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Heat Rate Improvement Guidelines for Indian Power Plants

Volume I

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PREFACE

This report is both an ending, and a beginning. It is the culmination of a several year long partnership between the United States Tennessee Valley Authority (TVA) and the Government of India's National Thermal Power Corporation (NTPC). This partnership was formed to transfer proven and economically feasible technology on improving thermal performance (heat rate) of coal fired power plants, to the Indian power sector. These improvements reduce the amount of coal burned per unit of generation, thereby resulting in reduced emissions to the environment, including carbon dioxide (CO_2).

This report is also a beginning, as its purpose is to document and disseminate to all utilities in India the knowledge that has been demonstrated and proven during this partnership. TVA and NTPC have worked together to transfer procedures, software, skills and technology that are used in the United States to maintain and improve the heat rate of plants there, and to demonstrate their validity in India. This report documents this information. It includes procedures on monitoring, testing, documenting losses, training, typical areas where improvements can usually be made with minimal cost, recommendations for new plant specifications, and how to most efficiently dispatch multiple units.

The origins of this project go back to 1992, and the United Nation's Framework Convention on Climate Changes (FCCC) that was held in Rio-de-Janeiro. As a result of this conference, a funding mechanism for environmental protection called Global Environmental Facility (GEF) was established. The Greenhouse Gas Pollution Prevention Project (GEP) represents the United States commitment to GEF. The purpose of the project is to reduce CO_2 emissions per kWh of electricity produced by improved efficiency of coal fired thermal plants. In 1994, a protocol of intent was signed between the United States and the Government of India. The GEP project has two components, the Alternative Bagasse Cogeneration (ABC) component and the Efficient Coal Conversion (ECC) component.

The National Thermal Power Corporation of India (NTPC) was selected as the implementing agency for the ECC component of the GEP project. NTPC created the Centre for Power Efficiency and Environmental Protection (CenPEEP), to address efficiency and environmental improvements at power stations of NTPC and various State Electricity Boards (SEBs). The United States Agency for International Development (USAID) entered into an agreement with the US Department of Energy (DOE) National Energy Technology Laboratory (NETL) to implement the ECC component with NTPC. In October 1995, the Tennessee Valley Authority, the largest utility in the United States, whose heat rates are historically among the best in the US, began work on this task, under the direction of DOE and USAID.

Some of the major activities that have been carried out during this multiyear program include:

- Performance testing (resulting in recommendations for improvements) at NTPC's Dadri, Singrauli, and Rihand plants; Gujarat SEB's Wanakbori plant and Maharashtra SEB's Koradi plant including :
    - Pulverizer
    - Burner Balancing
    - Boiler Cycle Optimization
    - Feedwater Heaters
    - HP / IP Turbines
    - Condensers
    - Pumps
    - Turbine Cycle
- Transfer of and training on, high accuracy performance test equipment
- Training at US power plants and corporate offices for 11 NTPC personnel
- Walkdowns and recommendations for heat rate improvements at several plants.
- Demonstration of new condenser tube cleaning techniques
- Demonstration of new techniques for locating sources of condenser air in-leakage
- Training on thermal modeling software
- Training seminars and workshops on subjects such as pulverizer optimization, condenser performance, and variable pressure operation

The culmination of these activities, and additional activities carried out independently by CenPEEP, has made a significant impact on the thermal efficiency and CO_2 emissions in Indian power plants.

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EXECUTIVE SUMMARY

This guideline documents the various components of a heat rate improvement program. The activities described here apply to any power plant, from small units (<100MW) to large plants. They also apply to both newer units and to older units. The components of the heat rate improvement program listed here have been successfully implemented in the United States and have now been demonstrated in the Indian power sector.

Key Points:

- Heat rate improvements are possible at any plant, and at many plants the improvement can easily be in the range of 1-2 percentage points.
- A heat rate improvement program can start small, concentrating on one or two areas, (i.e., condenser performance and/or burner balancing) or by concentrating on one or two activities (i.e., monitoring primary process indicators, testing, etc.)
- Heat rate improvement must have the full support of the plant management.
- Improvements do not "just happen", there must be a concerted effort made just to maintain current status, and even more effort expended to make improvements. Each plant should have a detailed plan listing specific activities that are to be carried out to make improvements, and the plan must be reviewed periodically to ensure that the plan is being carried out.
- Heat rate improvement is a continuing process. It must become part of the normal work activities, it must be considered along with, and at par with, reliability, safety, environmental concern, etc., when operating the unit, scheduling maintenance, and all other routine activities.
- Heat rate improvement is the responsibility of everyone. It cannot be assigned to one individual or organization, but requires the active involvement of the entire plant staff.

Typical Savings from Heat Rate Improvements (Rupees / year)

Plant Location	Pit Head Station		Rail Fed Station	
Unit Size (MW)	210	500	210	500
1 % Overall Heat Rate Improvement (from 2500 kcal/kWh)	5,518,800	13,140,000	14,716,800	35,040,000
1% Air Heater Inlet Oxygen	1,635,772	3,894,696	4,362,060	10,385,856
1 mm Hg Condenser Pressure	448,127	1,066,968	1,195,004	2,845,248
1 °C Air Heater Exit Gas Temperature	264,902	630,720	706,406	1,681,920
1 °C Main Steam Temperature	141,281	336,384	376,750	897,024
1 °C HRH at Intercept Valve Temperature	130,244	310,104	347,316	826,944
1000 kg/hr Reheat Attemporation Flow	54,305	129,298	144,813	344,794
Assumptions :				
PLF (%)	70	70	70	70
Fuel HHV (kcal/kg)	3500	3500	3500	3500
Fuel Cost (Rs/ton)	600	600	1600	1600

From the preceding analysis it is evident that even a modest change in the several unit operating parameters can result in substantial savings in fuel cost. Based on the collaborative efforts that were carried out in India on several units, by CenPEEP and TVA it is firmly believed that it is possible to improve the level of performance even in the newest and best performing plants, and that the financial rewards are well worth the time and resources expended in improving heat rate.

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SECTION 1 INTRODUCTION

The heat rate of a conventional fossil-fueled power plant is a measure of how efficiently it converts the chemical energy contained in the fuel into electrical energy. This conversion is accomplished in four major steps. First, the chemical energy in the fuel is converted into thermal energy, then the thermal energy is converted into kinetic energy, then the kinetic energy is converted in mechanical energy, and last the mechanical energy is converted to electrical energy. In each of these sub-processes, some energy is lost to the environment. Some of the fuel is not burned completely, some of the thermal energy is lost out the stack and rejected to the cooling water, some of the kinetic and mechanical energy produces heat instead of electricity, and last, some of the electricity that is produced is used by these sub-processes. The heat rate of a power plant is the amount of chemical energy that must be supplied to produce one unit of electrical energy. Put another way, it is the required input divided by the desired output, or the reciprocal of the efficiency. However, since chemical energy is usually measured in kilocalories (kcal) (or sometimes kilojoules, kJ) and electrical energy is usually measured in kilowatt-hours (kWh), the units of heat rate are normally kcal/kWh (or kJ/kWh).

If a power plant converted 100% of the chemical energy in the fuel into electricity, the plant would have a heat rate of 860 kcal/kWh. Unfortunately, due to the losses described above, a modern conventional power plant might have at best a design full load heat rate of 2200 kcal/kWh, which is about 39% efficient.

Many factors affect the actual, operating heat rate of a plant:

- The initial design. Some units are designed to be more efficient than others. Some units have more stages of feedwater heaters, resulting in better efficiency; or some units have steam driven pumps and fans instead of electric drives; some units operate at higher initial temperature and/or pressure, which results in a lower heat rate.
- Ambient conditions. A plant is more efficient when the condenser circulating water temperature is cooler.
- Load Factor. Conventional power plants are most efficient at full load, and the efficiency decreases as the load decreases.
- The fuel that is supplied. A plant may have been designed to burn a low moisture bituminous coal, but it is receiving a high moisture sub- bituminous coal. This will adversely affect the heat rate.
- How well the plant is operated and maintained. Power plants can be operated efficiently or inefficiently. Equipment deterioration may not be corrected speedily or completely, valves that should normally be shut may be left open, unnecessary auxiliary equipment may be run, the initial steam temperature may not be continuously maintained as high as it should be, etc.

It is this last category, that this guideline focuses on (although there is one section that discuss some initial design considerations.). This guideline concentrates on how to

monitor and maintain/improve the heat rate of a plant through operation and maintenance practices.

1.1 Heat Rate Improvement Program

1.1.1 Definition

A heat rate improvement program encompasses all the activities that monitor the thermal performance of the unit or have an impact on the thermal performance of the unit. This broad definition includes many "routine" activities that are carried out at most plants. Examples include maintaining statistics on coal burned and electricity generated, instrument calibrations, and routine maintenance on critical equipment (pulverizers etc.) Many plants also have some system to monitor the heat rate or parameters on a real time basis. Does this mean most plants have a heat rate improvement program? Not necessarily.

The activities listed above are part of a heat rate improvement program (HRIP) but, in order to have an effective program, the following questions must be answered "yes":

1. Is the actual heat rate of the unit known accurately and tracked at some set frequency?
2. Is the true "expected" heat rate of the unit calculated and compared to the actual?
3. Is the gap between actual and expected heat rate further broken down into areas or charged to various parameters and is this information tracked over time and comparisons made between similar units?
4. Are gaps that cannot be immediately corrected, or that continually reoccur, assigned to an engineer to determine the root cause and "best" solution?
5. Is there a written action plan developed (and periodically amended) listing specific verifiable actions to improve the heat rate, including for each action: schedules, responsible person, and anticipated change in heat rate and a measurable indicator, etc.?
6. Is the action plan periodically reviewed during the year, checking that activities are carried out and the anticipated results obtained?
7. Do the station head and department heads support the plan (expect parameters with large deviations to have specific actions identified, review progress of the action plan, provide manpower and other resources to support the plan, consider a rupee spent for fuel equal in importance to a rupee spent for labor, material, etc.)

If the answers to any one of these questions is "no", then that plant does not have a first class heat rate improvement program.

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1.1.2 Purpose

The purpose of a heat rate improvement program is to reduce the heat rate of a unit. This results in several benefits:

The amount of money spent for fuel will be reduced. This lowers the cost of producing the electricity.

The amount of emissions to the environment will be reduced. Improving the heat rate reduces the amount of greenhouse gas that is produced. Less fuel burned means less ash to be disposed of, and less particulates go out the stack.

Because less fuel is required to produce the same amount of electricity, the amount of wear on equipment such as pulverizers, coal pipes and nozzles, etc., is reduced. Also, along with reduced fuel flows, the air flow is reduced, reducing velocities through the boiler, resulting in less erosion and reduced fan power.

Frequently, heat rate improvement (such as in the condenser pressure or auxiliary power area) results in an increase in the net generation of the unit, allowing the unit to run at a higher plant load factor. This advantage can be very valuable, especially in the summer when the condenser cooling water temperatures and ambient air temperatures are high, which sometimes results in generation being limited due to condenser pressure or running out of fan(s) capacity.

A heat rate improvement program frequently assists the maintenance department by eliminating the need to "open and inspect" equipment (or calibrate instruments) based on a time based schedule. Frequently, maintenance can be postponed if the thermal performance of the equipment is acceptable. The converse is also true, the HRIP may also determine that the equipment should be inspected before the normal overhaul schedule. If an unusual degradation is seen, the cost and benefits of advanced maintenance can be calculated and the most economical decision can be made concerning when to make repairs. Another advantage is that sometimes the HRIP can help determine the cause of equipment degradation before the inspection, allowing spares to be obtained early, thereby reducing the duration of the maintenance. Also the equipment performance testing portion of the program also provides for a periodic "spot check' of station instrument calibrations. Errors in station instrument readings can impact unit efficiency and/or load capability, and also yield false statistical data, which is being used to analyze plant heat rate and individual parameter performance. When station instrument errors are found during a performance test on a piece of equipment, these errors can be corrected by having maintenance recalibrate the faulty instrument.

1.1.3 How To Start a Heat Rate Improvement Program

This guideline describes various processes of a heat rate improvement program. Many utilities or plants already have parts of these activities in place, other utilities or plants may be starting a heat rate improvement program for the first time. Regardless of the starting point, a utility or plant can begin or expand their HRIP in two ways.

One approach is to start applying one or more processes (such as monitoring primary process indicators, or calculating heat rate deviations by parameter) to all systems in the plant and, over time, add more processes to their program.

The second approach would be to concentrate on one or two systems (i.e., condenser or pulverizer/fuel feed) and work on multiple processes (monitoring primary process indicators, calculating heat rate deviations, testing, developing an improvement plan, etc.) for those systems first. After people become comfortable with this smaller program, it can be expanded, including more and more systems.

Regardless of the approach that is used, it is essential that the program be well designed. Too often, a heat rate improvement program is put together quickly without considering all the factors. Then when the results are questionable, or there is no sustainability or accountability, the program fades away.

1.1.4 Mistakes To Avoid

Over the last several years, many utilities have started heat rate improvement programs, only to see most of them become ineffective or disappear. Some utilities go through cycles, starting a new heat rate improvement program every few years, with a lot of initial fanfare, but after a couple of years everything returns to "business as usual." After one or two cycles, even the initial fanfare is ignored.

Some of the things that should <u>not</u> be done include:

1. Start without support from the Plant Management (Station Head) and Executives.

   Heat rate improvement must have the full support of the head of the station and all department heads. A single engineer (or even department) working alone, cannot improve the efficiency of the plant. It takes personnel from all departments working together to make improvements.

2. Assign responsibility for heat rate improvement to an engineer (or engineers) with little or no support from operations and maintenance or upper management.

   As was stated above, heat rate improvement is everyone's business. An engineer or manager may be assigned responsibility for tracking performance and other activities or to take the lead in developing the improvement plan, but he alone cannot improve heat rate. To improve the thermal efficiency, the way the unit is

operated has to change, or the way maintenance is scheduled and performed has to change. The "performance" engineer may know what needs to be done, but frequently he does not have the authority to ensure that the appropriate actions are carried out.

3. Develop "action plans" with:
   a) Vague or no specific action
   b) No single individual assigned responsibility
   c) No "firm" schedule

   Too often, action plans are vague and therefore nothing happens. Activities must be specific so it is clear what is to be done, and a check can be made to see if the action was carried out. Some examples of vague activities are "more operator awareness to steam temperatures", what does this mean? How can this be measured? How will this be checked to see if it was done?. Another example of a vague activity is "clean condensers more often." The obvious question is "how often?" An example of a more definitive action would be "Brush clean condensers in March, June and September". Another problem with plans is if no specific individual is assigned the responsibility (or if a group of people is assigned responsibility), then no one has "ownership" of this activity. At the end of the year, if the action has not been done, then who should be held accountable? And last, if there is not a schedule, then the activity probably will not be done.

4. Develop plans/action items and never follow them up.

   Even with good plans, if there is no follow up, the plan probably will not be carried out. At least quarterly (and perhaps monthly), there should be a review where the following questions are asked:
   - Were activities that were scheduled to start, actually started? If not, why not?
   - Were activities that were scheduled to be finished, actually completed? If not, why not?
   - For completed activities, did the anticipated change (to some measurable indicator) show the predicted change? If not, why not?

   When everyone knows that these questions are going to be asked, the chances of developing a good plan and following through with the improvement activities increases greatly.

5. Make a Heat Rate Expenditure less important than an O&M Expenditure.

   In some utilities, the rupees spent on labor and materials comes from "one budget", the rupees spent on fuel come from a separate "budget", and the two are considered separately. What should be done is to minimize the total amount of money spent from both pots. Instead, one person may be responsible for one pot

of money, and will not increase his spending, even if it would save twice that amount of money from the other pot. What is required is for all organizations to work together so that the total amount of money spent from both pots is minimized

A related issue is the tariff structure. Some utilities are worried that if they improve their heat rate, the tariff will be lowered proportionally. This leads them to believe that they are better off not to spend the additional O&M money to gain heat rate improvements. This is a valid concern, and one that should be addressed with the regulatory authorities. However, even if that constraint is present, the other gains (greenhouse gas emission reduction, reduced maintenance, etc.) justify a heat rate improvement program.

6. Bad or no data

Data is critical to a heat rate improvement program. If the data that is collected is erroneous, or if important data is missing, then decisions cannot be made, or worse, the wrong decision may be made. Bad data can be caused by human error, poor collection procedures (for example, using a snapshot at one point in time to represent the entire day's data), sensors drifting (for example thermocouples in high temperature environments tend to drift low over time), poor test procedures, and other causes.

Another problem with unreliable data, is that the results will be unreliable. One way this can appear is if the heat rate or one or more parameter deviations has a lot of scatter. Another indication of poor quality data is if the "unaccountable" heat rate deviations are excessive. If results are unreliable, people quickly lose confidence in them.

7. Install real time heat rate/controllable loss monitor with:
   a) poorly defined "expected" levels
   b) poor quality sensors (frequently out of service or out of calibration)
   c) no follow-up/check by management

Over the last several years, real time heat rate monitors and real time controllable loss monitors have been installed on many units. Some of the units have realized large improvements in the heat rate. However, in other cases, these systems have been abandoned.

One reason for the ineffectiveness of some parameter monitoring systems is that the unit operators were being held to unrealistic or unattainable expected or target values. After some time, having never being able to reach the goal set up for them, the operators lose interest.

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A very common problem, especially with heat rate monitors, is erroneous data. Most heat rate monitors require a large number of sensors to be functioning accurately to be able to estimate the real time heat rate of the unit. When one or more of these instruments goes out of service or out of calibration, the results become unbelievable, and as above, the intended users lose interest. Another cause of bad data is that most systems require that some data be "estimated", since it cannot be measured in real time (such as coal analysis, carbon content of fly ash, etc.). Sometimes these assumptions do not cause large errors in calculated values, but sometimes they can and do.

Another common reason why many plants have not achieved the improvements they expected with these systems is that there is no "follow-up" by management. There are no rewards or recognition of personnel who use the system to make improvements, there are no questions from management when the systems shows large losses. Eventually, the operators come to believe that the system isn't valuable if management is not paying attention to it, so they also ignore it.

8. Spend more time discussing "expected levels" and/or trying to account for every kcal of heat input, than working on lowering the heat rate.

The better indicator of how well a unit is being operated and maintained, is to track the sum of its "heat rate deviations from expected" for all parameters, instead of the actual heat rate. However this can cause a problem. In order to "look better", there are two options:

- Decrease the heat rate deviations by performing improvement activities.
- Raise the expected parameter levels.

The easier option is the second, and unfortunately, there will be some persons who will want to take this route. Two indications of this is are when more time is spent discussing the "expected levels" than activities that will actually improve the heat rate or if an heat rate improvement plan includes more "Accounting and Expected Level" activities than "Improvement" activities.

There are times when it is appropriate to change "expected levels", but this should not be the primary focus of a heat rate improvement plan.

Another activity that uses up valuable resources is trying to account for every kcal/kWh. This is not to say that trying to locate all the heat rate losses (or accounting for the losses that are usually grouped together as "unaccountable") is wrong. But our focus should stay on minimizing losses, not just accounting for them. Too often personnel get more concerned with the accounting, and forget that the accounting is only a tool and not the final product.

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These are some of the common mistakes that should be avoided when starting or re-starting a heat rate improvement program.

1.2 Definitions

While the general term "heat rate" was defined above, it is used in different ways and means different things, depending on several factors. For an operating unit, the most important heat rate is the "net heat rate". However, frequently "gross heat rate" is commonly listed. Turbogenerator manufacturers are not interested in the boiler and boiler losses, so they refer to "turbine cycle heat rate." Other common factors causing confusion are:

- Is the heat rate the "operating" heat rate or is it an average over some time period and includes electricity used when the unit was off-line and fuel used in startups?
- For net heat rates, is station auxiliary power prorated (coal handling, water plant, etc.) and "charged" to the unit(s)?
- Besides power used to drive equipment in the powerhouse, what other items are part of station service? Such items include power used in townships built as part of the station, or power used for construction (either new units at the station or major additions such as a bigger precipitator), power used for coal preparation and ash handling.

It is critical that when referring to a number for heat rate that a specific, clearly understood heat rate term be used.

1.2.1 Generation

Before defining the various heat rate terms, the two generation terms should be defined.

1.2.1.1 Gross Generation

The electrical energy produced by all generators on a unit. This is the integrated power leaving the generator, usually measured between the generator and main power transformer. (Some caution should be used when this term is seen, as some turbogenerator manufacturers use this term for the power supplied to the electrical generator (as torque on the shaft), and to them net generation is the output of the generator, with the difference being generator losses.)

1.2.1.2 Net Generation

The electrical energy that a unit is able to supply to the distribution system. It is the gross generation minus all electrical energy used in the process of producing electricity. This

includes the unit auxiliary energy (i.e. BFP, FDF, etc.) and station service energy (coal handling, service water, etc.). At plants with multiple units, the station service energy must be prorated between the units. As a general rule (but not universally adopted) energy used by townships is not subtracted from the gross (or added to the station service), and neither is energy used for construction of new units or construction of major additions (coal wash facilities, larger precipitators, scrubbers, etc.).

1.2.1.3 Comparison of Gross and Net

Most heat rates should be calculated using the "net" generation. There are certain times when a gross heat rate is the appropriate value to consider. However, power plants exist to supply electricity to the distribution system, and only the energy that goes to the grid has value to the plant's customer.

1.2.2 Heat Rate

There are several different definitions of heat rate that are used. Each definition is valid and useful when properly applied and clearly explained to anyone who could come across it. Unfortunately, too often only the term "heat rate" is used, and it is not always clear exactly which "heat rate" is being used. It is very important that additional terms be used so that it is perfectly clear what is meant.

The following list defines various terms, some are exclusive (i.e., a heat rate is either net or gross, but not be both) but many are not exclusive, and should be combined to provide a clear understanding of the heat rate that is being discussed (i.e., design gross turbine cycle heat rate or actual net heat rate)

1.2.2.1 Net versus Gross Heat Rate

A "net" heat rate is calculated using a heat input and the "net" electrical generation. A "gross" heat rate is calculated using a heat input and the "gross" electrical generation.

1.2.2.2 Unit versus Turbine Cycle Heat Rate

A "unit" heat rate includes all heat input to the boiler. A "turbine cycle" heat rate includes only heat input to the turbine cycle (the difference being the boiler efficiency). The heat input to the boiler should include all forms of chemical energy supplied (coal, oil, gas, or other fuels such as tires, wood, biogas, etc.) not just the primary fuel (i.e., coal).

For most functions (daily/monthly/annual reporting, comparison/benchmarking of units) net unit or net plant heat rate should be used. One place where gross turbine cycle heat rate will appear is in performance guarantee test reports.

1.2.2.3 Actual Heat Rate

The true heat rate of a unit is usually calculated from the amount of fuel (all fuel, coal, oil, gas, etc.) burned during the period, the higher heating value of the fuel, and the generation for the period. The integrated readings of fuel burned and generation are (i.e. daily, weekly, or monthly) used to calculate the actual heat rate. Actual heat rate includes fuel burned during startups, and includes the negative net generation during off-line periods.

1.2.2.4 Operating Heat Rate

This is the same as the actual heat rate, but it excludes any heat input during startups (or until the unit reached minimum stable load), and for operating net, it also excludes the electrical energy used when the unit was not synchronized (or was below minimum stable load). Some utilities define the operating heat rate as the heat rate when the unit is synchronized, while other utilities define operating heat rate as the heat rate when the unit is at or above its minimum stable load.

1.2.2.5 Contract Heat Rate

A "contract" heat rate is the operating heat rate, corrected for certain conditions. For example, a turbogenerator manufacturer may supply his equipment, guaranteeing a certain heat rate, with specified boundary conditions (initial steam conditions, final feedwater temperature, TTDs and DCAs on feedwater heaters, pressure drop through the reheater, etc.). When the PG test is run, the actual heat rate is first measured, then using correction factors (or a thermodynamic model) the "contract" heat rate is calculated. It is the heat rate that would have been measured if the boundary conditions were all at design specifications.

1.2.2.6 Design Heat Rate

The heat rate on a unit varies with both load and ambient conditions (air and cooling water temperature). However, one "design" heat rate number is frequently published for a unit. It is the heat rate the designer anticipates will occur at the design condenser pressure and a certain load. Whenever it is given, the unit condition should also be specified (such as "maximum continuous rating" (MCR) or the load when the turbine valves are wide open (VWO), etc.).

1.2.2.7 Reference Heat Rate

This is the heat rate the unit is capable of obtaining, based on the initial design configuration. It is usually derived from the turbogenerator and boiler performance guarantee or acceptance test results.

1.2.2.8 Expected Heat Rate

The heat rate that the unit should be capable of obtaining based on the present design configuration (but this heat rate may not be achievable with the present condition of that equipment). It is calculated by adjusting the reference heat rate for the effects of permanent modifications to the unit. Expected heat rate is almost always a net unit heat rate.

1.2.2.9 Target Heat Rate

The heat rate that is set as a goal. This is usually based on historical performance along with anticipated changes resulting from improvement activities and uncorrected degradation.

1.3 Descriptions of Each Section of the Guidelines

1.3.1 Section 2 - Monitoring Primary Process Indicators

Section 2 "Monitoring Primary Process Indicators", describes how to monitor "low level" indicators (temperatures, pressures, etc.) using trend and control charts. By monitoring these primary process indicators, many problems can be detected and corrected before they cause a significant heat rate deviation.

1.3.2 Section 3 - Calculating Heat Rate Deviations by Parameter

Section 3, "Calculating Heat Rate Deviations by Parameter", describes in detail, how to calculate an "expected" heat rate and heat rate deviations due to parameter deviations. It includes a discussion of what baseline should be used, data to be collected, and the calculation procedure.

1.3.3 Section 4 - Performance Testing

Section 4 describes a performance testing program, when tests should be run, what data should be collected, what calculations should be made and how to do them, what should be a pretest checksheet, etc.

1.3.4 Section 5 - Heat Rate Action Plan

Section 5 discusses Heat Rate Action Plans, a formal description of the all activities that will affect the unit. It describes how to build the plan, how to maintain the plan, and how to periodically review the plan to ensure that activities are being carried out and the anticipated benefits are being realized.

1.3.5 Section 6 - Tools

Section 6, "Tools" describes some of the information that should be available to the persons working on heat rate improvement. A description of the drawings, specification sheets, test and operating data that should be maintained is listed.

1.3.6 Section 7 - Manpower and Training Programs

Section 7, Manpower and Training Programs" discusses the human resources required to maintain a heat rate improvement program, both at the plants and in the corporate office. It also describes the various type of training that is usually provided to personnel involved in heat rate improvement.

1.3.7 Section 8 - Heat Rate Surveys and Audits

Section 8 describes how to survey a plant to collect background design information and to determine the current level of its heat rate program.

1.3.8 Section 9 - New Plant Specifications

Section 9 "New Plant Specifications" describes some areas, that if addressed in the specification, can either improve the efficiency or improve the monitoring and problem solving capabilities, of the plant. Some of these directly improve heat rate, some make the monitoring of the heat rate easier, and some allow easier continual maintenance of peak performance. This section also describes some test equipment that should be furnished with any plant, and some points that should be considered when specifying the performance guarantee test procedures.

1.3.9 Section 10 - Carbon Dioxide Emissions

Section 10, "Carbon Dioxide Emissions," describes three calculation procedures and the inputs that are required for tracking the amount of carbon dioxide emitted from a power plant.

1.3.10 Section 11 - Economic Dispatch of Multiple Units

Section 11, "Economic Dispatch of Multiple Units," describes the theory and calculation procedure to minimize the total cost of producing the required amount of power supplied

from two or more generating units. The approach is that all contributing units should be at the same "incremental cost." The incremental cost curves are driven primarily by the "incremental heat rate," but also includes the *variable* O&M costs. This approach can be applied to the units at a plant, to multiple units in an area, or to all units providing power to a grid.

1.3.11 Section 12 - Common Areas of Heat Rate Improvement

Section 12, "Common Areas of Heat Rate Improvement", discusses eleven methods that can be used at many plants for immediate heat rate improvements. Not all areas apply to all plants, and some plants may not have any problems in these areas. However, many of these topics discussed here can be applied to most plants. This section should not be used in place of the activities discussed in Sections 2-5, including doing root cause analysis of deviations. However, these areas usually provide a quick and inexpensive way to improve the performance of a unit until a formal Heat Rate Improvement Program is in place and accepted.

SECTION 2 MONITORING PRIMARY PROCESS INDICATORS

2.1 Purpose

Historically, heat rate management has been focused on identifying heat rate problems and then identifying and implementing corrective actions to eliminate the efficiency loss (see "Section 3 Calculating Heat Rate Deviations by Parameter"). In this approach, heat rate deviations from expected or design levels are identified and quantified. Then using fault trees or other resources the cause(s) are identified (hopefully drilling down to the root-cause, and not just a symptom). This method is useful, but the fact is, the heat rate should not be allowed to deteriorate to the point that unexpected or unknown deviations exist. This is a reactionary, "top down" approach to business that utility management should not consider acceptable.

The responsible operations and maintenance personnel should be aware of "problems" and take corrective action before the problem becomes sever enough to significantly impacting the heat rate. If they are unable to have the problems corrected that quickly, at least they have identified the problem so the heat rate deviation is predicted and accounted for from the "bottom up." The personnel closest to the equipment should be aware of equipment deterioration and have it corrected (or accounted for) prior to the point of it significantly impacting the heat rate. By following good O&M practices and proactively attacking maintenance, heat rate deteriorations can be avoided (or at least accounted for) before they are reflected in the overall performance statistics. Stated another way, attention to the basics of operation and maintenance will result in the plant's availability and efficiency statistical results taking care of themselves.

This pro-active approach is accomplished by identifying key "primary process indicators", monitoring them and taking corrective actions immediately as indicated. If this is done, the improvement in higher level performance parameters (heat rate, availability, etc.) will follow.

The required steps in this process are:

1) Defining the "Primary Process Indicators" to be monitored.
2) Determining the data collection methodology for each indicator.
3) Setting appropriate "limits" for each indicator.

A secondary purpose for monitoring primary process indicators is to assist in instrument validation/instrument calibration. By closely monitoring critical instruments, drifts or irregularities can be quickly identified and the instrument calibrated or replaced. Also, if a good program exists where these sensors are monitored and compared, then the calibration schedule for those instruments can be extended, allowing that manpower to be used to check instruments that have no other verification method.

2.2 Definition of Primary Process Indicators

What is a "Primary Process Indicator." It is the information to be monitored, measured, or tracked that will give an indication that the process is in trouble and needs corrective action (or that there has been a change in the process) prior to seeing adverse impacts to a higher indicator, such as a heat rate deviation.

Some indicators are also "parameters" used to calculate heat rate deviations, such as hot reheat steam temperature (see "Section 3 Calculating Heat Rate Deviations by Parameter"). However, most indicators are at a much lower level, such as mill outlet temperature. Normally, heat rate deviations are not calculated for mill outlet temperature, even though it affects the thermal performance (and other characteristics) of the unit. Another example of a primary process indicator is condenser waterbox differential pressure. If it increases (due to fouling of the inlet tube sheet, for example), it can reduce the condenser circulating water flow, resulting in a higher condenser pressure and a higher heat rate. Monitoring this differential pressure will immediately show when a change has occurred, and signal for corrective action to be taken, before it adversely affects the unit heat rate.

Each system in the power plant (fuel supply, combustion, feedwater, condenser circulating water, etc.) has multiple Primary Process Indicators that can be used to determine if the process is functioning as designed, (or if there has been a change to the process).

There are several available sources that can be used to help develop a list of indicators, such as original equipment manufacturers manuals, heat rate deviation fault trees, etc. Section 2.4 is a partial list of indicators that could be considered.

2.3 Data Collection and Analysis Methods

The second step is to determine the data collection methodology for each indicator. This step may start with verifying that the correct type of sensor/range is being used at each critical location and that the instrument is correctly installed. Instruments should be installed as per ASME PTC 19.2 and 19.3. However, in existing installations, if thermowells, etc. are not at the proper location, they still can be used for trending, even if they would not be considered representative for performance testing.

Some Primary Process Indicators may be monitored continuously via a computer system or a recorder or indicator in the unit control room. Others may have a site/auxiliary control room/local board indication that is only periodically checked, or only measured during routine or special tests.

Indicators whose readings are collected via a data acquisition system can be monitored continuously. Fortunately, with the large scale use of computers, huge amounts of information can be collected and managed relatively quickly and easily. However, this does not exclude other relevant data from being collected and analyzed. A second technique in use today involves using hand-held data collectors (roughly the size of a large calculator) to take periodic, manual readings. The data stored in these devices can then be downloaded to a computer for further

analysis. Third, data can be collected manually and entered into a computer spreadsheet for further analysis. A fourth method, used over many years, is to collect and plot data manually.

The quality of the data is extremely important. With any sensor, there is the possibility of it drifting, or being out of service. If data is read manually, gross errors can be excluded by the person taking the readings. However, with manual readings, the possibility of human errors cannot be ruled out. One crude method of checking for errors that has been used for several years is to put upper and lower limits for values. The drawback to this method is that the resulting range (that will be good over all possible scenarios, will exclude only the extreme errors. An improvement on this method is to plot one variable against another, and to put limits above and below this plot. This will help find more errors, but is a very time consuming process. A third method is to use pattern recognition or neural net software for data validation. These methods will result in greatly improved data integrity, and require less time to setup.

Once the data is obtained and is validated, it may require "filtering" before meaningful information can be obtained. For example, plotting air leakage into the condenser over time would be useful to determine if new leaks are occurring (that could adversely affect the condenser pressure). However, if the amount of leakage is measured (and plotted) at random or at specified time intervals (i.e., every two hours), the resulting plot will be meaningless as air inleakage will vary with the load on the unit. (At low loads some parts of the cycle are under vacuum, that are pressurized at full load.) Therefore, condenser air inleakage should be either measured only when the unit is operating in a specified load range or multiple plots should be kept, with each plot for a specified load range.

Another filtering method is to calculate a "deviation from expected" for a point, where the "expected" value could be a function of load, flow or some other variable(s). Instead of plotting the indicator's value, the plot is of the "indicator's deviation from expected." For example, condenser pressure will vary with the condenser duty (the amount of heat being rejected in the condenser), the circulating water inlet temperature, and the circulating water flow rate. Since at least two, and in some cases, all three of these can change, trending condenser pressure will not result in a meaningful plot. It would be better to calculate an "expected condenser pressure" and trend the difference between the "actual condenser pressure" and the "expected condenser pressure." However, this method should only be used when the "expected level" that will be used is well defined and valid.

Other data may need some simple arithmetic to be performed before plotting. For example, the inlet pressure to the intermediate pressure turbine (IPT) will vary with the load on the unit. The IPT exhaust pressure will also vary with load. However, the ratio of these two should be constant (unless there is a change to the flow path in the turbine). Therefore instead of plotting these two values individually, their ratio should be plotted. Another example of a computed point would be the air heater x-ratio, which is calculated from the air and gas temperatures in and out of the air heater.

The purpose of "filtering" data is to eliminate spikes, or other "noise" from the plots that are generated, so that any "out of control or specification limits" points on the plot are due to some

"real" disturbance to the process and not from normal variations (such as low load periods, different mill combinations, etc.).

Another consideration is the frequency and type of readings. The frequency sets how often a data point is stored in the database (or manually graphed). If the data is read automatically via a computer, it might be read as frequently as once a minute. For manual readings, once an hour, once or twice a shift, or daily are typical frequencies. The type of reading could be a "snapshot" or an "average." A snapshot is the value at a single point in time. Most manual readings (for example, where someone reads a gauge once a shift) are snapshots. Averaged readings may be stored in the database (or manually graphed), but each individual point is the average of snapshot values over some time period. For example, a data acquisition system (DAS) might read the condenser's waterbox differential pressure every 15 seconds. For the purpose of tracking this primary process indicator, the DAS might average these 15 second values over an hour, and the hourly averages would be trended over time. A special type of snapshot is the reading of an integrator. The value read is a snapshot, the value at that point in time, but the value reflects some total over time. An example of readings where an integrator is appropriate would be common station service or makeup water flow. Since they are not steady values, an hourly integrator reading would be more useful.

For indicators that are relatively stable (there are only small changes over time), snapshots taken once a day may be sufficient. Example of indicators that fit this category are:

- Turbine pressure ratios
- Number of startup drains that are leaking through

For indicators that can change relatively rapidly, either frequent snapshots, or averaged readings should be taken. If two (or more) points are used to calculate a parameter (such a pressure ratio or a temperature rise), all readings used in the calculation must be taken in the same manner (either snapshot or averaged) and at the same time.

A final consideration is how to analyze the data so that useful information can be easily obtained, without requiring excessive time or other resources.

One method is to draw a "control chart" for an indicator. A control chart is used to determine if a process is stable or not. A control chart has three lines (in addition to the indicator's values), an upper control limit, a center line, and a lower control limit. The control limits are not targets or specification limits, but are calculated using the rules of "Statistical Process Control." The process is in statistical control if the indicator's values fall inside the calculated limits, and do not show certain patterns. Some patterns that are looked for include a run of points all above or all below the center line; a run of points all ascending or all descending; or any repeating patterns. Statistical control only implies that the process is stable. A process can be in control, but not meeting the specification limits or targets. There are several references on Statistical Process Control that are available giving details on constructing and interpreting control charts.

A trend graph can also be used to track indicators. In addition to plotting the indicators value over time, the trend graph usually also shows specification limits and/or target values.

Whichever type of chart is used, (one type may be used for some indicators and the other type for other indicators) it must be updated continually, and reviewed. Whenever a trigger occurs, either an out-of-control-condition for control charts or out-of-specification/target-limits for trend charts, the "operator of the process" should immediately determine if it is due to "Common Cause" or "Special Cause". (It is critical that the "operator of the process", the person that is controlling the system(s) and most familiar with it, generate or review these charts. This does not mean that higher level engineers and managers do not need to monitor these charts, they should. However the personnel in direct control of the systems must take responsibility for the efficient operation of the systems.) A "Special Cause" event is due to a process upset or operator error, and only requires that the cause be noted on the chart. However a "Common Cause" event, one that is due to a problem with the process, should initiate immediate action to determine the problem's root cause and to correct the situation, so that a significant loss of efficiency does not occur.

The immediate action may be taken by the "operator of the process" solely, or it may require help from other departments.

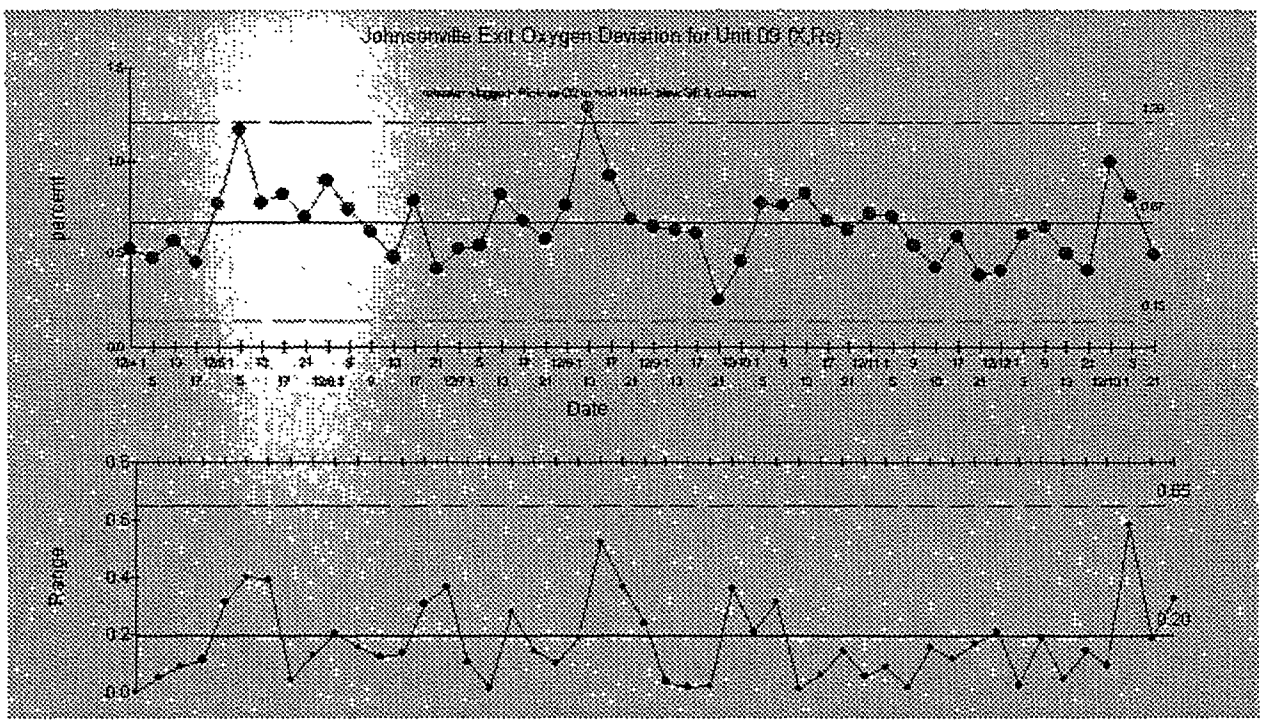

Figure 2.1 Example of a Control Chart

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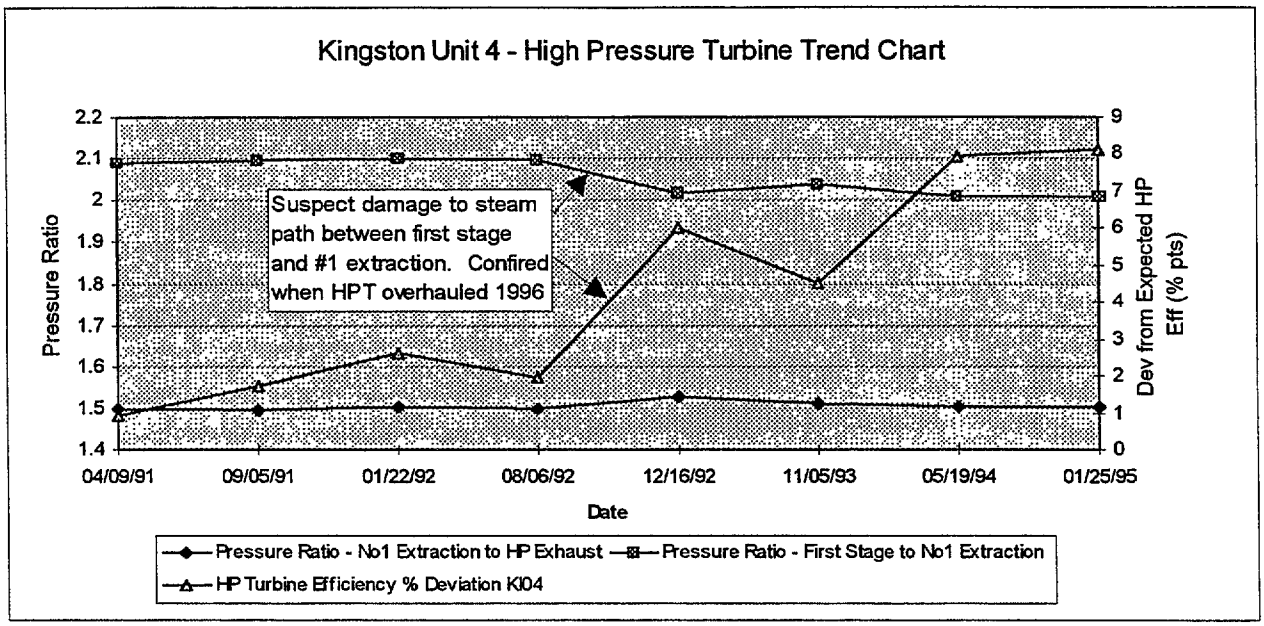

Figure 2.2 Example of a Trend Chart

Where a DAS system is continually monitoring and generating trend and control charts, the system can also be programmed to generate "exception reports." An exception report can be generated for each unit, to notify the responsible operator of indicators that are out of specification (trend chart) or out of control (control chart), so that appropriate action can be initiated. Exception reports can also be generated at a higher level, such as the plant wide report shown below.

