTD) 4.0 (kJ/kWhr/°C)

Utility Range: 1.3-2.3 Btu/kWh/°F (TTD) (2.5 - 4.4 kJ/kWhr/°C)

Next to Top High Pressure Heater Performance:

Utility Average: .54 Btu/kwh/°F (TTD) (1.0 kJ/kWhr/°C)

Utility Range: .33-.63 Btu/kWh/°F (TTD) (.63 - 1.2 kJ/kWhr/°C)

Third from Top High Pressure Heater Performance:

Utility Average:	.65 Btu/kWh/°F (TTD) (1.2 kJ/kWhr/°C)
Utility Range:	.57-.8 Btu/kWh/^{0}F (TTD) (1.1 - 1.5 kJ/kWhr/°C)

High Pressure Heater Out of Service:

Top Heater:	94	Btu/kWh (99 kJ/kWh)
Second Heater:	70	Btu/kWh (74 kJ/kWh)
Third Heater:	70	Btu/kWh (74 kJ/kWh)

Plant-Specific Value: The effect on heat rate of changes in feedwater heater TTD's and DCS's should be calculated using a heat balance code for accurate results. Terminal temperature differences have a more significant effect on heat rate than do drain cooler approach temperatures.

B.15.2 Possible Causes of Deviation

- Changes in heater level
- Changes in extraction line pressure drop
- Reduced condensate flow through the heater
- Heater baffle leaks
- Failure to vent noncondensable gases
- Tube fouling
- Too large a vent flow drawing excess steam

B.15.3 Possible Corrections Operator Controllable:

- Set feedwater heater levels Maintenance Correctable:
- Optimize feedwater heater level
- Maintain heater vent valves and line orifices

- Repair baffle leaks
- Clean tube bundles

B.16 Reduced Load Operation

B.16.1 Effect on Heat Rate

Utility Average: None available

Utility Range: None available

Plant-Specific Value: The effect on unit heat rate of unit load reduction can be calculated from the following equation:

$$HR\ (DEV) = Xi * HRi - NUHR \qquad \text{(eq. B-12)}$$

Xi = Time at power level i divided by time at power during the time period

where

Xi = Fraction of time spent at power level i

HRi = Heat rate at power level i

NUHR = Net unit heat rate.

B.17 Start-Up

B.17.1 Effect on Heat Rate

Utility Average: 7.33 Btu/kWh/start-up (7.73 kJ/kWh/startup)

Utility Range: .5-19 Btu/kWh/start-up (.5 - 20kJ/kW/startup

Plant-Specific Value: The effect on heat rate of unit start-up can be determined by the following equation:

$$HR\ (DEV) = [\frac{((SU * HOURS * SUHR) + NUHR * (720 - SU * HOURS))}{720}] - NUHR \qquad \text{(eq. B-13)}$$

where

SU	-	Number of start-ups per month
HOURS	-	Hours per start-up
SUHR	-	Average heat rate during start-up period
NUHR	-	Net unit heat rate.

B.17.2 Possible Causes of Deviation

- Forced outages
- Unscheduled outages

B.17.3 Possible Corrections Operator Controllable: None Maintenance Correctable:

- Eliminate unscheduled outages through effective predictive and preventative maintenance

C

PERFORMANCE PARAMETER DIAGNOSTICS

Table C-1
Boiler Losses
Performance Parameter Diagnostics

Performance Parameter	Causes for Degradation	Further Testing for Confirmation	Corrective Action
Throttle Pressure (Too high)	• Instrument error	• Compare other related pressures (superheater outlet, drum) with throttle pressure.	• Calibrate instrumentation
	• Not adhering to setpoints	• Verify that turbine control valves are not fully open. If the valves are fully open and the pressure is still the HP turbine bowl flow coefficient I too low.	• Check throttle pressure set point curve in computer; check control system.
	• Bowl Flow coefficient is high	• Check bowl flow coefficient (a turbine performance parameter) Note, that bowl flow coefficient will only affect throttle pressure if the turbine control valves are fully open.	• Follow instructions for turbine bowl flow coefficient.
Throttle Temperature (too low)	• Instrument error	• Compare throttle temperature with superheater outlet temperature. The throttle temperature should be zero to 5^{0}F lower	• Calibrate instrumentation
	• High throttle pressure	• Throttle pressure is a separate performance parameter.	
	• High superheater spray flow	• Check spray flow. This is a separate performance parameter.	