Johnsonville Fossil Plant Exception Report for Primary Process Indicators - 12/15/99 0700-1500

Primary Process Indicator	Unit 01	Unit 02	Unit 03	Unit 04	Unit 05	Unit 06	Total
Condenser Pressure Dev. from Expected						X High	1
AH Exit Gas Temp Dev. from Expected	SpecHi	X Low		X_UCL	Spec/XRs		4
Boiler Exit Oxygen Deviation					X_UCL		1
Throttle Pressure Deviation from Expected						R Trend	1
Throttle Temperature Deviation from Exp.				Spec/XRs			1
HRH at IV Temp							0
Mill "A" Outlet Temp	RS_UCL						1
Mill "B" Outlet Temp						X_LCL	1
Mill "C" Outlet Temp							0
Mill "D" Outlet Temp					RS Low		1
DT between the two SSH outlets							0
AH x-ratio							0
Left Waterbox Differential Pressure					X_UCL		1
Right Waterbox Differential Pressure							0
Total	2	1		2	4	3	12

Figure 2.3 Example of an Exception Report

2.4 List of Primary Process Indicators

The following is a partial list of Primary Process Indicators that should be monitored. Because each unit has different designs, this list is only a guide, and items should be added or deleted as necessary. Items marked "(H)" should be given the highest priority, and should be control charted/trend charted on all units. Items marked "(M)" are a medium priority and should be control charted/trend charted wherever practical. Items marked "(L)" are the lowest priority and should be control charted/trend charted where these points are available on a DAS and/or if there are known problems.

1) Boiler - Water/Steam Side
   - a) Steam Temperatures:
     - i) Secondary superheat outlet temperature(s) (H)
     - ii) ΔT between the two SSH outlets (M)
     - iii) Hot Reheat outlet temperature(s) (H)
     - iv) ΔT between the two HRH outlets (M)
     - v) ΔT across attemperation station when the steam temperature is below set point (H)
     - vi) ΔT between SSH outlet and HPT inlet (L)
     - vii) ΔT between HRH outlet and IPT inlet (L)
   - b) Attemperation Flows
     - i) Reheat (H)
     - ii) Superheat (H)
   - c) FW temperature at economizer outlet (M)
   - d) Water/Steam temperature rise across each sections (Econ, WW, PSH, SSH, RH, etc.) (M)
   - e) Pressure Drop
     - i) economizer (L)
     - ii) Waterwalls (drum units) (L)
     - iii) Economizer outlet - SSH outlet (once through units) (M)
2) Boiler - Fire Side
   - a) Flue gas temperatures
     - i) Air heater inlet gas temperature(s) (H)
     - ii) Air heater outlet gas temperature(s) (H)
   - b) Primary air header pressure (H)
   - c) Windbox pressure (H)
   - d) Furnace draft (L)
   - e) Windbox to furnace differential (M)
   - f) Draft loss through each bank of convection pass (i.e. SSH, LTSH, RH) (L)
   - g) Furnace exit gas temperature (M)
   - h) Flue gas oxygen percentage rise - nose arch to air heater inlet. (M)
   - i) Flue gas oxygen percentage rise - air heater outlet to ID fan inlet (M)
   - j) Damper positions (fuel/air, aux. air, overfire, AA, etc.) (M)
   - k) Burner tilt position (M)
   - l) Sootblowing steam usage (M)

m) Ratio of fan amps to flow (FDF, IDF, PAF, GRF) (M)

3) Air Heater
   - a) Air temperature entering (units with steam coils or hot air recirculation) (L)
   - b) ΔT at AH outlet/cold air bypass junction (H)
   - c) Flue gas analysis - oxygen at AH inlet (H)
   - d) Flue gas analysis - carbon monoxide (M)
   - e) Draft loss through AH
     - i) Air side (L)
     - ii) Gas Side (L)
   - f) AH x-ratio (M)
   - g) AH gas side efficiency (M)
   - h) Leakage (H)
   - i) Ratio of total primary air flow to total secondary air flow (M)

4) Pulverizer
   - a) Mill outlet temperatures (mill in service) (H)
   - b) Mill outlet temperatures (mill out of service) (H)
   - c) Temperature drop at air / tempering air junction (M)
   - d) Ratio of mill amps to coal flow (M)
   - e) Ratio of mill ΔP to coal flow (M)
   - f) Mill Fineness:
     - i) % passing through a 50 mesh (H)
     - ii) % passing through a 200 mesh (H)
   - g) For each coal pipe, the difference between it's air flow and the average for all pipes from that mill. (M)

5) Turbine
   - a) Throttle pressure (H)
   - b) Pressure ratio
     - i) HP - First Stage / HP exhaust (H)
     - ii) IP - inlet / outlet (H)
     - iii) LP - inlet / last extraction (M)
   - c) Ratio of corrected first stage pressure to corrected flow (H)
   - d) Ratio of turbine valve position to flow (L)
   - e) Section efficiencies (M)

6) Feedwater (Heaters and Feed Pump)
   - a) Terminal temperature differences (M)
   - b) Drain cooler approaches (L)
   - c) Water side pressure drop (L)
   - d) Pressure drop in extraction piping (L)
   - e) Ratio of drain flow to feedwater flow (L)
   - f) Final feedwater temperature (H)
   - g) T at feedwater heater outlet / FW bypass junction (M)
   - h) Heater level (L)
   - i) Heater vent positions (L)

7) Boiler Feed Pumps
   - a) BFP balancing drum leakoff flow or static pressure (if no differential) (H)

	b)	Ratio of pump amps to feedwater flow through the pump	(M)
	c)	% of total FW flow carried by each pump	(L)
8)	Condenser		
	a)	CCW Flow	(H)
	b)	Air Inleakage	(H)
	c)	Waterbox Differential Pressure	(H)
	d)	Terminal Temperature Difference	(M)
	e)	Temperature Rise	(M)
	f)	Deviation from expected condenser pressure	(M)
	g)	Max Temperature Difference in each Outlet Pipe (if outlet temp is measured at more than one point in a pipe)	(L)
	h)	Pressure (or elevation) drop across trash rack/traveling water screens	(M)
	i)	Condensate subcooling	(L)
	j)	Air removal subcooling	(L)
	k)	Tube Cleanliness factor	(M)
9)	Other		
	a)	Unit Station service, from unit board(s)	(H)
	b)	Common Station Service	(L)
	c)	Coal size leaving the crushers	(L)
	d)	Number of leaking valves (may want to have this divided into three categories, based on heat rate impact)	(M)
	e)	Sootblower availability (by area, WW, PSH, SSH, RH, Econ, etc.)	(M)
	f)	Makeup water flow	(M)
	g)	Steam flow to auxiliary equipment (BFPT, FDFT, other units, etc.)	(M)

2.5 References

Donald J. Wheeler and David S. Chambers, Understanding Statistical Process Control, 2nd Edition, SPC Press, Inc., 1992

Gary K. Griffith, Statistical Process Control Methods for Long and Short Runs, 2nd Edition, ASQC Quality Press, 1996

Donald J. Wheeler, Understanding Variation, the Key to Managing Chaos, SPC Press, Inc., 1993

SECTION 3 CALCULATING HEAT RATE DEVIATIONS BY PARAMETER

3.1 Reasons For Tracking Heat Rate Deviations By Parameter

Why are heat rate deviations calculated for various parameters? Someone might say that trending/control charting "heat rate parameter values" and "primary process indicators" is sufficient. ("Heat Rate parameters" are operating data or calculated values from operating data for which heat rate deviations can be calculated relatively easily, such as condenser pressure or turbine efficiency. "Primary process indicators" are operating data, such as pressure drop across the condenser waterbox or mill outlet temperature, which affect performance but for which heat rate deviations are not usually calculated) In a perfect or simplistic world, where there are only infrequent problems, that might be sufficient. In that world, as soon as something deviates from its historical value, corrective action would be taken, and the unit would be restored to its peak performance. But in an imperfect world, there are usually multiple problems. Also, some problems may have existed since initial operation of the unit, some problems may be masking other problems, and some problems may be causing multiple operating statistics to show deviations.

If only heat rate parameter values and/or primary process indicators are tracked, there is no way to determine the severity of different problems. For example, if the throttle temperature is 10 °C too low, and the condenser circulating water flow is 3000 m3/h too low, which problem should be addressed first? Also, how much resources should be committed to correct any given problem without spending more money than would be returned from the improved performance?

This is not to say that trending heat rate parameter values and primary process indicators should not be done. It is important to do that, however this alone does not supply enough information to monitor performance or to make decisions affecting the performance of the unit.

Another approach would be to monitor performance by trending the actual heat rate. A refinement for this would be to comparing the actual heat rate to some "baseline" (see Section 3.3 for a discussion of various baselines) and trending the deviation. By reviewing this trend, we can determine if the performance of the unit is improving or deteriorating. One advantage of this process over the above process is that the cost of the overall performance deterioration can be calculated. If this process is used exclusively, when there is a deterioration, there is no indication of the cause or causes. If this process is used along with the trending of heat rate parameters and primary process indicators, one or more areas causing the performance deterioration can be identified. However if there are multiple problems (as is usually the case) the relative severity of each is unknown.

As above, this argument does not invalidate trending the total heat rate deviation from some "expected" value, only that it is not sufficient for a heat rate improvement program.

The better solution is to combine both the above processes with an additional process. For as many heat rate parameters as possible, calculate a heat rate deviation due to each. The advantages of this additional step are:

a) The magnitude (heat rate deviation) of each problem area can be determined. On an individual unit, the problem areas can be ranked, and the most costly can be addressed first.
b) The heat rate deviation can be readily converted to a cost. When a corrective action is determined, the cost/benefit of the action can be calculated.
c) Trade-off between parameters can be calculated. For example, if a unit is having problems maintaining steam temperatures, a temporary solution might be to increase the air flow to the boiler. Unless you can calculate the heat rate deviation for the low steam temperature and compare it to the heat rate deviation due to the additional air flow, you cannot determine if it is more economical to run at reduced temperature or at higher air flows.
d) Problems at different units can be compared and ranked. By doing this, the problems areas at multiple units of a plant or over an entire system can be ranked. For example, a 200 MW unit might have a condenser pressure 25 mmHg higher than the "expected" value, a 500 MW unit might have an exit gas temperature 25 ^{0}C higher than expected. Unless the impact of both problems are converted to a heat rate deviation (then cost), you cannot determine which problem should be addressed first.
e) Once the heat rate deviations for each parameter are known, they can be summed to give a "accountable" heat rate deviation. Usually this value is smaller than the total heat rate deviation. The difference is the "unaccountable" heat rate deviation, and represents problems areas that are not tracked by a parameter (for example LP turbine efficiency, cycle isolation problems, etc.)

3.2 What Parameters Should Be Tracked?

In determining how to track heat rate deviations, there are two methods. First, deviations could be assigned to each piece of equipment, i.e., a heat rate deviation would be computed for the high pressure turbine, the air preheater, the boiler, the feedwater heaters, the BFP, the condenser, the CCW pumps, etc. There are definite advantages to this method, in that it would be known precisely which piece of equipment is not operating properly. However, the overall performance of most equipment is not continuously monitored, but instead the steam/air/flue gas conditions around the equipment are monitored. For example, if the steam temperature at the intercept valve is low, it is relatively easy to calculate a heat rate deviation for it. It is much harder to routinely determine the equipment that is causing the deviation, it could be the attemperator valve leaking, the reheater could be fouled or slagged, the gas recirculating fans or high pressure feedwater heaters could be out of service, etc.

The second method of tracking heat rate deviations would then be by steam/air/flue gas conditions, i.e., a heat rate deviation would be computed for the Throttle temperature, the steam

temperature at the intercept valve, the condenser pressure, the flue gas temperature leaving the AH, the flow rate to the APH coils, etc.

What is usually done, is a combination of these two methods. Heat rate deviations are calculated for some equipment (i.e., the turbine sections), but most losses are calculated from the conditions of the steam/air or flue gas at various locations (i.e. Throttle, condenser pressure, AH exit gas temperature). If one of the steam/air/flue gas conditions is causing a significant deviation, then the next level of analysis is performed to determine the root cause.

Parameters that may be monitored, and have heat rate deviations computed include:

3.2.1 Turbine Cycle

3.2.1.1 Auxiliary Steam Usage

Several pieces of auxiliary equipment may be present in the plant that are supplied with steam. The steam that is used may be main steam, from a turbine extraction, from the drum, or some other location. Regardless of the source of the steam, its use comes at some price. If the steam is not used by the auxiliary equipment, it would flow through the turbine, producing kilowatts. Therefore the use (or excessive use) of this steam causes a heat rate deviation. At some plants, it may be worthwhile to monitor the flow rate of steam to each auxiliary. At other plants, only the total usage may be tracked. The flow rate may be measured with orifice plates, nozzles, or annubars. Or the flow may be derived from a pressure reading (i.e. the bowl pressure of a drive turbine can be related to the flow rate to the turbine). Some equipment, such as air preheating coils, have a relatively constant steam flow, and only the number of hours of use is tracked. Some potential auxiliary steam users are:

a) APH coils
b) BFP turbines
c) Building heat
d) FDF turbines
e) Steam Jet Air Ejectors (SJAE)
f) Steam supplied to/received from other units
g) Misc. auxiliary steam headers

3.2.1.2 Condenser Pressure

Condenser pressure is usually one of the largest heat rate deviations at a plant. As the pressure in the condenser rises, the amount of energy extracted from each kilogram of steam is reduced. Approximately 1% of the output of the IP/LP turbine will be lost for every 6 mm Hg increase in condenser pressure. Usually a heat rate deviation is calculated due to the difference between the expected and actual condenser pressure. It is also possible to divide this deviation into two parts. The first heat rate deviation is due to the cooling water flow rate being different from design, and the second heat rate deviation due to cleanliness and other problems.

a) Deviation due to CCW flow
b) Deviation due to cleanliness & other reasons

How the condenser pressure is measured is critical. ASME PTC 12.2 describes how the condenser pressure should be measured. Due to the high velocity of the steam above the condenser tubes, a "basket tip" is required. This device allows only the static pressure to be sensed, and eliminates the velocity pressure component. Figure 3.1 shows an basket tip.

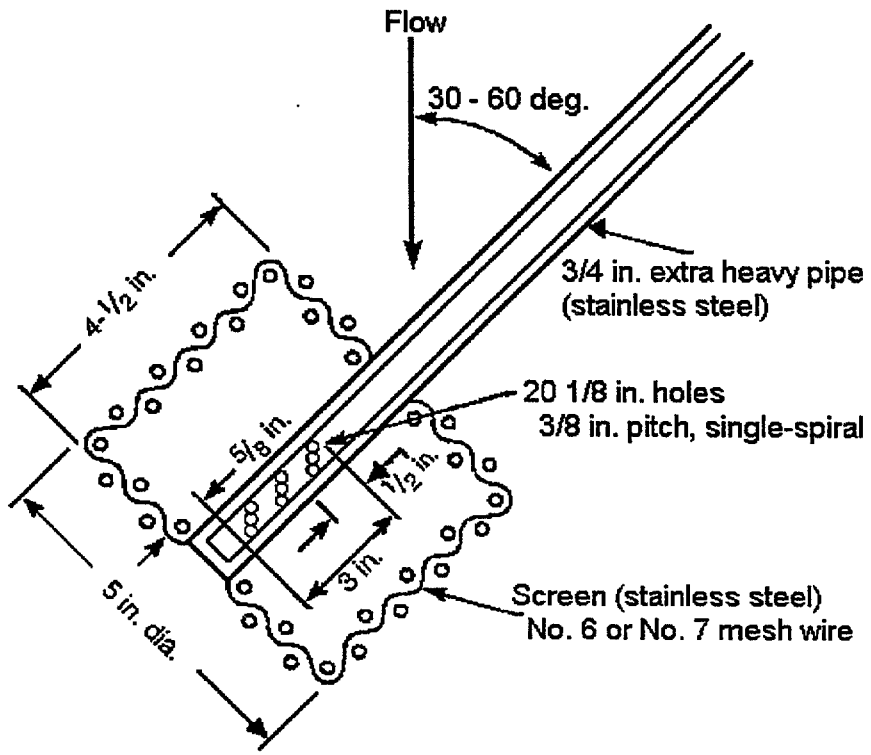

Figure 3.1 Basket Tip for Condenser Pressure Measurement

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An alternative to the basket tip is the "Guide Plate" shown in Figure 3.2.

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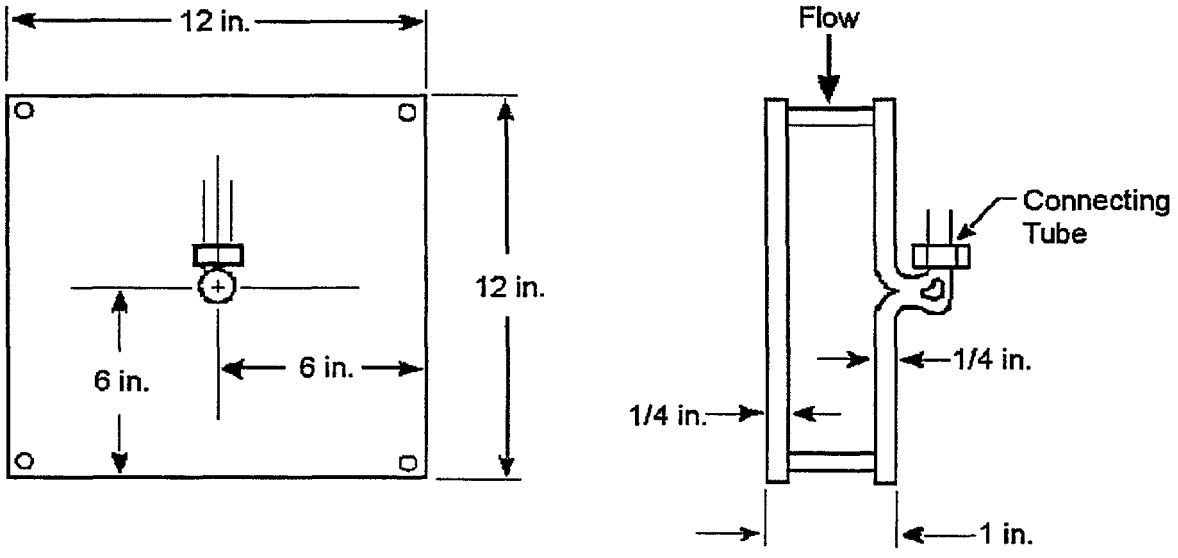

Figure 3.2 Guide Plate for Condenser Pressure Measurement

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The piping between the basket tip or guide plate must slope upwards continuously to prevent condensation from collecting in the piping. Figure 3.3 shows the proper piping arrangements for both gas and liquid filled piping.

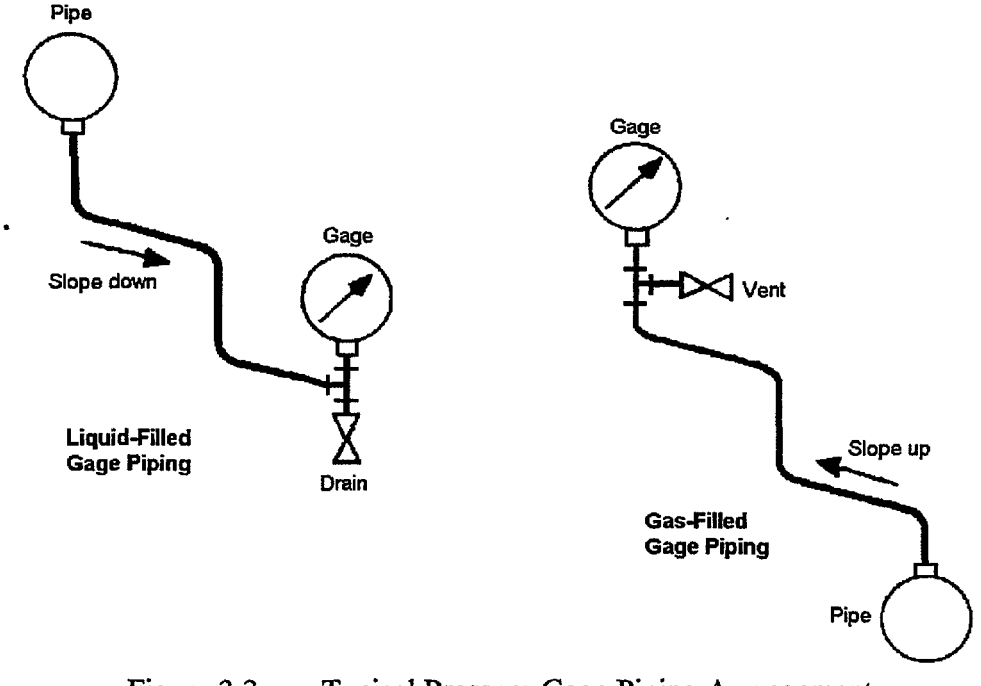

Figure 3.3 Typical Pressure Gage Piping Arrangement

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3.2.1.3 Final Feedwater Temperature & High Pressure Heaters Bypassed

There are three ways of calculating deviations due to the feedwater performance. First, a deviation can be calculated due to the final feedwater temperature being lower than expected. (For a typical unit, for every 1°C, decrease in final feedwater temperature, the boiler has to absorb 0.24 % more heat.) Second, a deviation can be calculated from the amount of time a heater or a string of heaters are bypassed. (Bypassing heaters may reduce final feedwater temperature, or it can cause the flow rate from higher pressure extractions to increase, thereby decreasing the output of the turbine, or other effects.) Third, both methods can be combined, where the final feedwater temperature deviation is used when all heaters are in service and the deviation for heaters being bypassed is used (and the final feedwater temperature is ignored) when one or more heaters is bypassed. The reason the final feedwater temperature cannot be used alone, is two fold. First, the heat rate deviation for the top heater being bypassed is different from the deviation that would be calculated based on the resulting final feedwater temperature. Second, if a lower pressure heater is bypassed, the final feedwater temperature may not change, but there is a heat rate deviation due to the higher extraction flow to higher pressure heaters.

3.2.1.4 Makeup

Makeup is the quantity of water that is lost from the cycle during operation. It is not simply the water supplied to a unit, as water is also supplied for boiler hydrostatic tests, chemical cleanings, etc. Water supplied for these purposes should be excluded. The heat rate deviation for makeup is an approximation, as the location in the cycle of each loss is not known, therefore the exact heat rate deviation is not known. Losses from main steam are the most expensive, because so much energy has been added to it. Losses from LP heaters are relatively inexpensive. (Some parts of the total makeup may be known and measurable, such as sootblowing. For these areas, specific losses can be calculated, based on correction factors from thermodynamic modeling software.) Typically, an assumption is made that the loss is from the boiler drum, half at saturated liquid condition and half at saturated vapor.

3.2.1.5 Turbines

The loss due to reduced turbine section efficiency is calculated for each turbine that operates entirely in the superheated steam region. This can be done by continuously calculating each section's efficiency (from station temperatures and pressures) and comparing to an "expected" value. However, it is more common to periodically test the turbines (usually at valves wide open) and to compare that efficiency to the expected VWO efficiency, and to assume that the heat rate deviation is constant over the load range. The heat rate correction factors for turbine efficiency are usually obtained from the manufacturer, or they can be calculated with the use of a thermodynamic model of the plant. The actual deviation can be due to various factors, depending on the cause of the efficiency loss. If the steam path is worn or rough, the exhaust temperature increases, so less energy per kilogram of steam flow is converted to mechanical work. In the HP turbine this loss is partially offset because less energy will be required to be added in the reheater. This small gain can also be lost if the amount of reheat attemperation must be increased. If the

stage pressures are also affected, then there are additional effects due to change in extraction flow rates. Usually heat rate deviations are calculated for the following sections:

a) High Pressure Section Efficiency
b) Intermediate Pressure Section Efficiency
c) Reheat Section Efficiency

3.2.2 Boiler Cycle

3.2.2.1 Boiler Differential Pressure

For once through boilers (boilers without a drum), the pressure drop through the circuit can increase significantly, causing the boiler feedwater pumps to work harder than normal.

3.2.2.2 Combustibles in Bottom Ash

Periodically, ash samples can be collected from the bottom ash and the carbon content of the ash determined. The unburned carbon represents chemical energy that was supplied to the boiler, but was not converted to thermal energy, extracted, rather it was lost. To convert this to a heat rate deviation some assumption about how much ash goes to fly ash and how much ash goes to bottom ash must be made. Typically, for pulverized coal fired unit it is assumed that 80 to 90% of the ash in the coal goes to fly ash and 20 to 10% of the ash goes to bottom ash. Because of the assumption that only 15% of the ash goes to bottom ash and because of problems with collecting representative samples, this parameter is frequently ignored. If the 50 mesh fineness from pulverizer testing is not good, this ash stream should be sampled periodically.

3.2.2.3 Combustibles in Fly Ash

As with bottom ash, fly ash can be sampled, and analyzed to determine the carbon content. Two important considerations in fly ash sampling are the location and the frequency. As the fly ash leaves the convection pass, some ash is usually collected on hoppers between the economizer and the air preheater. There are additional hoppers collecting some ash after the air preheater. Most of the ash is collected in the precipitators or baghouse. At any time the carbon content of the ash at any of these locations can vary considerably. The preferred method to ensure that the ash collected is representative of the fly ash leaving the convection pass is to collect the ash at that location. The drawback to this method is that this area is extremely hot and it is not easy to collect the sample here. For this reason, many plants collect their fly ash samples further downstream. If this is done, ash samples should be periodically collected from both places and the relationship between the two sampling ports determined and applied to subsequent samples drawn at the downstream location. For example, at the economizer outlet the fly ash may be 3% carbon. At the same time the % carbon in the ash collected at the first precipitator hopper is 2%. Until the next comparison is made, the % carbon for the unit can be determined by increasing the % carbon from the first precipitator hopper by 50%.

The second consideration is the sampling frequency. The method of choice is continuous sampling. As mills are removed from or placed in service, burner tilts moved, etc., the carbon content of the ash can change.

3.2.2.4 Exit Gas Temperature

The exit gas temperature is one of the two parts of dry gas loss, the energy that is lost out the stack. The temperature of the flue gas leaving the air preheater is measured. The temperature at this location is stratified, therefore it is important to measure the temperature at enough locations to get a representative temperature. Normally the Total Air is measured at the air preheater inlet. Since both the quantity of gas and its temperature are used in calculating this loss, the standard practice is to use the gas flow based on the O_2 at the AH inlet and correct the measured exit gas temperature to the temperature that would have been measured had there not been any leakage or the "EGT - no leakage." This is further corrected to account for the difference between the actual inlet air temperature and some constant "reference" inlet air temperature. It is this "EGT-no leakage, corrected to reference air temperature" that is used to calculate the heat rate deviation.

3.2.2.5 Total Air at the Boiler Outlet/Air Heater Inlet

The other part of dry gas loss is the quantity of gas that leaves the boiler. Normally, rather than directly measuring the amount of gas flow, it is calculated from the O_2 concentration of the gas and the coal analysis. Due to air in-leakage in the convection pass, the percentage of oxygen in the flue gas varies from point to point in the duct. There are two methods used to determine the representative O_2 in the duct. First, multiple probes can be used. (An advantage of this method is that if one probe fails, there are other probes in the duct supplying the operator with information to run the unit.) The second method is to periodically traverse the duct sampling the flue gas at multiple locations, at different loads or mill combinations. For each test, determine the average O_2 in the duct, and locate the station probe at a location that represents the duct average under the different load or mill combinations.

3.2.2.6 Fuel Characteristics

As the amount of moisture in the fuel increases, the boiler efficiency decreases as some of the heat released goes into heating and vaporizing the moisture. Also, as the hydrogen content of the fuel increases, the boiler losses increase as the water is formed from the burning of the hydrogen. At some plants these two losses are treated as heat rate deviations, i.e., there is an expected level for each which is compared to the actual value and the difference is converted to a heat rate deviation. At other plants, this is treated as uncontrollable, as is the condenser circulating water temperature, and the expected values for moisture and hydrogen are continually set equal to the actual value and the expected heat rate changes with the fuel analysis.

a) Moisture
b) Hydrogen

3.2.2.7 Throttle Pressure

If the unit is operating at a pressure at the stop valves different from the "expected", there is a heat rate penalty. This is due to the fact that the higher the pressure the higher the available energy. For a unit operating under "Full Pressure", the "Expected" value is almost always the constant "Design" pressure of the unit. (Occasionally a unit may permanently lower its throttle pressure due to reliability issues, then their "expected" value would be some value less than the "design" value.) For a unit operating under "Variable Pressure", the "expected" value will vary as a function of the steam flow to the turbine.

3.2.2.8 Throttle Temperature

As with the throttle pressure, this is measured just before the turbine stop valves. If there are multiple lines between the superheater outlet and the turbine, the temperature should be measured in each, as there can be differences pipe to pipe.

3.2.2.9 Hot Reheat at Intercept Valve (HRHIV) Temperature

Because there is usually some small temperature drop between the hot reheat outlet and the turbine, this temperature should be measured as close to the IP valves as possible. As with the throttle temperature, it should be measured in each pipe.

3.2.2.10 Pulverizer Pyrite Loss

To determine the heat rate of the unit the coal flow is usually measured before the pulverizers. Some material, called pyrites, is removed from the pulverizer. If the mill is operating properly, almost all of this material is rock, not coal. However, some mills will "spill coal" or a significant portion of the flow out the pyrite gate will be coal, resulting in a heat rate deviation. If the quantity of material removed is measured, and if the heating value of the pyrites are determined periodically, this loss can be determined.

3.2.2.11 Reheat Attemperation

Reheat attemperation (expressed as a percent of Main Steam Flow), used to control the hot reheat temperature, is normally determined by measuring the pressure drop across a flow orifice. This flow rate is then divided by the feedwater or main steam flow and expressed as a percentage. Reheat attemperation hurts the unit heat rate two ways. First, the flow bypasses the high pressure feedwater heaters (and therefore reduced the extraction flows) and second, because that flow did not pass through the high pressure turbine.

3.2.2.12 Reheater Pressure Drop

The pressure drop through the reheater causes a heat rate deviation. During performance guarantee tests, this is usually measured. However since it rarely changes, for routine monitoring, it is usually ignored.

3.2.2.13 Start-up Fuel

Normally, the reported heat rate of a unit is calculated using the entire quantity of fuel that is supplied to the unit during the time in question (this includes fuel burned when on-line, as well as fuel burned during startups). In order to account for the heat rate deviation due to fuel burned prior to synchronization, the quantity of fuel burned before the unit is synchronized is measured. If this deviation and the deviation for "Station Service (Prorated) - Idle" are subtracted from the actual heat rate, the result is the "operating" heat rate.

3.2.2.14 Superheat Attemperation

Superheat attemperation (expressed as a percent of Main Steam Flow), used to control the boiler outlet temperature, is normally determined by measuring the pressure drop across a flow orifice. This flow rate is then divided by the feedwater or main steam flow and expressed as a percentage. If the flow is taken downstream of the high pressure heaters, the heat rate deviation is negligible. If the flow is taken from the BFP discharge, then there is a heat rate deviation because the flow bypasses the high pressure feedwater heaters.

3.2.3 Auxiliary and Station Power

The heat rate deviation due to auxiliary power usage is divided into three areas: unit auxiliary power, station service-operation, and station service-idle.

3.2.3.1 Unit Auxiliary Power

This is the deviation due to higher than expected unit board (auxiliary) power usage. This source of power only applies during the time when the unit is on line. This is the power supplied directly to major auxiliaries for a single unit.

3.2.3.2 Station Service - Operating

This second deviation is due to prorated station service power usage while the unit is on line. This includes power used by the coal and ash handling equipment, makeup water plant, office and maintenance shops, etc.

3.2.3.3 Station Service - Idle

The third deviation is due to prorated station service power usage while the unit is off line. When the unit is off line, in addition to being charges for "common" equipment, the unit's pumps, fans, mills, etc. are fed from this source.

3.2.4 Unaccountables

The last category of losses are called "unaccountables." As the name suggests, it is not truly measured, but is the difference between the actual heat rate and the sum of the expected heat rate and all "accountable" heat rate deviations. There are two types of problems that contribute to this area. First is true losses that are not able to be measured, such as low pressure turbine efficiency, cycle isolation, boiler radiation losses, low pressure feedwater heater performance, etc. The second types of problems are errors. All errors end up in the unaccountable heat rate deviation group. Examples of such errors are coal flow measurements, coal heating value measurements, any instrument errors that are measuring parameters such as throttle temperature, condenser pressure or boiler outlet O_2, etc. Other types of, errors include calculation procedure errors, i.e. the calculated heat rate deviation of a 2% point change in HP turbine efficiency could be different from the true heat rate deviation. Because the unaccountable category includes real problems that should be addressed, unaccountables cannot be ignored. But because it can be misleading due to any errors affecting its value, it should not be the primary focus of a heat rate improvement program. A very rough rule-of-thumb is that the heat rate deviation for unaccountables should be less than 100 kJ/kWh or 25 kcal/kWh.

3.2.5 Categories of Parameters

Frequently, when analyzing heat rate deviations, they are divided into two categories, accountables and unaccountables. The accountables are then sub-divided into two groups, operator controllable and uncontrollable. This heading is somewhat misleading in that parameters that are listed in the operator controllable group are not *entirely* in his control, and some items that are normally listed in uncontrollable are somewhat in his control. Parameters normally considered as operator controllable are:

a) Condenser pressure
b) Final feedwater temperature
c) Combustibles in ash
d) Exit gas temperature
e) Total air (O_2) at the boiler outlet
f) Throttle temperature
g) Throttle pressure
h) Hot reheat at intercept valve temperature
i) Reheat attemperation flow
j) Superheat attemperation flow
k) Unit auxiliary power

Heat rate deviations for these parameters are frequently calculated in real-time (except for combustibles in ash that is not easily measured in real-time) and displayed to the unit operator. The operator can then see the effect of varying one parameter on the other parameters, as well as on the total heat rate deviation for these parameters and try to control the unit so that the total heat rate deviation is minimized.

3.3 What "Baseline" Should Be Used To Compare With The Actual Performance?

To calculate a heat rate deviation for a parameter, its actual value must be compared to some standard. There are several options that may be used. They are design, performance guarantee test, historical operating or test data, benchmarking similar units, and current "expected"

3.3.1 Design

One baseline that the current performance of the unit could be compared to is the "design" heat rate (or heat rate curve). From the point of view of setting up the tracking system, this is the easiest baseline to obtain. However, there are several problems with this method:

a) The actual initial performance of the unit may have never reached the design, or even if it did, the performance of some areas may have been better, while others were worse.
b) The "design" heat rate is based on some ambient conditions that vary during the year. Frequently the design heat rate is calculated at some design condenser pressure (or inlet water temperature). During some parts of the year, the condenser pressure should be lower than the design, therefore the heat rate should be lower than the design.
c) Over time, as changes are made to the plant, the "original design" may not represent the "current design" performance of each parameter or the overall heat rate.

3.3.2 Performance Guarantee Test

A better baseline to use would be the performance guarantee (acceptance) test results. It represents the performance level that was achieved when the unit was new, and as such, it is a level that should be achievable. (It may not be economically feasible to return all equipment to this level, but it is possible.) There are three problems with this method:

a) While the results of the acceptance test are achievable, they may not be the best possible. For example, the vendor is usually only interested in demonstrating that the guarantee can be met. Sometimes, the performance can be improved by minor adjustments to operating set points, to achieve even better performance.
b) The operating conditions under which the acceptance test were run may not be possible for everyday operation. Frequently, acceptance tests are run with the makeup isolated, no soot blowing, no sluicing, etc.
c) Over time, as changes are made to the plant, the "original design" may not represent the "current design" performance of each parameter or the overall heat rate.

3.3.3 Historical Operating/Test Data

Sometimes, a unit (or group of similar units) did not have an acceptance test. For those units, historical operating/test data, taken from a certain time period when the unit operation was considered to be "good", can be reviewed to determine what level of performance is possible. A large amount of data should be collected when the unit is operating at steady state conditions, and using some "engineering judgment" a curve can be drawn through the "better" points. There are two primary problems with this method:

a) The available operating data may not represent the best achievable performance
b) The "best" data for one parameter (which occurred at one point in time), when combined with the "best" data for another parameter (which occurred at a different point in time), may not be achievable together, or may not result in the best overall unit performance. For example, for some load, at some time, the amount of combustible matter in the fly ash may have been very low. At some other time the amount of excess air at the same load may have been low. However these two levels cannot occur at the same time.

3.3.4 Benchmarking Similar Units

A fourth method that can be used is similar to the previous method but, instead of looking at the performance of a particular unit, all similar units in the utility or in the industry are surveyed. Then operating or test data from the better or best performing unit(s) is reviewed and used to set expected levels. Potential problems with this method include the two problems mentioned above. Also, care must be taken when picking units to use. Just because the design of a unit is the same as another, other factors can cause differences in their performance. An obvious example would be the type of coal being burned and ash characteristics. A low ash fusion temperature coal will be fired differently, and the expected exit gas temperatures, total air, etc., for the two should be different. Another example is the condenser cooling water source at one location may allow that plant to maintain a certain cleanliness factor, whereas the water at another location may cause faster fouling and a lower average cleanliness factor.

3.3.5 Current "Expected"

The last method, and the one that is recommended, is to compare the actual current performance to an achievable "expected" value, based on the current configuration of the unit. To do this, start with the actual heat rate curve when the unit was new (typically from the performance guarantee test). Tabulate, from the performance test data, the value of each parameter (Throttle temperature, exit gas temperature, reheat attemperation flow, etc.) as a function of gross load or steam flow. These are called reference curves. If a permanent modification is made to the plant (such as adding additional air preheater baskets, removing reheat surface, retubing feedwater heaters, etc.), then a new curve is drawn for whatever parameters are affected, showing the new "expected" parameter value (i.e. if new air preheater baskets are added, the expected exit gas temperature curve might be 8°C lower than the reference curve). In this method, there is no expected heat rate curve. Instead, for each parameter, the heat rate deviation due to the

difference between the reference parameter curve and the expected parameter is calculated, and added to the reference heat rate curve, to obtain the expected heat rate. In this method, when some change is made to the plant, instead of having to develop new expected heat rate curves, only the affected parameter curves need to be modified, and that is a much easier task.

To calculate heat rate deviations for each parameter, the actual parameter value is compared to the "expected" parameter value. These expected parameter values are maintained as modifications are made to the unit, so the comparison is always to a realistic and achievable value.

Questions that frequently arise are, "When should an expected parameter level be adjusted?" and, "Should the expected levels be routinely changed to take into account "normal" degradation?" Usually, expected parameter curves should be changed only when there is a permanent change to the unit. One reason why this is done differently at different locations, is that heat rate deviations are used two ways. First, it may be used by operating personnel to attempt to operate the unit in the most efficient manner. They typically believe expected parameter values should be changed to reflect the current condition of the equipment. For example, if the air preheater is corroded, nothing they do will allow them to reach the "expected" value for exit gas temperature, so they believe it should be raised. However, another use of the data is to look at the heat rate deviations and to see to what extent the current condition of the equipment has degraded, so appropriate actions can be taken (if economically justified). If the expected level is changed, the fact that there is a potential for improvement (and the amount of the potential improvement) by cleaning or replacing baskets is lost. To account for the first group of users, "target" parameters can be set, and the heat rate deviations due to difference between the actual parameters and the "target" parameters can be calculated and tracked. See Section 5 for a discussion on how targets are set.

The following are some examples:

a) The high pressure turbine efficiency has dropped 2% points, should the expected HPT efficiency curve be changed? Normally not, as this is a potentially recoverable loss. Some of the loss may not be economically recoverable, but as fuel costs change, what is not economical today may be economical tomorrow. For this reason, heat rate deviations that are not currently economically recoverable should not be "hidden" by changing the expected levels.
b) The high pressure feedwater heaters are retubed with a different material for reliability concerns. The new material has a poorer heat transfer coefficient, resulting in a final feedwater temperature 2°C lower. For this case the expected final feedwater temperature curve would be lowered 2°C, as this is a "permanent" change.
c) Because of frequent boiler tube failures, the throttle (and/or hot reheat) temperature has been lowered from 540°C to 530°C. Normally the expected steam temperature would not be lowered because this heat rate deviation could be recovered if the tubes that are failing are replaced.

3.4 Overview of Methodology

Having decided that it is desirable to compare the current actual performance of a unit to a performance level that could be reached with the present ambient conditions (air and water temperature), generation level, and plant configuration, the first step is to determine the current "expected" heat rate (and parameter values). This could be done by maintaining a set of expected heat rate curves. This would require that these curves be updated whenever a change is made (i.e., air heater baskets are modified, the condenser tubes are retubed with a different material or wall thickness, etc.).

Instead of continuously modifying heat rate curves, a better method is to calculate the expected heat rate using several independent curves. The first curves that must be developed are a family of "reference" heat rate curves. Usually three or four curves make up this family. Each curve in the family represents the net unit heat rate of the unit when it was new, corrected to a design ambient air temperature. (Normally data from the Performance Guarantee Test is used to make these curves.) A family of curves is developed, each curve representing the net unit heat rate corrected to one condenser circulating water temperature. Usually one curve is corrected to the normal minimum CCW inlet temperature that occurs at the site, another curve is corrected to the maximum CCW inlet temperature, and one or two others are developed for CCW inlet temperatures between the minimum and maximum. (Another choice of temperatures could be the average CCW inlet temperature during each season of the year). Once these curves are developed, they are never changed.

Another set of curves is developed from the Acceptance/PG test data. One plot is made for each parameter (for which heat rate deviations are to be determined, such as total air at the boiler outlet and reheat attemperation flow). Using the data from each test, the value of each parameter is plotted against either gross generation or main steam/feedwater flow. These curves are the "reference" parameter curves which, like the "reference" heat rate curves, are never changed.

Once the "reference" parameter curves are developed, they are duplicated, and the duplicate curves are called "expected" parameter curves. Unlike the reference parameter curves that are never changed, the expected parameter curves may be modified. Whenever a change is made to the plant (i.e. additional AH baskets are added, additional precipitator fields are added, etc.) the "expected" parameter curve for the appropriate parameter is modified. (For example, when additional AH baskets are added, the "expected" EGT curve will be lowered somewhat, or if additional fields are installed in the precipitator, the "expected" station service curve will be raised.) Note that just the "expected" parameter curves are modified, not the "reference" heat rate or the "reference" parameter curves. For some modifications, more than one expected parameter curve may change. For example if new burners are installed, the "expected" parameter curves for CombFA, EGT, TA, TT, SHA, HRHIV, and RHA could all change. In general, the "expected" parameters are single curves, plotted against either main steam flow or gross load. However, a family of curves can be made for an "expected" parameter. For example, station service could be plotted against main steam flow, with multiple curves for different grades of coal.

To calculate the "expected" heat rate, the reference heat rate is first calculated. Then, for each parameter, heat rate deviations are calculated due to the difference between the "expected" parameter value and the "reference" parameter value. The "expected" heat rate is calculated by

adding the "reference" heat rate to the sum of the heat rate deviations due to differences between the "reference" parameter values and the "expected" parameter values. Note that there is no "expected" heat rate curve, instead an "expected" heat rate is a calculated value, using the family of "reference" heat rate curves, the set of "reference" parameter curves, and the set of "expected" parameter curves.

To calculate the heat rate deviation for most parameters, the difference between the "actual" parameter value and the "expected" parameter value is multiplied by a heat rate correction factor (% change in heat rate/100/unit change in parameter), and multiplied by the expected heat rate. For some parameters (such as boiler losses and condenser) a more complicated calculation procedure is normally used. However, in place of these detailed calculations, "rules of thumb" such as those listed in Appendix D (i.e. 1°C exit gas temperature is worth 1.2 kcal/kWh) can be used. The results from this simplified method will not be as accurate as the detailed procedure that follows, but where computer systems are not available, this method may be a better option.

The heat rate correction factors are obtained from various sources, including vendors thermal kits, thermodynamic modeling programs, and calculation procedures.

A sign convention to follow is that positive heat rate deviations indicates an increase in heat rate, resulting in poorer performance. A negative heat rate deviation indicates a decrease in heat rate, resulting in better thermal performance.

Once the procedure is setup, i.e., the required curves are developed, (family of reference heat rate curves, reference parameter curves, and expected parameter curves) and the calculation procedure is laid out, these are the general steps to follow:

1) Collect actual operating data for the time period (weekly, daily, hourly, etc.)
   - A) Parameter values (condenser pressure, exit gas temperature, station use, etc.)
   - B) Miscellaneous operating data, such as unit load, CCW temperature, ambient air temperature, fuel analysis, etc.
2) Calculate reference net unit heat rate ($NUHR_{Ref}$) for the period in question given average gross load ($GLoad_{Act}$) and condenser cooling water temperature ($CCWTI_{Act}$)
   - A) For the period in question, calculate the reference heat rate at each of the 3 or 4 seasonal CCW temperatures, at the periods average gross load. ($NUHR_{Ref,S1}$; $NUHR_{Ref,S2}$; $NUHR_{Ref,S3}$)
   - B) Perform a spline fit from the three or four seasonal heat rates to determine the reference heat rate at the periods gross generation and the period's actual inlet water temperature.
3) Using the given period's gross load or feedwater flow, for each parameter, determine its reference value from the reference value curves ($CombFA_{Ref}$, EGT_{Ref}, RHA_{Ref}, TT_{Ref}, TP_{Ref}, etc.).

4) Using the given period's gross load or feedwater flow, for each parameter, determine its expected value from the expected value curves ($CombFA_{Exp}$, EGT_{Exp}, RHA_{Exp}, TT_{Exp}, TP_{Exp}, etc.).

5) For each parameter (CombFA, EGT, RHA, TT, TP, etc.) calculate the heat rate deviation due to the difference between the expected parameter value and the reference parameter value. ($HRD_{CombFA,\ Exp\text{-}Ref}$, $HRD_{EGT,\ Exp\text{-}Ref}$, $HRD_{RHA,\ Exp\text{-}Ref}$, $HRD_{TT,\ Exp\text{-}Ref}$, $HRD_{TP,\ Exp\text{-}Ref}$, etc.).

6) Calculate the expected heat rate as the reference heat rate plus the sum of the heat rate deviations for all parameters, due to differences between the reference parameter values and the expected parameter values:

$$NUHR_{Exp} = NUHR_{Ref} + HRD_{CombFA,\ Exp\text{-}Ref} + HRD_{EGT,\ Exp\text{-}Ref} + HRD_{RHA,\ Exp\text{-}Ref} + HRD_{TT,\ Exp\text{-}Ref} + HRD_{TP,\ Exp\text{-}Ref} + \ldots$$

7) Calculate the total heat rate deviation, as the difference between the actual net unit heat rate for the period, and the expected net unit heat rate for the period:

$$HRD_{Tot,\ Act\text{-}Exp} = NUHR_{Act} - NUHR_{Exp}$$

8) For each parameter (CombFA, EGT, RHA, TT, TP, etc.) calculate the heat rate deviation due to the difference between the actual parameter value and the expected parameter value. ($HRD_{CombFA,\ Act\text{-}Exp}$, $HRD_{EGT,\ Act\text{-}Exp}$, $HRD_{RHA,\ Act\text{-}Exp}$, $HRD_{TT,\ Act\text{-}Exp}$, $HRD_{TP,\ Act\text{-}Exp}$, etc.).

9) Calculate the total accountable heat rate deviation by summing the heat rate deviations for each parameter:

$$HRD_{Acc,\ Act\text{-}Exp} = HRD_{CombFA,\ Act\text{-}Exp} + HRD_{EGT,\ Act\text{-}Exp} + HRD_{RHA,\ Act\text{-}Exp} + HRD_{TT,\ Act\text{-}Exp} + HRD_{TP,\ Act\text{-}Exp} + \ldots$$

10) The unaccountable heat rate deviation, which includes the deviation due to parameters that are not tracked (i.e. LP turbine efficiency, cycle isolation, etc.) and any errors in parameter values, quantity of fuel burned, the higher heating value of the fuel, etc. is calculated by taking the difference between the total heat rate deviation and the total accountable heat rate deviation:

$$HRD_{UnAcc,\ Act\text{-}Exp} = HRD_{Tot,\ Act\text{-}Exp} - HRD_{Acc,\ Act\text{-}Exp}$$

11) Calculate the cost of the heat rate deviation due to the difference between the actual and the expected value for each parameter. To do this, multiply the heat rate deviation (in kcal/kWh) times the fuel cost times the actual generation (in kWh) .

CALCULATION OF HEAT RATE DEVIATIONS

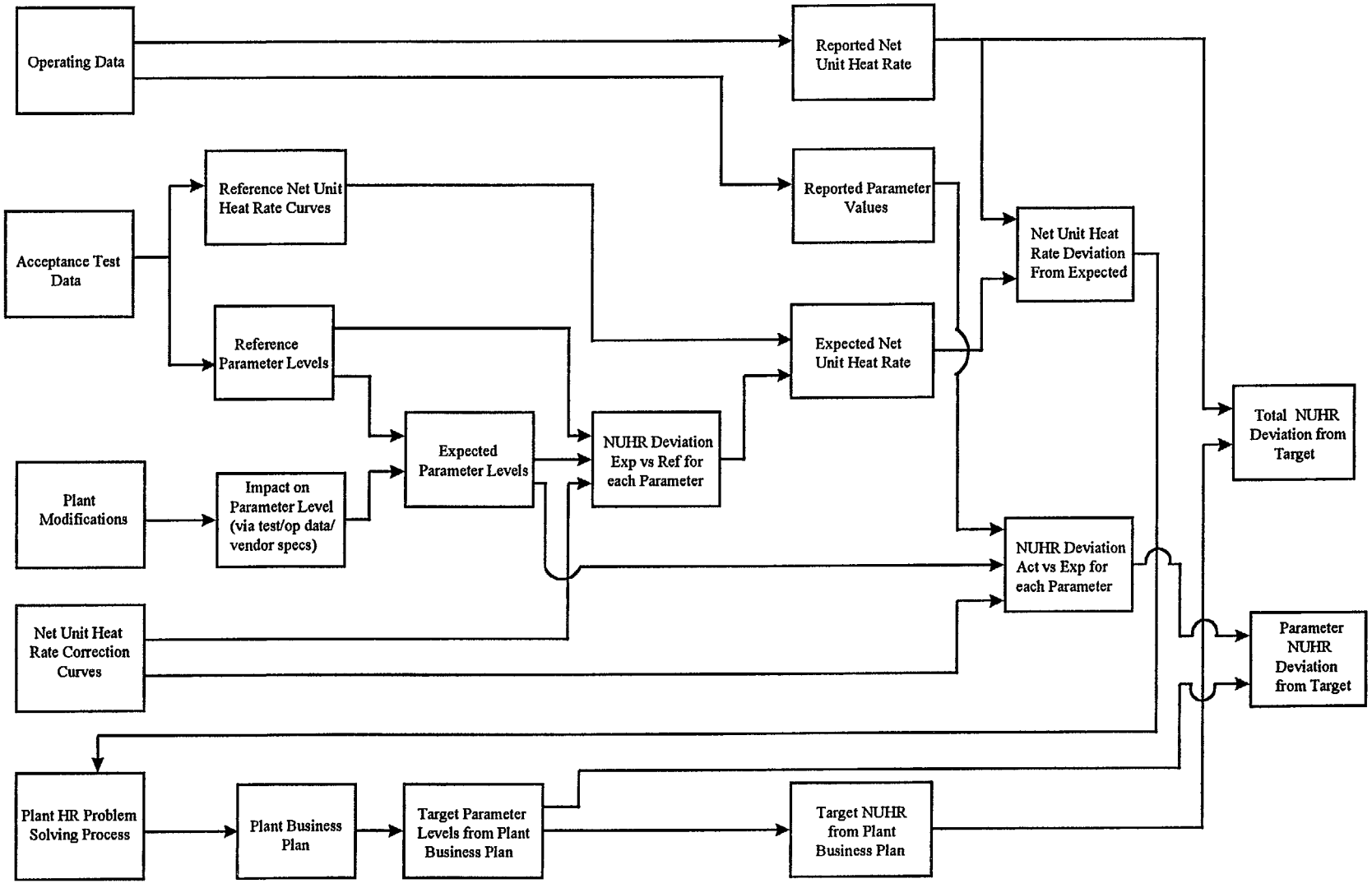

Figure 3.4 Heat Rate Deviation Accounting Flowchart

3.5 Data To Be Collected To Set Up the Procedures

The following data (constants and curves) is required to be collected once. They will be used, along with operating data from a unit) to calculate the heat rate deviations for various parameters.

3.5.1 Condenser Characteristics Database

This database includes both "reference" (the values at the time the reference curves were established, once determined, they are never changed) and "expected" (the current configuration, initially, the same values as the "reference", but as modifications are made, the expected values and curves should be changed) values for the following:

a) External diameter of tubes ($TubeOD_{Ref}$, $TubeOD_{Exp}$) in mm.
b) Internal diameter of tubes ($TubeID_{Ref}$, $TubeID_{Exp}$) in mm.
c) Tube wall thickness ($TubeBWG_{Ref}$, $TubeBWG_{Exp}$) in BWG.
d) Number of tubes ($TubeNo_{Ref}$, $TubeNo_{Exp}$)
e) Tube Material ($TubeMat_{Ref}$, $TubeMat_{Exp}$)
f) Heat Exchange Institute Material & Gauge Correction Factor ($TubeMGC_{Ref}$, $TubeMGC_{Exp}$)
g) Surface Area of Condenser ($CSArea_{Ref}$, $CSArea_{Exp}$) in m^2. The surface area is calculated by multiplying the outer circumference by the "effective" length of the tubes. The effective length of the tube is the actual length minus two times the tube sheet thickness. The part of the tubes covered by the tube supports is not excluded.
h) Circulating Water Flow Rate in kg/h. (This may be constant value, or it may be a curve as a function of the elevation of the water source ($CCWF_{Ref}$ & $CCWF_{Exp}$).
i) Cleanliness factor in percent (CF_{Ref}) This value is the cleanliness factor that can be maintained with a reasonable amount of effort. Typically 85% is used.

3.5.2 Turbine Cycle Database

a) From T/G performance guarantee test, a curve of gross turbine cycle heat rate and net turbine cycle heat rate, kcal/kWh (corrected to design condenser pressure) verses gross load, MW is developed. ($GTCHR_{REF, CorDBP}$)
b) Heat added by Boiler Feedwater Pumps, in kcal/h, this may be a constant, or more commonly, a curve as a function of load or feedwater flow, (PumpQ)
c) Heat rejected in auxiliary condensers (if any), in kcal/h, a curve as a function of load or feedwater flow, (AuxDuty)
d) Reference and expected parameter values or curves for:
   i) Throttle temperature (°C) (TT_{Ref}, TT_{Exp})
   ii) Throttle pressure (kg/cm^2) (TP_{Ref}, TP_{Exp})
   iii) Steam temperature at the IP turbine intercept valve (°C) ($HRHIV_{Ref}$, $HRHIV_{Exp}$)
   iv) Final feedwater temperature (°C) ($FFWT_{Ref}$, $FFWT_{Exp}$)

v) Turbine section efficiencies (%) (HPT_{Ref}, HPT_{Exp}; IPT_{Ref}, IPT_{Exp}; etc.)
vi) Makeup water flow (%) (MU_{Ref}, MU_{Exp})
vii) Unit Auxiliary Power (MW) (UAP_{Ref}, UAP_{Exp})
viii) Station Service (Prorated) Operating (MW) (SSO_{Ref}, SSO_{Exp})
ix) Superheat attemperation flow (%) (SHA_{Ref}, SHA_{Exp})
x) Reheat attemperation flow (%) (RHA_{Ref}, RHA_{Exp})
xi) Steam Supplied to APH coils (%) ($SCAPH_{Ref}$, $SCAPH_{Exp}$)

3.5.3 Boiler Database

The boiler database contains both "reference" (at the time of the reference curves were established) and "expected" (the current configuration) values for:

a) Coal Analysis
   i) Carbon (fraction) ($UltC_{Ref}$, $UltC_{Exp}$)
   ii) Hydrogen (fraction) ($UltH_{Ref}$, $UltH_{Exp}$)
   iii) Oxygen (fraction) ($UltO_{Ref}$, $UltO_{Exp}$)
   iv) Nitrogen (fraction) ($UltN_{Ref}$, $UltN_{Exp}$)
   v) Sulfur (fraction) ($UltS_{Ref}$, $UltS_{Exp}$)
   vi) Ash (fraction) ($FuelA_{Ref}$, $FuelA_{Exp}$)
   vii) Moisture (fraction) ($FuelTM_{Ref}$, $FuelTM_{Exp}$)
   viii) Higher Heating Value (kcal/kg) (HHV_{Ref}, HHV_{Exp})

b) Air Heater
   i) No Leakage, Exit Gas Temperature (°C) (corrected to reference inlet air temperature) ($EGT_{Ref, CorTNL, CorRAT}$, $EGT_{Exp, CorTNL, CorRAT}$)
   ii) Leakage (%) (this could be a constant or a curve as a function of flow) (AHL_{Ref}, AHL_{Exp})

   Leakage is defined as 100*the mass of wet air leaking to the gas side divided by the mass of wet gas entering the air heater. However, it is usually approximated by one of the two equations using measurements of gas analysis, by volume, on the gas side:

   % Leakage = 90* (%CO_2 entering - %CO_2 leaving) / %CO_2 leaving

   or a slightly less accurate method:

   % Leakage = 100* (%O_2 leaving - %O_2 entering) / (20.95-%O_2 leaving)

   iii) Air Heater Air Outlet Temperature (°C) ($AHAOT_{Ref}$, $AHAOT_{Exp}$)
   iv) Air Heater Gas Inlet Temperature (°C) ($AHGIT_{Ref}$, $AHGIT_{Exp}$)

c) Boiler Outlet O_2 (%) (this would be a curve as a function of flow) ($BOO2_{Ref}$, $BOO2_{Exp}$)

d) Boiler Outlet Sum of O_2 and CO_2 (%) ($BO2CO2_{Ref}$, $BO2CO2_{Exp}$)

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e) Combustibles in Ash (%) ($CombTA_{Ref}$, $CombTA_{Exp}$)
f) Moisture in Inlet Air (kg/kg) (Wma_{Ref}, Wma_{Exp})
g) Miscellaneous Boiler Losses (%) ($BLMisc_{Ref}$, $BLMisc_{Exp}$)
h) Boiler Efficiency (%) (a reference curve only, not expected) (BE_{Ref})
i) Air Inlet Temperature (°C) (a reference value only, not expected) ($AirT_{Ref}$)
j) Pulverizer Reject rate (kg rejected per kg of as-fired fuel) (Wre_{Ref}, Wre_{Exp})
k) Pulverizer Reject Heating Value (kcal/kg) (Hr_{Ref}, Hr_{Exp})

3.5.4 Miscellaneous Unit Characteristics Database

a) Use the two curves developed above (Section 3.5.2 step a - reference net turbine cycle heat rate corrected to design condenser pressure and Section 3.5.3 step h - reference boiler efficiency) to obtain a net unit heat rate curve corrected to design condenser pressure.
b) Reference condenser circulating water temperatures (3 or 4) to be used with the reference heat rate curves ($CCWTI_{Ref,\,S1}$, $CCWTI_{Ref,\,S2}$, $CCWTI_{Ref,\,S3}$)
c) From this net unit heat rate corrected to design condenser pressure curve, develop 3 or 4 curves, for 3 or 4 condenser cooling water temperatures ($NUHR_{Ref,S1}$; $NUHR_{Ref,S2}$; $NUHR_{Ref,S3}$).
d) Typical heat input supplied during a cold startup up to synchronization (in kcal) (HICold).
e) Typical heat input supplied during a hot startup up to synchronization (in kcal) (HIHot).
f) Turbogenerator manufacturer's generator loss curve, plotted as total losses, kW (mechanical and electrical) versus gross load, MW. ($GLoss_{Ref}$)
g) Curve fit the HEI Inlet Water Temperature Correction Factor (HEITCF), as a function of the inlet water temperature (CCWTI)

3.5.5 Heat Rate Correction Curves

For each parameter (excluding exit gas temperature, boiler outlet O_2, combustibles in ash, moisture in fuel, pulverizer rejects, and hydrogen in fuel), a heat rate correction curve or family of curves of percent change in net heat rate versus load or main steam flow (possibly with multiple curves for different parameter values) or % change in net heat rate versus the parameter value (possibly with multiple curves for different loads or steam flows). For example, the condenser pressure curves are usually plotted as a family of curves (each curve being for a different gross load or main steam flow) of percent change in net heat rate (%) plotted against condenser pressure (mm Hg). At the "design" condenser pressure, the curve or curves will pass through 0. At higher condenser pressures, the % change in heat rate will be positive values, and at pressures below "design," the values will be negative.

3.6 Operating Data to be Collected to Calculate Heat Rate Deviations

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3.6.1 Parameter Values

In order to calculate heat rate deviations and costs for various parameters (exit gas temperature, condenser pressure, reheat attemperation flow, etc.), the value for these parameters must be known:

a) Auxiliary Steam Usage to APH Coils (SCAPH) (% or kg/h or hours)
b) Boiler Differential Pressure Drop (BDP) (%)
c) Combustibles in Ash (CombTA) (%) (Fly Ash (CombFA) only or a weighted average of both Fly Ash and Bottom Ash (CombBA))
d) Condenser Pressure (CP) (mm Hg)
e) AH Exit Gas Temperature (EGT) (°C)
f) Boiler Outlet O_2 (BOO2) (%)
g) Final Feedwater Temperature (FFWT) (°C)
h) Hours each possible combination of Heaters is Bypassed
i) Throttle Pressure (TP) (kg/cm^2)
j) Throttle Temperature (TT) (°C)
k) Makeup (MU) (%)
l) Reheat Attemperation Flow (RHA) (%)
m) Station Service - Idle (SSI) (MW)
n) Station Service - Operating (SSO) (MW)
o) Steam Temperature at Intercept Valve (HRHIV) (°C)
p) Superheat Attemperation (SHA) (%)
q) Turbine Efficiencies (HPT, IPT, etc.) (%)
r) Moisture In Fuel (FuelTM) (fraction)
s) Hydrogen In Fuel (ULTH) (fraction)
t) Pulverizer reject rate (Wre) (kg rejected / kg of as-fired fuel)
u) Unit Auxiliary Power (UAP) (MW)

3.6.2 Miscellaneous Values

In addition to the parameter values, other items that must be measured, calculated, or estimated include:

a) Ambient air temperature (AirT) (°C)
b) Condenser Cooling water inlet temperature (CCWTI) (°C)
c) Gross Generation in MWH (GG_{Act}) and length of time the unit was synchronized during the period in question in hours (HOU_{Act}) or Average Gross Load in MW ($GLoad_{Act}$)
d) Main Steam Flow (MSF) or Feedwater Flow (FWF) (kg/h)
e) Condenser Circulating Water Flow (CCWF) (kg/h)
f) Amount of Fuel Burned (FS_{Act}) (kg)
g) Higher Heating Value of Fuel burned (HHV_{Act}) (kcal/kg)
h) Ultimate (or Proximate) Fuel Analysis
i) Cost of Fuel (FC) (Rs/kcal)

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j) Air Heater Leakage (AHL) (%)
k) Amount of moisture in air (Wma) (kg/kg) or relative humidity
l) Heating value of pulverizer rejects (Hr) (kcal/kg)
m) Flue gas temperature entering the air heater (AHGIT) (°C)
n) Air temperature leaving the air heater (AHAOT) (°C) (if the unit has a tri-sector air heater, this is the weighted average temperature)
o) Number of Cold Startups during the period (NoCold)
p) Number of Hot Startups during the period (NoHot)

3.6.3 Data Reduction

It is preferable if all the data is collected from instrumentation fed directly to the computer system performing these calculations. A problem with this method is that erroneous data can easily slip into the calculations. This results in erroneous, and sometimes ridiculous results. This leads to disbelief in the system, and its failure to be utilized.

Data is also manually recorded and entered into the system. Two drawbacks with this are that it is labor intensive, and the calculations cannot be in real-time. Second, whereas the electronic readings are essentially continuous, manual reading may be hourly, bi-hourly, or even less frequently. As the frequency of readings decreases, the reading will be less representative of the true value for the period. This can lead to errors.

Ideally, the reading should be fed into the data reduction system electronically. If this is not feasible, then the frequency of manual reading should be as high as is reasonably feasible. Second, there should be some error checking of the raw data, before it is used in any calculations. As a minimum, the critical items should have some limits, and values out of limits should be flagged. Currently more tools are being developed and becoming commercially available, such as pattern recognition and neural nets, that can locate and substitute for, bad data.

How should multiple readings at a location be combined to a period (15 minute, hourly, shift, etc.) average to be passed to the heat rate deviation calculation procedure? The easiest method is to merely average the values (sum the readings and divide the sum by the number of readings). Another method is to weight-average the values by some other value. For example, if the Throttle temperature during a period of time varied, it is common to weight average the temperatures based on the steam flow. In this method the instantaneous parameter values at high loads (or flows) affect the average parameter values more than instantaneous parameter values at low flows. This is the method that is normally used.

Certain reading are summed or integrated, before passing them to the deviation calculation procedure. This is done for:

a) Generation (MW)
b) Auxiliary Power (MW)
c) Main Steam or Feedwater Flow (kg)

d) Attemperation Flows (kg)
e) Makeup flow (kg)
f) Fuel Supplied (kg)

3.6.4 Fuel Characteristics

Two critical values, listed in Section 3.6.2, that are used in the calculations are the amount of fuel burned, and the heating value of the fuel (items "f" and "g"). Every plant must have a regular program to maintain the coal weighing equipment in good condition and to periodically check its calibration. This includes belt scales feeding bunkers, batch coal scales, volumetric feeders, or gravimetric feeders. If there are belt scales at two or more locations, it is advisable to periodically run the same coal across both as a check. For example, if a plant has belt scales for both coal received and for coal going to the bunkers, then periodically run both belts empty (and if this is done early in the shift, run the belts long enough to warm up) and then run some amount of coal across the received scales and immediately across the bunker scale and compare the two readings. If there is a significant difference (more than 0.5%), the cause for the difference should be investigated and corrected. Another method of cross-checking is to compare the measured amount of coal received over some time period (such as three months), the change in the amount of coal in the stockpile(s), and the measured amount of coal burned. As above, if there is a significant discrepancy, the cause should be determined and corrected.

Not only is the quantity of fuel burned critical to the determination of heat rate, but the quality is equally important. In order to know the higher heating value and coal analysis of the fuel being burned, a continuous sampling program should exist at each plant. Preferably automatic samplers should be installed in the bunker room, so that all the coal going to the bunkers can be sampled according to ASTM D2234-98. Some common problems encountered with sampling include:

- Non-representative sampling, this is especially true if the samples are being collected manually. The ASTM procedures go into great detail to ensure that the sample collected is not biased by the way the sample is obtained.
- Contamination of the sample, if the sampling/analysis equipment and containers are not cleaned between samples.
- Loss of moisture between the sampling location and the analysis location. The sample containers must be sealed to prevent the loss of moisture, which will make the indicated HHV higher than it really is.

3.7 Detailed Procedures

The following are detailed procedures for developing the curves and calculating losses. Section 3.7.1 through 3.7.2 are done once for a unit. Section 3.7.3 must be repeated anytime there is a permanent change made to the unit. Sections 3.7.4 through 3.7.25 is usually done at some frequency on all units. For plants that have data acquisition systems, this step may be performed

every minute. Where it is done via manual inputs to a computer spreadsheet, it might be done each shift or daily. If done completely manually, it may only be done weekly. If it is done in real-time, there are some inputs that will have to be "estimated." For example, the coal analysis and heating value will not be known in real-time, unless extremely expensive equipment is provided.

3.7.1 Develop Reference Heat Rate and Reference Parameter Curves

Before any heat rate deviations can be calculated, a family of reference heat rate curves must be developed and a family of reference parameter level curves must be developed. This is usually done using data from a performance guarantee test. (See Section 3.3)

3.7.1.1 Tabulate the Necessary Data From Each Run of a Turbine Cycle PG Test

The data to be tabulated is the "actual" values of measured or calculated data, not the values that have been corrected to design or contract conditions. The data to be tabulated includes all parameters for which heat rate deviations are to be calculated:

a) Throttle Temperature (°C)
b) Throttle Pressure (kg/cm^2)
c) Hot Reheat Intercept Valve Temperature (°C)
d) Condenser Pressure (mm Hg)
e) Final Feedwater Temperature (°C)
f) Turbine Efficiency (%) for each section with a dry exhaust
g) Makeup Water Flow (% of FWF)
h) Unit Auxiliary Power (MW)
i) Prorated Station Service Power (MW)

Other data to be tabulated includes:

a) Average Gross Load (MW)
b) Gross Turbine Cycle Heat Rate (kcal/kWh)
c) Net Turbine Cycle Heat Rate (kcal/kWh)
d) Main Steam or Feedwater Flow (use the test measurement from whichever locations is measured by station instrumentation) (t/h)
e) Condenser Circulating Water (CCW) Inlet Temperature (°C)
f) Amount of condensate subcooling (°C)
g) LP Turbine Exhaust Steam Flow to the Condenser (t/h)
h) Temperature (°C), Pressure (kg/cm^2), and Flowrate (t/h) of the Extraction to the Lowest Pressure Feedwater Heater (also tabulate the enthalpy from the temperature and pressure (h_{EXTR} to be used in step 4, kcal/kg)
i) Drain Temperature of the Lowest Pressure Feedwater Heater (°C) and enthalpy (h_{DRAIN} to be used in step 4, kcal/kg)
j) Temperature of the Condensate Leaving the Lowest Pressure Feedwater Heater (°C) (and enthalpy $h_{HTR\ OUT}$ to be used in step 4, kcal/kg)
k) Condensate Flow to the Lowest Pressure Feedwater Heater (t/h)

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l) Heat Rejected in the Condenser from Miscellaneous Drains (kcal/h)
m) Heat Added to the Feedwater by Motor Driven BFPs (kcal/h)

The last thing to do with the Performance Guarantee data is to plot the LP Turbine expansion line on a Mollier diagram.

3.7.1.2 Tabulate the Necessary Data From Each Run of a Boiler Cycle PG Test

As in Section 3.7.1.1, the data to be tabulated is the "actual" values of measured or calculated data, not the values that have been corrected to design or contract conditions (except for the exit gas temperature, which should be corrected to the reference inlet air temperature and to no AH leakage). The data to be tabulated includes all parameters for which heat rate deviations are to be calculated:

a) Superheat Attemperation Flow (% of FWF)
b) Reheat Attemperation Flow (% of FWF)
c) Flue Gas O_2 at the Air Heater Inlet
d) Boiler Differential Pressure (kg/cm^2)
e) Combustibles in Fly Ash
f) Combustibles in Bottom Ash
g) Steam Supplied to APH Coils (kg/h)
h) Pressure Drop Through the Boiler (%)
i) AH Exit Gas Temperature (corrected to no-leakage) (°C)
j) Pulverizer Reject Rate (kg/kg)

Other data to be tabulated includes:

a) Average Gross Load (MW)
b) Main Steam or Feedwater Flow (use the test measurement from whichever locations is measured by station instrumentation) (t/h)
   Note: for control purposes, first stage pressure is often used as the indication of main steam flow. For control purposes this is permissible, but it should not be used as the accurate flow indication. This is because problems in the turbine (deposits, mechanical damage, SPE, etc.) will cause the first stage pressure to change, even though the main steam flow does not.
c) Reference Inlet Air Temperature (this will be a constant value) (°C)
d) Boiler Efficiency (100 - Sum of all losses) (%)
e) Miscellaneous Boiler Losses (Sum of small losses that will not be tracked in the heat rate deviation procedure (items such as radiation losses, moisture in air, sensible heat in ash, etc.)
f) Heating Value of Pulverizer Rejects (kcal/kg)
g) Air Heater Gas Inlet Temperature (°C)
h) Air Heater Air Outlet Temperature (°C)
i) Fuel Analysis (fraction)
j) Moisture in Air (kg/kg)

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3.7.1.3 Develop Reference Parameter Curves

Once this data is collected and tabulated, for each item that is intended to be a "Parameter" for which heat rate deviations will be calculated, plot the actual value versus gross load, or flow (main steam or feedwater) except for condenser pressure. Most parameters are more a function of flow than generation, and should be plotted that way. The curves drawn through these points become the "reference parameter curves." Usually first, second and, in some cases, third order polynomial curve fits are made. For some parameters, a single curve may not fit the data well. If so, one curve may be used to represent the data over one range, and another curve over another range. For example Hot Reheat Steam at Intercept Valve Temperature typically is broken into two curves. At high loads (perhaps 70% and above) it is a constant 538°C . But below 70% it drops off.

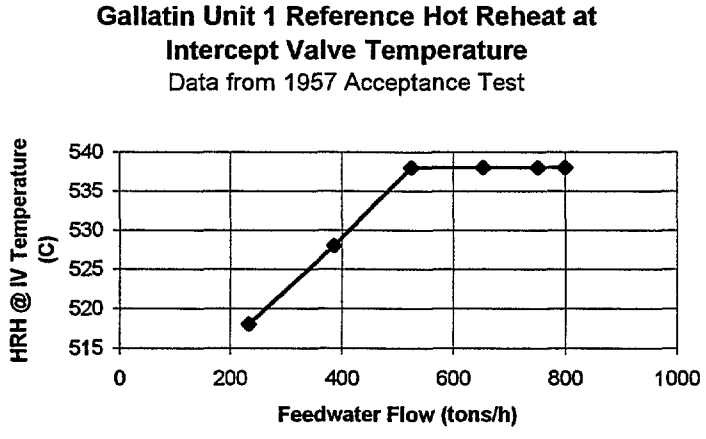

Figure 3.5 Example of Reference Parameter Curve Requiring Two Segments

Once the reference parameter curves are established, they are never changed.

In addition to plotting the "reference parameter curves", other curves that should be plotted are:

- Net Turbine Cycle Heat Rate (kcal/kWh) versus Gross Load (MW).
- Reference Boiler efficiency (%) versus flow.

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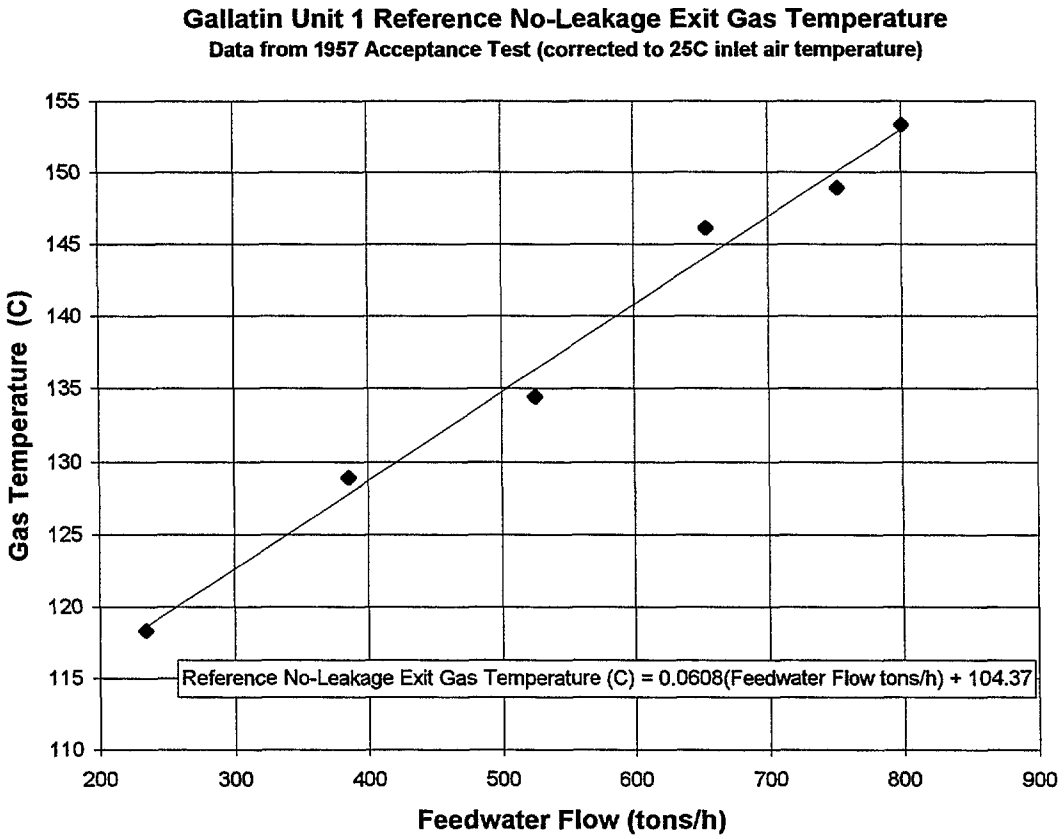

Figure 3.6 Example of Reference Parameter Curve with Equation

3.7.1.4 Develop a Family of "Reference" Net Unit Heat Rate Curves

The next step is to develop a Family of "Reference" Net Unit Heat Rate Curves (versus gross load) for 3 or 4 condenser circulating water temperatures. For each turbine cycle heat rate test, record the measured flow (main steam or feedwater). Using the curve of boiler efficiency versus flow from Section 3.7.1.2, calculate the reference boiler efficiency for that flow, and therefore the reference boiler efficiency for that turbine cycle test. Divide the net turbine cycle heat rate by this calculated (from the curve) boiler efficiency to get the net unit heat rate for that test.

Select three or four condenser circulating water temperatures. One method is to choose the average water temperature for each season. Another method is to choose the normal yearly minimum and maximum temperatures, and pick one or two temperatures between those two.

Select a 'reference" condenser cleanliness factor. Typically 85% is used. This is the cleanliness that can be maintained with a reasonable maintenance effort, and may vary from unit to unit.

Select a "reference" condenser circulating water flow rate ($CCWF_{Ref}$). This should be average flow from the performance guarantee tests.

The net unit heat rate from each performance guarantee test (calculated from step 4A) must be corrected back to the reference cleanliness factor, and to each of the selected water temperatures. If thermal modeling software is available, it can be used to do this, or it can be done by hand.

An iterative calculation is required because as the inlet water temperature changes, the condenser pressure and condensate temperature changes. The change in condensate temperature changes the extraction flow to the lowest pressure feedwater heater. This changes the LP turbine exhaust flow. The change in condenser pressure changes the enthalpy of the steam entering the condenser. This and the change in exhaust flow changes the heat rejection in the condenser. The following illustration shows which process variables can normally be held constant, and which are calculated based on the new inlet water temperature:

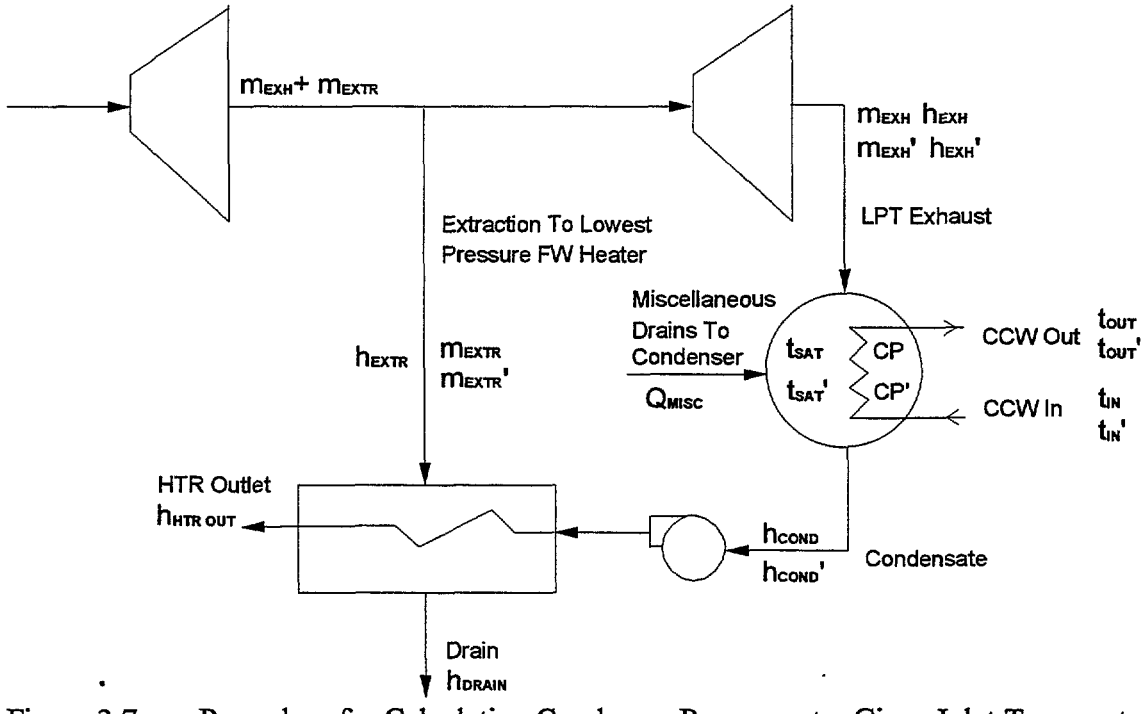

Figure 3.7 Procedure for Calculating Condenser Pressure at a Given Inlet Temperature

Note: variables shown with a ' denote the variable corrected to the desired water temperature, variables without a ' are the original test values.

a) Assume a condenser pressure (CP') that you think would result from the condenser circulating water temperature you want to correct the actual net unit heat rate to.

b) Using the condenser pressure from step "a" and the steam tables, calculate the saturation temperature (t_{Sat}') in the condenser.

c) Using the condenser pressure from step "a", the steam tables, and the amount of subcooling, calculate the condensate temperature and enthalpy (h_{COND}').

d) Using the condenser pressure from step "a" and the LP Turbine expansion line calculate the enthalpy of the exhaust steam entering the condenser (h_{EXH}').

e) Calculate the change in the extraction flow to the lowest pressure feedwater heater, due to the change in condenser pressure, from the actual test value. There are two methods depending on if the lowest pressure feedwater heater drain flow is pumped forward, or drains to the condenser.

i) If the drain goes to the condenser, the condensate flow is unchanged therefore:

$$m_{EXTR}' = (m_{COND} * (h_{HTR\ OUT} - h_{COND}')) / (h_{EXTR} - h_{DRAIN})$$

ii) If the drain is pumped forward, the calculation is slightly more complicated:

Let $\Delta h_{STM} = h_{EXTR} - h_{DRAIN}$

Let $\Delta h_{COND}' = h_{HTR\ OUT} - h_{COND}'$

$$m_{EXTR}' = (m_{EXH} + m_{EXTR}) * \Delta h_{COND}' / (\Delta h_{COND}' + \Delta h_{STM})$$

f) Calculate the new exhaust steam flow $m_{EXH}' = m_{EXH} + m_{EXTR} - m_{EXTR}'$

g) Calculate the new condenser duty Q' (using the above calculations and the tabulated miscellaneous heat rejected value Q_{MISC})

$$Q' = Q_{MISC} + m_{EXH}' * (h_{EXH}' - h_{COND}')$$

h) Calculate the temperature rise of the condenser circulating water, using the reference condenser circulating water flow (m_{CCW})

$$t_{RISE}' = Q' / (m_{CCW} * Cp)$$

i) Assume an inlet water temperature to the condenser t_{IN}'

j) Calculate the outlet water temperature $t_{OUT}' = t_{IN}' + t_{RISE}'$

k) Calculate the Log Mean Temperature Difference

$$LMTD' = t_{RISE}' / (LN((t_{Sat}' - t_{IN}')/(t_{Sat}' - t_{OUT}'))$$

l) Calculate the Heat Transfer Coefficient U', at the assumed inlet water temperature, and reference cleanliness factor

m) Calculate the condenser duty from the heat transfer relationship

$Q' = U'*A*LMTD'$, where A is the surface are of the condenser.

n) If the Q' from step "m" is not equal to the Q' from step "g", return to step "i", and repeat steps "j-m" with a different inlet water temperature.

o) If the condenser duties match, but the inlet temperature is not the temperature that is desired, return to step "a", using a different condenser pressure

p) When the duties match, and the inlet water temperature is the desired value, correct the test's net unit heat rate using the heat rate correction curves for condenser pressure

Let CF_{Act} = the % change in heat rate for the difference between the actual and design condenser pressure.

Let $CF_{Ref\ T}$ = the % change in heat rate for the difference between the calculated condenser pressure at the desired inlet temperature and design condenser pressure.

$$NUHR_{Ref} = NUHR_{Act} * (1+ CF_{Ref\,T}/100) / (1+ CF_{Act}/100)$$

q) When this has been done for the three or four inlet water temperatures, for a single test run, repeat the procedure for the other test runs.

r) Plot the three or four curves (one for each inlet water temperature) of net unit heat rate versus gross load.