Performance Parameter	Causes for Degradation	Further Testing for Confirmation	Corrective Action
	• Superheater surface dirty	• Perform test of boiler surface cleanliness factor.	• Blow soot • Clean surface
	• Low excess air	• Check excess air, this is a separate performance parameter.	
	• Setpoint is too low	• Check setpoint	• Raise setpoint
Superheater Spray Flow (too high)	• Instrument error	• Calculate flow from steam temperatures entering and leaving desuperheater.	
	• Steam temperature setpoint is too low.	• Check steam temperature setpoint	• Raise temperature setpoint.
	• Spray flow is in manual, setpoint is too high	• Check controls	• Adjust controls
	• Excess air is too high	• Check excess air. This is a separate performance parameter	
	• Feedwater temperature is low due to:		
	• a) HP heater bypass	• Check temperatures before and after bypass mix.	• Close bypass valve. Repair if leaking
	• b) HP Heater TD is high	• Check heater TD. This is a separate performance parameter	
	• c) IP turbine extraction flow coefficient is high.	• Check flow coefficient. This is a separate performance parameter	
	• Throttle pressure is too	• Check throttle pressure. This is a	

Performance Parameter	Causes for Degradation	Further Testing for Confirmation	Corrective Action
	low.	separate performance parameter	
	• Spray Control valve is leaking	• Close block valve • Check whether line is hot. A hot line indicates leakage.	• Repair valve
Reheater Spray Flow (too high)	• Instrument error	• Calculate flow from steam temperatures entering and leaving desuperheater.	
	• Steam temperature setpoint is too low.	• Check steam temperature setpoint.	• Raise temperature setpoint
	• Spray flow is in manual, setpoint is too high	• Check controls	• Adjust controls
	• Excess air is too high	• Check excess air. This is a separate performance parameter	
	• Feedwater temperature is low due to:		
	• a) HP heater bypass	• Check temperatures before and after bypass mix	• Close bypass valve. Repair if leaking
	• b) HP Heater TD is high	• Check heater TD. This is a separate performance parameter.	
	• c) IP turbine extraction flow coefficient is high.	• Check flow coefficient. This a separate performance parameter.	
	• Throttle pressure is too low.	• Check throttle pressure. This is a separate performance parameter.	

Performance Parameter	Causes for Degradation	Further Testing for Confirmation	Corrective Action
	• Spray control valve is leaking.	• Close block valve. • Check whether line is hot. A hot line indicates leakage.	• Repair valve
	• Throttle temperature is too high	• Check throttle temperature. It is a separate performance parameter..	
	• HP turbine efficiency is too low.	• Check HP turbine efficiency. It is a separate performance parameter.	
Reheat Temperature (too low)	• Instrument error	• Compare reheat temperature with reheater outlet temperature. The reheat temperature should be zero to 5^0 F lower.	• Calibrate instrumentation
	• Excessive reheater spray flow	• Check spray flow. This is a separate performance parameter.	
	• Setpoint is too low	• Check setpoint	• Raise setpoint
	• Reheater surface dirty.	• Perform test of boiler surface cleanliness factor.	• Blow soot • Clean surface
	• High throttle pressure	• Throttle pressure is a separate performance parameter.	
	• Low excess air	• Check readings of multiple 0_2 analyzers relative to each other. • Check with manual orsat or analyzer.	• Calibrate or repair instrumentation.
	• Setpoint is too high.	• Check setpoint.	• Adjust setpoint.
	• Poor atomization	• Observe flame patterns	

Performance Parameter	Causes for Degradation	Further Testing for Confirmation	Corrective Action
	• Low fuel oil temperature	• Check fuel oil temperature vs. specs for viscosity of fuel being burned.	
	• Poor distribution of air among individual burners	• Reduce excess air while observing flame. Identify flames that begin smoking first.	• Reset burner dampers.
	• Leaking air through burners out of service.	• Check shutoff dampers.	• Repair dampers.
Reheater Pressure Drop (Steam side) (Too High)	• Instrument error	• Install a differential pressure gage across the reheater.	• Calibration
	• Increased pipe resistance due to internal scale deposits.		• Chemically clean reheater tubes.
Auxiliary Power (Too high)	• Instrument error		• Calibration
	• Poor fan efficiency	• Test fan. A first test involves measuring flow and head, and comparing these with the fan curve. A second test involves the same, but also checking the efficiency by measuring motor power.	
		• Inspect fan for excessive clearances, damage, lack of cleanliness, etc.	
	• High draft resistance	• Measure boiler draft resistance (fan discharge pressure). Compare with value in boiler specification after adjusting for flow.	

Performance Parameter	Causes for Degradation	Further Testing for Confirmation	Corrective Action
	• High excess air	• Check excess air. It is a separate performance parameter.	
Carbon Monoxide (Too high)	• Instrument error	• Check CO analyzer against portable analyzer, at several points in flue gas duct.	• Calibrate repair or reposition analyzer.
	• Low excess air	• Check excess air.	• Increase excess air.
	• Poor distribution of air	• Reduce excess air while watching flame. Note which flames begin smoking first.	• Reset burner dampers.
	• Poor atomization	• Observe flame pattern.	
Unburned Carbon (Too high)	• Incomplete combustion	• See above.	• See above.
Fuel oil temperature (Too Low)	• Insufficient steam flow to fuel oil heater.	• Check position of temperature control valve.	
	• Fouling of fuel oil	• Check steam drain temperature	• Inspect fuel oil heater.
Economizer Approach (Too high)	• Instrument error	• Check feedwater temperature by checking outlet temperature of HP heater with economizer inlet temperature.	• Calibrate Instrumentation
	• Fouled economizer surface	• Calculate heat transfer coefficient of economizer, and compare with design value check for heat balance around economizer.	• Blow soot • Clean boiler
	• High economizer entrance gas	• Check for heat balance around economizer. Check entrance gas	• Blow soot • Clean boiler.

Performance Parameter	Causes for Degradation	Further Testing for Confirmation	Corrective Action
	temperature	temperature with design.	
Air Heater Leakage (Too High)	• Instrument error	• Use redundant instrumentation	• Calibrate instrumentation • Find and eliminate any leaks in ORSAT instrument tubing
		• Check for stratified flow by traversing duct..	• Take enough sample points to get representative results
Air Heater Air Side Effectiveness (Too low)	• Instrument error	• Check air heater pressure drop resistance.	• Clean air heater or replace baskets
	• Corroded air heater	• Check air heater leakage, a separate parameter. • Inspect air heater.	• Replace baskets
	• Baskets are being short circuited.	• Check air heater pressure drop resistance. It may be low if some baskets are short circuited.	• Repair.

Table C-2
Turbine Losses
Performance Parameter Diagnostics

Performance Parameter	Causes for Degradation	Further Testing for Confirmation	Corrective Action
HP Turbine Bowl Flow coefficient (Too Low)	• Instrument error	• Compare throttle flow calculated from fuel oil flow with throttle flow calculated from feedwater, measurements. • Check bowl pressure with valve wide open. It should be approximately 4 percent below throttle pressure.	• Calibrate instrumentation.
	• Deposits on first stage nozzles.	• Deposits will reduce the flow coefficient	• Inspect and overhaul turbine. • Correct water chemistry
	• Erosion of first stage nozzles.	• Correct water chemistry.	• Inspect and overhaul turbine.
HP Turbine Blading Efficiency (Too Low)	• Instrument error	• Check redundant measurements at main steam and cold reheat. • Check bowl pressure with wide open valves. It should be approximately 4 percent below	• Calibrate instrumentation.

Performance Parameter	Causes for Degradation	Further Testing for Confirmation	Corrective Action
		throttle pressure.	
	• Deposits	• Check HP Turbine bowl flow coefficient. Deposits on first stage nozzles will reduce this parameter. Also check first stage flow coefficient; deposits on the second stage nozzles will reduce this parameter.	• Inspect and overhaul HP turbine. • Correct water chemistry. • Examine trend to determine time and cause. • Inspect turbine.
	• Erosion	• Same as above, except that erosion will increase the flow coefficient.	• Correct water chemistry.
	• Internal leakage through casing joints.	• None available	• Inspect and overhaul HP turbine.
	• Rubbed shaft seals or spill strips	• Check N_2 packing leakage flow coefficient. An increase N2 flow coefficient could be coincident with other increase leakages.	• Inspect and overhaul HP turbine. • Examine data trends to determine time and cause of degradation.
	• Internal damage	• Examine records damage usually occurs suddenly, often coincident with high vibration.	• Inspect and overhaul turbine. • Inspect and overhaul HP turbine.