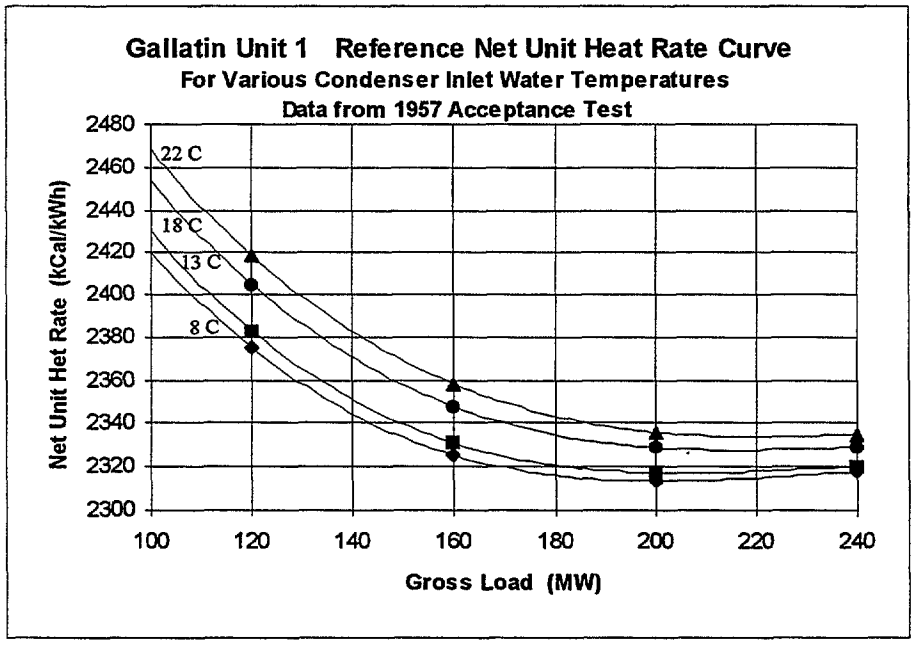

Figure 3.8 Example of a Family of Reference Heat Rate Curves

3.7.2 Generate Heat Rate Correction Factor Curves

The next step is to generate curves that show how much the unit heat rate is affected by a change in an individual parameter. This is done for every parameter that is to be tracked excluding exit gas temperature, Boiler Outlet O2, combustibles, moisture in fuel, and hydrogen in fuel (the heat rate change due to changes in these parameters is calculated from ASME equations, and not correction factor curves). The heat rate correction for all three auxiliary power/station service

power parameters is calculated, and therefore no correction factor is needed for these parameters either. Most curves should be plotted as % change in Net Unit Heat Rate versus the parameter value. Note that many turbogenerator manufacturers plot % change in turbine cycle heat rate. If that curve is available, it is acceptable to convert the % change in turbine cycle heat rate to % change in net unit heat rate using a design or performance guarantee test boiler efficiency and station use curves. Also, sometimes the thermal kits will have a family of curves, with each curve for a given gross load. If multiple curves are given, to get the correction factor at any load, interpolate or use a spline fit between the curves.

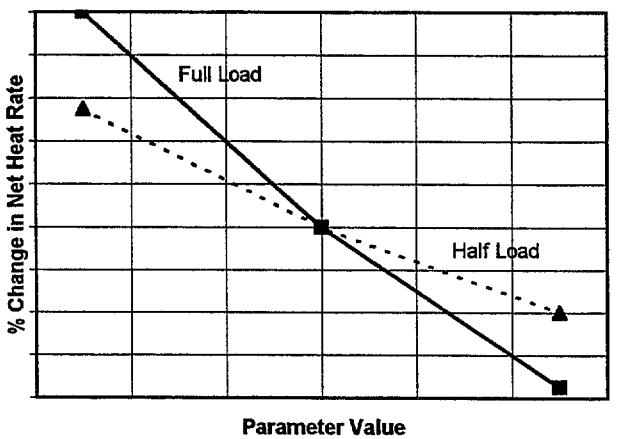

Figure 3.9 Example of a Family of Heat Rate Correction Factor Curves w/ the Parameter Value as the X Axis

Some curves are plotted as % change in Net Unit Heat Rate versus gross load. For example a curve of % change in heat rate for 1% attemperation flow versus gross load may be given. Sometimes a single curve will be given (or developed) for parameter. Or, a family of curves may be given, each curve for a specific parameter value (i.e. the graph may have gross load on the x axis and % change in net unit heat rate on the y axis, with one curve for 1% attemperation, another curve for 3% attemperation, etc.)

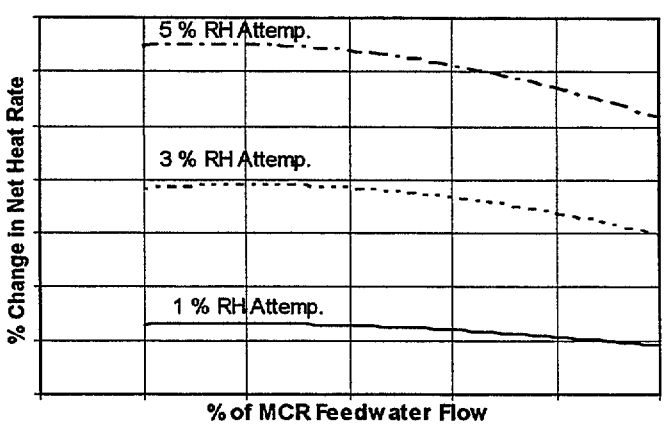

Figure 3.10 Example of a Family of Heat Rate Correction Factor Curves w/ the Parameter Value as the Z Axis

Some heat rate correction factors will be constants. For example, the heat rate correction factors for turbine section efficiencies are usually a single constant: x % change in net unit heat rate for a 1% point change in section efficiency.

Sections 3.7.2.1 and 3.7.2.2 contain information on where to obtain heat rate correction factors.

3.7.2.1 Turbine Vendors Thermal Kit

The supplier of the turbine almost always provides a set of heat rate correction curves for several conditions.

a) Throttle Pressure
b) Throttle Temperature
c) Steam Temperature at Inlet to Intercept Valve
d) Condenser Pressure
e) Final Feedwater Temperature (sometimes)
f) Reheat Attemperation (sometimes)
g) Superheat Attemperation (sometimes)

3.7.2.2 Thermodynamic Model of the Plant

A very useful tool to develop heat rate correction curves are the computer based thermodynamic modeling programs that are available. Once a plant has been properly modeled, changes in the cycle can be simulated, and the effects on heat rate (and other parameters) can be seen. There are now both boiler and turbine cycle modeling programs available, however the turbine cycle models, while complicated, are much easier and quicker to use than the boiler models, therefore these are used much more often.

a) Makeup
b) Auxiliary Steam Usage
c) Final Feedwater Temperature / Feedwater Heaters Out of Service
d) Reheat Attemperation
e) Superheat Attemperation

3.7.3 Develop "Expected" Parameters Curves

When the reference heat rate curves and parameter curves are first developed, the expected parameter curves will be identical to the reference parameter curves (see section 3.7.1.4). Over time, there will be modifications and additions to the unit that permanently change the performance of the unit. When any permanent change(s) to the plant are made, the change must be studied, and the applicable curves should be modified as appropriate. Examples of modifications that are sometimes made, and the parameters that are affected include:

- Adding additional auxiliary equipment (precipitators, larger ID fans, etc.) This increases station service.
- Operating procedures are changed (i.e., changing from full pressure to hybrid variable pressure operation). This would change the expected throttle pressure, and possibly the expected steam temperature and attemperation flow curves.
- A significant change in fuel characteristics (burning 40% moisture fuel instead of 10%). Some "expected" parameter curves may need to be changed, including exit gas temperature.

There are various sources for the data that is required to update the curves.

- A performance/optimization test after the modification. This method is recommended when there have been any changes in the boiler area for two reasons. First, because the effects of fuel switches, furnace or convection pass modifications, etc. are very difficult to accurately calculate. Second, boiler modifications usually affect several parameters (exit gas temperature, combustibles in ash, Boiler Outlet O2, steam temperatures, and attemperation flow rates).
- Manufacturers design data. This method may be acceptable for changes that affect station service for example. If additional fields are added to a precipitator, the manufacturer should be required to provide information on the change in auxiliary power that is to be expected.
- Engineering calculations/assumptions.

3.7.4 Calculation of Reference Net Heat Rate

The Reference Net Heat Rate is calculated at the period's average gross load, and condenser inlet water temperature, by calculating a heat rate at each of the three or 4 reference heat rate curves, and then interpolating between theses values based on the water temperature.

1) Calculate average gross load for time interval ($GLoad_{Act}$).

$$GLoad_{Act} = GG_{Act} / HOU_{Act}$$

2) Calculate average cooling water temperature for time interval ($CCWTI_{Act}$).
3) Calculate the heat rate from each of the three or four reference heat rate curves (each curve being at a different condenser water inlet temperature), at the average gross load.

$$NUHR_{Ref,\,S1} = f(GLoad_{Act},\ CCWTI_{Ref,\,S1})$$

$$NUHR_{Ref,\,S2} = f(GLoad_{Act},\ CCWTI_{Ref,\,S2})$$

$$NUHR_{Ref,\,S3} = f(GLoad_{Act},\ CCWTI_{Ref,\,S3})$$

4) Perform a spline fit (see Appendix F) of the heat rates from step "C", using the actual average cooling water temperature, to determine the reference net heat rate $NUHR_{Ref}$.

$$NUHR_{Ref} = f(NUHR_{Ref,\,S1},\ CCWTI_{Ref,\,S1},\ NUHR_{Ref,\,S2},\ CCWTI_{Ref,\,S2},\ NUHR_{Ref,\,S3},\ CCWTI_{Ref,\,S3},\ CCWTI_{Act})$$

3.7.5 Actual Net Heat Rate

1) Determine total actual fuel burned in kilograms for the time interval (FS_{Act})
2) Determine the Higher Heating Value (HHV_{Act}) of the fuel (if multiple samples are collected during the period, weight average by amount of coal burned for the period from the coal analysis).
3) Calculate the actual net unit heat rate by multiplying the total quantity of fuel burned by the fuel's higher heating value, and then dividing by the net generation.

$$NUHR_{Act} = (FS_{Act}*HHV_{Act}) / (GG_{Act} - (SSO_{Act} + UAP_{Act})*HOU_{Act} - SSI_{Act}*(PH-HOU_{Act}))/1000$$

3.7.6 General Comments on Calculating Heat Rate Deviations for All Parameters

For calculating heat rate deviations for parameters in the turbine cycle, the standard procedure is relatively straight-forward. For each parameter, the heat rate deviation that is calculated is the amount the net unit heat rate would increase by if its value *only* changed from its expected value to its actual value. In other words, when calculating the heat rate deviation for Throttle temperature, all other parameters are assumed to be at their expected level, and the net unit heat rate is at its expected level. The only turbine cycle exception to this rule is condenser pressure. Its *expected* level is adjusted up if there are other problems in the cycle. This way, if there are problems in the cycle resulting in additional heat load on the condenser, the expected condenser pressure is the condenser pressure that should be obtainable with the additional heat load.

The boiler cycle losses are calculated in a similar manner, but with an important difference. In addition to the question of when calculating the heat rate deviation for one parameter, should the other parameters be held at their expected value or actual value; there is another pair of questions. Because some parameters have a significant influence on other parameters, should those influences be considered? For example, if the exit gas temperature (no leakage, corrected to reference inlet air temperature) increases, not only does the dry gas loss increase, but both the moisture in fuel loss and the hydrogen loss also increases (because the water vapor leaving the stack will be at a higher enthalpy).

There is not any clear "right" or "wrong" way of looking at these issues. In this procedure, boiler losses are calculated assuming other parameters are at their expected levels (which is consistent with the turbine cycle losses), but if there are additional losses in other areas, they are taken into account. Continuing with the previous example, the heat rate deviation this procedure would calculate for an increase in exit gas temperature would include the increase in dry gas loss (from the expected to actual exit gas temperature), the increase in moisture in fuel loss (based on the actual versus expected fuel moisture at the actual exit gas temperature) and the increase in

hydrogen loss (based on the actual versus expected fuel hydrogen and the actual fuel hydrogen at the actual exit gas temperature). The heat rate deviation due to moisture in fuel loss would be calculated at the expected exit gas temperature, and would account for the change (if any) in fuel moisture only.

This procedure will at times slightly under account for some losses. For example, if both the exit gas temperature and the boiler outlet O_2 are above expected, the heat rate deviation for exit gas temperature is calculated using the expected (lower than actual) boiler outlet O_2, and the boiler outlet O_2 heat rate deviation is calculated using the expected (lower than actual) exit gas temperature. If the total heat rate deviation was calculated in one step using the actual exit gas temperature and the actual boiler outlet O_2, that heat rate deviation would be larger than the sum of the two deviations calculated individually .

3.7.7 Air Preheating Steam Coils

The heat rate deviation for air preheating coils is usually based on the flow rate to the coils, or if the flow is relatively constant when they are in service, the number of hours of operation. If multiple sources of steam can be used, then a heat rate deviation must be calculated for each.

1) Establish $SCAPH_{Ref}$ value from reference curve (based on acceptance test or design data, this could be a function of main steam/feedwater flow and or ambient air temperature).

$$SCAPH_{Ref} = f(GLoad_{Act}) \text{ or } f(FWF_{Act}) \text{ or } f(MSF_{Act}) \text{ or } f(FWF_{Act}, AmbT_{Act})$$

2) Establish $SCAPH_{Exp}$ value from expected level curve (based on current plant equipment design.

$$SCAPH_{Exp} = f(GLoad_{Act}) \text{ or } f(FWF_{Act}) \text{ or } f(MSF_{Act}) \text{ or } f(FWF_{Act}, AmbT_{Act})$$

3) From the curve of heat rate correction factor due to air coil usage versus gross load or flow, determine the $HRCF_{SCAPH}$ at the period's average gross load.

$$HRCF_{SCAPH} = f(GLoad_{Act}) \text{ or } f(FWF_{Act})$$

4) Determine the heat rate deviation due to the difference between the expected makeup and the reference makeup.

$$HRD_{SCAPH,\ Exp\text{-}Ref} = (SCAPH_{Exp} - SCAPH_{Ref}) * NUHR_{Ref} * HRCF_{SCAPH}$$

5) After calculating the expected net unit heat rate $NUHR_{Exp}$ (see Section 3.7.22), determine the heat rate deviation due to the difference between the actual makeup and the expected makeup

$$HRD_{SCAPH,\ Act\text{-}Exp} = (SCAPH_{Act} - SCAPH_{Exp}) * NUHR_{Exp} * HRCF_{SCAPH}$$

3.7.8 Boiler Differential Pressure

The heat rate deviation for the boiler differential pressure is based on the percentage pressure drop between the economizer inlet to the superheater outlet.

1) Establish BDP_{Ref} value from reference curve (based on acceptance test or design data).

$$BDP_{Ref} = f(GLoad_{Act}) \text{ or } f(FWF_{Act}) \text{ or } f(MSF_{Act})$$

2) Establish BDP_{Exp} value from expected level curve (based on current plant equipment design.

$$BDP_{Exp} = f(GLoad_{Act}) \text{ or } f(FWF_{Act}) \text{ or } f(MSF_{Act})$$

3) From the curve of the heat rate correction factor due to 1% boiler differential pressure versus gross load (or feedwater flow), determine the $HRCF_{BDP}$ at the periods average gross load.

$$HRCF_{BDP} = f(GLoad_{Act}) \text{ or } f(FWF_{Act})$$

4) Determine the heat rate deviation due to the difference between the expected and the reference boiler differential pressure.

$$HRD_{BDP,\ Exp\text{-}Ref} = (BDP_{Exp} - BDP_{Ref}) * NUHR_{Ref} * HRCF_{BDP}$$

5) After calculating the expected net unit heat rate $NUHR_{Exp}$ (see Section 3.7.22), determine the heat rate deviation due to the difference between the actual and the expected boiler differential pressure.

$$HRD_{BDP,\ Act\text{-}Exp} = (BDP_{Act} - BDP_{Exp}) * NUHR_{Exp} * HRCF_{BDP}$$

3.7.9 Condenser Pressure

This is a two step procedure. First, the reference condenser pressure is calculated, using the actual unit load, heat rejected to the condenser based on a reference turbine cycle heat rate curve, the reference condenser parameters (tube materials, thickness, etc.), and the reference condenser pressure. Since the reference condenser pressure is calculated using the reference condenser pressure, this is an iterative process. An initial guess is made for the reference condenser pressure. This pressure is then used to determine the heat rejection in the condenser, by adjusting the reference turbine cycle heat rate. Then the condenser characteristics are used to calculate the condenser pressure. If there is a difference between the two pressure, the initial guess is changed, and the calculations are repeated.

A second iterative procedure is used to calculate the expected condenser pressure. The expected condenser pressure is calculated using the actual heat rejected to the condenser (adjusted to the expected condenser pressure and for startup fuel), and the expected condenser parameters.

Steps "1" through "4" are calculating some miscellaneous values that will be needed in the later, iterative steps:

1) Calculate heat added by the boiler feed water pumps ($PumpQ_{Ref,Exp}$). Obtain from design or acceptance test curves.

$$PumpQ_{Ref,Exp} = f(MSF_{Act}, \text{ or } FWF_{Act}, \text{ or } GLoad_{Act})$$

2) If the unit has any auxiliary equipment condensers, calculate their total duty or rate of heat transferred out of the turbine cycle, ($AuxDuty_{Ref,Exp}$) from curves developed across the flow or load range using thermal modeling software.

$$AuxDuty_{Ref,Exp} = f(MSF_{Act}, \text{ or } FWF_{Act}, \text{ or } GLoad_{Act})$$

3) Calculate Generator Loss ($GLoss_{Ref, Exp}$) from manufacturer's curves.

$$GLoss_{Ref, Exp} = f(GLoad_{Act})$$

4) Calculate HEI Inlet Water Temperature Correction Factor ($HEITCF_{Ref, Exp}$), based on the actual circulating water inlet temperature ($CCWTI_{Act}$)

$$HEITCF_{Ref, Exp} = f(CCWTI_{Act})$$

Steps "5" through "24" is an iterative procedure for calculating the reference condenser pressure, based on the "reference" turbine cycle heat rate curve and "reference" condenser characteristics (tube material, tube wall thickness, etc.)

5) Establish reference value for Condenser Circulating water flow rate ($CCWF_{Ref}$) from the condenser characteristics (some constant value), or from a curve (i.e., $CCWF_{Ref}$ vs. lake elevation).

$$CCWF_{Ref} = \text{constant or } f(\text{water elevation})$$

6) Calculate the reference value for Water Velocity through the tubes ($TubeWV_{Ref}$), given the flow rate ($CCWF_{Ref}$), the number of tubes ($TubeNo_{Ref}$), and the internal diameter of the tubes ($TubeID_{Ref}$).

$$TubeWV_{Ref} = CCWF_{Ref} \ / \ 1000 \ kg/m^3 \ / \ (3.14159 * (TubeID_{Ref}/2/1000)^2 * TubeNo_{Ref}) \ / \ 3600 \ sec/hr)$$

7) Look up the reference value for the Heat Exchange Institute "Material and Gauge Correction Factor" ($TubeMGC_{Ref}$) as a function of the material of the condenser tubes ($TubeMat_{Ref}$) and the tube wall thickness ($TubeBWG_{Ref}$).

$$TubeMGC_{Ref} = f(TubeMat_{Ref}, TubeBWG_{Ref})$$

8) Establish reference value for HEI's "Uncorrected Heat Transfer Coefficient" ($TubeU_{Ref}$) based on tube external diameter ($TubeOD_{Ref}$) and water velocity through the tubes ($TubeWV_{Ref}$).

$$TubeU_{Ref} = f(TubeOD_{Ref}, TubeWV_{Ref})$$

9) Establish reference value for Condenser Surface Area ($CSArea_{Ref}$).

10) Calculate Reference Gross Turbine Cycle Heat Rate (corrected to design condenser pressure) $GTCHR_{Ref, CorDCP}$ as a function of the period's average gross load ($GLoad_{Act}$)

$$GTCHR_{Ref, CorDCP} = f(GLoad_{Act})$$

11) Use design condenser pressure (CP_{Des}) as the first guess for calculating CP_{Ref}.

$$CP_{Ref} = CP_{Des}$$

12) From turbo/generator manufacturer's heat rate correction curve(s) for condenser pressure (percent change in heat rate vs. condenser pressure) determine the $HRCF_{CP, Ref\text{-}Des}$.

$$HRCF_{CP, Ref\text{-}Des} = f(CP_{Ref}, MSF_{Act}) \text{ or } f(CP_{Ref}, GLoad_{Act})$$

13) If the manufacturer provides multiple curves based on different steam flows or loads, then the $HRCF_{CP, Ref\text{-}Des}$ will have to be determined by interpolating or spline fitting between curves.

14) Calculate the gross turbine cycle heat rate, corrected to the reference condenser pressure, $GTCHR_{Ref,Ccp}$.

$$GTCHR_{Ref,CorRCP} = GTCHR_{Ref, CorDCP} * (1 + HRCF_{CP, Ref\text{-}Des} /100)$$

15) Calculate the rate of heat rejection in the condenser, Condenser Duty (Duty) by taking the total heat input to the turbine cycle, including the heat added by the pumps, and then subtracting the energy converted to electricity, the losses in the generator, and heat rejected in auxiliary condensers (if any).

$$Duty_{Ref} = (GTCHR_{Ref,CorRCP} * GLoad_{Act} * 1000kW/MW) - (860{,}000 \text{ kcal/h/MW} * GLoad_{Act}) - (860 \text{ kcal/h/kW} * GLoss_{Ref, Exp}) + PumpQ_{Ref,Exp} - AuxDuty_{Ref,Exp}$$

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16) Calculate Heat Transmittance (HTran) by dividing the heat load on the condenser by the surface area of the condenser

$$HTran_{Ref} = Duty_{Ref} / CSArea_{Ref}$$

17) Calculate HEI Heat Transfer Coefficient ($HEIHTC_{Ref}$), by multiplying the Uncorrected Heat transfer Coefficient by the Tube Material and Gauge Correction Factor and by multiplying by the Inlet Water Temperature Correction Factor. This is the heat transfer coefficient for a "clean" tube.

$$HEIHTC_{Ref} = TubeU_{Ref} * TubeMGC_{Ref} * HEITCF_{Ref, Exp}.$$

18) Calculate the Reference Heat Transfer Coefficient (HTC_{Ref}) by multiplying the HEI Heat Transfer Coefficient by the cleanliness factor that could be maintained. Normally a cleanliness factor of 85% is used.

$$HTC_{Ref} = HEIHTC_{Ref} * CF_{Ref} / 100$$

19) Calculate Logarithmic Mean Temperature Difference ($LMTD_{Ref}$) by dividing the heat transmittance by the reference heat transfer coefficient

$$LMTD_{Ref} = HTran_{Ref} / HTC_{Ref}.$$

20) Calculate the Outlet Water Temperature (CCWTO) by the equation Duty is equal to CCW flow rate multiplied by the heat capacity of the water multiplied by the temperature rise of the water. This equation can be rearranged to give:

$$CCWTO_{Ref} = (Duty_{Ref} / (CCWF_{Ref} * 1.0 \text{ kcal/kg C})) + CCWTI_{Act}$$

21) Calculate Circulating Water Temperature Rise

$$CCWTR_{Ref} = CCWTO_{Ref} - CCWTI_{Act}.$$

22) Calculate the reference Exhaust Steam Temperature

$$EST_{Ref} = (CCWTO_{Ref} * e^{(CCWTR / LMTD)} - CCWTI_{Act}) / (e^{(CCWTR / LMTD)} - 1).$$

23) Calculate CP using EST_{Ref} and steam tables.

24) Check to see if the condenser pressure used in steps "11" through "22" is within 0.2mm of the calculated condenser pressure (CP from step "23"). If the difference is greater than 0.2mm, reset CP_{Ref} = CP and return to step "12". If the difference is smaller than 0.2mm, continue with the next step.

Steps "25" through "47" is an iterative procedure for calculating the expected condenser pressure.

25) Establish the typical heat input used on a cold startup (HICold).
26) Establish the typical heat input used on a hot startup (HIHot).
27) Adjust the period's actual gross unit heat rate by removing the fuel supplied during startups.

$$GUHR_{Act,\ CorSF} = (FS_{Act} * HHV_{Act} - NoCold*HICold - NoHot*HIHot) / (GG_{Act} * 1000kWh/MWh)$$

28) Establish expected value for Condenser Circulating water flow rate ($CCWF_{Exp}$) from the condenser characteristics, or from a curve (i.e. $CCWF_{Exp}$ vs. lake elevation).

$$CCWF_{Exp} = \text{constant or } f(\text{water elevation})$$

29) Calculate the expected value for Water Velocity through the tubes ($TubeWV_{Exp}$), given the flow rate ($CCWF_{Exp}$), the number of tubes ($TubeNo_{Exp}$), and the internal diameter of the tubes ($TubeID_{Exp}$).

$$TubeWV_{Exp} = CCWF_{Exp} / 1000\ kg/m^3 / (3.14159 * (TubeID_{Exp}/2/1000)^2 * TubeNo_{Exp} * 3600\ sec/h)$$

30) Look up the expected value for the Heat Exchange Institute "Material and Gauge Correction Factor" ($TubeMGC_{Exp}$) as a function of the material of the condenser tubes ($TubeMat_{Exp}$) and the tube wall thickness ($TubeBWG_{Exp}$).

$$TubeMGC_{Exp} = f(TubeMat_{Exp}, TubeBWG_{Exp})$$

31) Establish expected value for HEI's "Uncorrected Heat Transfer Coefficient" ($TubeU_{Exp}$) based on tube external diameter ($TubeOD_{Exp}$) and water velocity through the tubes ($TubeWV_{Exp}$).

$$TubeU_{Exp} = f(TubeOD_{Exp}, TubeWV_{Exp})$$

32) Establish expected value for Condenser Surface Area ($CSArea_{Exp}$).
33) From manufacturer's heat rate correction curve for condenser pressure (Percent change in heat rate vs. condenser pressure), determine the percent change in heat rate due to the actual condenser pressure (verses design).

$$HRCF_{CP,\ Act\text{-}Des} = f(CP_{Act}, MSF_{Act}) \text{ or } f(CP_{Act}, GLoad_{Act})$$

34) Use the actual condenser pressure (CP_{Act}) as the first guess for calculating the expected condenser pressure CP_{Exp}.

$$CP_{Exp} = CP_{Act}$$

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35) From manufacturer's heat rate correction curve for condenser pressure (Percent change in heat rate vs. condenser pressure) determine the percent change in heat rate due to the expected condenser pressure (versus design)

$$HRCF_{CP,\ Exp\text{-}Des} = f(CP_{Exp},\ MSF_{Act}) \text{ or } f(CP_{Exp},\ GLoad_{Act})$$

36) Ratio the expected and actual heat rate correction fractional multipliers ($HRCF_{CP,\ Exp\text{-}Des}$ and $HRCF_{CP,\ Act\text{-}Des}$)

$$HRCFR_{CP,\ Act\text{-}Exp} = (1+ HRCF_{CP,\ Exp\text{-}Des}/100) / (1+ HRCF_{CP,\ Act\text{-}Des}/100)$$

37) Calculate the actual gross unit heat rate adjusted for the "assumed" expected condenser pressure (and also adjusted to eliminate startup fuel).

$$GUHR_{Act,\ CorSF,\ CorECP} = GUHR_{Act,\ CorSF} * HRCFR_{CP,\ Act\text{-}Exp}.$$

38) Calculate the expected condenser duty, which is the actual condenser duty, corrected for startup fuel and to expected condenser pressure ($Duty_{Exp} = Duty_{Act,\ CorSF,\ CorECP}$) by taking the total heat input to the steam plus the heat added by the pumps and then subtracting the energy converted to electricity and the energy lost in the generator.

$$Duty_{Exp} = (GUHR_{Act,\ CorSF,\ CorECP} * GLoad_{Act} * 1000kW/MW * BE_{Act}/100) - (860{,}000\ kcal/h/MW * GLoad_{Act}) - (860\ kcal/h/kW * GLoss_{Ref,\ Exp}) + PumpQ_{Ref,Exp} - AuxDuty_{Ref,Exp}$$

39) Calculate Heat Transmittance ($HTran_{Exp}$) by dividing the heat rejection rate in the condenser by the area available for heat transfer.

$$HTran_{Exp} = Duty_{Exp} / CSArea_{Exp}$$

40) Calculate HEI Heat Transfer Coefficient ($HEIHTC_{Exp}$), by multiplying the Uncorrected Heat transfer Coefficient by the Tube Material and Gauge Correction Factor and by multiplying by the Inlet Water Temperature Correction Factor

$$HEIHTC_{Exp} = TubeU_{Exp} * TubeMGC_{Exp} * HEITCF_{Ref,\ Exp}.$$

41) Calculate the expected Heat Transfer Coefficient (HTC_{Exp}), based on the expected tube cleanliness factor (usually the same value as the reference, which is usually 85%).

$$HTC_{Exp} = HEIHTC_{Exp} * CF_{Exp} / 100$$

42) Calculate the expected Logarithmic Mean Temperature Difference ($LMTD_{Exp}$) by dividing the heat transmittance by the heat transfer coefficient

$$LMTD_{Exp} = HTran_{Exp} / HTC_{Exp}.$$

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43) Calculate the expected Condenser Circulating Water Temperature Rise ($CCWTR_{Exp}$)

$$CCWTR_{Exp} = (Duty_{Exp} / CCWF_{Exp} / (1kcal/kg\ C))$$

44) Calculate the expected Outlet Water Temperature ($CCWTO_{Exp}$)

$$CCWTO_{Exp} = CCWTR_{Exp} + CCWTI_{Act}.$$

45) Calculate Expected Exhaust Steam Temperature (EST_{Exp})

$$EST_{Exp} = (CCWTO_{Exp} * e^{(CCWTR/LMTD)} - CCWTI_{Act}) / (e^{(CCWTR/LMTD)} - 1).$$

46) Calculate CP using EST_{Exp} and steam tables.

47) Check to see if the condenser pressure used in steps "35" through "45" is within 0.2mm of the calculated condenser pressure (CP from step "46"). If the difference is greater than 0.2mm, reset CP_{Exp} = CP and return to step "35". If the difference is smaller than 0.2mm, continue with the next step.

48) Ratio the reference and expected heat rate correction fractional multipliers ($HRCF_{CP, Exp-Des}$ and $HRCF_{CP, Act-Des}$)

$$HRCFR_{CP, Exp-Ref} = (1 + HRCF_{CP, Ref-Des} / 100) / (1 + HRCF_{CP, Exp-Des}/100)$$

49) Determine the heat rate deviation due to the difference between the expected and reference condenser pressure ($HRD_{CP, Exp-Ref}$)

$$HRD_{CP, Exp-Ref} = (1 - HRCFR_{Exp-Ref}) * NUHR_{Ref}$$

50) Add this and all other heat rate deviations (expected verses reference) to the reference net unit heat rate ($NUHR_{Ref}$) to obtain the expected net unit heat rate ($NUHR_{Exp}$) see Section 3.7.22.

$$NUHR_{Exp} = NUHR_{Ref} + \Sigma\ HRD_{I, Exp-Ref}.$$

51) Determine the heat rate deviation due to the difference between the actual and expected heat rate ($HRD_{CP, Exp-Ref}$).

$$HRD_{CP, Act-Exp} = (1 - HRCFR_{CP, Act-Exp}) * NUHR_{Exp}$$

3.7.10 Final Feedwater Temperature & High Pressure Feedwater Heater(s) Out of Service

There are three ways to calculate heat rate deviations for lower than expected final feedwater temperature. The first is to simply calculate a deviation based on the final feedwater temperature. For example if the FFW temperature is 5 °C too low, multiply that by a heat rate correction factor

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to get the heat rate deviation. A second method is to calculate a heat rate deviation only when one or more feedwater heaters is out of service. The program would have a heat rate deviation factor for all possible combinations of heaters that can be bypassed (the highest pressure heater only, all high pressure heaters, the first and second heater, etc.). The third method, and the one that is recommended, is to calculate a deviation for the high pressure heaters using the final feedwater temperature when all heaters are in service, and to calculate a heat rate deviation for the high pressure heaters using the appropriate correction factor (depending on which heater(s) are out of service) when any high pressure heater(s) is out of service.

When the heaters are in service:

1) Establish $FFWT_{Ref}$ value from reference curve (based on acceptance test or design data).

$$FFWT_{Ref} = f(GLoad_{Act}) \text{ or } f(FWF_{Act}) \text{ or } f(MSF_{Act})$$

2) Establish $FFWT_{Exp}$ value from expected level curve (based on current plant equipment design.

$$FFWT_{Exp} = f(GLoad_{Act}) \text{ or } f(FWF_{Act}) \text{ or } f(MSF_{Act})$$

3) From the curve of heat rate correction factor due to 1°C final feedwater temperature difference versus gross load (or feedwater flow), determine the $HRCF_{FFWT}$ at the periods average gross load.

$$HRCF_{FFWT} = f(GLoad_{Act}) \text{ or } f(FWF_{Act})$$

4) Determine the heat rate deviation due to the difference between the expected makeup and the reference makeup.

$$HRD_{FFWT,\ Exp\text{-}Ref} = (FFWT_{Ref} - FFWT_{Exp}) * NUHR_{Ref} * HRCF_{FFWT}$$

5) After calculating the expected net unit heat rate $NUHR_{Exp}$ (see Section 3.7.22), determine the heat rate deviation due to the difference between the actual and the expected final feedwater temperature

$$HRD_{FFWT,\ Act\text{-}Exp} = (FFWT_{Exp} - FFWT_{Act}) * NUHR_{Exp} * HRCF_{FFWT}$$

When the heaters are out of service:

1) For each combination of heaters that is out of service, determine the heat rate correction factor for that combination. (For example, if for part of the period the top heater only was bypassed, and for part of the period all HP heaters were bypassed, determine the heat rate correction factor for both.)

$$HRCF_{HtrBP} = f(GLoad_{Act}) \text{ or } f(FWF_{Act})$$

2) Calculate the heat rate deviation for each combination of heaters bypassed:

$HRD_{HtrBP, Act-Exp} = NUHR_{Exp} * HRCF_{HtrBP} * (\text{Hours bypassed} / PH_{Act})$

3.7.11 Makeup

1) Establish MU_{Ref} value from reference curve (based on acceptance test or design data).

$MU_{Ref} = f(GLoad_{Act})$ or $f(FWF_{Act})$ or $f(MSF_{Act})$

2) Establish MU_{Exp} value from expected level curve (based on current plant equipment design.

$MU_{Exp} = f(GLoad_{Act})$ or $f(FWF_{Act})$ or $f(MSF_{Act})$

3) From the curve of heat rate correction factor due to 1% makeup versus gross load, determine the $HRCF_{MU}$ at the periods average gross load.

$HRCF_{MU} = f(GLoad_{Act})$

4) Determine the heat rate deviation due to the difference between the expected makeup and the reference makeup.

$HRD_{MU, Exp-Ref} = (MU_{Exp} - MU_{Ref}) * NUHR_{Ref} * HRCF_{MU}$

5) After calculating the expected net unit heat rate $NUHR_{Exp}$ (see Section 3.7.22), determine the heat rate deviation due to the difference between the actual makeup and the expected makeup

$HRD_{MU, Act-Exp} = (MU_{Act} - MU_{Exp}) * NUHR_{Exp} * HRCF_{MU}$

3.7.12 Reheat Attemperation

1) Establish the reference value for reheat attemperation flow (based on acceptance test or design) at the period's feedwater flow, main steam flow, or gross load.

$RHA_{Ref} = f(GLoad_{Act})$ or $f(FWF_{Act})$ or $f(MSF_{Act})$

2) Establish the expected value for reheat attemperation flow (based on current plant equipment and operating procedures) at the period's feedwater flow, main steam flow, or gross load.

$RHA_{Exp} = f(GLoad_{Act})$ or $f(FWF_{Act})$ or $f(MSF_{Act})$

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3) Determine the heat rate correction factor (% change in net unit heat rate) for the reference reheat attemperation rate (as a percent of feedwater flow).

$HRCF_{RHA,\ Ref} = f(GLoad_{Act})$ or $f(FWF_{Act})$ or $f(MSF_{Act})$ & RHA_{Ref} (from a curve or family of curves from the t/g manufacturer or from thermal modeling software).

4) Determine the heat rate correction factor (% change in net unit heat rate) for the expected reheat attemperation rate (as a percent of feedwater flow).

$HRCF_{RHA,\ Exp} = f(GLoad_{Act})$ or $f(FWF_{Act})$ or $f(MSF_{Act})$ & RHA_{Exp}

5) Determine the heat rate correction factor (% change in net unit heat rate) for the actual reheat attemperation rate (as a percent of feedwater flow).

$HRCF_{RHA,\ Act} = f(GLoad_{Act})$ or $f(FWF_{Act})$ or $f(MSF_{Act})$ & RHA_{Act}

6) Determine the HRDZ (expected % attemperation - reference % attemperation)*Ref HR * HRCF.

$HRD_{RHA,\ Exp\text{-}Ref} = NUHR_{Ref} * (HRCF_{RHA,\ Exp} - HRCF_{RHA,\ Ref}) / 100$

7) After calculating the expected net unit heat rate $NUHR_{Exp}$ (see Section 3.7.22), determine the heat rate deviation due the difference between the actual reheat attemperation and the expected reheat attemperation.

$HRD_{RHA,\ Act\text{-}Exp} = NUHR_{Exp} * (HRCF_{RHA,\ Act} - HRCF_{RHA,\ Exp}) / 100$

3.7.13 Station Service (Prorated) - Idle

Normally, both the reference and the expected idle time station service are zero. Calculate the heat rate deviation (from expected) due to station service used when the unit was off-line during the period in question.

$$HRD_{SSI,\ Act\text{-}Exp} = \{ (FS_{Act} * HHV_{Act}) / (GG_{Act} - (UAP_{Exp}+SSO_{Exp})*HOU_{Act} - SSI_{Act}*(PH-HOU_{Act})) \}/1000 - \{ (FS_{Act} * HHV_{Act}) / (GG_{Act} - (UAP_{Exp}+SSO_{Exp})*HOU_{Act})\}/1000$$

3.7.14 Station Service (Prorated) - Operating

1) Determine SSO_{Ref} value from reference curve, (plotted as station use in MW versus steam flow, feedwater flow or gross load, based on acceptance test or design data) at the period's steam flow, feedwater flow or gross load

$SSO_{Ref} = f(GLoad_{Act})$ or $f(FWF_{Act})$ or $f(MSF_{Act})$

2) Establish SSO_{Exp} value from expected level curve, (plotted as station use in MW verses steam flow, feedwater flow or gross load, based on current plant equipment design) at the period's steam flow, feedwater flow or gross load.

$SSO_{Exp} = f(GLoad_{Act})$ or $f(FWF_{Act})$ or $f(MSF_{Act})$

3) Calculate the heat rate deviation due to the difference between the expected and reference station service using either equation:

$$HRD_{SSO,\ Exp\text{-}Ref} = \{ (FS_{Act} * HHV_{Act}) / (GG_{Act} - (UAP_{Ref} + SSO_{Exp}) * HOU_{Act})\}/1000 - \{ (FS_{Act} * HHV_{Act}) / (GG_{Act} - (UAP_{Ref} + SSO_{Ref})*HOU_{Act})\}/1000$$

4) After calculating the expected net unit heat rate $NUHR_{Exp}$ (see Section 3.7.22), determine the heat rate deviation due to the difference between the actual operating station use and the expected operating station use.

$$HRD_{SSO,\ Act\text{-}Exp} = \{ (FS_{Act} * HHV_{Act}) / (GG_{Act} - (UAP_{Exp} +SSO_{Act})*HOU_{Act})\}/1000 - \{ (FS_{Act} * HHV_{Act}) / (GG_{Act} - (UAP_{Exp} +SSO_{Exp})*HOU_{Act})\}/1000$$

3.7.15 Hot Reheat at Intercept Valve Temperature

1) Establish $HRHIV_{Ref}$ value from reference curve (based on acceptance test or design data).

$HRHIV_{Ref} = f(GLoad_{Act})$ or $f(FWF_{Act})$ or $f(MSF_{Act})$

2) Establish $HRHIV_{Exp}$ value from expected level curve (based on current plant equipment design.

$HRHIV_{Exp} = f(GLoad_{Act})$ or $f(FWF_{Act})$ or $f(MSF_{Act})$

3) From the heat rate correction curve(s) for reheat steam temperature (plotted as percent change in heat rate vs. steam temperature) determine the percent change in heat rate due to the reference, expected and actual steam temperatures versus design, $HRCF_{HRHIV,\ Ref\text{-}Des}$, $HRCF_{HRHIV,\ Exp\text{-}Des}$, $HRCF_{HRHIV,\ Act\text{-}Des}$.

a) $HRCF_{HRHIV,\ Ref\text{-}Des} = f(HRHIV_{Ref})$ or $f(HRHIV_{Ref}, GLoad_{Act})$
b) $HRCF_{HRHIV,\ Exp\text{-}Des} = f(HRHIV_{Exp})$ or $f(HRHIV_{Exp}, GLoad_{Act})$
c) $HRCF_{HRHIV,\ Act\text{-}Des} = f(HRHIV_{Act})$ or $f(HRHIV_{Act}, GLoad_{Act})$

(If the manufacturer provides multiple curves based on different loads, $HRCF_{HRHIV, Ref\text{-}Des}$, $HRCF_{HRHIV, Exp\text{-}Des}$, and $HRCF_{HRHIV, Act\text{-}Des}$ will have to be determined by interpolating or using a spline fit between curves.)

4) Determine the heat rate deviation due to the difference between the expected steam temperature and the reference steam temperature.

$$HRD_{HRHIV, Exp\text{-}Ref} = \{ (1+ HRCF_{HRHIV, Exp\text{-}Des}/100) / (1+ HRCF_{HRHIV, Ref\text{-}Des}/100) - 1\} * NUHR_{Ref}$$

5) After calculating the expected net unit heat rate $NUHR_{Exp}$ (see Section 3.7.22), determine the heat rate deviation due to the difference between the actual steam temperature and the expected steam temperature

$$HRD_{HRHIV, Act\text{-}Exp} = \{ (1+ HRCF_{HRHIV, Act\text{-}Des}/100) / (1+ HRCF_{HRHIV, Exp\text{-}Des}/100) - 1\} * NUHR_{Exp}.$$

3.7.16 Superheat Attemperation

1) Establish the reference value for superheat attemperation flow (based on acceptance test or design) at the period's feedwater flow, main steam flow, or gross load.

$$SHA_{Ref} = f(GLoad_{Act}) \text{ or } f(FWF_{Act}) \text{ or } f(MSF_{Act})$$

2) Establish the expected value for superheat attemperation flow (based on current plant equipment and operating procedures) at the period's feedwater flow, main steam flow, or gross load.

$$SHA_{Exp} = f(GLoad_{Act}) \text{ or } f(FWF_{Act}) \text{ or } f(MSF_{Act})$$

3) Determine the heat rate correction factor (% change in net unit heat rate) for the reference superheat attemperation rate (as a percent of feedwater flow).

$$HRCF_{SHA, Ref} = f(GLoad_{Act}) \text{ or } f(FWF_{Act}) \text{ or } f(MSF_{Act}) \ \& \ SHA_{Ref}$$

(from a curve or family of curves supplied by the turbogenerator manufacturer or from thermal modeling software).

4) Determine the heat rate correction factor (% change in net unit heat rate) for the expected superheat attemperation rate (as a percent of feedwater flow).

$$HRCF_{SHA, Exp} = f(GLoad_{Act}) \text{ or } f(FWF_{Act}) \text{ or } f(MSF_{Act}) \ \& \ SHA_{Exp}$$

5) Determine the heat rate correction factor (% change in net unit heat rate) for the actual superheat attemperation rate (as a percent of feedwater flow).

$HRCF_{SHA,\ Act} = f(GLoad_{Act})$ or $f(FWF_{Act})$ or $f(MSF_{Act})$ & SHA_{Act}

6) Determine the HRDZ ((expected % attemperation - reference % attemperation)*Ref HR * HRCF).

$$HRD_{SHA,\ Exp\text{-}Ref} = NUHR_{Ref} * (HRCF_{SHA,\ Exp} - HRCF_{SHA,\ Ref}) / 100$$

7) After calculating the expected net unit heat rate $NUHR_{Exp}$ (see Section 3.7.22), determine the heat rate deviation due the difference between the actual superheat attemperation and the expected superheat attemperation.

$$HRD_{SHA,\ Act\text{-}Exp} = NUHR_{Exp} * (HRCF_{SHA,\ Act} - HRCF_{SHA,\ Exp}) / 100$$

3.7.17 Throttle Pressure

1) Establish TP_{Ref} value from reference curve (based on performance guarantee test or design data).