Performance Parameter	Causes for Degradation	Further Testing for Confirmation	Corrective Action
IP Turbine Bowl Flow Coefficient	• Instrument error	• Compare reheat flow calculated from fuel oil flow with reheat flow calculated from feedwater, measurements. • Check reheater pressure drop. Design value is 10 percent.	• Calibrate instrumentation.
(Too Low)	• Deposits	• Deposits will reduce the flow coefficient.	• Inspect and overhaul turbine. • Correct water chemistry.
(Too High)	• Erosion	• Erosion will increase the flow coefficient	• Inspect and overhaul turbine. • Correct water chemistry.
IP Turbine Blading Efficiency	• Instrument error	• Check redundant measurements at hot reheat. Feedpump turbine inlet pressure and temperature should be close to crossover values.	• Calibrate instrumentation.
	• Deposits	• Check IP turbine inlet, IP turbine first extraction, coefficients These are all separate performance parameters. Deposits will decrease their value.	• Inspect and overhaul turbine. • Correct water chemistry.

Performance Parameter	Causes for Degradation	Further Testing for Confirmation	Corrective Action
	• Erosion	• Same as above, except that erosion will have the opposite effects.	• Inspect and overhaul turbine. • Correct water chemistry.
	• Internal leakage through casing joints.	• None available	• Inspect and overhaul turbine.
	• Internal damage	• Examine records. Damage usually occurs suddenly, often coincident with high vibration.	• Inspect and overhaul turbine.
	• Rubbed seals	• Check N_2 packing leakage flow coefficient, and IP extraction stage tip leakage flow. These are separate performance parameters that will increase due to rubbed seals.	• Inspect and overhaul turbine. • Identify operating condition that resulted in rubs and alter procedures.
		• Examine efficiency trends: Loss of efficiency due to rubbed seals usually occurs suddenly, particularly during turbine trips, startup, and other transients.	• Identify operating condition that resulted in rubs, and alter procedures.
(Too High)	• N2 packing leakage flow coefficient.	• This is a separate performance parameter. An increase in its value, due to rubs, will result in a perception of improved IP	• Inspect and overhaul HP turbine. • Examine data trends to determine time and cause

Performance Parameter	Causes for Degradation	Further Testing for Confirmation	Corrective Action
		efficiency.	of degradation.
IP Turbine First Extraction Flow Coefficient	• Instrument error	• Compare extraction pressure at turbine with pressure at heater. The two should be within 5 percent. If greater than 5 percent, there could be a flow restriction such as a jammed non-return value in the pipe.	• Calibrate instrumentation. • Correct flow.
		• Check flow coefficients for other stages (HP 1st stage, IP inlet, LP inlet) and other flowmeters (main steam, feedwater, and condensate). If all these are in error by a similar amount, the calculation of steam cycle flows based on fuel oil flow is suspect.	• Confirm measurement of fuel heating value.
	• Deposits or Erosion	• Erosion increased the flow co-efficient, deposits decrease it.	• Inspect and overhaul turbine.
		• Check IP turbine blading efficiency, which is a separate performance parameter.	
LP Turbine efficiency ratio (Too Low)	• Isolation n - Leaking valves anywhere in the cycle which cause the degradation of	• Confirm tight closure of all valves on isolation list. Test leaking valves by comparing upstream and downstream temperatures.	• Repair leaking valves.

Performance Parameter	Causes for Degradation	Further Testing for Confirmation	Corrective Action
	energy will be perceived as a loss in LP turbine efficiency.		
		• If the efficiency ration worsens as load decreases, the leakage low is constant, and probably from the main steam line.	
	• Instrument Error • The most important measurements are: • a) Fuel flow • b) heating value • c) generator output.	• Compare calculated flows, feedwater, etc. with measured turbine flow coefficients with design values. • A discrepancy indicates an error in fuel flow or heating valve.	• Calibrate fuel flow meter • Check heating values.
		• Compare redundant generator-output meters.	• Calibrate watt-hour meter.
	• Erosion due to moisture.	• Visually inspect last and next to last stage buckets during shutdown through manhole.	• Replace or repair buckets.
	• Damage	• Check efficiency trends. Efficiency loss due to damage will usually be sudden.	• Inspect and overhaul turbine.

Performance Parameter	Causes for Degradation	Further Testing for Confirmation	Corrective Action
	• Rubbed Seals	• Check first IP and extraction tip leakage flow, separate performance parameters. This flow will usually experience an increase when rubs occur.	• Inspect and overhaul turbine. • Determine operating condition that caused rubs and change procedures to prevent reoccurrence.
	• Deposits	• Check first and second LP extraction flow coefficients, separate performance parameters. They will decrease due to deposits.	• Inspect and overhaul turbine. • Correct water chemistry.
	• Erosion	• Same as above, except that flow coefficients increase due to erosion.	• Inspect and overhaul turbine.
Extraction Temperatures (Too high)	• Instrument error	• Compare extraction temperature at heater with extraction temperature at turbine. They should be virtually the same. Also check temperature at inlet and outlet of feedwater heater. With temperatures at adjacent heaters.	• Calibrate
	• Rubbed spill strips	• Inspect turbine	• Repair
	• Turbine casing joint leakages	• Inspect turbine	• Repair

Performance Parameter	Causes for Degradation	Further Testing for Confirmation	Corrective Action
N2 Packing Leakage Flow Coefficient	• Instrument error	• Determination of N_2 packing leakage flow involves delicate testing with some very exacting instrument accuracy requirements. Read procedure developed for this test.	• Modify instrument or procedure
	• Worn seals	• Inspect turbine	• Repair
Other Packing Leakage Flow Coefficients (Too High)	• Instrument error		• Inspect and calibrate
	• Worn seals	• Inspect turbine.	• Repair

Table C-3
Condenser Losses
Performance Parameter Diagnostics

Performance Parameter	Causes for Degradation	Further Testing for Confirmation	Corrective Action
Condenser Cleanliness (Too Low)	• Instrument error	• Calibration • Compare hotwell temp with back pressure • Verify circ water outlet temp by measuring flow with pump curve	• Replace instrument
	• Macrofouling	• Inspect inlet tube sheet • Compare actual and expected tube bundle pressure drop. Macrofouling increases pressure drop significantly	• Backflush • Clean inlet tube sheet
	• Microfouling	• Inspect tubes • Compare actual and expected pressure drop microfouling does not significantly increase pressure drop	• Clean tubes • Chlorinate • Thermal shock
	• Excess air inleakage	• Measure air removal pump discharge flow to determine air in-leakage. Compare to past records.	• Start up second ejector • Repair leak
	• Inefficient air removal	• Measure air removal pump shutoff head. Compare to past records.	• Start up second pump • Repair leak
Condenser Cooling Water Flow	• Macrofouling	• Inspect inlet tube sheet	• Backflush • Clean inlet tube sheet

Performance Parameter	Causes for Degradation	Further Testing for Confirmation	Corrective Action
(Too Low)			
		• Compare actual and expected tube bundle pressure drop. Macrofouling does not significantly increase pressure drop	
	• Microfouling	• Inspect tubes • Compare actual and expected pressure drop microfouling does not significantly increase pressure drop	• Clean tubes • Chlorinate
	• Pump Wear • Insufficient water flow	• Measure motor current, compare with design	• Inspect and rebuild • Place additional circ. water pumps in service.
Condenser Cooling Water Side Pressure Drop (Too High)	• Macrofouling	• Inspect inlet tube sheet • Compare actual and expected tube bundle pressure drop. Macrofouling increases pressure drop significantly	• Backflush • Clean inlet tube sheet
	• Microfouling	• Inspect tubes • Compare actual and expected pressure drop microfouling does not significantly increase pressure drop	• Clean tubes • Chlorinate • Thermal shock
Air Ejector Discharge Flow (See Note) (Too Low)	• Instrument Error	• Use a different ejector with a different discharge flow meter.	• Repair flow meter.
	• Transient response of condenser	• If during startup, wait for unit to come to full load. If at load, wait 15 minutes and repeat measurement.	