   $TP_{Ref} = f(GLoad_{Act})$ or $f(FWF_{Act})$ or $f(MSF_{Act})$

2) Establish TP_{Exp} value from expected level curve (based on current equipment, fuel and operating procedures.

   $TP_{Exp} = f(GLoad_{Act})$ or $f(FWF_{Act})$ or $f(MSF_{Act})$

3) From the heat rate correction curve(s) for throttle pressure (plotted as percent change in heat rate vs. steam pressure) determine the percent change in heat rate due to the reference, expected and actual steam pressures versus design steam pressure,

   a) $HRCF_{TP,\ Ref\text{-}Des} = f(TP_{Ref})$ or $f(TP_{Ref},\ GLoad_{Act})$
   b) $HRCF_{TP,\ Exp\text{-}Des} = f(TP_{Exp})$ or $f(TP_{Exp},\ GLoad_{Act})$
   c) $HRCF_{TP,\ Act\text{-}Des} = f(TP_{Act})$ or $f(TP_{Act},\ GLoad_{Act})$

   (If the turbo-generator manufacturer provides multiple curves, with each curve for a different load or flow, $HRCF_{TP,\ Ref\text{-}Des}$, $HRCF_{TP,\ Exp\text{-}Des}$, and $HRCF_{TP,\ Act\text{-}Des}$ will have to be determined by interpolating or using a spline fit between curves.)

4) Determine the heat rate deviation due to the difference between the expected steam pressure and the reference steam pressure

$$HRD_{TP,\ Exp\text{-}Ref} = ((1+ HRCF_{TP,\ Exp\text{-}Des}/100) / (1+ HRCF_{TP,\ Ref\text{-}Des}/100) - 1) * NUHR_{Ref}.$$

5) After calculating the expected net unit heat rate $NUHR_{Exp}$ (see Section 3.7.22), determine the heat rate deviation due to the difference between the actual steam pressure and the expected steam pressure.

$$HRD_{TP,\ Act\text{-}Exp} = ((1 + HRCF_{TP,\ Act\text{-}Des}/100) / (1+ HRCF_{TP,\ Exp\text{-}Des}/100) - 1) * NUHR_{Exp}$$

3.7.18 Throttle Temperature

1) Establish TT_{Ref} value from reference curve (based on acceptance test or design data).

$$TT_{Ref} = f(GLoad_{Act}) \text{ or } f(FWF_{Act}) \text{ or } f(MSF_{Act})$$

2) Establish TT_{Exp} value from expected level curve (based on current plant equipment design.

$$TT_{Exp} = f(GLoad_{Act}) \text{ or } f(FWF_{Act}) \text{ or } f(MSF_{Act})$$

3) From the heat rate correction curve(s) for throttle temperature (plotted as percent change in heat rate vs. steam temperature) determine the percent change in heat rate due to the reference, expected and actual steam temperatures versus design steam temperature, $HRCF_{TT,\ Ref\text{-}Des}$, $HRCF_{TT,\ Exp\text{-}Des}$, $HRCF_{TT,\ Act\text{-}Des}$.

a) $HRCF_{TT,\ Ref\text{-}Des} = f(TT_{Ref})$ or $f(TT_{Ref}, GLoad_{Act})$
b) $HRCF_{TT,\ Exp\text{-}Des} = f(TT_{Exp})$ or $f(TT_{Exp}, GLoad_{Act})$
c) $HRCF_{TT,\ Act\text{-}Des} = f(TT_{Act})$ or $f(TT_{Act}, GLoad_{Act})$

(If the manufacturer provides multiple curves, with each curve for a different load or flow, $HRCF_{TT,\ Ref\text{-}Des}$, $HRCF_{TT,\ Exp\text{-}Des}$, and $HRCF_{TT,\ Act\text{-}Des}$ will have to be determined by interpolating or using a spline fit between curves.)

4) Determine the heat rate deviation due to the difference between the expected steam temperature and the reference steam temperature.

$$HRD_{TT,\ Exp\text{-}Ref} = ((1+ HRCF_{TT,\ Exp\text{-}Des}/100) / (1+ HRCF_{TT,\ Ref\text{-}Des}/100) - 1) * NUHR_{Ref}$$

5) After calculating the expected net unit heat rate $NUHR_{Exp}$ (see Section 3.7.22), determine the heat rate deviation due to the difference between the actual steam temperature and the expected steam temperature.

$$HRD_{TT,\ Act\text{-}Exp} = ((1 + HRCF_{TT,\ Act\text{-}Des}/100) / (1+ HRCF_{TT,\ Exp\text{-}Des}/100) - 1) * NUHR_{Exp}.$$

3.7.19 Turbine Efficiency (any section, HP, IP, etc.)

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Usually, instead of continuously monitoring the efficiency of a turbine section, the efficiency is determined periodically at a repeatable valve position, usually all valves wide open. This actual VWO efficiency is then compared to an expected efficiency, at the same valve position.

1) Establish the reference section efficiency at VWO (a constant value from acceptance test or design data).

$$TE_{Ref} = \text{constant}$$

2) Establish the expected section efficiency at VWO based on current plant equipment design.

$$TE_{Exp} = \text{constant}$$

3) Establish the heat rate correction factor (usually a constant) for the % change in net unit heat rate for a 1% point change in section efficiency.

$$HRCF_{TE} = \text{constant}$$

4) Determine the heat rate deviation due to the difference between the reference and expected section efficiency.

$$HRD_{TE,\ Exp\text{-}Ref} = (TE_{Ref} - TE_{Exp}) * NUHR_{Ref} * HRCF_{TE} / 100.$$

5) After calculating the expected net unit heat rate $NUHR_{Exp}$ (see Section 3.7.22), determine the heat rate deviation due to the difference between the actual section efficiency and the expected section efficiency.

$$HRD_{TE,\ Act\text{-}Exp} = (TE_{Exp} - TE_{Act}) * NUHR_{Exp} * HRCF_{TE} / 100$$

3.7.20 Unit Auxiliary Power

1) Determine UAP_{Ref} value from reference curve, (plotted as station use in MW versus steam flow, feedwater flow or gross load, based on acceptance test or design data) at the period's steam flow, feedwater flow or gross load

$$UAP_{Ref} = f(GLoad_{Act}) \text{ or } f(FWF_{Act}) \text{ or } f(MSF_{Act})$$

2) Establish UAP_{Exp} value from expected level curve, (plotted as station use in MW versus steam flow, feedwater flow or gross load, based on current plant equipment design) at the period's steam flow, feedwater flow or gross load.

$$UAP_{Exp} = f(GLoad_{Act}) \text{ or } f(FWF_{Act}) \text{ or } f(MSF_{Act})$$

3) Calculate the heat rate deviation due to the difference between the expected and reference unit auxiliary power using either equation:

$$HRD_{UAP,\ Exp\text{-}Ref} = \{ (FS_{Act} * HHV_{Act}) / (GG_{Act} - HOU_{Act}(SSO_{Ref}+UAP_{Exp})) \}/1000 - \{ (FS_{Act} * HHV_{Act}) / (GG_{Act} - HOU_{Act}(SSO_{Ref}+UAP_{Ref})) \}/1000$$

4) After calculating the expected net unit heat rate $NUHR_{Exp}$ (see Section 3.7.22), determine the heat rate deviation due to the difference between the actual unit auxiliary power and the expected unit auxiliary power:

$$HRD_{UAP,\ Act\text{-}Exp} = \{ (FS_{Act} * HHV_{Act}) / (GG_{Act} - HOU_{Act}(SSO_{Ref}+UAP_{Act})) \}/1000 - \{ (FS_{Act} * HHV_{Act}) / (GG_{Act} - HOU_{Act}(SSO_{Ref}+UAP_{Exp})) \}/1000$$

3.7.21 Losses That Affect Boiler Efficiency (General Procedure)

All parameters that affect boiler efficiency (Exit Gas Temperature, Boiler Outlet O_2, Combustibles in Ash, Pulverizer Rejects, Moisture in Fuel, Hydrogen in Fuel, etc.) are handled in a similar manner. The general procedure is similar to that for the turbine cycle parameters. First, for each parameter, the heat rate deviation must be determined due to the difference between expected parameter value and reference parameter value. This is done for each parameter, by calculating the ratio of the reference boiler efficiency to the boiler efficiency calculated with all parameters at reference values, except the parameter in question (for that parameter the expected value is used). From this ratio, 1.0 is subtracted, then that difference is multiplied by the reference heat rate. Second, the heat rate deviations (expected versus reference) for all parameters are summed and added to the reference net heat rate to determine the expected net heat rate. Third, an expected boiler efficiency is calculated, using all expected values. Fourth, for each parameter, the heat rate deviation due to the difference between actual parameter value and expected parameter value is determined by calculating the ratio of the expected boiler efficiency to the boiler efficiency calculated with all parameters at expected values, except the parameter in question (for that parameter the actual value is used). From this ratio, 1.0 is subtracted, then that difference is multiplied by the expected heat rate.

It can be seen by the discussion above, that "boiler efficiency" will be calculated several times using various combinations of values. Rather than repeat these steps under each parameter (and a boiler efficiency is calculated twice under each parameter), the following steps show the procedure used to calculate a boiler efficiency.

1) If an ultimate coal analysis is not given, but only a proximate analysis (and sulfur and higher heating value), the proximate analysis will be used to estimate the as-fired percentages of carbon, hydrogen, nitrogen, and oxygen.

   a) $UltC = (1-0.01Z)*Cp + (5.0*FuelA) - (50.0*UltS)$
      Where $Z = FuelTM*100 + 110*FuelA + 10*UltS$

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$Qp = (100 * HHV) / (100 - Z)$
$Vp = 100*(100*FuelVM - 10*FuelA - 10*UltS) / (100 - Z)$
$Cp = 0.0015782 * Qp - 0.2226 * Vp + 37.69$

b) $UltH = (1-0.01Z)*Hp + (1.0*FuelA) - (1.5*UltS)$
Where $Z = FuelTM*100 + 110*FuelA + 10*UltS$
$Qp = (100 * HHV) / (100 - Z)$
$Vp = 100 * (100*FuelVM - 10*FuelA - 10*UltS) / (100 - Z)$
$Hp = 0.0001707 * Qp +0.0653 * Vp -2.92$

c) $UltN = 2.10 - 1.2*FuelVM$

d) $UltO_{Act} = 100 - (UltC_{Act} + UltH_{Act} + UltN_{Act} + UltS_{Act} + FuelA_{Act} + FuelTM_{Act})$

2) For any flue gas O_2 reading, if it is based on a wet analysis, as is usually the case, it must be converted to a dry analysis. If the %moisture in the flue gas is unknown, for coal fired units, 8% is typically used.

O_2 dry = O_2 wet / (1-%Moisture/100)

3) If the given exit gas temperature has not been corrected to the "reference inlet air temperature, correct it.

a) x1 = Reference Air Temp. * (Gas Temp In - Gas Temp Out) + Gas Temp In * (Gas Temp Out - Air Temp In) or

$x1 = AirT_{Ref} * (AHGIT - EGT) + AHGIT * (EGT - AirT)$

b) x2 = Gas Temp In - Air Temp In or

$x2 = AHGIT - AirT$

c) Exit gas temperature has not been corrected to the "reference inlet air temperature

EGT_{CorRAT} (°C)= x1 / x2

4) Calculate the specific heat of the air at the average air temperature in the AH (Inlet Temperature + Outlet Temperature) / 2. This is done by using ASME's PTC 4.1 Steam Generating Units, Figure 5.19-1 (see Appendix I, Figure I.1). = Cp_{AIR} (kcal/kg/K)

5) Calculate the amount of CO_2 in the flue gas by subtracting the O_2 percentage from the constant for (O_2 + CO_2).

CO_2 (%) = (O_2 + CO_2) - O_2 = $BO2CO2 - {}_{O2}$

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6) Calculate the specific heat of the flue gas using average gas temperature (Gas inlet and no-leakage gas outlet), carbon content of fuel, hydrogen content of fuel, and amount of CO_2 in the flue gas. See ASME's PTC 4.1 Steam Generating Units, Figure 5.19-3 (see Appendix I, Figure I.2). = Cp_{GAS} (kcal/kg/K)

7) If the exit gas temperature has not been corrected for AH leakage, correct it to zero leakage by:

$$EGT_{CorTNL} = (Leakage/100) * (Cp_{AIR} / Cp_{GAS}) * (Gas\ Outlet\ Temp. - Air\ Inlet\ Temp) + Gas\ Outlet\ Temp \quad or$$

$$EGT_{CorRAT,\ CorTNL}\ (°C) = (AHL\ /100) * (Cp_{AIR} / Cp_{GAS}) * (EGT_{CorRAT} - AirT) + EGT_{CorRAT}$$

8) Calculate the heating value of the dry refuse:

a) Hdp = (Weighted average LOI in ash/100) * 8055.6 or

Hdp (kcal/kg) = (CombTA/100) * 8055.6

9) Calculate the amount of dry refuse (kilograms) per kilogram of as-fired fuel

a) Wdp = Amount of ash in fuel / (100 - Weighted average LOI in ash) or

Wdp = (100 * FuelA) / (100 - CombTA)

10) Calculate the amount of carbon burned (kilograms) per kilogram of as-fired fuel

a) Cb = (Carbon in fuel) - (Wdp * Hdp / 8055.6) or

Cb = (UltC) - (Wdp * Hdp / 8055.6)

11) Calculate the amount of dry gas produced (kilograms) per kilogram of as-fired fuel

$$N_2 = 100 - CO_2 - O_2$$

$$X3 = (44.01 * CO_2 + 32.00 * O_2 + 28.02 * N_2) / (12.01 * CO_2)$$

$$Wg\ (kg/kg) = (Cb + (12.01/32.07) * UltS) * X3$$

12) Calculate the amount of dry air supplied (kilograms) per kilogram of as-fired fuel

$$N_2 = 100 - CO_2 - O_2$$

$$X4 = Cb + (12.01 / 32.07) * UltS$$

$$X5 = 12.01 * CO_2$$

$$X6 = 28.02 * N_2 * X4$$

$$Wa\ (kg/kg) = \{ (X6 / X5) - UltN \} / 0.7685$$

13) Calculate the enthalpy of the water vapor in the flue gas

$$Mg = 8.936 * UltH + Wma * Wa * FuelTM$$

$$X7 = 100 * 1.5 * Cb / (Mg*CO_2)$$

$$Pmg = Ambient\ Pressure / (1 + X7)$$

FGH (kcal/kg) = from steam tables at pressure of Pmg and temperature of EGT

14) Calculate the dry gas loss

$$BLDG\ (\%) = 100 * Wg * Cp_{GAS} * (EGT_{CorRAT,\ CorTNL} - AirT_{Ref}) / HHV$$

15) Calculate the combustible in ash loss

$$BLComb\ (\%) = 100 * Wdp * Hdp / HHV$$

16) Calculate the moisture in fuel loss

$$BLMC\ (\%) = \{100 * FuelTM * (FGH - Sat\ Liquid\ h\ at\ AirT) \} / HHV$$

17) Calculate the hydrogen in fuel loss

$$BLH2\ (\%) = \{100 * 8.936 * UltH * (FGH - Sat\ Liq\ h\ at\ AirT) \} / HHV$$

18) Calculate the moisture in air loss

$$BLMA\ (\%) = \{100 * Wma * Wa * (FGH - Sat\ Liq\ h\ at\ AirT) \} / HHV$$

19) Calculate the pulverizer reject loss

$$BLPyr\ (\%) = 100 * Wre * Hr / HHV$$

20) Boiler Efficiency = 100 - ΣLosses

$$BE = 100 - (BLDG + BLComb + BLMC + BLH2 + BLMA + BLPyr)$$

There are many variations on this procedure that individual utilities may use. For example, instead of tabulating reference and expected AH gas inlet temperatures, these could be calculated by reference and expected gas side efficiencies. Another variation would be to always set the "expected" moisture value to the current "actual" value, which would have the effect of moving

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the expected net heat rate up or down as the moisture content changes, and no heat rate deviation for moisture in fuel would ever be shown. A third variation would be include the moisture in air and/or pulverizer reject loss with the miscellaneous boiler losses. At some plants, the combustible loss is separated into two parameters, combustibles in fly ash and combustibles in bottom ash.

3.7.21.1 Boiler Outlet O_2

1) Establish BE_{Ref} value from reference curve (based on acceptance test or design data).

$$BE_{Ref} = f(FWF_{Act}) \text{ or } f(MSF_{Act}) \text{ or } F(GLoad_{Act})$$

2) Calculate a new boiler efficiency, using all reference values, except for using the expected boiler outlet O_2:

$$BE_{Ref+BOO2} = f(EGT_{CorTNL,\ CorRAT,\ Ref},\ BOO2_{Exp},\ CombTA_{Ref},\ BLMisc_{Ref},\ UltC_{Ref},\ UltH_{Ref},\ UltO_{Ref},\ UltS_{Ref},\ UltN_{Ref},\ FuelTM_{Ref},\ HHV_{Ref},\ BO2CO2_{Ref},\ AHGIT_{Ref},\ AHL_{Ref},\ Wre_{Ref},\ Hr_{Ref},\ \text{etc.})$$

3) Determine the heat rate deviation due to the difference between the reference and expected boiler outlet O_2:

$$HRD_{BOO2,\ Exp\text{-}Ref} = NUHR_{Ref} * \{ (BE_{Ref} / BE_{Ref+BOO2}) - 1 \}$$

4) After calculating all heat rate deviations due to differences between the expected and reference values, calculate the expected net unit heat rate $NUHR_{Exp}$ (see Section 3.7.22):

$$NUHR_{Exp} = \Sigma HRD_{i,\ Exp\text{-}Ref} + NUHR_{Ref}$$

5) Calculate the expected boiler efficiency, using all expected values:

$$BE_{Exp} = f(EGT_{CorTNL,\ CorRAT,\ Exp},\ BOO2_{Exp},\ CombTA_{Exp},\ BLMisc_{Exp}\ UltC_{Exp},\ UltH_{Exp},\ UltO_{Exp},\ UltS_{Exp},\ UltN_{Exp},\ FuelTM_{Exp},\ HHV_{Exp},\ BO2CO2_{Exp},\ AHGIT_{Exp},\ AHL_{Exp},\ Wre_{Exp},\ Hr_{Exp},\ \text{etc.})$$

6) Calculate a new boiler efficiency, using all expected values, except for using the actual boiler outlet O_2:

$$BE_{Exp+BOO2} = f(EGT_{CorTNL,\ CorRAT,\ Exp},\ BOO2_{Act},\ CombTA_{Exp},\ BLMisc_{Exp}\ UltC_{Exp},\ UltH_{Exp},\ UltO_{Exp},\ UltS_{Exp},\ UltN_{Exp},\ FuelTM_{Exp},\ HHV_{Exp},\ BO2CO2_{Exp},\ AHGIT_{Exp},\ AHL_{Exp},\ Wre_{Exp},\ Hr_{Exp},\ \text{etc.})$$

7) Calculate the heat rate deviation due to the difference between the actual and the expected boiler outlet O_2:

$$HRD_{BOO2,\ Act\text{-}Exp} = NUHR_{Exp} * \{ (BE_{Exp} / BE_{Exp+BOO2}) - 1 \}$$

3.7.21.2 Combustibles in Ash

1) Establish BE_{Ref} value from reference curve (based on acceptance test or design data).

$$BE_{Ref} = f(FWF_{Act}) \text{ or } f(MSF_{Act}) \text{ or } F(GLoad_{Act})$$

2) Calculate a new boiler efficiency, using all reference values, except for using the expected combustibles in ash:

$$BE_{Ref+CombTA} = f(EGT_{CorTNL,\ CorRAT,\ Ref},\ BOO2_{Ref},\ CombTA_{Exp},\ BLMisc_{Ref},\ UltC_{Ref},\ UltH_{Ref},\ UltO_{Ref},\ UltS_{Ref},\ UltN_{Ref},\ FuelTM_{Ref},\ HHV_{Ref},\ BO2CO2_{Ref},\ AHGIT_{Ref},\ AHL_{Ref},\ Wre_{Ref},\ Hr_{Ref},\ \text{etc.})$$

3) Determine the heat rate deviation due to the difference between the reference and expected combustibles in ash:

$$HRD_{CombTA,\ Exp\text{-}Ref} = NUHR_{Ref} * \{ (BE_{Ref} / BE_{Ref+CombTA}) - 1 \}$$

4) After calculating all heat rate deviations due to differences between the expected and reference values, calculate the expected net unit heat rate $NUHR_{Exp}$ (see Section 3.7.22):

$$NUHR_{Exp} = \Sigma HRD_{i,\ Exp\text{-}Ref} + NUHR_{Ref}$$

5) Calculate the expected boiler efficiency, using all expected values:

$$BE_{Exp} = f(EGT_{CorTNL,\ CorRAT,\ Exp},\ BOO2_{Exp},\ CombTA_{Exp},\ BLMisc_{Exp}\ UltC_{Exp},\ UltH_{Exp},\ UltO_{Exp},\ UltS_{Exp},\ UltN_{Exp},\ FuelTM_{Exp},\ HHV_{Exp},\ BO2CO2_{Exp},\ AHGIT_{Exp},\ AHL_{Exp},\ Wre_{Exp},\ Hr_{Exp},\ \text{etc.})$$

6) Calculate a new boiler efficiency, using all expected values, except for using the actual combustibles in ash:

$$BE_{Exp+CombTA} = f(EGT_{CorTNL,\ CorRAT,\ Exp},\ BOO2_{Exp},\ CombTA_{Act},\ BLMisc_{Exp}\ UltC_{Exp},\ UltH_{Exp},\ UltO_{Exp},\ UltS_{Exp},\ UltN_{Exp},\ FuelTM_{Exp},\ HHV_{Exp},\ BO2CO2_{Exp},\ AHGIT_{Exp},\ AHL_{Exp},\ Wre_{Exp},\ Hr_{Exp},\ \text{etc.})$$

7) Calculate the heat rate deviation due to the difference between the actual and the expected exit gas temperature (no-leakage, corrected to reference inlet air temperature):

$$HRD_{CombTA,\ Act\text{-}Exp} = NUHR_{Exp} * \{ (BE_{Exp} / BE_{Exp+\text{-}CombTA}) - 1 \}$$

3.7.21.3 Exit Gas Temperature

1) Establish BE_{Ref} value from reference curve (based on acceptance test or design data).

$BE_{Ref} = f(FWF_{Act})$ or $f(MSF_{Act})$ or $F(GLoad_{Act})$

2) Calculate a new boiler efficiency, using all reference values, except for using the expected exit gas temperature (corrected to reference inlet air and no-leakage)

$BE_{Ref+EGT}$ = f($EGT_{CorTNL,\ CorRAT,\ Exp}$, $BO2_{Ref}$, $CombTA_{Ref}$, $BLMisc_{Ref}$, $UltC_{Ref}$, $UltH_{Ref}$, $UltO_{Ref}$, $UltS_{Ref}$, $UltN_{Ref}$, $FuelTM_{Ref}$, HHV_{Ref}, $BO2CO2_{Ref}$, $AHGIT_{Ref}$, AHL_{Ref}, Wre_{Ref}, Hr_{Ref}, etc.)

3) Determine the heat rate deviation due to the difference between the reference and expected exit gas temperature (no-leakage, corrected to reference air temperature):

$HRD_{EGT,\ Exp-Ref} = NUHR_{Ref} * \{ (BE_{Ref} / BE_{Ref+EGT}) - 1 \}$

4) After calculating all heat rate deviations due to differences between the expected and reference values, calculate the expected net unit heat rate $NUHR_{Exp}$ (see Section 3.7.22):

$NUHR_{Exp} = \Sigma HRD_{i,\ Exp-Ref} + NUHR_{Ref}$

5) Calculate the expected boiler efficiency, using all expected values:

BE_{Exp} = f($EGT_{CorTNL,\ CorRAT,\ Exp}$, $BO2_{Exp}$, $CombTA_{Exp}$, $BLMisc_{Exp}$ $UltC_{Exp}$, $UltH_{Exp}$, $UltO_{Exp}$, $UltS_{Exp}$, $UltN_{Exp}$, $FuelTM_{Exp}$, HHV_{Exp}, $BO2CO2_{Exp}$, $AHGIT_{Exp}$, AHL_{Exp}, Wre_{Exp}, Hr_{Exp}, etc.)

6) Calculate a new boiler efficiency, using all expected values, except for using the actual exit gas temperature (corrected to reference inlet air and no-leakage)

$BE_{Exp+EGT}$ = f($EGT_{CorTNL,\ CorRAT,\ Act}$, $BO2_{Exp}$, $CombTA_{Exp}$, $BLMisc_{Exp}$ $UltC_{Exp}$, $UltH_{Exp}$, $UltO_{Exp}$, $UltS_{Exp}$, $UltN_{Exp}$, $FuelTM_{Exp}$, HHV_{Exp}, $BO2CO2_{Exp}$, $AHGIT_{Exp}$, AHL_{Exp}, Wre_{Exp}, Hr_{Exp}, etc.)

7) Calculate the heat rate deviation due to the difference between the actual and the expected exit gas temperature (no-leakage, corrected to reference inlet air temperature):

$HRD_{EGT,\ Act-Exp} = NUHR_{Exp} * \{ (BE_{Exp} / BE_{Exp+EGT}) - 1 \}$

3.7.21.4 Hydrogen in Fuel

1) Establish BE_{Ref} value from reference curve (based on acceptance test or design data).

$BE_{Ref} = f(FWF_{Act})$ or $f(MSF_{Act})$ or $F(GLoad_{Act})$

2) Calculate a new boiler efficiency, using all reference values, except for using the expected hydrogen fraction in fuel:

$BE_{Ref+UltH}$ = f($EGT_{CorTNL, CorRAT, Ref}$, $BO2_{Ref}$, $CombTA_{Ref}$, $BLMisc_{Ref}$, $UltC_{Ref}$, $UltH_{Exp}$, $UltO_{Ref}$, $UltS_{Ref}$, $UltN_{Ref}$, $FuelTM_{Ref}$, HHV_{Ref}, $BO2CO2_{Ref}$, $AHGIT_{Ref}$, AHL_{Ref}, Wre_{Ref}, Hr_{Ref}, etc.)

3) Determine the heat rate deviation due to the difference between the reference and expected hydrogen fraction in fuel:

$$HRD_{UltH, Exp-Ref} = NUHR_{Ref} * \{ (BE_{Ref} / BE_{Ref+UltH}) - 1 \}$$

4) After calculating all heat rate deviations due to differences between the expected and reference values, calculate the expected net unit heat rate $NUHR_{Exp}$ (see Section 3.7.22):

$$NUHR_{Exp} = \Sigma HRD_{i, Exp-Ref} + NUHR_{Ref}$$

5) Calculate the expected boiler efficiency, using all expected values:

BE_{Exp} = f($EGT_{CorTNL, CorRAT, Exp}$, $BO2_{Exp}$, $CombTA_{Exp}$, $BLMisc_{Exp}$ $UltC_{Exp}$, $UltH_{Exp}$, $UltO_{Exp}$, $UltS_{Exp}$, $UltN_{Exp}$, $FuelTM_{Exp}$, HHV_{Exp}, $BO2CO2_{Exp}$, $AHGIT_{Exp}$, AHL_{Exp}, Wre_{Exp}, Hr_{Exp}, etc.)

6) Calculate a new boiler efficiency, using all expected values, except for using the actual hydrogen fraction in fuel:

$BE_{Exp+UltH}$ = f($EGT_{CorTNL, CorRAT, Exp}$, $BO2_{Exp}$, $CombTA_{Exp}$, $BLMisc_{Exp}$ $UltC_{Exp}$, $UltH_{Act}$, $UltO_{Exp}$, $UltS_{Exp}$, $UltN_{Exp}$, $FuelTM_{Exp}$, HHV_{Exp}, $BO2CO2_{Exp}$, $AHGIT_{Exp}$, AHL_{Exp}, Wre_{Exp}, Hr_{Exp}, etc.)

7) Calculate the heat rate deviation due to the difference between the actual and the expected hydrogen fraction in fuel:

$$HRD_{UltH, Act-Exp} = NUHR_{Exp} * \{ (BE_{Exp} / BE_{Exp+UltH}) - 1 \}$$

3.7.21.5 Moisture in Fuel

1) Establish BE_{Ref} value from reference curve (based on acceptance test or design data).

$$BE_{Ref} = f(FWF_{Act}) \text{ or } f(MSF_{Act}) \text{ or } F(GLoad_{Act})$$

2) Calculate a new boiler efficiency, using all reference values, except for using the expected moisture in fuel:

$BE_{Ref+FuelTM}$ = f($EGT_{CorTNL, CorRAT, Ref}$, $BO2_{Ref}$, $CombTA_{Ref}$, $BLMisc_{Ref}$, $UltC_{Ref}$, $UltH_{Ref}$, $UltO_{Ref}$, $UltS_{Ref}$, $UltN_{Ref}$, $FuelTM_{Exp}$, HHV_{Ref}, $BO2CO2_{Ref}$, $AHGIT_{Ref}$, AHL_{Ref}, Wre_{Ref}, Hr_{Ref}, etc.)

3) Determine the heat rate deviation due to the difference between the reference and expected moisture in fuel:

$$HRD_{FuelTM,\ Exp\text{-}Ref} = NUHR_{Ref} * \{ (BE_{Ref} / BE_{Ref+FuelTM}) - 1 \}$$

4) After calculating all heat rate deviations due to differences between the expected and reference values, calculate the expected net unit heat rate $NUHR_{Exp}$ (see Section 3.7.22):

$$NUHR_{Exp} = \Sigma HRD_{i,\ Exp\text{-}Ref} + NUHR_{Ref}$$

5) Calculate the expected boiler efficiency, using all expected values:

$$BE_{Exp} = f(EGT_{CorTNL,\ CorRAT,\ Exp},\ BOO2_{Exp},\ CombTA_{Exp},\ BLMisc_{Exp}\ UltC_{Exp},\ UltH_{Exp},\ UltO_{Exp},\ UltS_{Exp},\ UltN_{Exp},\ FuelTM_{Exp},\ HHV_{Exp},\ BO2CO2_{Exp},\ AHGIT_{Exp},\ AHL_{Exp},\ Wre_{Exp},\ Hr_{Exp},\ \text{etc.})$$

6) Calculate a new boiler efficiency, using all expected values, except for using the actual moisture in fuel:

$$BE_{Exp+FuelTM} = f(EGT_{CorTNL,\ CorRAT,\ Exp},\ BOO2_{Exp},\ CombTA_{Exp},\ BLMisc_{Exp}\ UltC_{Exp},\ UltH_{Exp},\ UltO_{Exp},\ UltS_{Exp},\ UltN_{Exp},\ FuelTM_{Act},\ HHV_{Exp},\ BO2CO2_{Exp},\ AHGIT_{Exp},\ AHL_{Exp},\ Wre_{Exp},\ Hr_{Exp},\ \text{etc.})$$

7) Calculate the heat rate deviation due to the difference between the actual and the expected moisture in fuel:

$$HRD_{FuelTM,\ Act\text{-}Exp} = NUHR_{Exp} * \{ (BE_{Exp} / BE_{Exp+FuelTM}) - 1 \}$$

3.7.21.6 Pulverizer Rejects

1) Establish BE_{Ref} value from reference curve (based on acceptance test or design data).

$$BE_{Ref} = f(FWF_{Act}) \text{ or } f(MSF_{Act}) \text{ or } F(GLoad_{Act})$$

2) Calculate a new boiler efficiency, using all reference values, except for using the expected pulverizer reject rate and heating value:

$$BE_{Ref+Wre+Hr} = f(EGT_{CorTNL,\ CorRAT,\ Ref},\ BOO2_{Ref},\ CombTA_{Ref},\ BLMisc_{Ref},\ UltC_{Ref},\ UltH_{Ref},\ UltO_{Ref},\ UltS_{Ref},\ UltN_{Ref},\ FuelTM_{Ref},\ HHV_{Ref},\ BO2CO2_{Ref},\ AHGIT_{Ref},\ AHL_{Ref},\ Wre_{Exp},\ Hr_{Exp},\ \text{etc.})$$

3) Determine the heat rate deviation due to the difference between the reference and expected pulverizer reject rate and heating value:

$$HRD_{Wre,\ Exp\text{-}Ref} = NUHR_{Ref} * \{ (BE_{Ref} / BE_{Ref+Wre+Hr}) - 1 \}$$

4) After calculating all heat rate deviations due to differences between the expected and reference values, calculate the expected net unit heat rate $NUHR_{Exp}$ (see Section 3.7.22):

$$NUHR_{Exp} = \Sigma HRD_{i,\ Exp\text{-}Ref} + NUHR_{Ref}$$

5) Calculate the expected boiler efficiency, using all expected values:

$$BE_{Exp} = f(EGT_{CorTNL,\ CorRAT,\ Exp},\ BO2_{Exp},\ CombTA_{Exp},\ BLMisc_{Exp}\ UltC_{Exp},\ UltH_{Exp},\ UltO_{Exp},\ UltS_{Exp},\ UltN_{Exp},\ FuelTM_{Exp},\ HHV_{Exp},\ BO2CO2_{Exp},\ AHGIT_{Exp},\ AHL_{Exp},\ Wre_{Exp},\ Hr_{Exp},\ \text{etc.})$$

6) Calculate a new boiler efficiency, using all reference values, except for using the actual pulverizer reject rate and heating value:

$$BE_{Exp+Wre+Hr} = f(EGT_{CorTNL,\ CorRAT,\ Exp},\ BO2_{Exp},\ CombTA_{Exp},\ BLMisc_{Exp}\ UltC_{Exp},\ UltH_{Exp},\ UltO_{Exp},\ UltS_{Exp},\ UltN_{Exp},\ FuelTM_{Exp},\ HHV_{Exp},\ BO2CO2_{Exp},\ AHGIT_{Exp},\ AHL_{Exp},\ Wre_{Act},\ Hr_{Act},\ \text{etc.})$$

7) Calculate the heat rate deviation due to the difference between the actual and the expected pulverizer reject rate:

$$HRD_{Wre,\ Act\text{-}Exp} = NUHR_{Exp} * \{ (BE_{Exp} / BE_{Exp+Wre+Hr}) - 1 \}$$

3.7.22 Expected Net Heat Rate

After calculating the heat rate deviations (expected versus reference) for each parameter ($HRD_{i,\ Exp\text{-}Ref}$), sum up the heat rate deviations (expected versus reference) for each parameter and add the sum to the reference net heat rate to calculate the expected net heat rate.

$$NUHR_{Exp} = NUHR_{Ref} + \Sigma HRD_{i,\ Exp\text{-}Ref}$$

3.7.23 Unaccountable

After calculating the heat rate deviations for all parameters that are tracked, the last parameter to calculate is "Unaccountables." This parameter includes the losses for any areas that are not tracked, such a low pressure turbine efficiency, cycle isolation, etc. It also includes errors in any measured parameters (such as exit gas temperature) as well as fuel measurements (quantity and quality) as well as electrical load and station use measurements.

$$HRD_{UnAcc,\ Act\text{-}Exp} = NUHR_{Act} - NUHR_{Exp} - \Sigma HRD_{i,\ Act\text{-}Exp}$$

3.7.24 Calculation of Cost Due to Heat Rate Deviations

The last step is a very important one that is often overlooked. Each heat rate deviation must be converted to a cost in rupees or lakhs. Engineers who are involved daily with heat rate deviations may know the magnitude of problems, but most people do not. Therefore it is imperative that the heat rate deviations be converted to cost. This is done by multiplying by the net generation for the period, and multiplying by the fuel cost. This will convert the heat rate deviation (kJ/kWh or kcal/kWh) to rupees or lakhs. If instead of multiplying by net generation, net load is used instead, the resulting cost will be a rate (rupees per hour or lakhs per hour).

Table 3.1 Example of a Heat Rate Deviation Report

	Parameter Values			Heat Rate Deviation	
Parameter	**Actual**	**Expected**	**Variance**	**kcal/kWh**	**Cost Rs**
Throttle Pressure, kg/cm2	241.00	241.00	0.00	0.0	0
Throttle Temperature, deg C	535.70	538.00	-2.30	1.6	2,951
Reheat Temperature, deg C	539.30	538.00	1.30	-0.8	-1,505
Station Service Operating Hours, %	3.36	3.11	0.25	5.8	10,621
Station Service Idle Hours., %	0.00			0.0	0
Superheat Attemperation, %	2.62	2.98	-0.36	0.0	0
Reheat Attemperation, %	0.03	0.50	-0.47	-2.4	-4,482
Exhaust Pressure, mm Hg	1.69	1.50	0.19	6.4	11,782
Makeup Water, %		0.50		0.0	0
Exit Gas Temperature, deg C	310.81	289.20	21.61	11.2	20,446
Total Air, %	116.80	118.42	-1.62	-1.3	-2,425
Combustibles in Flyash, %	2.94	3.00	-0.06	-0.1	-262
Moisture in Coal, %	5.11	6.40	-1.29	-2.8	-5,178
Pulverizer Pyrite Loss, %	0.10	0.20	-0.10	-2.4	-4,366
HP Turbine Efficiency, %	84.30	90.51	-6.21	33.2	60,879
IP Turbine Efficiency, %	90.89	91.00	-0.11	0.3	538
Building Heat, %	0.02	0.00	0.02	0.1	114
Both #1 Heaters, hrs. out of service	8.00			0.5	856
Unaccountable Heat Rate Dev.				12.0	21,972
Total Btu Dev.				61.1	111,941

Example of a report format showing parameter values, heat rate deviations, and cost of heat rate deviations.

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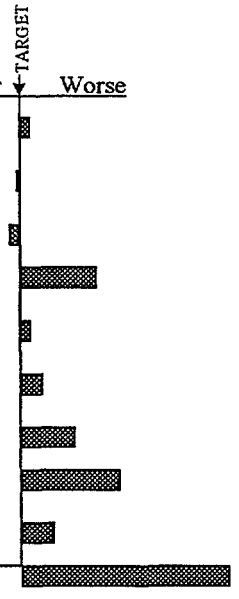
OPERATOR ON SHIFT: R.C. Helton

Marvil Fossil Plant - Unit 1 - Controllable Losses - 1999/01/01 13:00 to 13:05

Load: 210 MW
CCW Inlet Temp: 20 C
Boiler Inlet Air Temp: 25 C
Fuel Cost: Rs 171/10^6 kcal

	Expected	Target	Actual	HR Dev from Target (kcal/kWh)	Cost of HR Dev Rs/ h
Throttle Temperature (C)	535	530	528	1.3	46
Throttle Pressure (kg/cm^2)	150.0	150.0	151.5	- 0.1	-5
Hot Reheat Temperature (C)	535	535	537	-1.2	-42
Exit Gas Temp, NL (C)	136	140	145	6.0	215
Reheat Attemporation (%)	0	0	0.3	1.9	68
Boiler Outlet O_2 (%)	2.5	2.5	2.8	2.2	80
Unit Auxiliary Power (MW)	10.5	11.0	11.5	5.2	185
Condenser Pressure (mm Hg)	43	43	48	10.2	364
Final Feedwater Temp (C)	246	245	240	2.7	95
Sum of current deviations/costs				28.2	1006



Cost last shift (Rs) 7654
Cost this shift (Rs) 6314
Cost past Hour (Rs) 1052

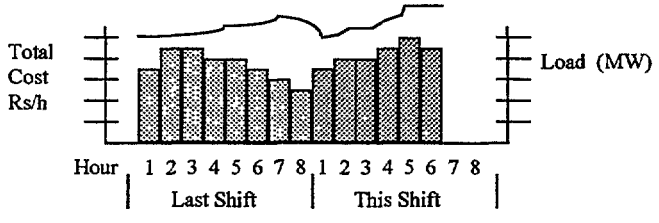


Figure 3.11 Example of a Control Room Heat Rate Display - Operator Controllables

Notice that the Operator Controllable display (Figure 3.11) compares the current parameter values (the "Actual" column) to the "Target" value and not the "Expected" value. These targets are set during the Heat Rate Improvement Plan process (see Section 5), and are not necessarily the same as the "Expected" parameter values. In Figure 3.11, the "Expected" EGT is 136 ^{0}C, where the Target is 140 ^{0}C. This could be due to deterioration of the AH baskets. Since this is a recoverable loss, the "Expected" value is not changed. Perhaps the baskets are going to be replaced next year. This year's target then is higher than the expected, and the operator is attempting to reach the "Target" value, not the "Expected" value.

In order for "controllable loss monitoring" to be effective, the values the operators are supposed to obtain must be realistic. This level is called a "Target". However, for engineers and managers doing long term planning, they need to compare current performance to a different standard, the best achievable value based on the configuration of the unit, excluding any deterioration. These are called "Expected" levels.

3.8 Example

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3.8.1 Unit Characteristics - Reference and Expected

3.8.1.1 Condenser Reference and Expected Characteristics - Marvil Unit 1

Name	*Abbrev.*	*Units*	*Ref.*	*Expected*
External Diameter of Tubes	TubeOD	mm	22.2	22.2
Internal Diameter of Tubes	TubeID	mm	19.7	20.4
Tube Wall Thickness	TubeBWG	BWG	18	20
Number of Tubes	TubeNo	-	11010	11010
Tube Material	TubeMat	-	Admiralty	Stain. St.
HEI Material & Gauge Correction Factor	TubeMGC	-	1.00	0.82
Surface Area	CSArea	m^2	6970	6970
Condenser Circulating Water Flow Rate	CCWF	kg/h	25.76E6	25.76E6
Cleanliness Factor	CF	%	85	85

3.8.1.2 Turbine Cycle Reference and Expected Characteristics - Marvil Unit 1

Name	*Abbrev.*	*Units*	*Ref.*	*Expected*
Heat Added by Boiler Feedwater Pumps	PumpQ	kcal/h	2.613E6	2.613E6
Heat Rejected by Auxiliary Condensers	AuxDuty	kcal/h	0	0
Throttle Temperature	TT	°C	565	565
Throttle Pressure	TP	kg/cm^2a	127.59	127.59
HP Turbine Efficiency	HPT	%	83.27	83.27
IP Turbine Efficiency	IPT	%	90.57	90.57
Makeup Water Flow	MU	%	0	0.8
Station Service (Prorated) Operating	SSO	MW	0.5	0.75
Superheat Attemperation	SHA	%	0.4	1.0
Reheat Attemperation	RHA	%	0	0
Steam Supplied to APH Coils	SCAPH	%	0	0

3.8.1.3 Boiler Cycle Reference and Expected Characteristics - Marvil Unit 1

Name	*Abbrev.*	*Units*	*Ref.*	*Expected*
Coal Analysis - Carbon	UltC	-	0.5351	0.5351
Coal Analysis - Hydrogen	UltH	-	0.05	0.05
Coal Analysis - Oxygen	UltO	-	0.1882	0.1882
Coal Analysis - Nitrogen	UltN	-	0.0115	0.0115
Coal Analysis - Sulfur	UltS	-	0.0101	0.0101
Coal Analysis - Ash	FuelA	-	0.1004	0.1004
Coal Analysis - Moisture	FuelTM	-	0.1047	0.1047
Coal Analysis - Higher Heating Value	HHV	kcal/kg	5847	5847
Air Heater Inlet Air Temperature	AirT	°C	27	-------

Air Heater Leakage	AHL	%	7	12
Air Heater Air Outlet Temperature	AHAOT	°C	300	300
Air Heater Gas Inlet Temperature	AHGIT	°C	340	340
Boiler Outlet Sum of O_2 and CO_2	BO2CO2	%	18	18
Combustibles in Ash	CombTA	%	2	2.2
Moisture in Inlet Air	Wma	kg/kg	0.011	0.011
Miscellaneous Boiler Losses	BLMisc	%	1.0	1.0
Pulverizer Reject Rate	Wre	kg/kg	0.003	0.003
Pulverizer Reject Heating Value	Hr	kcal/kg	2000	2000
Heat Input for Hot Startup	HIHot	kcal	-	96E6
Heat Input for Cold Startup	HICold	kcal	-	210E6

3.8.1.4 Reference and Expected Curves of Marvil Unit 1

Name	*Abbrev.*	*Units*	*X Axis*		*Curve Coeff.* $y = C_0 + C_1X + C_2X^2 + C_{3X}^3$			
			Name	*Units*	C_0	C_1	C_2	C_3
Ref. Net Unit HR at CCWT = 8 °C	$NUHR_{REF, S1}$	kcal/kWh	GLoad	MW	2608	-2.185	0.0051	0
Ref. Net Unit HR at CCWT = 13 °C	$NUHR_{REF, S2}$	kcal/kWh	GLoad	MW	2670	-2.7148	0.0064	0
Ref. Net Unit HR at CCWT = 18 °C	$NUHR_{REF, S3}$	kcal/kWh	GLoad	MW	2710.2	-2.7485	0.0061	0
Ref. Net Unit HR at CCWT = 22 °C	$NUHR_{REF, S4}$	kcal/kWh	GLoad	MW	2786	-3.2936	0.0073	0
Ref. Gross Turbine Cycle Heat Rate corrected to design condenser press.	$GTCHR_{REF, CorDCP}$	kcal/kWh	GLoad	MW	2315.9	-3.1987	0.008	0
Ref. Net Unit Heat Rate corrected to design condenser pressure	$NUHR_{REF, CorDCP}$	kcal/kWh	GLoad	MW	3035.8	-5.6864	0.0127	0
Ref. Final Feedwater Temperature	$FFWT_{REF}$	°C	FWF	kg/h	169	2.26E-04	-9.31E-11	0
Exp. Final Feedwater Temperature	$FFWT_{EXP}$	°C	FWF	kg/h	171	2.147E-04	-8.85E-11	0
Ref. Unit Auxiliary Power	UAP_{REF}	MW	FWF	kg/h	7.999	-9.513E-6	1.754E-11	0
Exp. Unit Auxiliary Power	UAP_{EXP}	MW	FWF	kg/h	8.294	-1.131E-5	2.239E-11	0
Ref. Steam Temperature at Intercept Valve (FWF above 409,000kg/h)	$HRHIV_{REF}$	°C	FWF	kg/h	565	0	0	0
Ref. Steam Temperature at Intercept Valve (FWF below 409,000kg/h)	$HRHIV_{REF}$	°C	FWF	kg/h	470.61	2.3077E-4	0	0
Exp. Steam Temperature at Intercept Valve (FWF above 409,000kg/h)	$HRHIV_{EXP}$	°C	FWF	kg/h	565	0	0	0
Exp. Steam Temperature at Intercept Valve (FWF below 409,000kg/h)	$HRHIV_{EXP}$	°C	FWF	kg/h	470.61	2.3077E-4	0	0
Ref. No Leakage Exit Gas Temp, corr. to $AirT_{Ref}$	$EGT_{REF, CorTNL, CorRAT}$	°C	FWF	kg/h	128.33	2.796E-5	0	0
Exp. No Leakage Exit Gas Temp, corr. to $AirT_{Ref}$	$EGT_{EXP, CorTNL, CorRAT}$	°C	FWF	kg/h	124.33	2.796E-5	0	0
Ref. Boiler Outlet O_2	$BOO2_{REF}$	%	FWF	kg/h	6.525	-1.279E-5	8.797E-12	0
Exp. Boiler Outlet O_2	$BOO2_{EXP}$	%	FWF	kg/h	6.275	-1.279E-5	8.797E-12	0

Ref. Boiler Efficiency	BE_{Ref}	%	FWF	kg/h				
HEI Temp Correction Factor <21°C	HEITCF	-	CCWTI	°C	+ 6.7000E-01	+ 1.5769E-02	+ 1.2813E-04	-6.0304E-06
HEI Temp Correction Factor >21°C	HEITCF	-	CCWTI	°C	6.5184E-01	+ 2.5560E-02	- 5.1698E-04	4.0969E-06
Generator Loss Curve	GLoss	kW	GLoad	MW	-88325.1	1455.56	-7.77646	0.01386

3.8.2 Operating Data From Marvil Unit 1 - May 01 - 31, 2000

3.8.2.1 Parameter Values

Name	*Abbrev.*	*Units*	*Value*
Aux. Steam Usage (to APH Coils)	SCAPH	%	n/a
Boiler Differential Pressure	BDP	%	n/a
Combustibles in Ash	CombTA	%	2.4
Condenser Pressure	CP	mm Hg	68
AH Exit Gas Temperature	EGT	°C	148
Boiler Outlet O2	BOO2	%	2.3
Final Feedwater Temperature (w/ no heaters b/p)	FFWT	°C	265
Feedwater Heater #8 - Hours bypassed	FWH8	h	100
Feedwater Heater #7 - Hours bypassed	FWH7	h	0
Feedwater heater #6 - Hours bypassed	FWH6	h	0
Throttle Pressure	TP	kg/cm^2a	125.00
Throttle Temperature	TT	°C	562
Makeup	MU	%	1.1
Reheat Attemperation Flow	RHA	%	1.3
Station Service Idle	SSI	MW	1.829109
Station Service Operating	SSO	MW	0.524376
Auxiliary Power Usage	UAP	MW	11.21565
Hot Reheat Steam Temperature at Intercept Valve	HRHIV	°C	557
Superheat Attemperation Flow	SHA	%	2.0
HP Turbine Efficiency (at VWO)	HPT	%	82.1
IP Turbine Efficiency	IPT	%	87.2
Pulverizer Reject Rate	Wre	kg/kg	0.004
Coal Analysis - Moisture	FuelTM	-	0.2566
Coal Analysis - Hydrogen	UltH	-	0.0441

3.8.2.2 Miscellaneous Values

Name	*Abbrev.*	*Units*	*Value*
Ambient Air Temperature		°C	18.8
Condenser Cooling Water Inlet Temperature	CCWTI	°C	20
Gross Generation	GG	MWh	133492
Hours of Use	HOU	h	667.46
Feedwater Flow	FWF	kg/h	596,000
Condenser Circulating Water Flow	CCWF	kg/h	25.76E6
Amount of Fuel Burned	FS	kg	92,558,895

Coal Analysis - Carbon	UltC	-	0.2815
Coal Analysis - Oxygen	UltO	-	0.2692
Coal Analysis - Nitrogen	UltN	-	0.0039
Coal Analysis - Sulfur	UltS	-	0.0016
Coal Analysis - Ash	FuelA	-	0.1432
Coal Analysis - Higher Heating Value	HHV	kcal/kg	3599
Fuel Cost	FC	Rs/Mkcal	170
Air Heater Leakage	AHL	%	14
Moisture in Air	Wma	kg/kg	0.013
Heating Value of Pulverizer Rejects	Hr	kcal/kg	1800
Flue Gas Temperature Entering Air Heater	AHGIT	°C	340
Air Temperature Leaving Air Heater	AHAOT	°C	310
Number of Cold Start-ups	NoCold	-	0
Number of Hot Startups	NoHot	-	1
Air Heater Inlet Sum of Flue Gas O_2 and CO2	BO2CO2	%	18
Miscellaneous Boiler Losses	BLMisc	%	1
Ambient Pressure		bar a	1

3.8.3 Heat Rate Correction Factors

Abbrev	Units	Z Axis			X Axis					
		Name	Value	Units	Name	Units	C0	C1	C2	C3
$HRCF_{CP}$	Note 1	FWF	596000	kg/h	CP	mmHg	- 2.2499	-4.1176E-2	2.3196E-03	-1.2713E-5
$HRCF_{CP}$	Note 1	FWF	375000	kg/h	CP	mmHg	6.2185	6.1820E-2	1.6981E-3	-1.0224E-5
$HRCF_{RHA}$	Note 1	RHA	1	%	GLoad	MW	-0.153929	0.92557E-2	-0.6643E-4	0.14583E-6
$HRCF_{SHA}$	Note 1	SHA	2	%	GLoad	MW	0.517304	-0.3421E-2	-0.6608E-5	0.62502E-7
$HRCF_{TT}$	Note 1	-	-	-	TT	°C	12.216	-0.021600	0	0
$HRCF_{HRHIV}$	Note 1	FWF	596000	kg/h	HRHIV	°C	13.234	-0.0234	0	0
$HRCF_{HRHIV}$	Note 1	FWF	270000	kg/h	HRHIV	°C	16.288	-0.0288	0	0
$HRCF_{TP}$	Note 1	FWF	596000	kg/h	TP	kg/cm^2	7.621740	-0.059738	0	0
$HRCF_{TP}$	Note 1	FWF	270000	kg/h	TP	kg/cm^2	17.05818	-0.133699	0	0
$HRCF_{MU}$	Note 1	MU	1	%	FWF	kg/h	0.5409	- 8E-08	2E-13	0
$HRCF_{HPT}$	Note 1	ΔHPT	1 % pt	-	-	-	0.2152	0	0	0
$HRCF_{IPT}$	Note 1	ΔIPT	1 % pt	-	-	-	0.1958	0	0	0
$HRCF_{FFWT}$	Note 1	ΔFFWT	2.78	C	FWF	kg/h	9.1667E-03	1.8456E-7	0	0
$HRCF_{FWH8}$	Note 1	-	-	-	FWF	kg/h	0.714429	0	0	0

Note 1: % Change in Net Unit Heat Rate

3.8.4 Calculations

3.8.4.1 Calculation of Reference Net Heat Rate

Calculate average gross load for time interval ($GLoad_{Act}$).

$$GLoad_{Act} = GG_{Act} / HOU_{Act}$$
$$= 133492 \text{ MWh} / 667.46 \text{ h}$$
$$= 200\text{MW}$$

Calculate the heat rate from each of the four reference heat rate curves (each curve being at a different condenser water inlet temperature), at the average gross load.

$$NUHR_{Ref,\ 8\,°C} = f(GLoad_{Act}) = f(200\text{MW}) = 2375 \text{ kcal/kWh}$$

$$NUHR_{Ref,\ 13\,°C} = f(GLoad_{Act}) = f(200\text{MW}) = 2383 \text{ kcal/kWh}$$

$$NUHR_{Ref,\ 18\,°C} = f(GLoad_{Act}) = f(200\text{MW}) = 2405 \text{ kcal/kWh}$$

$$NUHR_{Ref,\ 22\,°C} = f(GLoad_{Act}) = f(200\text{MW}) = 2419 \text{ kcal/kWh}$$

Perform a spline fit (see Appendix F) of the heat rates from above, using the actual average cooling water temperature (20 °C), to determine the reference net heat rate $NUHR_{Ref}$.

$$NUHR_{Ref} = f(NUHR_{Ref,\ S1},\ CCWTI_{Ref,\ S1},\ NUHR_{Ref,\ S2},\ CCWTI_{Ref,\ S2},\ NUHR_{Ref,\ S3},\ CCWTI_{Ref,\ S3},\ CCWTI_{Act})$$

$$=2413.2 \text{ kcal/kWh}$$

3.8.4.2 Actual Net Unit Heat Rate

$$NUHR_{Act} = (FS_{Act}*HHV_{Act}) / (GG_{Act} - (SSO_{Act} + UAP_{Act})*HOU_{Act} - SSI_{Act}*(PH-HOU_{Act}))/1000$$

$$= (92558895\text{kg}*3599\text{kcal/kg}) / (133492\text{MWh} - (0.5244\text{MW} + 11.2156\text{MW})*667.46\text{h} - 1.8291\text{MW}*(744-667.46)\text{h})/1000$$

$$= (333119.46\text{E6kcal} / (133492 - 7836 - 140)\text{MWh}/1000$$
$$= 2654 \text{ kcal/kWh}$$

3.8.4.3 Condenser Pressure

Steps "1" through "4" calculate some values that will be needed in the following steps:

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1) Calculate heat added by the boiler feed water pumps ($PumpQ_{Ref,Exp}$). Obtain from design or acceptance test curves.

$$PumpQ_{Ref,Exp} = f(MSF_{Act}, \text{ or } FWF_{Act}, \text{ or } GLoad_{Act}) = 2.613E6 \text{ kcal/h}$$

2) If the unit has any auxiliary equipment condensers, calculate their total duty or rate of heat transferred out of the turbine cycle, ($AuxDuty_{Ref,Exp}$) from curves developed across the flow or load range using thermal modeling software.

$$AuxDuty_{Ref,Exp} = f(MSF_{Act}, \text{ or } FWF_{Act}, \text{ or } GLoad_{Act}) = 0 \text{ kcal/h}$$

3) Calculate Generator Loss ($GLoss_{Ref, Exp}$) from manufacturer's curves.

$$\begin{aligned} GLoss_{Ref, Exp} &= f(GLoad_{Act}) \\ &= -88325.1 + 1455.56*200 - 7.77646*200^2 + 0.01386*200^3 \\ &= 2608.5 \text{ kW} \end{aligned}$$

4) Calculate HEI Inlet Water Temperature Correction Factor ($HEITCF_{Ref, Exp}$), based on the actual circulating water inlet temperature ($CCWTI_{Act}$).

$$\begin{aligned} HEITCF_{Ref, Exp} &= f(CCWTI_{Act}) \\ &= + 6.7000E-1 + 1.5769E-2*20 + 1.2813E-4*20^2 - 6.0304E-6*20^3 \\ &= 0.988 \end{aligned}$$

Steps "5" through "22" are calculating the reference condenser pressure, based on the "reference" turbine cycle heat rate curve and "reference" condenser characteristics (tube material, tube wall thickness, etc.).

5) Establish reference value for Condenser Circulating water flow rate ($CCWF_{Ref}$) from the condenser characteristics (some constant value), or from a curve (i.e., $CCWF_{Ref}$ vs. lake elevation).

$$CCWF_{Ref} = \text{constant or f(water elevation)} = 25.76E6 \text{ kg/h}$$

6) Calculate the reference value for Water Velocity through the tubes ($TubeWV_{Ref}$), given the flow rate ($CCWF_{Ref}$), the number of tubes ($TubeNo_{Ref}$), and the internal diameter of the tubes ($TubeID_{Ref}$).

$$\begin{aligned} TubeWV_{Ref} &= CCWF_{Ref} \ / \ 1000 \text{ kg/m}^3 \ / \ (3.14159 * (TubeID_{Ref}/2/1000)^2 * TubeNo_{Ref}) / 3600 \text{ sec/hr}) \\ &= 25.76E6\text{kg/h} / 1000 \text{ kg/m}^3 / (3.14159 * (19.7\text{mm}/2/1000)^2 * 11010 * 3600 \text{ sec/hr}) \\ &= 2.132 \text{ m/s} \end{aligned}$$

7) Look up the reference value for the Heat Exchange Institute "Material and Gauge Correction Factor" ($TubeMGC_{Ref}$) as a function of the material of the condenser tubes ($TubeMat_{Ref}$) and the tube wall thickness ($TubeBWG_{Ref}$).

$$TubeMGC_{Ref} = f(TubeMat_{Ref}, TubeBWG_{Ref}) = 1.00$$

8) Establish reference value for HEI's "Uncorrected Heat Transfer Coefficient" ($TubeU_{Ref}$) based on tube external diameter ($TubeOD_{Ref}$) and water velocity through the tubes ($TubeWV_{Ref}$).

$$TubeU_{Ref} = f(TubeOD_{Ref}, TubeWV_{Ref}) = 3396 \text{ kcal/(h m}^2\text{ K)}$$

9) Establish reference value for Condenser Surface Area ($CSArea_{Ref}$) = 6970 m^2

10) Calculate Reference Gross Turbine Cycle Heat Rate (corrected to design condenser pressure) $GTCHR_{Ref, CorDCP}$ as a function of the period's average gross load ($GLoad_{Act}$).

$$\begin{aligned} GTCHR_{Ref, CorDCP} &= f(GLoad_{Act}) \\ &= 2315.9 - 3.1987*(200MW) + 0.008*(200MW)^2 \\ &= 1996.2 \text{ kcal/kWh} \end{aligned}$$

11) Use design condenser pressure (CP_{Des}) as the first guess for calculating CP_{Ref}.

$$CP_{Ref} = CP_{Des} = 50 \text{ mmHg}$$

12) From turbo/generator manufacturer's heat rate correction curve(s) for condenser pressure (percent change in heat rate vs. condenser pressure) determine the $HRCF_{CP, Ref\text{-}Des}$.

$$\begin{aligned} HRCF_{CP, Ref\text{-}Des} &= f(CP_{Ref}, MSF_{Act}) \text{ or } f(CP_{Ref}, GLoad_{Act}) \\ &= f(50 \text{ mm Hg}, 596{,}000\text{kg/h FWF}) \\ &= -2.2499 - 4.1176E\text{-}2*(50) + 2.3196E\text{-}3*(50)^2 - 1.2713E\text{-}5*(50)^3 \\ &= -0.099 \end{aligned}$$

13) Calculate the gross turbine cycle heat rate, corrected to the reference condenser pressure, $GTCHR_{Ref,CorRCP}$.

$$\begin{aligned} GTCHR_{Ref,CorRCP} &= GTCHR_{Ref, CorDCP} * (1 + HRCF_{CP, Ref\text{-}Des} /100) \\ &= 1996.2 \text{ kcal/kWh} * (1 - 0.099/100) \\ &= 1994.2 \text{ kcal/kWh} \end{aligned}$$

14) Calculate the rate of heat rejection in the condenser, Condenser Duty (Duty) by taking the total heat input to the turbine cycle, including the heat added by the pumps, and then

subtracting the energy converted to electricity, the losses in the generator, and heat rejected in auxiliary condensers (if any).

$$Duty_{Ref} = (GTCHR_{Ref,CorRCP} * GLoad_{Act} * 1000kW/MW) - (860{,}000\ kcal/h/MW * GLoad_{Act}) - (860\ kcal/h/kW * GLoss_{Ref,\ Exp}) + PumpQ_{Ref,Exp} - AuxDuty_{Ref,Exp}$$

$$= (1994.2\ kcal/kWh * 200MW*1000kW/MW) - (860{,}000\ kcal/h/MW * 200MW) - (860\ kcal/h/kW * 2608.5\ kW) + 2.613E6\ kcal/h - 0$$

$$= 227{,}209{,}690\ kcal/h$$

15) Calculate Heat Transmittance (HTran) by dividing the heat load on the condenser by the surface area of the condenser.

$$HTran_{Ref} = Duty_{Ref} / CSArea_{Ref}$$
$$= 227{,}209{,}690\ kcal/h / 6970\ m^2$$
$$= 32{,}598\ kcal/h/\ m^2$$

16) Calculate HEI Heat Transfer Coefficient ($HEIHTC_{Ref}$), by multiplying the Uncorrected Heat transfer Coefficient by the Tube Material and Gauge Correction Factor and by multiplying by the Inlet Water Temperature Correction Factor. This is the heat transfer coefficient for a "clean" tube.

$$HEIHTC_{Ref} = TubeU_{Ref} * TubeMGC_{Ref} * HEITCF_{Ref,\ Exp}.$$
$$= 3396\ kcal/(h\ m^2\ K) * 1.0 * 0.988$$
$$= 3355.2\ kcal/(h\ m^2\ K)$$

17) Calculate the Reference Heat Transfer Coefficient (HTC_{Ref}) by multiplying the HEI Heat Transfer Coefficient by the cleanliness factor that could be maintained. Normally a cleanliness factor of 85% is used.

$$HTC_{Ref} = HEIHTC_{Ref} * CF_{Ref} / 100$$
$$= 3355.2\ kcal/(h\ m^2\ K) * 85/100$$
$$= 2852.0\ kcal/(h\ m^2\ K)$$

18) Calculate Logarithmic Mean Temperature Difference ($LMTD_{Ref}$) by dividing the heat transmittance by the reference heat transfer coefficient.

$$LMTD_{Ref} = HTran_{Ref} / HTC_{Ref}.$$
$$= 32{,}598\ kcal/h/\ m^2/\ 2852.0\ kcal/(h\ m^2\ K)$$
$$= 11.43\ °C$$

19) Calculate the Outlet Water Temperature (CCWTO) by the equation Duty is equal to CCW flow rate multiplied by the heat capacity of the water multiplied by the temperature rise of the water. This equation can be rearranged to give:

$$CCWTO_{Ref} = (Duty_{Ref} / (CCWF_{Ref} * 1.0 \text{ kcal/kg C})) + CCWTI_{Act}$$
$$= (227{,}209{,}690 \text{ kcal/h} / (25.76E6 \text{ kg/h} * 1.0 \text{ kcal/kg C})) + 20 \text{ °C}$$
$$= 28.82 \text{ °C}$$

20) Calculate Circulating Water Temperature Rise.

$$CCWTR_{Ref} = CCWTO_{Ref} - CCWTI_{Act}.$$
$$= 28.82 \text{ °C} - 20 \text{ °C}$$
$$= 8.82 \text{ °C}$$

21) Calculate the reference Exhaust Steam Temperature.

$$EST_{Ref} = (CCWTO_{Ref} * e^{(CCWTR / LMTD)} - CCWTI_{Act}) / (e^{(CCWTR / LMTD)} - 1).$$
$$= (28.82 \text{ °C} * e^{(8.82 \text{ °C}/11.43 \text{ °C})} - 20 \text{ °C}) / (e^{(8.82 \text{ °C}/11.43 \text{ °C})} - 1 \text{ °C})$$
$$= 36.4 \text{ °C}$$

22) Calculate CP using EST_{Ref} and steam tables.

$$CP_{REF} = f(EST(36.4)) = 45.54 \text{ mm Hg}$$

23) Check to see if the condenser pressure used in steps "11" through "21" is within 0.2mm of the calculated condenser pressure (CP from step "22"). If the difference is greater than 0.2mm, reset CP_{Ref} = CP and return to step "12". If the difference is smaller than 0.2mm, continue with step 24. The initial guess was 50 mmHg, and the result of the first iteration was 45.54. Returning to step 12 and reiterating will eventually result in a final solution. The following table shows the convergence:

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		Iteration			
		1	2	3	4
CP_{Ref}	mm Hg	50.00	45.54	45.24	45.22
$HRCF_{CP, Ref\text{-}Des}$		-0.0988	-0.5155	-0.5426	-0.5443
$GTCHR_{Ref, CorRCP}$	kcal/kWh	1994.2	1985.9	1985.4	1985.3
$Duty_{Ref}$	kcal/h	2.272E8	2.256E8	2.254E8	2.254E8
$HTran_{Ref}$	kcal/h/m^2	32599	32360	32345	32344
$HEIHTC_{Ref}$	kcal/(h m^2 K)	3356.6	3356.6	3356.6	3356.6
HTC_{Ref}	kcal/(h m^2 K)	2853.1	2853.1	2853.1	2853.1
$LMTD_{Ref}$	°C	11.43	11.34	11.34	11.34
$CCWTO_{Ref}$	°C	28.82	28.76	28.75	28.75
$CCWTR_{Ref}$	°C	8.82	8.76	8.75	8.75
EST_{Ref}	°C	36.4	36.28	36.27	36.27
CP_{Ref}	mm Hg	45.54	45.24	45.22	45.22
Difference	mm Hg	4.464	0.298	0.019	0.001

Steps "24" through "45" are calculating the expected condenser pressure.

24) Establish the typical heat input used on a cold startup (HICold) = 210E6 kcal

25) Establish the typical heat input used on a hot startup (HIHot) = 86E6 kcal

26) Adjust the period's actual gross unit heat rate by removing the fuel supplied during startups.

$$GUHR_{Act, CorSF} = (FS_{Act} * HHV_{Act} - NoCold*HICold - NoHot*HIHot) / (GG_{Act} * 1000kWh/MWh)$$

$$= (92{,}558{,}895kg * 3599kcal/kg - 0*210E6kcal - 1*96E6kcal) / (133492MWh * 1000kWh/MWh)$$

$$= 2494.7 \text{ kcal/kWh}$$

27) Establish expected value for Condenser Circulating water flow rate ($CCWF_{Exp}$) from the condenser characteristics, or from a curve (i.e. $CCWF_{Exp}$ vs. lake elevation).

$$CCWF_{Exp} = \text{constant or f(water elevation)} = 25.76E6 \text{ kg/h}$$

28) Calculate the expected value for Water Velocity through the tubes ($TubeWV_{Exp}$), given the flow rate ($CCWF_{Exp}$), the number of tubes ($TubeNo_{Exp}$), and the internal diameter of the tubes ($TubeID_{Exp}$).

$TubeWV_{Exp}$ = $CCWF_{Exp}$ / 1000 kg/m^3 / (3.14159 * ($TubeID_{Exp}$/2/1000)^2 * $TubeNo_{Exp}$ * 3600 sec/h)

= 25.76E6 kg/h / 1000 kg/m^3 / (3.14159*(20.4mm/2/1000mm/m)^2 * 11010 * 3600sec/h)

= 1.988 m/s

29) Look up the expected value for the Heat Exchange Institute "Material and Gauge Correction Factor" ($TubeMGC_{Exp}$) as a function of the material of the condenser tubes ($TubeMat_{Exp}$) and the tube wall thickness ($TubeBWG_{Exp}$).

$TubeMGC_{Exp}$ = f($TubeMat_{Exp}$, $TubeBWG_{Exp}$) = 0.82

30) Establish expected value for HEI's "Uncorrected Heat Transfer Coefficient" ($TubeU_{Exp}$) based on tube external diameter ($TubeOD_{Exp}$) and water velocity through the tubes ($TubeWV_{Exp}$).

$TubeU_{Exp}$ = f($TubeOD_{Exp}$, $TubeWV_{Exp}$) = 2900.8 kcal/(h m^2 K)

31) Establish expected value for Condenser Surface Area ($CSArea_{Exp}$) = 6970 m^2

32) From manufacturer's heat rate correction curve for condenser pressure (Percent change in heat rate vs. condenser pressure), determine the percent change in heat rate due to the actual condenser pressure (verses design).

$HRCF_{CP,\ Act\text{-}Des}$ = f(CP_{Act}, MSF_{Act}) or f(CP_{Act}, $GLoad_{Act}$)
= f(68 mm Hg, 596,000kg/h FWF)
= -2.2499 - 4.1176E-2*(68) + 2.3196E-3*(68)^2 - 1.2713E-5*(68)^3
= 1.678

33) Use the actual condenser pressure (CP_{Act}) as the first guess for calculating the expected condenser pressure CP_{Exp}.

CP_{Exp} = CP_{Act} = 68mmHg

34) From manufacturer's heat rate correction curve for condenser pressure (Percent change in heat rate vs. condenser pressure) determine the percent change in heat rate due to the expected condenser pressure (versus design)

$HRCF_{CP,\ Exp\text{-}Des}$ = f(CP_{Exp}, MSF_{Act}) or f(CP_{Exp}, $GLoad_{Act}$)
= f(68 mm Hg, 596,000kg/h FWF)
= -2.2499 - 4.1176E-2*(68) + 2.3196E-3*(68)^2 - 1.2713E-5*(68)^3
= 1.678

35) Ratio the expected and actual heat rate correction fractional multipliers ($HRCF_{CP,\ Exp\text{-}Des}$ and $HRCF_{CP,\ Act\text{-}Des}$)

$$HRCFR_{CP,\ Act\text{-}Exp} = (1+ HRCF_{CP,\ Exp\text{-}Des} / 100) / (1+ HRCF_{CP,\ Act\text{-}Des}/100)$$
$$= (1+ 1.678/ 100) / (1+ 1.678/100)$$
$$= 1$$

36) Calculate the actual gross unit heat rate adjusted for the "assumed" expected condenser pressure (and also adjusted to eliminate startup fuel).

$$GUHR_{Act,\ CorSF,\ CorECP} = GUHR_{Act,\ CorSF} * HRCFR_{CP,\ Act\text{-}Exp}.$$
$$= 2494.7 \text{ kcal/kWh} * 1$$
$$= 2494.7 \text{ kcal/kWh}$$

37) Calculate the expected condenser duty, which is the actual condenser duty, corrected for startup fuel and to expected condenser pressure ($Duty_{Exp} = Duty_{Act,\ CorSF,\ CorECP}$) by taking the total heat input to the steam plus the heat added by the pumps and then subtracting the energy converted to electricity and the energy lost in the generator.

$$Duty_{Exp} = (GUHR_{Act,\ CorSF,\ CorECP} * GLoad_{Act} * 1000\text{kW/MW} * BE_{Act}/100) - (860{,}000 \text{ kcal/h/MW} * GLoad_{Act}) - (860 \text{ kcal/h/kW} * GLoss_{Ref,\ Exp}) + PumpQ_{Ref,Exp} - AuxDuty_{Ref,Exp}$$

$$= (2494.7 \text{ kcal/kWh} * 200\text{MW} * 1000\text{kW/MW} * 80.812/100) - (860{,}000 \text{ kcal/h/MW} * 200\text{MW}) - (860 \text{ kcal/h/kW} * 2608.5 \text{ kW}) + 2.613\text{E}6 \text{ kcal/h} - 0$$

$$= 2.316\text{E}8 \text{ kcal/h}$$

38) Calculate Heat Transmittance ($HTran_{Exp}$) by dividing the heat rejection rate in the condenser by the area available for heat transfer.

$$HTran_{Exp} = Duty_{Exp} / CSArea_{Exp}$$
$$= 2.316\text{E}8 \text{ kcal/h} / 6970 \text{ m}^2$$
$$= 33224 \text{ kcal/h/m}^2$$

39) Calculate HEI Heat Transfer Coefficient ($HEIHTC_{Exp}$), by multiplying the Uncorrected Heat transfer Coefficient by the Tube Material and Gauge Correction Factor and by multiplying by the Inlet Water Temperature Correction Factor.

$$HEIHTC_{Exp} = TubeU_{Exp} * TubeMGC_{Exp} * HEITCF_{Ref,\ Exp}.$$
$$= 2900.8 \text{ kcal/(h m}^2 \text{ K)} * 0.82 * 0.988$$
$$= 2351.0 \text{ kcal/(h m}^2 \text{ K)}$$

40) Calculate the expected Heat Transfer Coefficient (HTC_{Exp}), based on the expected tube cleanliness factor (usually the same value as the reference, which is usually 85%).

$$HTC_{Exp} = HEIHTC_{Exp} * CF_{Exp} / 100$$
$$= 2351.0 \text{ kcal/(h m}^2\text{ K)} * 85/100$$
$$= 1998.4 \text{ kcal/(h m}^2\text{ K)}$$

41) Calculate the expected Logarithmic Mean Temperature Difference ($LMTD_{Exp}$) by dividing the heat transmittance by the heat transfer coefficient.

$$LMTD_{Exp} = HTran_{Exp} / HTC_{Exp}.$$
$$= 33224 \text{ kcal/h/m}^2 / 1998.4 \text{ kcal/(h m}^2\text{ K)}$$
$$= 16.63 \text{ °C}$$

42) Calculate the expected Condenser Circulating Water Temperature Rise ($CCWTR_{Exp}$).

$$CCWTR_{Exp} = (Duty_{Exp} / CCWF_{Exp} / (1\text{kcal/kg C}))$$
$$= (2.316E8 \text{ kcal/h} / 25.76E6 \text{ kg/h} / 1\text{kcal/kg C})$$
$$= 8.99 \text{ °C}$$

43) Calculate the expected Outlet Water Temperature ($CCWTO_{Exp}$)

$$CCWTO_{Exp} = CCWTR_{Exp} + CCWTI_{Act}.$$
$$= 8.99 \text{ °C} + 20 \text{ °C}$$
$$= 28.99 \text{ °C}$$

44) Calculate Expected Exhaust Steam Temperature (EST_{Exp})

$$EST_{Exp} = (CCWTO_{Exp} * e^{(CCWTR/LMTD)} - CCWTI_{Act}) / (e^{(CCWTR/LMTD)} - 1).$$
$$= (28.99 \text{ °C} * e(8.99 \text{ °C}/16.63 \text{ °C}) - 20 \text{ °C}) / (e(8.99 \text{ °C}/16.63 \text{ °C}) - 1)$$
$$= 41.52 \text{ °C}$$

45) Calculate CP using EST_{Exp} and steam tables.

$$CP_{Exp} = f(EST_{Exp})$$
$$= f(41.52 \text{ °C})$$
$$= 59.97 \text{ mmHg}$$

46) Check to see if the condenser pressure used in steps "33" through "44" is within 0.2mm of the calculated condenser pressure (CP from step "45"). If the difference is greater than 0.2mm, reset CP_{Exp} = CP and return to step "34". If the difference is smaller than 0.2mm, continue with step "47". The initial assumption of 68mm Hg did not match with the resulting pressure of 59.97. Returning to step "34" and reiterating will eventually result in a final solution. The following table shows the convergence:

		Iteration			
		1	2	3	4
CP_{Exp}	mm Hg	68.00	59.97	59.05	58.94
$HRCF_{CP, Exp-Des}$		1.6786	0.8812	0.7890	0.7785
$HRCFR_{CP, Act-Exp}$		1.0000	0.9922	0.9913	0.9911
$GUHR_{Act, CorSF, CorECP}$	kcal/kWh	2494.7	2475.1	2472.9	2472.6
$Duty_{Exp}$	kcal/h	2.316E8	2.284E8	2.280E8	2.280E8
$HTran_{Exp}$	kcal/h/m^2	33224	32771	32718	32712
$HEIHTC_{Exp}$	kcal/(h m^2 K)	2351.0	2351.0	2351.0	2351.0
HTC_{Exp}	kcal/(h m^2 K)	1998.4	1998.4	1998.4	1998.4
$LMTD_{Exp}$	°C	16.63	16.40	16.37	16.37
$CCWTR_{Exp}$	°C	8.99	8.87	8.85	8.85
$CCWTO_{Exp}$	°C	28.99	28.87	28.85	28.85
EST_{Exp}	°C	41.52	41.23	41.20	41.19
CP_{Exp}	mm Hg	59.97	59.05	58.94	58.93
Difference	mm Hg	8.029	0.923	0.106	0.012

47) Ratio the reference and expected heat rate correction fractional multipliers ($HRCF_{CP, Exp-Des}$ and $HRCF_{CP, Act-Des}$).

$$HRCFR_{CP, Exp-Ref} = (1+ HRCF_{CP, Ref-Des} / 100) / (1+ HRCF_{CP, Exp-Des}/100)$$
$$= (1 + -0.5443/100) / (1 + 0.7785/100)$$
$$= 0.9869$$

48) Determine the heat rate deviation due to the difference between the expected and reference condenser pressure ($HRD_{CP, Exp-Ref}$).

$$HRD_{CP, Exp-Ref} = (1 - HRCFR_{Exp-Ref}) * NUHR_{Ref}$$
$$= (1 - 0.9869) * 2413.2 \text{ kcal/kWh}$$
$$= 31.7 \text{ kcal/kWh}$$

49) Add this and all other heat rate deviations (expected verses reference) to the reference net unit heat rate ($NUHR_{Ref}$) to obtain the expected net unit heat rate ($NUHR_{Exp}$) see Section 3.8.4.17.

$$NUHR_{Exp} = NUHR_{Ref} + \Sigma\, HRD_{I, Exp-Ref}.$$

50) Determine the heat rate deviation due to the difference between the actual and expected heat rate ($HRD_{CP, Exp-Ref}$).

$$HRD_{CP, Act-Exp} = (1 - HRCFR_{CP, Act-Exp}) * NUHR_{Exp}$$
$$= (1 - 0.9911) * 2471.0 \text{ kcal/kWh}$$

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$= 22.0$ kcal/kWh

3.8.4.4 Final Feedwater Temperature & High Pressure Feedwater Heater(s) Out of Service

For the time when all heaters were in service:

Establish $FFWT_{Ref}$ value from reference curve (based on acceptance test or design data).

$$FFWT_{Ref} = f(FWF_{Act}) = f(169 + 2.26E\text{-}4*596000 - 9.31E\text{-}11*(596000)^2 = 270.6\ °C$$

Establish $FFWT_{Exp}$ value from expected level curve (based on current plant equipment design.

$$FFWT_{Exp} = f(FWF_{Act}) = f(171 + 2.147E\text{-}4*596000 - 8.85E\text{-}11*(596000)^2 = 267.5\ °C$$

From the curve of heat rate correction factor due to 2.78°C final feedwater temperature difference versus gross load (or feedwater flow), determine the $HRCF_{FFWT}$ at the periods average gross load.

$$HRCF_{FFWT} = f(FWF_{Act}) = f(9.1667E\text{-}3 + 1.8456E\text{-}7*596000 = 0.119\ \%\ \text{change in NUHR} / 2.78\ °C$$

Determine the heat rate deviation due to the difference between the expected final feedwater temperature and the reference final feedwater temperature.

$$HRD_{FFWT,\ Exp\text{-}Ref} = (FFWT_{Ref} - FFWT_{Exp}) * NUHR_{Ref} * HRCF_{FFWT} / 2.78$$

$$= (270.6\ °C - 267.5\ °C) * 2413.2\text{kcal/kWh} * (0.119\ \%\ \text{change in NUHR} / 2.78\ °C) / 2.78\ °C$$

$$= 3.2\ \text{kcal/kWh}$$

After calculating the expected net unit heat rate $NUHR_{Exp}$ determine the heat rate deviation due to the difference between the actual and the expected final feedwater temperature.

$$HRD_{FFWT,\ Act\text{-}Exp} = (FFWT_{Exp} - FFWT_{Act}) * NUHR_{Ref} * HRCF_{FFWT} / 2.78 * (HOU_{ACT}\text{-Bypassed Hours}_{Act})/HOU_{Act}$$

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= (267.5 °C - 265 °C) * 2471.0kcal/kWh * (0.119 % change in NUHR / 2.78 °C) / 2.78 °C * (667.46-100)h/667.46h

= 2.2 kcal/kWh

When the heaters are out of service:

$HRCF_{HtrBP}$ = $f(FWF_{Act})$
= 0.714429 % change in NUHR

Calculate the heat rate deviation for each combination of heaters bypassed:

$HRD_{HtrBP, Act-Exp}$ = $NUHR_{Exp}$ * $HRCF_{HtrBP}$ * (Hours bypassed / PH_{Act})
= 2471.0kcal/kWh * 0.714429/100 * (100h/667.46h)
= 2.6 kcal/kWh

3.8.4.5 Makeup

Establish MU_{Ref} value from reference curve.

$MU_{Ref} = f(FWF_{Act}) = 0\%$

Establish MU_{Exp} value from expected level curve.

$MU_{Exp} = f(FWF_{Act}) = 0.8\%$

From the curve of heat rate correction factor due to 1% makeup versus feedwater flow, determine the $HRCF_{MU}$ at the period's feedwater flow (596000 kg/h).

$HRCF_{MU}$ = $f(FWF_{Act})$
= 0.5409 - 8E-8*596000 + 2E-13*596000^2
= 0.564 % change in NUHR / 1 % MU

Determine the heat rate deviation due to the difference between the expected makeup and the reference makeup.

$HRD_{MU, Exp-Ref}$ = (MU_{Exp} - MU_{Ref}) * $NUHR_{Ref}$ * $HRCF_{MU}$
= (0.8 - 0)% * 2413.2kcal/kWh*0.564 % ΔNUHR / 1 % MU/100
= 10.9 kcal/kWh

After calculating the expected net unit heat rate $NUHR_{Exp}$ (Section 3.8.4.17) determine the heat rate deviation due to the difference between the actual makeup and the expected makeup.

$HRD_{MU, Act-Exp}$ = (MU_{Act} - MU_{Exp}) * $NUHR_{Exp}$ * $HRCF_{MU}$
= (1.1 - 0.8)% * 2471.0kcal/kWh*0.564 % ΔNUHR / 1 % MU/100

$= 4.2$ kcal/kWh

3.8.4.6 Reheat Attemperation

Establish the reference value for reheat attemperation flow.

$$RHA_{Ref} = f(FWF_{Act}) = 0\%$$

Establish the expected value for reheat attemperation flow.

$$RHA_{Exp} = f(FWF_{Act}) = 0\%$$

Determine the heat rate correction factor (% change in net unit heat rate) for 1% reference reheat attemperation (as a function of gross load MW).

$$\begin{aligned} HRCF_{RHA,\ Ref} &= f(GLoad_{Act}) \\ &= -0.153929 + 0.92557E\text{-}2*200 - 0.6643E\text{-}4*200^2 + 0.14583E\text{-}6*200^3 \\ &= 0.2066\ \%\ \text{change in NUHR/1\% RHA} \end{aligned}$$

Determine the heat rate correction factor (% change in net unit heat rate) for reference reheat attemperation

$$= (0.2066\ \%\ \text{change in NUHR/1\% RHA}) * 0\% = 0$$

Determine the heat rate correction factor (% change in net unit heat rate) for the expected reheat attemperation rate (as a percent of feedwater flow).

$$= (0.2066\ \%\ \text{change in NUHR/1\% RHA}) * 0\% = 0$$

Determine the heat rate correction factor (% change in net unit heat rate) for the actual reheat attemperation rate (as a percent of feedwater flow).

$$\begin{aligned} &= (0.2066\ \%\ \text{change in NUHR/1\% RHA}) * 1.3\% \\ &= 0.2686\ \%\ \text{change in NUHR} \end{aligned}$$

Determine the HRDZ (expected % attemperation - reference % attemperation)*Ref HR * HRCF.

$$\begin{aligned} HRD_{RHA,\ Exp\text{-}Ref} &= NUHR_{Ref} * (HRCF_{RHA,\ Exp} - HRCF_{RHA,\ Ref}) / 100 \\ &= 2413.2\ \text{kcal/kWh} * (0 - 0) / 100 \\ &= 0\text{kcal/kWh} \end{aligned}$$

After calculating the expected net unit heat rate $NUHR_{Exp}$ (see Section 3.8.4.17), determine the heat rate deviation due the difference between the actual reheat attemperation and the expected reheat attemperation.

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$HRD_{RHA, Act-Exp}$ $= NUHR_{Exp} * (HRCF_{RHA, Act} - HRCF_{RHA, Exp}) / 100$
$= 2471.0$ kcal/kWh * (0.2686 - 0) / 100
$= 6.6$ kcal/kWh

3.8.4.7 Station Service (Prorated) - Idle

Both the reference and the expected idle time station service are zero.

Calculate the heat rate deviation (from expected) due to station service used when the unit was off-line during the period in question.

$HRD_{SSI, Act-Exp} = \{ (FS_{Act} * HHV_{Act}) / (GG_{Act} - (UAP_{Exp}+SSO_{Exp})*HOU_{Act} - SSI_{Act}*(PH-HOU_{Act})) \}/1000 - \{ (FS_{Act} * HHV_{Act}) / (GG_{Act} - (UAP_{Exp}+SSO_{Exp})*HOU_{Act})\}/1000$

$= \{ (92558895kg * 3599kcal/kg) / (133492MWh - (9.507+0.75)MW*667.46h - 1.8291MW*(744-667.46)h) \}/1000 - \{(92558895kg * 3599kcal/kg) / (133492MWh - (9.507+0.75)MW*667.46h) \}/1000$

$= 2.9$ kcal/kWh

3.8.4.8 Station Service (Prorated) - Operating

Determine SSO_{Ref} value from reference curve.

$SSO_{Ref} = f(FWF_{Act}) = 0.5MW$

Establish SSO_{Exp} value from expected level curve.