Performance Parameter	Causes for Degradation	Further Testing for Confirmation	Corrective Action
	• Insufficient steam flow and/or pressure	• Check supply steam pressure.	
	• Excessive Air In-leakage	• Locate leaks. Determine if leakage decreases as load is decreased. If this is the case, leakage may be in low pressure extraction lines, whose pressure becomes sub-atmospheric at low loads.	• Locate leaks and repair equipment as required.
Air Ejector Discharge Flow (See Note) (Too High) Note: The air ejector discharge flow is equal to the air in-leakage flow. It is not an indicator of ejector performance.			

Table C-4
Auxiliary
Performance Parameter Diagnostics

Performance Parameter	Causes for Degradation	Further Testing for Confirmation	Corrective Action
Heater Terminal Differences (Too High)	• Instrument error	• Check steam pressure at heater with pressure at turbine. The turbine pressure should be about 3 percent higher. Check the outlet temperature from heater with the inlet to the downstream heater.	• Calibrate
	• Fouled tubes	• Feedwater pressure drop will be higher than expected.	• Clean heater
	• Leaking channel partition plate	• Feedwater pressure drop will be higher than expected.	• Repair
	• Poor venting of non condensable	• Vent line is cold.	• Repair
Heater Drain Cooler Approaches (Too High)	• Instrument error	• Check that feedwater temperature entering heater is equal to that leaving previous heater.	• Calibrate
	• Fouled surface	• Fouling tends to cause relatively small increases in DC. Changing heater level does not materially affect the DC.	• Clean heater
	• Low heater level	• Raising heater level improves the DC.	• Raise level

Performance Parameter	Causes for Degradation	Further Testing for Confirmation	Corrective Action
	• Holes in drain cooler shroud	• Inspect heater	• Repair heater
Pump Volumetric Efficiency (Too Low)	• Instrument error	• Usually it will be flow that is in question. Check validity of the flow measurement used by comparing with other available flow measurements (turbine stage pressures, fuel flow, etc.)	• Calibrate instrument
	• Erosion and wear of pump impeller	• Check records for gradual deterioration in performance	• Inspect pump
	• Open recirculation line	• Walkdown system and verify that manual recirculation valves are shut, and automatic valves are seated	• Shut valves and take maintenance action where needed
	• Internal damage	• Examine records for sudden change in performance, usually accompanied by the onset by high vibration	• Inspect and overhaul pump
Full Load Component Cooling System Temperature Rise (Too High)	• Instrument error	• If no redundant instruments available, pull existing thermocouples and check with test instrument in well	• Calibrate instruments
	• Flow of cooling (circ.) water too low	• None	• When the equipment is available, inspect waterbox and inlet piping for blockage or macrofouling

Performance Parameter	Causes for Degradation	Further Testing for Confirmation	Corrective Action
	• Duty too high	• Determine each component which is cooled by this system, and verify that bearing temperatures and vibration levels are not excessive	• Inspect and repair if bearing problems found
	• Tubes are fouled	• Check history of microfouling on condenser, relate this to level of cleaning (e.g., chlorination) on condenser compared to component cooling heat exchanger	• Inspect tubes for microfouling, clean tubes
Feed Pump and Driver Effectiveness Ratio (Too Low)	• Instrument error	• Check other related pressures and temperatures (crossover)	• Calibrate instruments
	• Deposits on pump turbine	• Check expected extraction pressure. Deposits will cause an increase in this parameter	• Inspect turbine • Correct water chemistry
	• Erosion of pump turbine	• Same as above, but opposite effect	• Inspect turbine • Correct water chemistry
	• Internal damage	• Examine records and trends of performance parameters, since damage is usually caused by a sudden event, often coincident with the onset of high vibration	• Inspect and overall turbine

Table C-5
Electrical Losses
Performance Parameter Diagnostics

Performance Parameter	Causes for Degradation	Further Testing for Confirmation	Corrective Action
Generator Hydrogen Pressure (Too Low)	• Instrument Error	• Compare with other available measurements elsewhere on system.	• Calibrate instrument.
	• Hydrogen supply inadequate.	• Check pressure in hydrogen supply tanks. If pressure is sufficient, verify that all valving between hydrogen supply and generator is aligned correctly.	• Correct valving, replace hydrogen supply as needed.
	• Leak in system.	• Check frequency of replacement of hydrogen supply to system; walk down hydrogen system with portable gas detection equipment.	• Locate and repair leak., depending upon unit and system status (may require system purge to repair).