$SSO_{Exp} = f(FWF_{Act}) = 0.75MW$

Calculate the heat rate deviation due to the difference between the expected and reference station service:

$HRD_{SSO, Exp-Ref} = \{ (FS_{Act} * HHV_{Act}) / (GG_{Act} - (UAP_{Ref} +SSO_{Exp})*HOU_{Act}/1000 - \{ (FS_{Act} * HHV_{Act}) / (GG_{Act} - (UAP_{Ref} +SSO_{Ref})*HOU_{Act}/1000$

$= \{ (92558895kg * 3599kcal/kg) / (133492MWh - (8.56+0.75)MW*667.46h \}/1000 - \{(92558895kg * 3599kcal/kg) / (133492MWh - (8.56+0.5)MW*667.46h)\}/1000$

$= 3.4$ kcal/kWh

Determine the heat rate deviation due to the difference between the actual operating station use and the expected operating station use.

$$HRD_{SSO,\ Act\text{-}Exp} = \{ (FS_{Act} * HHV_{Act}) / (GG_{Act} - (UAP_{Exp} + SSO_{Act})*HOU_{Act}/1000 - \{ (FS_{Act} * HHV_{Act}) / (GG_{Act} - (UAP_{Exp} + SSO_{Exp})*HOU_{Act}/1000$$

$$= \{ (92558895kg * 3599kcal/kg) / (133492MWh - (9.507+0.5244)MW*667.46h \}/1000 - \{(92558895kg * 3599kcal/kg) / (133492MWh - (9.507+0.75)MW*667.46h)\}/1000$$

$$= -3.1\ kcal/kWh$$

3.8.4.9 Hot Reheat at Intercept Valve Temperature

Establish $HRHIV_{Ref}$ value from reference curve.

$$HRHIV_{Ref} = f(FWF_{Act}) = 565\ °C$$

Establish $HRHIV_{Exp}$ value from expected level curve.

$$HRHIV_{Exp} = f(FWF_{Act}) = 565\ °C$$

From the heat rate correction curve(s) for reheat steam temperature (plotted as percent change in heat rate vs. steam temperature) determine the percent change in heat rate due to the reference, expected and actual steam temperatures verses design, $HRCF_{HRHIV,\ Ref\text{-}Des}$, $HRCF_{HRHIV,\ Exp\text{-}Des}$, $HRCF_{HRHIV,\ Act\text{-}Des}$.

$$HRCF_{HRHIV,\ Ref\text{-}Des} = f(HRHIV_{Ref}), GLoad_{Act})$$
$$= 13.234 - 0.0234(565\ °C) = 0\ \%\ change\ in\ NUHR$$

$$HRCF_{HRHIV,\ Exp\text{-}Des} = f(HRHIV_{Exp}), GLoad_{Act})$$
$$= 13.234 - 0.0234(565\ °C) = 0\ \%\ change\ in\ NUHR$$

$$HRCF_{HRHIV,\ Act\text{-}Des} = f(HRHIV_{Act}), GLoad_{Act})$$
$$= 13.234 - 0.0234(557\ °C) = 0.20\ \%\ change\ in\ NUHR$$

Determine the heat rate deviation due to the difference between the expected steam temperature and the reference steam temperature.

$$HRD_{HRHIV,\ Exp\text{-}Ref} = \{ (1+ HRCF_{HRHIV,\ Exp\text{-}Des}) / (1+ HRCF_{HRHIV,\ Ref\text{-}Des}) - 1\} * NUHR_{Ref}$$
$$= \{ (1 + 0/100) / (1 + 0/100) -1) * 2413.2\ kcal/kWh$$
$$= 0\ kcal/kWh$$

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After calculating the expected net unit heat rate $NUHR_{Exp}$ (Section 3.8.4.17) determine the heat rate deviation due to the difference between the actual steam temperature and the expected steam temperature.

$$HRD_{HRHIV,\ Exp\text{-}Ref} = \{ (1+ HRCF_{HRHIV,\ Act\text{-}Des}) / (1+ HRCF_{HRHIV,\ Exp\text{-}Des}) - 1\} * NUHR_{Exp}$$
$$= \{ (1 + 0.2/100) / (1 + 0/100) -1) * 2471.0 \text{ kcal/kWh}$$
$$= 4.9 \text{ kcal/kWh}$$

3.8.4.10 Superheat Attemperation

Establish the reference value for superheat attemperation flow.

$$SHA_{Ref} = f(FWF_{Act}) = 0.4\%$$

Establish the expected value for superheat attemperation flow.

$$SHA_{Exp} = f(FWF_{Act}) = 1.0\%$$

Determine the heat rate correction factor (% change in net unit heat rate) for the reference superheat attemperation rate (as a percent of gross load in MW).

$$HRCF_{SHA,\ Ref} = f(GLoad_{Act}) \ \& \ SHA_{Ref}$$
$$= (0.4/1)*(0.517304 - 0.3421E\text{-}2*200 - 0.6608E\text{-}5*200^2 + 0.62502E\text{-}7*200^3)$$
$$=0.028 \text{ \% change in NUHR}$$

Determine the heat rate correction factor (% change in net unit heat rate) for the expected superheat attemperation rate (as a percent of feedwater flow).

$$HRCF_{SHA,\ Exp} = f(GLoad_{Act}) \ \& \ SHA_{Exp}$$
$$= (1.0/1)*(0.517304 - 0.3421E\text{-}2*200 - 0.6608E\text{-}5*200^2 + 0.62502E\text{-}7*200^3)$$
$$=0.069 \text{ \% change in NUHR}$$

Determine the heat rate correction factor (% change in net unit heat rate) for the actual superheat attemperation rate (as a percent of feedwater flow).

$$HRCF_{SHA,\ Act} = f(GLoad_{Act}) \ \& \ SHA_{Act}$$
$$= (2.0/1)*(0.517304 - 0.3421E\text{-}2*200 - 0.6608E\text{-}5*200^2 + 0.62502E\text{-}7*200^3)$$
$$=0.138 \text{ \% change in NUHR}$$

Determine the HRDZ ((expected % attemperation - reference % attemperation)*Ref HR * HRCF).

$$HRD_{SHA,\ Exp\text{-}Ref} = NUHR_{Ref} * (HRCF_{SHA,\ Exp} - HRCF_{SHA,\ Ref}) / 100$$

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$= 2413.2$ kcal/kWh * (0.069 - 0.028) / 100
$= 1.0$ kcal/kWh

After calculating the expected net unit heat rate $NUHR_{Exp}$ (Section 3.8.4.17) determine the heat rate deviation due the difference between the actual and expected superheat attemperation.

$HRD_{SHA, Act-Exp} = NUHR_{Exp} * (HRCF_{SHA, Act} - HRCF_{SHA, Exp}) / 100$
$= 2471.0$ kcal/kWh * (0.138 - 0.069) / 100
$= 1.7$ kcal/kWh

3.8.4.11 Throttle Pressure

Establish TP_{Ref} value from reference curve.

$TP_{Ref} = f(FWF_{Act}) = 127.59$ kg/cm^2

Establish TP_{Exp} value from expected level curve.

$TP_{Exp} = f(FWF_{Act}) = 127.59$ kg/cm^2

From the heat rate correction curve(s) for throttle pressure (plotted as percent change in heat rate vs. steam pressure) determine the percent change in heat rate due to the reference, expected and actual steam pressures versus design steam pressure,

$HRCF_{TP, Ref-Des} = f(TP_{Ref})$
$= 7.62174 - 0.059738*127.59)$
$= 0$ % change in NUHR

$HRCF_{TP, Exp-Des} = f(TP_{Exp})$
$= 7.62174 - 0.059738*127.59)$
$= 0$ % change in NUHR

$HRCF_{TP, Act-Des} = f(TP_{Act})$
$= 7.62174 - 0.059738*125.0)$
$= 0.154$ % change in NUHR

Determine the heat rate deviation due to the difference between the expected steam pressure and the reference steam pressure

$HRD_{TP, Exp-Ref} = ((1+ HRCF_{TP, Exp-Des}/100) / (1+ HRCF_{TP, Ref-Des}/100) - 1) * NUHR_{Ref}$.
$= ((1 + 0/100) / (1+0/100) - 1) * 2413.2$ kcal/kWh
$= 0$ kcal/kWh

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After calculating the expected net unit heat rate $NUHR_{Exp}$ (see Section 3.8.4.17), determine the heat rate deviation due to the difference between the actual steam pressure and the expected steam pressure.

$$HRD_{TP,\ Act\text{-}Exp} = ((1+ HRCF_{TP,\ Act\text{-}Des} /100) / (1+ HRCF_{TP,\ Exp\text{-}Des} /100) - 1) * NUHR_{Ref} .$$
$$= ((1 + 0.154/100) / (1+0/100) - 1) * 2471.0 \text{ kcal/kWh}$$
$$= 3.8 \text{ kcal/kWh}$$

3.8.4.12 Throttle Temperature

Establish TT_{Ref} value from reference curve.

$$TT_{Ref} = f(FWF_{Act}) = 565\ °C$$

Establish TT_{Exp} value from expected level curve.

$$TT_{Exp} = f(FWF_{Act}) = 565\ °C$$

From the heat rate correction curve(s) for throttle temperature (plotted as percent change in heat rate vs. steam temperature) determine the percent change in heat rate due to the reference, expected and actual steam temperatures versus design steam temperature, $HRCF_{TT,\ Ref\text{-}Des}$, $HRCF_{TT,\ Exp\text{-}Des}$, $HRCF_{TT,\ Act\text{-}Des}$.

$$HRCF_{TT,\ Ref\text{-}Des} = f(TT_{Ref})$$
$$= 12.216 - 0.0216 * 565\ °C$$
$$= 0\ \%\ \text{change in NUHR}$$

$$HRCF_{TT,\ Exp\text{-}Des} = f(TT_{Exp})$$
$$= 12.216 - 0.0216 * 565\ °C$$
$$= 0\ \%\ \text{change in NUHR}$$

$$HRCF_{TT,\ Act\text{-}Des} = f(TT_{Act})$$
$$= 12.216 - 0.0216 * 562\ °C$$
$$= 0.077\ \%\ \text{change in NUHR}$$

Determine the heat rate deviation due to the difference between the expected steam temperature and the reference steam temperature.

$$HRD_{TT,\ Exp\text{-}Ref} = ((1+ HRCF_{TT,\ Exp\text{-}Des}/100) / (1+ HRCF_{TT,\ Ref\text{-}Des}/100) - 1) * NUHR_{Ref}$$
$$= ((1 + 0/100) / (1 + 0/100) - 1) * 2413.2 \text{ kcal/kWh}$$
$$= 0 \text{ kcal/kWh}$$

After calculating the expected net unit heat rate $NUHR_{Exp}$ (see Section 3.8.4.17), determine the heat rate deviation due to the difference between the actual steam temperature and the expected steam temperature.

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$$HRD_{TT,\ Act\text{-}Exp} = ((1 + HRCF_{TT,\ Act\text{-}Des}/100) / (1 + HRCF_{TT,\ Exp\text{-}Des}/100) - 1) * NUHR_{Exp}.$$
$$= ((1 + 0.077/100) / (1 + 0/100) - 1) * 2471.0 \text{ kcal/kWh}$$
$$= 1.9 \text{ kcal/kWh}$$

3.8.4.13 HP Turbine Efficiency

Establish the reference section efficiency at VWO.

$$HPT_{Ref} = \text{constant} = 83.27\%$$

Establish the expected section efficiency at VWO.

$$HPT_{Exp} = \text{constant} = 83.27\ \%$$

Establish the heat rate correction factor (usually a constant) for the % change in net unit heat rate for a 1% point change in section efficiency.

$$HRCF_{HPT} = \text{constant} = 0.2152\ \%\ \text{change in heat rate for a 1 \% pt change in HP efficiency}$$

Determine the heat rate deviation due to the difference between the reference and expected section efficiency.

$$HRD_{HPT,\ Exp\text{-}Ref} = (HPT_{Ref} - HPT_{Exp}) * NUHR_{Ref} * HRCF_{HPT} / 100.$$
$$= (83.27 - 83.27) * 2413.2 \text{ kcal/kWh} * 0.2152/100$$
$$= 0 \text{ kcal/kWh}$$

After calculating the expected net unit heat rate $NUHR_{Exp}$ (Section 3.8.4.17) determine the heat rate deviation due to the difference between the actual section efficiency and the expected section efficiency.

$$HRD_{HPT,\ Act\text{-}Exp} = (HPT_{Exp} - HPT_{Act}) * NUHR_{Exp} * HRCF_{HPT} / 100$$
$$= (83.27 - 82.1) * 2471.0\text{kcal/kWh} * 0.2152 / 100$$
$$= 6.2 \text{ kcal/kWh}$$

3.8.4.14 IP Turbine Efficiency

Establish the reference section efficiency.

$$IPT_{Ref} = \text{constant} = 90.57\%$$

Establish the expected section efficiency.

$$IPT_{Exp} = \text{constant} = 90.57\ \%$$

Establish the heat rate correction factor (usually a constant) for the % change in net unit heat rate for a 1% point change in section efficiency.

$HRCF_{IPT}$ = constant = 0.1958 % change in heat rate for a 1 % pt change in IP efficiency

Determine the heat rate deviation due to the difference between the reference and expected section efficiency.

$$HRD_{IPT,\,Exp\text{-}Ref} = (IPT_{Ref} - IPT_{Exp}) * NUHR_{Ref} * HRCF_{IPT} / 100.$$
$$= (90.57 - 90.57) * 2413.2 \text{ kcal/kWh} * 0.1958/100$$
$$= 0 \text{ kcal/kWh}$$

After calculating the expected net unit heat rate $NUHR_{Exp}$ (see Section 3.8.4.17), determine the heat rate deviation due to the difference between the actual section efficiency and the expected section efficiency.

$$HRD_{IPT,\,Act\text{-}Exp} = (IPT_{Exp} - IPT_{Act}) * NUHR_{Exp} * HRCF_{IPT} / 100$$
$$= (90.57 - 87.2) * 2471.0\text{kcal/kWh} * 0.1958 / 100$$
$$= 16.3 \text{ kcal/kWh}$$

3.8.4.15 Unit Auxiliary Power

Determine UAP_{Ref} value from reference curve, (plotted as station use in MW versus feedwater flow based on acceptance test or design data) at the period's steam flow, feedwater flow or gross load.

$$UAP_{Ref} = f(FWF_{Act})$$
$$= 7.999 - 9.513E\text{-}6*596000 + 1.754E\text{-}11*596000^2$$
$$= 8.56\text{MW}$$

Establish UAP_{Exp} value from expected level curve, (plotted as station use in MW versus steam flow, feedwater flow or gross load, based on current plant equipment design) at the period's steam flow, feedwater flow or gross load.

$$UAP_{Exp} = f(FWF_{Act})$$
$$= 8.294 - 1.131E\text{-}5*596000 + 2.239E\text{-}11*596000^2$$
$$= 9.507\text{MW}$$

Calculate the heat rate deviation due to the difference between the expected and reference unit auxiliary power using either equation:

$$HRD_{UAP,\,Exp\text{-}Ref} = \{ (FS_{Act} * HHV_{Act}) / (GG_{Act} - HOU_{Act}(SSO_{Ref}+UAP_{Exp}) \}/1000 - \{ (FS_{Act} * HHV_{Act}) / (GG_{Act} - HOU_{Act}(SSO_{Ref}+UAP_{Ref}) \}/1000$$

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= { (92558895kg*3599kcal/kg) / (133492 - 667.46h*(0.5+9.507)MW)/1000 - ((92558895kg*3599kcal/kg) / (133492 - 667.46h*(0.5+8.56)MW)/1000

= 13.0 kcal/kWh

After calculating the expected net unit heat rate $NUHR_{Exp}$ (3.8.4.17) determine the heat rate deviation due to the difference between the actual unit auxiliary power and the expected unit auxiliary power:

$HRD_{UAP,\ Act\text{-}Exp}$ = { $(FS_{Act} * HHV_{Act}) / (GG_{Act} - HOU_{Act}(SSO_{Ref}+UAP_{Act})$ }/1000 - { $(FS_{Act} * HHV_{Act}) / (GG_{Act} - HOU_{Act}(SSO_{Ref}+UAP_{Exp})$ }/1000

= { (92558895kg*3599kcal/kg) / (133492 - 667.46h*(0.5+11.21565)MW)/1000 - ((92558895kg*3599kcal/kg) / (133492 - 667.46h*(0.5+9.507)MW)/1000

= 23.8 kcal/kWh

3.8.4.16 Boiler Losses

The following tables shows how boiler efficiency is calculated for the reference conditions, the expected conditions, and the actual conditions. Then to determine the heat rate deviation, expected versus reference, one parameter at a time is changed from the reference to the expected value.

Second, to determine the heat rate deviation, actual versus expected, one parameter at a time is changed from the expected to the actual.

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		Reference	Expected	Actual	Note 1	Note 2	Note 3	Note 4	Note 5	Note 6
AirT	°C	27		18.8	27	27	27	27	27	27
EGT	°C	133.68	128.99	148	133.68	133.68	133.68	133.68	133.68	133.68
AHGIT	°C	340	340	340	340	340	340	340	340	340
AHAOT	°C	300	300	310	300	300	300	300	300	300
AHL	%	7	12	14	7	7	7	7	7	7
UltC	-	0.5351	0.5351	0.2815	0.5351	0.5351	0.5351	0.5351	0.5351	0.5351
UltH	-	0.0500	0.0500	0.0441	0.0500	0.0500	0.0500	**0.0500**	0.0500	0.0500
UltO	-	0.1882	0.1882	0.2692	0.1882	0.1882	0.1882	0.1882	0.1882	0.1882
UltN	-	0.0115	0.0115	0.0039	0.0115	0.0115	0.0115	0.0115	0.0115	0.0115
UltS	-	0.0101	0.0101	0.0016	0.0101	0.0101	0.0101	0.0101	0.0101	0.0101
FuelA	-	0.1004	0.1004	0.1432	0.1004	0.1004	0.1004	0.1004	0.1004	0.1004
FuelTM	-	0.1047	0.1047	0.2566	0.1047	0.1047	0.1047	0.1047	**0.1047**	0.1047
HHV	kcal/kg	5847	5847	3599	5847	5847	5847	5847	5847	5847
O2	%	2.027	1.777	2.3	**1.777**	2.027	2.027	2.027	2.027	2.027
O2 + CO2	%	18	18	18	18	18	18	18	18	18
CombTA	%	2	2.2	2.4	2	**2.2**	2	2	2	2
Wma	kg/kg	0.011	0.011	0.013	0.011	0.011	0.011	0.011	0.011	0.011
Wre	kg/kg	0.003	0.003	0.004	0.003	0.003	0.003	0.003	0.003	**0.003**
Hr	kcal/kg	2000	2000	1800	2000	2000	2000	2000	2000	**2000**
BLMisc	%	1	1	1	1	1	1	1	1	1
X1				49112						
X2				321.2						
$EGT_{CorrRAT}$	°C			152.90162						
CpAir	kcal/kg/K			0.239						
CO2	%	15.973005	16.223005	15.7	16.223	15.973	15.973	15.973	15.973	15.973
CpFG	kcal/kg/K	0.236	0.236	0.237	0.236	0.236	0.236	0.236	0.236	0.236
$EGT_{CorrRAT, CorTNL}$	°C	144.99	140.99	171.85	144.99	144.99	**140.99**	144.99	144.99	144.99
Hdp	kcal/kg	161.11	177.22	193.33	161.11	177.22	161.11	161.11	161.11	161.11

		Reference	Expected	Actual	Note 1	Note 2	Note 3	Note 4	Note 5	Note 6
Wdp	kg/kg	0.1024	0.1027	0.1467	0.1024	0.1027	0.1024	0.1024	0.1024	0.1024
Cb	kg/kg	0.5331	0.5328	0.2780	0.5331	0.5328	0.5331	0.5331	0.5331	0.5331
N2	%	82	82	82	82	82	82	82	82	82
X3		15.980	15.749	16.240	15.749	15.980	15.980	15.980	15.980	15.980
Wg	kg/kg	8.578	8.451	4.524	8.455	8.575	8.578	8.578	8.578	8.578
X4		0.5368	0.5366	0.2786	0.5368	0.5366	0.5368	0.5368	0.5368	0.5368
X5		191.836	194.838	188.557	194.838	191.836	191.836	191.836	191.836	191.836
X6		1233.4499	1232.9685	640.07168	1233.45	1232.969	1233.45	1233.45	1233.45	1233.45
Wa	kg/kg	8.352	8.219	4.412	8.223	8.348	8.352	8.352	8.352	8.352
Mg	kg/kg	0.4564	0.4563	0.4088	0.4563	0.4564	0.4564	0.4564	0.4564	0.4564
X7		10.968	10.798	6.497	10.802	10.963	10.968	10.968	10.968	10.968
Pmg	bar/bar	0.0835592	0.0847608	0.1333909	0.084731	0.083589	0.083559	0.083559	0.083559	0.083559
FGH	kcal/kg	657.161	655.018	663.622	657.159	657.161	657.161	657.161	657.161	657.161
h f(amb P & T)	kcal/gk	27.041	0.014	18.853	27.041	27.041	27.041	27.041	27.041	27.041
BLDG	%	4.303	4.093	4.515	4.241	4.301	4.157	4.303	4.303	4.303
BLComb	%	0.282	0.311	0.788	0.282	0.311	0.282	0.282	0.282	0.282
BLMC	%	1.128	1.173	4.597	1.128	1.128	1.128	1.128	1.128	1.128
BLH2	%	4.815	5.005	7.060	4.815	4.815	4.815	4.815	4.815	4.815
BLMA	%	0.990	1.013	1.028	0.975	0.990	0.990	0.990	0.990	0.990
BLPyr	%	0.103	0.103	0.200	0.103	0.103	0.103	0.103	0.103	0.103
BE	%	87.379	87.302	80.812	87.456	87.352	87.525	87.379	87.379	87.379

Note 1 All parameters at Reference Values, except Boiler Outlet O2 is at Expected Value
Note 2 All parameters at Reference Values, except percent combustibles in ash is at Expected Value
Note 3 All parameters at Reference Values, except Exit Gas Temperature, no leakage, corrected to ref. inlet air temp. is at Expected Value
Note 4 All parameters at Reference Values, except Hydrogen In Fuel is at Expected Value
Note 5 All parameters at Reference Values, except Moisture in Fuel is at Expected Value
Note 6 All parameters at Reference Values, except Pulverizer Reject Rate and Heating Value are at Expected Value

		Reference	Expected	Actual	Note 7	Note 8	Note 9	Note 10	Note 11	Note 12
AirT	°C	27		18.8						
EGT	°C	133.68	128.99	148	128.99	128.99	128.99	128.99	128.99	128.99
AHGIT	°C	340	340	340	340	340	340	340	340	340
AHAOT	°C	300	300	310	300	300	300	300	300	300
AHL	%	7	12	14	12	12	12	12	12	12
UltC	-	0.5351	0.5351	0.2815	0.5351	0.5351	0.5351	0.5351	0.5351	0.5351
UltH	-	0.0500	0.0500	0.0441	0.0500	0.0500	0.0500	**0.0441**	0.0500	0.0500
UltO	-	0.1882	0.1882	0.2692	0.1882	0.1882	0.1882	0.1882	0.1882	0.1882
UltN	-	0.0115	0.0115	0.0039	0.0115	0.0115	0.0115	0.0115	0.0115	0.0115
UltS	-	0.0101	0.0101	0.0016	0.0101	0.0101	0.0101	0.0101	0.0101	0.0101
FuelA	-	0.1004	0.1004	0.1432	0.1004	0.1004	0.1004	0.1004	0.1004	0.1004
FuelTM	-	0.1047	0.1047	0.2566	0.1047	0.1047	0.1047	0.1047	**0.2566**	0.1047
HHV	kcal/kg	5847	5847	3599	5847	5847	5847	5847	5847	5847
O2	%	2.027	1.777	2.3	**2.3**	1.777	1.777	1.777	1.777	1.777
O2 + CO2	%	18	18	18	18	18	18	18	18	18
CombTA	%	2	2.2	2.4	2.2	**2.4**	2.2	2.2	2.2	2.2
Wma	kg/kg	0.011	0.011	0.013	0.011	0.011	0.011	0.011	0.011	0.011
Wre	kg/kg	0.003	0.003	0.004	0.003	0.003	0.003	0.003	0.003	**0.003**
Hr	kcal/kg	2000	2000	1800	2000	2000	2000	2000	2000	**2000**
BLMisc	%	1	1	1	1	1	1	1	1	1
X1				49112						
X2				321.2						
EGTCorrRAT	°C			152.90162						
CpAir	kcal/kg/K			0.244						
CO2	%	15.973005	16.223005	15.7	15.7	16.223	16.223	16.223	16.223	16.223
CpFG	kcal/kg/K	0.249	0.248	0.249	0.248	0.248	0.248	0.248	0.248	0.248
EGTCorrRAT, CorTNL	°C	144.99	140.99	171.27	140.99	140.99	**171.27**	140.99	140.99	140.99
Hdp	kcal/kg	161.11	177.22	193.33	177.22	193.33	177.22	177.22	177.22	177.22

		Reference	Expected	Actual	Note 7	Note 8	Note 9	Note 10	Note 11	Note 12
Wdp	kg/kg	0.1024	0.1027	0.1467	0.1027	0.1029	0.1027	0.1027	0.1027	0.1027
Cb	kg/kg	0.5331	0.5328	0.2780	0.5328	0.5326	0.5328	0.5328	0.5328	0.5328
N2	%	82	82	82	82	82	82	82	82	82
X3		15.980	15.749	16.240	16.240	15.749	15.749	15.749	15.749	15.749
Wg	kg/kg	8.578	8.451	4.524	8.715	8.448	8.451	8.451	8.451	8.451
X4		0.5368	0.5366	0.2786	0.5366	0.5364	0.5366	0.5366	0.5366	0.5366
X5		191.836	194.838	188.557	188.557	194.838	194.838	194.838	194.838	194.838
X6		1233.4499	1232.9685	640.07168	1232.969	1232.485	1232.969	1232.969	1232.969	1232.969
Wa	kg/kg	8.352	8.219	4.412	8.494	8.216	8.219	8.219	8.219	8.219
Mg	kg/kg	0.4564	0.4563	0.4088	0.4566	0.4563	0.4563	0.4035	0.4700	0.4563
X7		10.968	10.798	6.497	11.150	10.794	10.798	12.209	10.482	10.798
Pmg	bar/bar	0.0835592	0.0847608	0.1333909	0.082305	0.084791	0.084761	0.075708	0.08709	0.084761
FGH	kcal/kg	657.161	655.018	663.622	655.023	655.018	655.018	655.037	655.013	655.018
h f(amb P & T)	kcal/gk	27.041	0.014	18.853	0.014	0.014	0.014	0.014	0.014	0.014
BLDG	%	4.303	4.093	4.515	4.220	4.091	5.180	4.092	4.093	4.093
BLComb	%	0.282	0.311	0.788	0.311	0.340	0.311	0.311	0.311	0.311
BLMC	%	1.128	1.173	4.597	1.173	1.173	1.173	1.173	2.875	1.173
BLH2	%	4.815	5.005	7.060	5.005	5.005	5.005	4.415	5.005	5.005
BLMA	%	0.990	1.013	1.028	1.047	1.012	1.013	1.013	1.013	1.013
BLPyr	%	0.103	0.103	0.200	0.103	0.103	0.103	0.103	0.103	0.103
BE	%	87.379	87.302	80.812	87.142	87.275	86.215	87.893	85.601	87.302

Note 7 All parameters at Expected Values, except Boiler Outlet O2 is at Actual Value
Note 8 All parameters at Expected Values, except percent combustibles in ash is at Actual Value
Note 9 All parameters at Expected Values, except Exit Gas Temperature, no leakage, corrected to Exp. inlet air temp. is at Actual Value
Note 10 All parameters at Expected Values, except Hydrogen In Fuel is at Actual Value
Note 11 All parameters at Expected Values, except Moisture in Fuel is at Actual Value
Note 12 All parameters at Expected Values, except Pulverizer Reject Rate and Heating Value are at Actual Values

$HRD_{BOO2,\ Exp\text{-}Ref}$ $= NUHR_{Ref} * ((BE_{Ref} / BE_{Ref+ExpBOO2}) - 1)$
$= 2413.2 kcal/kWh * ((87.379 / 87.456) - 1)$
$= -2.1$ kcal/kWh

$HRD_{CombTA,\ Exp\text{-}Ref}$ $= NUHR_{Ref} * ((BE_{Ref} / BE_{Ref+ExpCombTA}) - 1)$
$= 2413.2 kcal/kWh * ((87.379 / 87.352) - 1)$
$= +0.7$ kcal/kWh

$HRD_{EGT,\ Exp\text{-}Ref}$ $= NUHR_{Ref} * ((BE_{Ref} / BE_{Ref+ExpEGT}) - 1)$
$= 2413.2 kcal/kWh * ((87.379 / 87.525) - 1)$
$= -4.0$ kcal/kWh

$HRD_{UltH,\ Exp\text{-}Ref}$ $= NUHR_{Ref} * ((BE_{Ref} / BE_{Ref+ExpUltH}) - 1)$
$= 2413.2 kcal/kWh * ((87.379 / 87.379) - 1)$
$= 0$ kcal/kWh

$HRD_{FuelTM,\ Exp\text{-}Ref}$ $= NUHR_{Ref} * ((BE_{Ref} / BE_{Ref+ExpFuelTM}) - 1)$
$= 2413.2 kcal/kWh * ((87.379 / 87.379) - 1)$
$= 0$ kcal/kWh

$HRD_{Wre,\ Exp\text{-}Ref}$ $= NUHR_{Ref} * ((BE_{Ref} / BE_{Ref+ExpWre}) - 1)$
$= 2413.2 kcal/kWh * ((87.379 / 87.379) - 1)$
$= 0$ kcal/kWh

$HRD_{BOO2,\ Act\text{-}Exp}$ $= NUHR_{Exp} * ((BE_{Exp} / BE_{Exp+ActBOO2}) - 1)$
$= 2471.0 kcal/kWh * ((87.302 / 87.142) - 1)$
$= 4.5$ kcal/kWh

$HRD_{CombTA,\ Act\text{-}Exp}$ $= NUHR_{Exp} * ((BE_{Exp} / BE_{Exp+ActCombTA}) - 1)$
$= 2471.0 kcal/kWh * ((87.302 / 87.275) - 1)$
$= 0.8$ kcal/kWh

$HRD_{EGT,\ Act\text{-}Exp}$ $= NUHR_{Exp} * ((BE_{Exp} / BE_{Exp+ActEGT}) - 1)$
$= 2471.0 kcal/kWh * ((87.302 / 86.215) - 1)$
$= 31.2$ kcal/kWh

$HRD_{UltH,\ Act\text{-}Exp}$ $= NUHR_{Exp} * ((BE_{Exp} / BE_{Exp+ActUltH}) - 1)$
$= 2471.0 kcal/kWh * ((87.302 / 87.893) - 1)$
$= -16.6$ kcal/kWh

$HRD_{FuelTM,\ Act\text{-}Exp}$ $= NUHR_{Exp} * ((BE_{Exp} / BE_{Exp+ActFuelTM}) - 1)$
$= 2471.0 kcal/kWh * ((87.302 / 85.601) - 1)$
$= 49.1$ kcal/kWh

$HRD_{Wre,\ Act\text{-}Exp}$ $= NUHR_{Exp} * ((BE_{Exp} / BE_{Exp+ActWre}) - 1)$

$= 2471.0 kcal/kWh * ((87.302 / 87.302) -1)$
$+ 0$ kcal/kWh

3.8.4.17 Expected Net Heat Rate

The Expected Net Heat Rate is the Reference Net heat rate plus the sum of all deviations due to differences between the reference and expected levels:

$$NUHR_{Exp} = NUHR_{Ref} + \Sigma HRD_{i,\ Exp\text{-}Ref}$$

$$\begin{aligned} NUHR_{Exp} = {} & NUHR_{Ref} + HRD_{BOO2,\ Exp\text{-}Ref} + HRD_{CombTA,\ Exp\text{-}Ref} + HRD_{EGT,\ Exp\text{-}Ref} + \\ & HRD_{UltH,\ Exp\text{-}Ref} + HRD_{FuelTM,\ Exp\text{-}Ref} + HRD_{Wre,\ Exp\text{-}Ref} + \\ & HRD_{CP,\ Exp\text{-}Ref} + HRD_{MU,\ Exp\text{-}Ref} + HRD_{SSO,\ Exp\text{-}Ref} + \\ & HRD_{HRHIV,\ Exp\text{-}Ref} + HRD_{RHA,\ Exp\text{-}Ref} + HRD_{SHAe,\ Exp\text{-}Ref} + \\ & HRD_{TP,\ Exp\text{-}Ref} + HRD_{TT,\ Exp\text{-}Ref} + HRD_{HPT,\ Exp\text{-}Ref} + \\ & HRD_{IPT,\ Exp\text{-}Ref} + HRD_{FUAP,\ Exp\text{-}Ref} + HRD_{FFWT,\ Exp\text{-}Ref} \end{aligned}$$

$$\begin{aligned} NUHR_{Exp} &= 2413.2 \ -2.1 + 0.7 -4.0 + 0 + 0 + 0 + 31.7 + 10.9 + 3.4 + 0 + 0 + \\ &\quad 1.0 + 0 + 0 + 0 + 0 + 13.0 + 3.2 \\ &= 2471.0 \text{ kcal/kWh} \end{aligned}$$

3.8.4.18 Unaccountable Heat Rate Deviation

The unaccountable deviation is the difference between the total deviation (Actual - Expected Heat Rate) minus the sum of the accountable deviations:

$$\begin{aligned} HRD_{UnAcc} &= (NUHR_{Exp} - NUHR_{Exp}) - \Sigma HRD_{i,\ Act\text{-}Exp} \\ &= (2654 kcal/kWh - 2471 \text{ kcal/kWh}) - (22.0 + 2.2 + 2.6 + 4.2 + 6.6 + 2.9 - \\ &\quad 3.1 + 4.9 + 1.7 + 3.8 + 1.9 + 6.2 + 16.3 + 23.8 - 4.5 + 0.8 + 31.2 - \\ &\quad -16.6 + 49.1 + 0) \\ &= 183 \text{ kcal/kWh} - 156.0 \text{ kcal/kWh} \\ &= 27.0 \text{ kcal/kWh} \end{aligned}$$

3.8.4.19 Cost of Heat Rate Deviation

One method to approximate the annual cost of a heat rate deviation, is to multiply the heat rate deviation by the fuel cost, the plant load factor, the unit rating (used to calculate the PLF) and the hours in a year. For example the heat rate deviation for unit auxiliary power was 23.8 kcal/kWh. If the fuel cost was 170 Rs/E6kcal, the PLF was 90%, and the unit rating was 210MW:

$$\begin{aligned} Cost_{UAP} &= 23.8 \text{ kcal/kWh} * (90/100) * (210MW*8760h/yr*1000kW/MW) * \\ &\quad 170 \text{ Rs/E6kcal} \\ &= 6{,}700{,}000 \text{ Rs/year} \end{aligned}$$

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3.9 Parameter Heat Rate Deviations - Effects, Causes, and Corrective Actions

Appendix D gives typical heat rate deviations for most parameters, as well as typical causes and corrective actions. Appendix E shows several logic trees that can be used to determine the root cause of a deviation that is calculated from this procedure.

These tables and logic trees should be used with caution, as they are generic in nature and in some cases the equations are approximations. They should be modified and made plant specific and more detailed.

If these calculations are being performed in real-time by a data acquisition system that has access to more data, the heat rate deviation accounting system and the logic trees can be tied together. If this is done, whenever a heat rate deviation occurs, the operator or engineer can immediately go to the appropriate logic tree. The next enhancement is to map to the diagram, the computer addresses of other data that is needed to move through the logic tree, along with "expected" values to compare with. Then some possible causes of deviations can be immediately ruled out or others can be pin pointed.

SECTION 4 PERFORMANCE TESTING

4.1 Introduction

There are several method available to assess the condition of equipment (to determine if corrective actions are necessary) in a power plant. One method is to periodically "open and inspect" equipment on some schedule. There are several disadvantages to this. First, many times the equipment is inspected, at some expense, when it is not required (and unfortunately sometimes equipment is damaged as a result of the inspection). Second, if a problem occurs just after an inspection, it may not be detected for a long time, during which time, additional damage is done, and the equipment does not operate efficiently.

A second method is to use station instruments to provide data to a calculation routine to determine equipment condition. This is similar to the practices described in Section 2, where station data is control charted and trend charted, but the calculations in Section 2 are usually very limited and most test calculations are very complicated. While this method can be useful, there is one significant drawback. Unfortunately the accuracy of station instruments is, in most cases, not high enough to provide reliable results (see Figures 9.1 and 9.2). For example, for high pressure turbines, a 1 °C error in main steam temperature will result in an error in the calculated section efficiency of about 0.46 percentage points. The combination of the uncertainties of all the station instruments to the section efficiency calculation results in overall uncertainty of close to 2 percentage points. This is not good enough to base crucial decisions such as "Should this turbine section be inspected?", which will have large costs and outage time.

The third method, and the recommended one, is to periodically install high accuracy instrumentation, set certain unit conditions to ensure that the data collected is repeatable and comparable to other tests, and to collect and analyze this information.

Tests with high accuracy test instruments are done to compare the current performance with expected performance and to assess performance degradation. This is accomplished best by periodically running tests over time and trending the results. Performance guarantee tests can be used as the baseline, or the initial periodic test, conducted after an overhaul can be used where no baseline exists. Each future test result can be compared to a now established standard. By accurately knowing the current level of performance and the best achievable, decisions based on facts can be made regarding performance improvement activities. The trended test data is an invaluable input to the maintenance-planning engineer for predicting equipment maintenance or the modification needs and also for evaluation of the maintenance work done. Frequently performance tests show that scheduled maintenance is not necessary, and can be postponed, reducing maintenance costs. On the other hand, the results may show that the equipment has deteriorated to the point that an early overhaul will be cost effective, that the savings in fuel cost will more that offset the additional maintenance expense. Typical examples are air heater seal

adjustment, condenser cleaning, burner replacement, boiler feed pump rebuild, feedwater heater partition plate leak repairs or the turbine overhaul.

Post maintenance tests are used to measure the effectiveness of maintenance activities. The performance test can determine if the anticipated results of the work, such as mentioned above, were achieved. If not, then the cause can be determined, so that the next time similar work is performed, the results will be satisfactory.

Another objective of the periodic tests with high accuracy instruments is to verify the readings of the station instruments. Frequently, station instruments "drift", so that its output does not correspond to its input, and it needs to be calibrated. Some instruments are functioning properly, but are not placed at the right location. For example, due to large stratification in the APH outlet flue gas duct, a single point measurement of Oxygen may not be truly representative of the average level of O_2 in the duct, requiring more probes or repositioning of the probe. Sometimes the instrument may be functioning properly and is also positioned correctly, but still indicates an erroneous value. For example if just above the O_2 probe there is a hole in the duct work allowing air ingress, the probe will indicate much more O_2 than is in the duct. These deficiencies can only be detected by doing periodic tests using high accuracy instruments.

Another objective of these tests is to determine a safe and reliable operating regime, that would also result in the best overall unit heat rate. For example, in the boiler area, many combination of operator controllable parameters including excess air level, mill outlet temperatures, mill primary air flows, air damper positions, windbox to furnace differential pressure, sootblowing frequency etc. are possible. By performing an optimization test, the combination that results in minimizing the unit heat rate can be determined.

In summary, the periodic performance tests are conducted to:

- Compare the current performance with expected performance
- Predict when and what, equipment maintenance or modification may be justified or required.
- Evaluate the effectiveness of maintenance work
- Validate the readings of 'on line' instruments
- Determine the most efficient operating regime for equipment
- Account for heat rate losses (such as is due to poor turbine efficiency)

4.2 Pre-Test Activities

To ensure that a test achieves the required results, is reliable and repeatable, is properly documented, is safe (personnel and equipment), and to minimize the time required, it is imperative that various preliminary activities are performed.

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Most of these preliminary activities should be discussed in a meeting between the test crew and the (other) plant staff. Some of the high level items that should be discussed, and agreed to by all parties, include the intent and purpose of the test, the required unit conditions, and personnel requirements. All parties should clearly understand what is to be learned, as well as the limitations of the test. This meeting serves a dual purpose. It clarifies the roles of each organization and, if the test is being conducted by an organization from outside the plant, the meeting gets the plant staff to become a part of the team. Other, lower level details that are usually discussed include the number of tests to be conducted, the data to be collected, the methods of data collection and analysis, etc.

4.2.1 Set Specific Purpose(s)

It is important that the specific purpose of the test be clearly defined and understood not only by all members of the test crew but also by all concerned organizations. Some of the purposes of the test could be:

- Determining if a unit meets some guaranteed performance (Acceptance Tests)
- Demonstration of performance guarantees
- Creating Baseline Data
- Optimization of operating parameters
- Database for maintenance planning and evaluation
- Diagnostics
- Input / Output Curve Generation

Performance Guarantee tests are done to determine if the supplied equipment meets the guarantees given by manufacturer . Baseline tests generate a 'reference database' that can be compared to future test on the same equipment. Optimization tests are done to determine the settings that produce some minimum (i.e. heat rate) or maximum (i.e. final feedwater temperature) value of a result. Often in addition to trying to maximize or minimize some result, there are additional constraints. For example when running a boiler optimization, there may be a constraint to limit the furnace exit gas temperature to some value to prevent slagging, or when optimizing FW heater levels to maximize FFWT, to prevent damaging the drain valves and piping the constraint to maintain DCAs at not higher than 15 °C may be added.

Baseline data when trended over a period of time reveals performance degradation and helps the planning engineer in fixing maintenance schedules and ordering spare parts or even suggesting modifications. These tests could also be done to understand plant problems or identify the specific causes. Another objective could be to generate Input/Output Curves, which are then used to develop incremental heat rate curves used for economic dispatching.

Whatever the purpose of the test, it is important that it is specifically defined and understood by all concerned personnel. For example, if "Boiler optimization" tests are to

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be conducted, the purpose would include a list of the parameters that are to be varied and over what range and what the "optimization" criteria is (i.e. best boiler efficiency, best unit heat rate, best heat rate keeping FEGT below 1200 °C, etc.). When a test is to be conducted for generating Input/Output Curves, the specific purpose might be to generate a heat rate curve, between 65-100% of full load, with equipment in its normal operating mode (i.e. do not make unusual adjustments, settings, etc.).

4.2.2 Unit Walkdown

If this is the first time a particular type of test is to be conducted on the unit by this test crew, the first step in the test preparations is to conduct a 'walkdown' of the unit to understand the current physical arrangements and operating condition of the unit. One of the purposes of the walkdown is to identify problem areas that could affect the quality of the test, and to recommend corrective actions, if required, to be completed prior to the tests.

Another purpose is to physically check existing test ports and tappings for their suitability. Alternatively, for units without any test ports, possible locations for new ports and associated requirements are identified. Also checked are operating platforms, and the availability of auxiliary services including compressed air, power receptacle, lighting, and service water. A sketch of the configuration showing test locations, and auxiliary services should be made, and other information recorded such as the size of test ports, the type of flanges or pipe fittings available, etc.

4.2.3 Determining Points To Be Measured

The test engineer has to decide in advance as to what data is to be collected, and how it will be collected. This would depend on the objective of the test, as well as the design, layout/location, condition and operation of the equipment to be tested. There can not be a common, single set of parameters to be measured for any type of test. The requirement will vary depending on individual unit needs and circumstances as well as the required accuracy levels. Section 4.7 provides some guidance on the parameters to be measured for each type test. ASME PTC for each component can be referred for the criterion for determining exact number of points.

The test engineer has to consider the cost involved for measurement in context of the accuracy levels required by the test and then decide upon a technically practical and cost effective set of points to be measured.

4.2.3.1 What Data is Required

The objective of the test usually determines what data is to be collected. For example, if the objective is to determine the enthalpy drop efficiency of the IP turbine, then this can

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be done by measuring the temperature at two locations and the pressure at the same two locations. However, if the objective is to collect the efficiency and other data that could be used to diagnose problem areas, then additional data should be collected.

4.2.3.2 Decide How To Measure Each Point

Once the required data is identified, the measurement methodology must be determined for each required data point. The primary considerations are the cost (equipment cost as well as labor) versus the accuracy. What accuracy is necessary for a measurement? This is usually determined by performing an uncertainty analysis. This requires that the effect on the final results(s), of each measurement be calculated. Large changes in the value of some points may have very little change on the final results, therefore the station instrument may be accurate enough. Other points may be very critical, and multiple high accuracy sensors may be used at those locations.

The first decision that is usually made is what data can be obtained using station instruments, and what data requires higher accuracy test instruments? There are two additional considerations (in addition to the cost vs. accuracy mentioned above). First, is one purpose of the test a check of the accuracy of the station instruments? For many non-critical instruments, performance tests are used to determine when calibration is necessary. If this is the case, then even if the station instrument is "good enough", a test instrument may be used. Second, if a portable data acquisition system is to be used for most measurements, then it may be cost effective to use test instruments, connected to the test DAS, so as to minimize the amount of manual data entry.

A frequently overlooked option where test instruments are to be used, is to use multiple sensors at an individual location to increase the accuracy of the measurement and result. The possibility of stratification in individual pipes or differences between similar pipes must be considered. For example, if there are two steam pipes, such as at the HP exhaust, the temperature, pressure, etc. should be measured in both because it is not uncommon for the two to be significantly different. Also the temperature in the outlet water pipe from a condenser is frequently stratified, and measurements should be made in as many locations as possible to determine the true average temperature. Another consideration for critical measurements is the possibility of using more than one instrument at a location. For example, for critical pressure readings, two transmitters can be connected to the same tap. The more sensors that are used at a location, the lower the uncertainty of that reading.

For locations where a higher accuracy test instrument is to be used, the particular instrument must be identified (type, model, range, etc.) For example for temperature, several options are available including RTDs or thermocouples or thermometers. For pressures, a manometer, Bourdon gauge, or pressure transmitter may be used. If a pressure transmitter is chosen, then what range is needed, should a gauge or absolute pressure transmitter be used, etc.

Another decision that must be made for some measurements is, should the data be recorded manually or via a data acquisition system?

Last, a decision must be made as to how often to record each value. With data acquisition systems, there is a tendency to get very fast scan times. Before setting a very fast data collection speed, remember that the sensor takes some time to respond to changes (especially temperature sensors), therefore scanning all instruments every second, is not going to result in any higher accuracy.

4.2.4 Collection of Characteristic Data

When testing for the first time, all available information such as design details of the equipment, heat balance diagrams, acceptance test report, historical and current operating data should be studied and kept available. Modifications done in the unit, differences between design and current parameters such as the quality of the coal being fired or changes to the amount of superheat surface area should be recorded. In particular, certain data is required for calculations, and should be tabulated. For example, for a pump test, in order to convert the differential pressure across the flow nozzle/orifice, the pipe and nozzle/orifice material and diameters must be known. For condenser tests, to calculate the expected heat transfer coefficient, the tube material, thickness, length, etc. must be known.

4.2.5 Data Sheets

Manual data collection sheets are log sheets used to record operating parameters in the control room and in the plant, during the test. These sheets are also used to record any unusual conditions or disturbances that occur during the test. The person recording data should sign the sheet at the end of the test. These data sheets can have three purposes:

- To ensure stability and constancy of parameters during the test (and from one test to another) and to record the magnitude of deviation (or swing) in the data.
- Manually collected data can be averaged and used for calculation purposes if required.
- Data collected from manual collection sheet and the Data Acquisition System should be used to compare with the test data obtained using high accuracy test instruments. If there is a discrepancy, the cause should be investigated.

4.2.6 Pre-Test Check Sheets

Pretest check sheets list the details of unit conditions required for the test. This includes a cycle isolation list of valves to be open and closed, damper positions, a list of set points for controllers, etc. Before each test, the list of conditions is checked and initialed

showing that each required condition is met. The sheet is then turned over to the test leader. These sheets ensure that test to test conditions do not change (except for the specific items, if any) so that the results of two tests will be comparable. For example, for a BFP test, the check sheet will include the specified pump flow, isolation of recirculation valves, etc.

4.2.7 Develop Calculation Procedure

Once the test objective is set and the data points have been determined, the detailed calculation procedure is developed. This step by step procedure may be done by hand, or it may be done using computer programs (such as spreadsheets or specialized data analysis software).

Too often, the test calculations stop at calculating some performance parameter for the equipment being tested (i.e. for feedwater heaters it might be the TTD and DCA, for an air heater it might be the gas side efficiency and x-ratio). The calculation procedure should continue, comparing the actual test result with some "expected" level. Additionally the deviation between the actual results and the expected level should be converted to a heat rate deviation and finally to a cost (typically an annual cost).

4.2.8 Develop List of Required Support

The test crew frequently needs additional help from the plant or other organizations. This includes such things as:

- Building probes
- Providing ice (for ice baths) each day of testing
- Providing power for test instruments (where none currently exists)
- Providing laboratory services such as coal sampling or analysis, ash sampling or analysis, etc.
- Calibration gases
- Assistance with instrument installation (i.e. power meters, etc.)

4.3 Testing

4.3.1 Duties of the Test Leader

A test leader is identified for the coordination of all activities pertaining to the test. The test leader is the focal point for the test and all questions pertaining to the test should be directed to him. The test leader's responsibilities include:

- Planning each day's activities at least one day in advance.

- Ensuring that the operation staff is informed and concurs with the required test conditions in advance.
- Assigning duties to the test crew and delegating responsibilities for various activities.
- Ensures that each person in the test crew understands their duties and is trained and qualified to perform his assigned tasks.
- Assigning a specific number to each test and ensuring that all members of the test crew and data collectors if any, are aware of it.
- Ensuring that the pretest check sheets and manual data sheets are properly and completely filled in, checked and collected.
- Ensuring that the unit has reached steady state condition before beginning a test run, and for monitoring the overall unit conditions during the test
- Sets the start and finish time of the test.
- Staying in contact with all locations during the test and deciding on suspension or extension of the test if required.
- Review the data collected after each test, and determining if the test is valid or should be discarded or repeated.

4.3.2 Development of the Test Plan

The test plan defines the number of the tests to be conducted and the conditions for each. It also indicates what type of analysis will be conducted on coal and ash samples, if samples will be combined before analysis, the approximate length of each test, and other details. The first test is a trial test,. If there are problems with the trial test, additional trial tests may be required until all problems are resolved. For some tests, such as routine HP/IP turbine efficiency, may have only one non-trial (actual) test. Other tests, such as boiler optimization, may have several non-trial tests, including one or more "baseline" tests to document the current level of performance, followed by several parametric tests, each varying one parameter, followed by one or more final tests at the "optimized" conditions.

The test plan is both rigid and flexible at the same time. The tests necessary to accomplishment the stated objectives should be completed, and sometimes the plan must be modified based on the results of completed tests. Also, it is not unusual to receive request to perform more tests for additional purposes. When this occurs the test leader, and the other affected persons should discuss the proposed changes and come to a consensus.

4.3.3 Purpose of the Trial Test

Every test plan starts with a trial test. This is a very important component and should be considered with the same seriousness as an actual test. One purpose is to ensure that there are no problems getting the conditions listed on the pre-test checksheet. Another intent is

to familiarize the data takers and test crew with the methodology and ensure integration of test activities at all locations. The data collected is also analyzed and its veracity confirmed. The readings from the high accuracy test instruments should be carefully compared to the station instruments. If there are any discrepancies they should be investigated. It should not be assumed that the test instrument is correct and the station reading is incorrect.

4.3.4 Coordination of Data Collection from Multiple Areas

During most of the tests simultaneous sampling is done at many locations and data from multiple areas is collected. For example in a boiler efficiency test, flue gas temperature and composition is measured at air heater outlet, air inlet temperature is measured at air heater inlet, fly ash is sampled at economizer outlet, bottom ash sample is collected from ash disposal system and coal sample is collected from feeder inlet chutes. It is imperative that measurement and sampling at all these locations starts and stops at the same time. If data is collected at one location for an hour, and at another for 90 minutes, the results may not be reliable. This coordination of data collection from multiple areas / locations is the responsibility of the test leader.

4.3.5 Review of Data Before Equipment Removal

Data collected in each test is reviewed immediately after each test. Once all the tests as planned have been carried out, a final check is made and data reviewed before removing the equipment. Too often, a test crew will "tear down" their equipment as soon as the last data point is collected, only to later learn that there was a problem, and the test is invalid and must be repeated.

4.4 Post Test Activities

The preparation for the test and conducting the test is only part of the job. Unless something is done with the results of the test, there is no purpose to the test. Many times the result of a routine test may reveal that are no problems. But even in those cases, the results must be passed along to other organizations. Also, the information should be added to the test results database, so if a problem is detected in the next test, there will be a narrower time frame in which to look for the cause.

4.4.1 Develop Action Plan to Address Poor Performance

If the test reveals a deficiency in the performance of a piece of equipment, additional analysis needs to be performed. One of the first steps is to determine the effect of the deviation. For example, if the test result shows that the HP turbine section efficiency is

two percentage points below the "expected value". Is this a cause for concern? Should the unit be shut down and the turbine overhauled as soon as it can be scheduled?

To start to answer this question, the heat rate deviation and cost of that deviation should be calculated. That will give an indication of the severity of the problem. If the cost of the poor turbine efficiency is only 25% of the cost of an overhaul, it is not worthwhile to do the outage now, but the efficiency should be watched more closely (and perhaps frequently) to see if the degradation continues.

If the cost of the degradation is significant, additional tests or studies may be needed to determine the root cause of the problem, and to develop an action plan to correct the problem. Frequently, if the test is properly designed, the additional data necessary for the analysis of the problem has already been collected. In this manner, the test report can include not only the performance of the equipment, but the effect on the heat rate, the cost, the cause, and recommended corrective actions

4.4.2 Develop Action Plan to Address Instrumentation Problems

Another action item that should result from each test is to make a list of station instruments that appear to be giving erroneous data, and to initiate work request to have them checked. The criteria for when action should be taken will vary for different instruments. The more critical instruments (main steam temperature, condenser pressure, boiler outlet O_2, etc.) would have a small "dead band" where moderate deviations would trigger maintenance to rectify the discrepancy in the station instrument, where other indications (heater drain temperatures, IP turbine extraction pressures, etc.) would be allowed to continue as-is until the error gets worse before requesting maintenance.

4.4.3 Test Report

Promptly preparing a good test report is as important as ensuring that the test is of high quality. The test report should include a short summary of the purpose and results, and also include very detailed information on the unit setup, the method(s) of data collection, and all collected data for both in-depth analysis and to allow the test to be repeated in the future. The following is an example of a test report outline, showing the various sections:

I. Cover Sheet
II. Table of Contents
III. Executive Summary
IV. Recommendations
V. Acknowledgments
VI. Introduction
  A. Purpose (why was the test run)
     (i.e. boiler efficiency based on station instrumentation had dropped

130

2 percentage points over the last year)

B. Objectives of the test.

(i.e. determine actual boiler efficiency from test quality instrumentation, check accuracy of station gas temperatures and flue gas oxygen, and determine effect of excess air, primary air flow and fuel air damper position on boiler efficiency and net unit heat rate)

C. Constraints/limitations placed on the test program.

(i.e. the test had to be completed before an outage started on dd-mm-yyyy, or burner tilts were not functioning and are locked in the horizontal position, etc.)

D. Organizations and personnel involved in the test and what were their responsibilities.

VII. Test Setup

A. Unit

This section describes the unit conditions required for the test, and the reasons behind the settings, etc.

B. Equipment

This section includes sketches and descriptions of the locations of test measurements, what instruments were used at each location.

VIII. Test Plan and Log

This section would contain the original test schedule, followed by a brief daily journal of what was actually done, problems that were encountered, and why deviations from the schedule were intentionally or unintentionally made.

IX. Results

This section would cover in detail what was learned from the test, what (if any) expected results could not be determined and why.

X. Appendixes

A. Unit Description

This section includes details (model numbers, sizes, type, configuration, design temperatures, pressures and flow rates, materials, etc.) on the equipment (boiler, condenser, air preheater, etc.) and related control systems. This section also has the details (data) needed in the calculations (i.e. specifications on flow nozzles, water leg measurements, etc.)

B. Measurements.

Table of measured parameters (i.e. main steam temperature or feedwater flow nozzle differential pressure). For each parameter, list the number of sensor(s) used, the type of sensor(s), the sensor(s) location, and how the value(s) were recorded.

C. Calculations

Include all steps on how the "raw" data was converted into useful information. It is often good to include from one test, a complete detailed calculation.

D. Pre-test Check Sheets (blank form)
E. Data Sheets Used (blank form)
F. List of DAS Points Collected (point name, ID number, etc.)
G. Calibration Data

4.5 Equipment List

One of the distinctive features of the performance test program described in this section is the high accuracy test equipment that is used. Different utilities organize the test program in different ways. Some, have a centralized test crew that does all performance tests. Other utilities have a test crew at each plant. Many utilities have a combination, a small test crew at each plant that performs the smaller scale tests, and another corporate test crew for running large scale tests such as turbine cycle heat rate tests and boiler optimization tests.

Section 4.5.1, 4.5.2, and 4.5.3 list the test equipment in three categories. First, the equipment each plant definitely should have. Second, equipment each plant might consider obtaining after gaining some experience with, and showing some benefits, from performance tests. Third, equipment that a regional or utility test crew should have for use at multiple locations. The lists are not absolutes, they are just a guide.

4.5.1 Basic Equipment Every Plant Should Have

Certain activities are critical to maintaining high efficiency at a power plant. The equipment that is required to perform these activities should be available at each plant. Some of these activities are:

- Locating condenser air inleakage.
- Calibrating primary air flow indications.
- Running clean air tests to ensure that the air flows are balanced in each coal pipe of a mill.
- Measuring the gas temperature and composition (O_2 and CO) at the furnace exit.
- Collecting representative fly ash sample to determine unburned carbon in ash.

Some of the equipment that would be required for these activities would include:

- Helium leak detector
- Glass Thermometer
- Pitot tubes
    - "S" type, sheathed, long enough to traverse the primary air ducts
    - "L" type, 300-400 mm long for traversing coal pipes

- High Velocity Thermocouple probe, for measuring temperatures and extracting a gas sample from the furnace outlet.
- High volume sampler for carbon in fly ash
- Gas Analyzers, for at least O_2 and CO
- Miscellaneous items such as:
    - Vacuum pumps
    - Manometers
    - Tubing
    - Desiccant
    - Ice baths
- Condenser brushes and/or scrappers and the water guns to shoot them. While strictly not "test equipment", the tools to quickly clean a condenser should be available at every power plant, without exception.

4.5.2 Advanced Equipment Every Plant Should Consider

Once a plant has some experience using test equipment and realizes the benefits of running performance tests, the additional equipment required for more extensive testing can be obtained. This includes the equipment to conduct:

- Dirty air and fineness test on pulverizers.
- Air heater leakage
- Condenser performance tests
- Enthalpy drop turbine efficiency tests
- H.P Heater Performance Tests
- BFPs Performance Tests
- Portable radios to allow communication between the members of the test crew for properly coordinating the work.

These activities would require some additional equipment (typical specification in Vol. II) including:

- Dirty air test kit, consisting of a dirty air probe, iso-kinetic coal sampling probe, aspirator, hoses, dustless connectors, etc.
- Probes for measuring the air/gas temperature and gas analysis (O_2 CO) in and out of the air heater. These are usually fabricated at the plant.
- An assortment of RTDs, perhaps 15-30
- An assortment of pressure transmitters, gauge, absolute and differential for measuring pressures from condenser pressure to BFP discharge, again perhaps 15-25 pieces.
- Power/energy meters, one or two.
- High accuracy thermometers.
- Some method for determining CCW flow (annubar, pitot traverse, dP taps calibrated by dye-dilution, etc.) for measuring condenser performance.

- A small data acquisition system, capable of accepting various types of signals from 30-60 sensors.

4.5.3 Regional / Utility Equipment That Should Be Obtained

Additional equipment is occasionally needed, but perhaps not as often at any particular plant, as to justify an individual plant purchasing it. What can be done in these circumstances is to have a regional or utility wide "pool" of equipment that each plant can draw on.

The equipment in this category includes additional quantities of the previously mentioned items:

- RTDs
- Pressure transmitters
- Energy meters
- Gas analyzers
- Additional data acquisition channels

Other items would include:

- High volume fly ash samplers
- Coal/ash analysis equipment
  - Proximate Analysis
  - Grindability
  - Fineness
- Infrared camera for detecting hot spots or leaking valves.

4.6 Test Equipment Calibration

Critical to the reliability of the test results, is the accuracy of the test equipment that is used to collect the raw data. All too often, an organization will obtain good quality, high accuracy test equipment, but neglect to have it calibrated regularly. The ASME PTC for steam turbine acceptance tests emphasizes this to the point that pre and post test calibrations are required for many instruments. The following are very general guidelines that may be considered when setting up a calibration program for test instruments. Another important part of an instrument calibration program is the quality of the calibration facility and the personnel who work there. This must also be considered in the calibration procedure.

Usually a 6 to 12 point calibration is performed, with instruments that exhibit hysteresis being calibrated in both directions(i.e., pressure transmitters are calibrated under increasing and decreasing pressure).

4.6.1 Pressure

4.6.1.1 Manometers

U tube manometers are considered as a primary standard. Incline and well type manometers should be compared to an u-tube manometer when initially obtained.

4.6.1.2 Pressure Transmitters

Pressure transmitters should be calibrated at least every two years, and more often if they are handled roughly, exposed to extreme changes in temperature, or are used in many tests in a short period of time. If a transmitter is over-ranged, it should also be re-calibrated.

4.6.2 Temperature

4.6.2.1 Thermocouples

Most thermocouples should be calibrated annually. However, thermocouples that are cycled to elevated temperatures repeatedly (type E >400 °C, type K>1100 °C, type J > 700 C) may need more frequent calibration. This would include thermocouples when used in HVTs, main steam and hot reheat steam thermowells, hot primary air flow calibrations, etc.

Thermocouples made from extension wire (such as used in measuring flue gas temperatures in large grids (where accuracy is not as critical) show be functionally checked before each use. Also, thermocouples should be calibrated with the extension wire that it will use.

4.6.2.2 RTDs

Resistance Temperature Detectors (RTDs) should be calibrated every 12-24 months

4.6.3 Flow

4.6.3.1 Pitot Tubes

Pitot tubes should be inspected prior to each use, looking for mechanical damage (a dent or burr, etc.) or pluggage. If the pitot is damaged, it is usually discarded.

4.6.4 Data Acquisition System

The data acquisition system should be calibrated at least annually, checking each channel, for all types of inputs (mV, Ω, mA, etc.)

4.6.5 Gas Analyzers

Gas analyzers for CO, O_2 and CO_2 etc. should be checked in laboratory with standard gases of different concentration (at least at 3 points) every year. However, the gas analyzers need to be checked for calibration everyday before the start of the test for correcting the drifts if any.

4.6.6 Energy Meters

Energy meters have a direct bearing on the heat rate assessment. These should be got calibration every year against the standards maintained in-house or at a test lab.

4.7 Types of Tests

4.7.1 HP / IP Turbine Enthalpy Drop Efficiency Test

4.7.1.1 Purpose

The purpose of HP/IP Turbine Enthalpy Drop Test is to determine the efficiency of HP and IP sections of the turbine, trend their performance and generate data to pinpoint the probable cause of degradation. The trends can then be used for determination of the optimum point at which the costs incurred due to performance degradation exceed the cost to be incurred for restoration of equipment. It is also used for validation of online instruments.

4.7.1.2 Frequency

Time based, typically annually, but the other triggers can take precedence:

- Before and after turbine overhauls
- Change in pressure ratio of first stage to HP exhaust or HRH at Intercept Valve to IP Exhaust
- Increase in Steam flow, at constant valve position
- Degradation in cylinder efficiency based on line calculations
- Abnormal first stage pressure
- High extraction temperatures

4.7.1.3 Unit Conditions

The primary criteria is to here 'repeatability' of each test. Repeatability is obtained easiest at 'Valve Wide Open'- VWO condition for all machines and not determined by a

particular load, MW or steam flow. This is necessary to ensure that any change in efficiency from test to test is due to change in turbine condition and not due to changes in test conditions.

Other necessary conditions include feedwater heaters in service and extractions valves wide open, any other extractions (to auxiliary drive turbines, oil heating, etc. should be at the "normal" setting, etc.

The Test Engineer is responsible for ensuring that the unit has reached steady state before beginning a test run. Generally, the conditions listed in ASME PTC 6-1976-Steam Turbines, are followed, but the Test Engineer may, at his own discretion make this decision. If there is any deviation from the stipulated guidelines as listed in ASME PTC these have to be properly documented.

4.7.1.4 Data to be collected

For calculating the section efficiency of HP and IP turbine, the following minimum data needs to be collected by test instruments:

- Throttle Pressure
- Throttle Temperature
- Cold Reheat Pressure
- Cold Reheat Temperature
- Hot Reheat Pressure
- Hot Reheat Temperature
- IPT Exhaust Pressure
- IPT Exhaust Temperature

It is also highly desirable to measure the temperature and pressure (at the turbine) of any extractions, the "HP turbine "bowl" pressure (measured downstream of the control valves) and the first stage pressure (measured downstream of the first row of rotating blades).

Test of station reading should be taken of the main steam flow/feedwater flow, generator output, and condenser pressure.

In addition to the above, readings of the control room data and DAS printouts are also taken. Typical control room data and DAS data required are given in Vol. – II under the specific test procedures.

4.7.1.5 Calculations

HP/IP sectional efficiency is calculated as

$$\eta = \frac{(h_{in} - h_{out})}{(h_{in} - h_{isen})}$$

where h_{in}- enthalpy of steam at section inlet
h_{out}- enthalpy of steam at section outlet

h_{isen} - enthalpy of outlet steam calculated at outlet pressure and entropy determined by inlet conditions

Other calculations that should be made include:

- Corrected stage pressures (for initial steam conditions)
- Corrected main steam flow (for initial steam conditions)
- Corrected generator output (for boundary conditions, HP and IP turbine efficiency)
- Elevation of measured extraction temperature above the temperature from the expansion line.

4.7.1.6 Analysis

If section efficiency has deteriorated, the corrected main steam flow and stage pressures should be examined. If the corrected steam flow has decreased, it indicates a blockage in the turbine (such as by deposits or foreign body damage). If the flow has increased, it indicates an opening of the flow path, as occurs with erosion. To determine the location, examine the stage pressures. The stage pressures will have increased before blockage, or decreased where there has been erosion. If the section has an extraction, and the temperature there is elevated, it can indicate increased seal clearances (as from a rub).

If the corrected generator output has deceased, it indicated either poor LP turbine efficiency or a cycle isolation problem.

Probable causes of low HP turbine efficiency could be erosion of nozzle blocks, erosion of turbine blades, broken turbine blades, excessive gland leakage, strip seal leakage, an erratic control valve or deviations caused by deviations in operating parameters. Low IP turbine efficiency could be due to erosion of turbine blades, deposits on turbine blades, reheat bypass valve leakage, excess gland seal leakage and strip seal leaks.

4.7.1.7 References

ASME Performance Test Code – 6, Steam Turbines
ASME Performance Test Code – 19.1, Measurement Uncertainty
ASME Performance Test Code – 19.2, Pressure Measurement
ASME Performance Test Code – 19.3, Fundamentals of Temperature Measurement
For detailed test procedure refer Heat Rate Improvement Guidelines for Indian power Plants Vol. -II

4.7.1.8 Attachments

Table 4.1 Typical Enthalpy Drop Turbine Test Pre Test Check Sheet

HP / IP Turbine Efficiency Test Pre – Test Check Sheet

Station : _____ **Unit No:**_____
Date: ____/____/____ **Checks Made By:**_____

Description	Status	Initials
Control Valve Position "A" and "B"	%	
Auxiliary Steam Feed	Isolated	
Both NRVs to No ----- HP FWH:	Open	
Both NRVs to No ----- HP FWH:	Open	
Both NRVs to No ----- HP FWH:	Open	
Both NRVs to Deaerator:	Open	
Both NRVs to No ----- LPH	Open	
Gland Steam Pressure Controller:	Auto and set to ---- kg/cm2	
HP Bypass Valves "A" and "B"	Closed	
LP Bypass Valves "A" and "B"	Closed	
Main Steam Line Drains (----No. total)	Closed	
Hot Reheat Drains (----No. total)	Closed	
Cold Reheat Drains (---- No. total)	Closed	
HP Steam Chest Warming (---- No. total)	Closed	
HPH Drain Cooling Zone Steam Vent	Closed	
HPH Drip to Condenser	Closed	
HPH --- Extr. Drain to condenser (----No. total)	Closed	
HPH --- Extr. Drain to condenser (---- No. total)	Closed	
DA High Load Extr. Drain to Condenser (---- No. total)	Closed	
DA Low Load Extr. Drain to Condenser (----- No. total)	Closed	
LPH --- Extr. Drain to Condenser (----No. total)	Closed	
Trap Bypasses	Closed	

Notes / Special Conditions:

Table 4.2 Typical Enthalpy Drop Turbine Test Control Room Data Sheet

HP / TP Turbine Efficiency Test Control Room Data Sheet

Unit No. : **Date :** **Reading Taken by :**

Description	**Instrument***	**Unit**	**Time**				
			Hrs	Hrs	Hrs	Hrs	Hrs
Unit Load		MW					
Control Valve "A"		%Open					
Control Valve "B"		%Open					
Turbine Speed		rpm					
Main Steam Flow		t/h					
Feed Water Flow		t/h					
HP Turbine Inlet Temp (Left)		^{0}C					
HP Turbine Inlet Temp (Right)		^{0}C					
HP Turbine Inlet Press (Left)		kg/cm^2					
HP Turbine Inlet Press (Right)		kg/cm^2					
IP Turbine Inlet Temp (Left)		C					
IP Turbine Inlet Temp (Right)		C					
IP Turbine Inlet Press (Left)		kg/cm^2					
IP Turbine Inlet Press (Right)		kg/cm^2					
Condenser Vacuum Grid "A"		mmHg					
Condenser Vacuum Grid "B"		mmHg					
Generator Hydrogen Pressure		kg/cm^2					
LP Gland Steam Line Pressure		kg/cm^2					
LP Gland Steam Line Temperature		C					
Total Unit Aux. Power (Beginning and Ending Readings)							
Notes : 1. 2. 3.							

Table 4.3 Typical Enthalpy Drop Turbine Test DAS Data Sheet

DAS Points for HP / IP Turbine Efficiency Testing

Sl. No.	Point ID	Description	Unit
1.		Unit Load	MW
2.		Control Valve "A"	%Open
3.		Control Valve "B"	%Open
4.		Turbine Speed	rpm
5.		Main Steam Flow	t/h
6.		Feed Water Flow	t/h
7.		HP Turbine Inlet Temp (Left)	^{0}C
8.		HP Turbine Inlet Temp (Right)	^{0}C
9.		HP Turbine Inlet Press (Left)	kg/cm^2
10.		HP Turbine Inlet Press (Right)	kg/cm^2
11.		HP Turbine Ext. ----- Temp (turbine end)	^{0}C
12.		HP Turbine Ext. ----- Press. (turbine end)	kg/cm^2
13.		HP Turbine Exhaust Temp (Left)	^{0}C
14.		HP Turbine Exhaust Temp (Right)	^{0}C
15.		HP Turbine Exhaust Press (Left)	kg/cm^2
16.		HP Turbine Exhaust Press (Right)	kg/cm^2
17.		IP Turbine Inlet Temp (Left)	^{0}C
18.		IP Turbine Inlet Temp (Right)	^{0}C
19.		IP Turbine Inlet Press (Left)	kg/cm^2
20.		IP Turbine Inlet Press (Right)	kg/cm^2
21.		IP Turbine Ext. ----- Temp (turbine end)	^{0}C
22.		IP Turbine Ext. ----- Press. (turbine end)	kg/cm^2
23.		IP Turbine Ext. ----- Temp (turbine end)	^{0}C
24.		IP Turbine Ext. ----- Press. (turbine end)	kg/cm^2
25.		IP Turbine Exhaust Temp (Left)	^{0}C
26.		IP Turbine Exhaust Temp (Right)	^{0}C
27.		IP Turbine Exhaust Press (Left)	kg/cm^2
28.		IP Turbine Exhaust Press (Right)	kg/cm^2
29.		Condenser Vacuum Grid "A"	mmHg
30.		Condenser Vacuum Grid "B"	mmHg
31.		Superheater Spray	t/hr
32.		Reheater Spray	t/hr
33.		Generator Hydrogen Pressure	kg/cm^2
34.		LP Gland Steam Line Pressure	kg/cm^2
35.		LP Gland Steam Line Temperature	^{0}C

4.7.2 Condenser Performance Test

4.7.2.1 Purpose

Condenser tests are done to assess the thermal and hydraulic performance of the condenser in terms of tube fouling, air-in-leakage and to optimize condenser-cleaning schedules. It is also used to trend the performance parameters and generate data to pinpoint the probable cause of degradation. The trends can then be used for deciding the future course of action like tube cleaning schedules, attending air-in-leakage, adequacy of CW flow, tube replacement programs etc.

4.7.2.2 Frequency

Time based, typically every two or three months, but other triggers can take precedence:

- High condenser pressure
- Pre and post tube cleaning
- High water box dP
- Excessive CW temperature rise

4.7.2.3 Unit Conditions

Unit should be in operation at normal full load in steady state condition. Every effort should be made to conduct the test runs close to normal operating parameters. The unit should be operating under specified cycle condition:

- Normal cascading of H.P heater drips
- Isolation of high energy drains etc.
- No make up should be taken during the test period.
- Condenser water box are full (by operating water box vent)

Generally the conditions listed in ASME performance test code PTC 12.2 is followed prior to starting the test.

4.7.2.4 Data to be collected

- Cooling water inlet temperature (one point per pipe)
- Cooling water outlet temperature (multiple points per pipe)
- Condenser pressure
- Condensate temperature
- Air-steam mixture temperature (between condenser and vacuum pumps/steam jets)
- Cooling water flow to condenser (see Section 4.7.2.9)
- Pressure drop (dP) across condenser (inlet to outlet waterbox)

- Air-steam mixture flow from condenser (rotameter)

4.7.2.5 Calculations

Condenser Cleanliness factor (CF) is calculated to determine the degree of tube fouling. The Heat Exchange Institute (HEI) condenser cleanliness factor is a comparison of the as operating thermal performance of the condenser to the as designed thermal performance of the condenser with 100% clean tubes for a given set of operating conditions. The cleanliness factor as defined mathematically by HEI is given below.

$$CF = \frac{U\ actual}{U\ theoretical}$$

$$U\ actual = \frac{CCWF \times Cp \times (CCWTO - CCWTI) \times Density\ of\ water}{A\ condensing \times LMTD}$$

Where:

Uactual	= Kcal/hr m^2 oC
Density of water	= 1000 Kg/m^3
A condensing	= m^2 (condensing surface area)

$$LMTD = \frac{Tout - Tin}{Ln \frac{Tsat - Tin}{Tsat - Tout}}\ deg.C$$

U theoretical = Ubase x Tin correction factor x tube correction factor x 4.882428

Where:

U theoretical	= Kcal/hr m^2 deg. C
Ubase	= heat transfer coefficient in Btu/hrsqft. at 70°F C.W inlet temp.,18 BWG and Admiralty Metal (from HEI standard Appendix H table H.1)
Tin correction factor	= correction factor for actual C.W inlet temp (from HEI standard Appendix H table H.2)
tube correction factor	= correction factor for tube material and tube wall gauge (from HEI standard Appendix H table H.3)

Condensate Depression is calculate by subtracting the Condensate Temperature from the steam temperature corresponding to the condenser pressure:

Condensate Depression = Tsat - Condensate Temperature

The degree of subcooling of the air/steam mixture (going to the vacuum pumps/steam jets) is calculated by subtracting the air/steam mixture temperature from the steam temperature corresponding to the condenser pressure:

Air/steam Subcooling = Tsat - Air/steam mixture temperature

4.7.2.6 Analysis

Probable causes of low cleanliness factors are:

- Tube fouling
- Excessive air in leakage
- Low levels in waterboxes

These deviations could be corrected by starting an additional ejector or a vacuum pump, eliminating the air-in-leakage, filling the waterboxes, or cleaning the condenser at the next available opportunity.

If the cleanliness factor is acceptable, but he condenser pressure is high, possible causes include:

- Excess condenser load
- Low cooling water flow
- Increase in CW inlet temperature (which could be either due to changes in ambient conditions or problems with cooling tower performance)

These deviations could be corrected by increasing the CW flow (increasing the opening of the inlet or outlet valves, starting an additional CCW pump, cleaning the tube sheet, etc.), putting additional cooling tower in service, or locating an eliminating the source of the additional heat load on the condenser

Cooling water flow through the condenser will decrease if the condenser tubes become fouled or if a large number of tubes are plugged over a period of time. Any reduction in cooling water flow would result in a higher cooling water temperature rise across the condenser. But the same could also result from an increase in condenser heat load, generally caused by passing of high-energy drains. Waterbox priming should be continuously maintained so that the condenser tubes run full and the waterbox differential should be monitored.

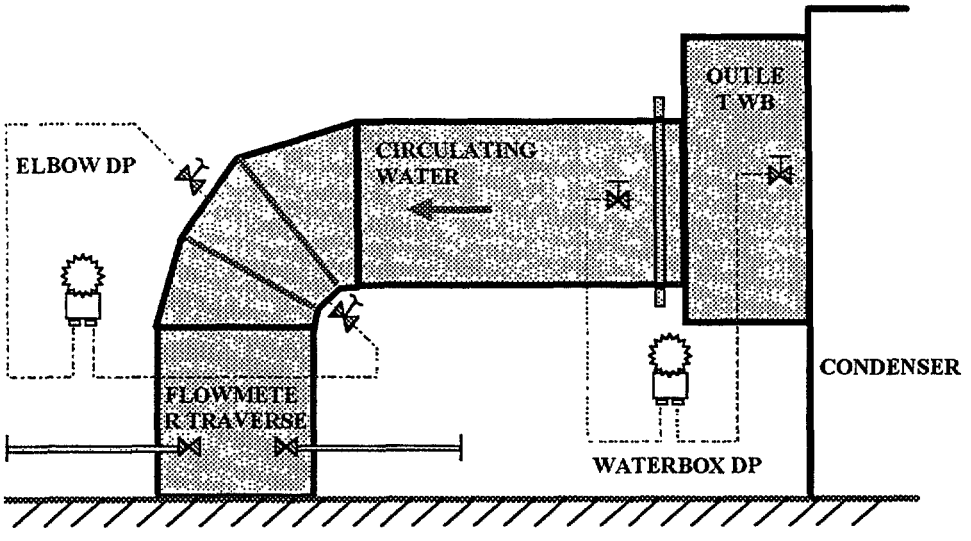
Also, the condensate temperature is compared to the saturation temperature for the operating condenser pressure to as a check of the condenser pressure measurement and to measure the amount of subcooling of the condensate. High subcooling can result in deteriorated heat rate, increased levels of dissolved oxygen and corrosion of bottom tubes of the tube bundle.

If the air/steam mixture is not subcooled approximately 4-5 °C, then the air removal equipment may be handling more steam than air, essentially reducing the air removal capacity of the equipment.

4.7.2.7 Reference Documents

- ASME Performance Test Code 12.2 on Steam Condensing Apparatus
- HEI Standards for Steam Surface Condensers
- For detailed test procedure refer Heat Rate Improvement Guidelines for Indian power Plants Vol. -II

4.7.2.8 Attachments



ELBOW TAP ARRGT

Figure 4.1 Dye-dilution Set-up

Table 4.4 Example of Condenser Test Pre-test Check Sheet

Condenser Performance Test Pre-Test Check Sheet

Unit: _____ Date: ____/____/____

Checks Made By:_____

		Initials:
Auxiliary Steam Feed :	Isolated	_____
Gland Steam Pressure Controller:	Auto and set to 0.07 kg/cm2	_____
HP Bypass Valves "A" and "B":	Closed	_____
LP Bypass Valves "A1", "A2", "B1", "B2":	Closed	_____
Main Steam Line Drains (12 total)	Closed	_____
Hot Reheat Drains (12 total)	Closed	_____
Cold Reheat Drains (8 total)	Closed	_____
HP Steam Chest Warming (4 total)	Closed	_____
HPH #6A Drip to Condenser	Closed	_____
HPH #6B Drip to Condenser	Closed	_____
HPH #5A Drip to Condenser	Closed	_____
HPH #5B Drip to Condenser	Closed	_____
HPH #3 Drip to Condenser	Closed	_____
DA High Load Drain to Condenser (3 total)	Closed	_____
DA Low Load Drain to Condenser (1 total)	Closed	_____
HPH #6 Extr. Drain to Condenser (6 total)	Closed	_____
HPH #5 Extr. Drain to Condenser (6 total)	Closed	_____
LPH #3 Extr. Drain to Condenser (3 total)	Closed	_____
BFP Gland Drains to Condenser	Auto	_____
HP Spray	Closed	_____
LP Spray	Auto - Open, but CV Closed	_____
CW Pumps	No Change in Number or Speed	_____
CW Inlet Valves	Full Open	_____

DAS Set-Up

Location	elevation	ID	Node #	Channels	
CW Outlet "A"	0m	_____	_____	_____	_____
CW Outlet "B"	0m	_____	_____	_____	_____

Notes / Special Conditions:

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Table 4.5 Example of Condenser Test Pre-test Control Room Data Sheet

Condenser Performance Test Control Room Data Sheet

Unit: _____ Date: ____/____/_ Readings Taken
___ By:_____

Time		
Unit Load	gauge	MW
Turbine Speed	digital	rpm
Main Steam Flow	gauge	kg/h
Feedwater Flow	gauge	kg/h
Condenser Vacuum "A"	gauge	mm Hg
Condenser Vacuum "B"	gauge	mm Hg
Hotwell Level	recorder	m
HP Gland Steam Line Pressure	BB gauge	kg/cm^2
HP Gland Steam Line Temperature	BB gauge	C
LP Gland Steam Line Pressure	BB gauge	kg/cm^2
LP Gland Steam Line Temperature	BB gauge	C
LP Cyc Rear Exh "A" Temp	BB dial	C
LP Cyc Rear Exh "B" Temp	BB dial	C
AEP "A" Pump Motor Current	BB gauge	Amps
AEP "B" Pump Motor Current	BB gauge	Amps
AEP "C" Pump Motor Current	BB gauge	Amps
CW Discharge Valve Position "A1"	BB dial	% Open
CW Discharge Valve Position "A2"	BB dial	% Open
CW Discharge Valve Position "B1"	BB dial	% Open
CW Discharge Valve Position "B2"	BB dial	% Open
CW Temp "A1" Inlet	BB dial	C
CW Temp "A1" Outlet	BB dial	C
CW Temp "A2" Inlet	BB dial	C
CW Temp "A2" Outlet	BB dial	C
CW Temp "B1" Inlet	BB dial	C
CW Temp "B1" Outlet	BB dial	C
CW Temp "B2" Inlet	BB dial	C
CW Temp "B2" Outlet	BB dial	C
AEP "A" Air Flow	local	kg/s
AEP "B" Air Flow	local	kg/s
AEP "C" Air Flow	local	kg/s

CW Discharge Valve Position "A1"	local	% Open
CW Discharge Valve Position "A2"	local	% Open
CW Discharge Valve Position "B1"	local	% Open
CW Discharge Valve Position "B2"	local	% Open

Notes: _____ _____ _____

4.7.2.9 Determination of CCW Flow / Condenser Duty

The condenser duty is the total heat load on the condenser from the LP turbine and all the drains that exhaust in the condenser. The duty may be calculated from the condenser-circulating waterside as the energy gain of the water as it flows through the condenser.

$$\text{Duty} = \text{CCWF} * \text{Cp} * (\text{CCWTO} - \text{CCWTI})$$

where Duty - Condenser duty kcal/hr
CCWF- mass flow rate of the circulating water kg/hr
Cp - mean specific heat of circulating water kcal/kg-C

The condenser duty can also be calculated by doing an energy balance around the steam side of the condenser, taking into account the exhaust steam from LP turbine and all the drains that exhaust into the condenser.

4.7.2.9.1 Direct Measurement of CCW Flow

Measurement of CW flow rate can provide indications of cooling system problems and can be used to calculate turbine cycle heat rate. Unfortunately, this parameter is often difficult to accurately measure in power plants. Some of the methods available for measurement of CCW flow are given below.

4.7.2.9.1.1 Pitot Traverse

Pitot tubes can be used to accurately measure CW flow provided there is a straight accessible section of circulating water pipe, which has been fitted with proper traverse valves. The valves should be installed so that two to three multi-point traverses can be performed. A multi-point traverse is essential to obtain a good flow measurement to offset the stratification effect. The traverse shows the velocity profile across the pipe and from this average flow can be estimated. Accuracy will be acceptable if the velocity profile is relatively flat.

4.7.2.9.1.2 Annubar

It is basically a fixed multiple hole pitot tube and is used for continuous measurement of CW flow. This has however one limitation that the chances of clogging of the sensing ports are high since the hole size are very small and on line cleaning is very difficult. It requires frequent cleaning during available opportunities.

4.7.2.9.1.3 Ultrasonic

Another method to measure CW flow is the use of ultrasonic flow devices. Pairs of ultrasonic transducers are mounted on the pipe. The acoustic propagation time between each pair of transducers is measured and a mean flow is determined.

4.7.2.9.1.4 Pipe Elbow or Waterbox to Pipe Transition dP calibrated by Dye-Dilution

The dye dilution technique is designed to know the discharge flows in closed turbulent pipes and open channels in various power plant applications. The flow measurement is based on the mass conservation of dye, which can be summarized by the relation

$$Qs = (Ci/Cs)Qi$$

Where Qs Stream flow rate (to be determined)
Qi Dye injection flow rate
Ci Dye injection concentration
Cs Diluted concentration of the dye

A known concentration and quantity of dye is injected at one point in the stream and is sampled for diluted concentration at a down stream point after proper mixing is ensured. The concentration of the diluted dye is measured with the help of a UV Flourometer which is set up at the sampling station. The quantity of CW flow can be calculated using the above relation. The exercise is repeated for different flows and the CW flow is calculated in each case.

To correlate this flow with the differential pressure, a dP transmitter is set up across the elbow or water box to outlet pipe (see Figure 4.1) so that dP transmitter can be calibrated for future flow monitoring. For each of the flows calculated above the corresponding dP is noted and a graph is plotted with dP versus flow. This curve can be used as a reference for calculating the CW flow in future by monitoring dP.

- This system estimates the mass flow rate unlike other techniques where velocity is estimated
- Specially suited for the turbulent flows since turbulence helps in better mixing
- Can be used for half filled pipes as well as open channels

4.7.2.9.2 Turbine Cycle Energy Balance

CW flow can be calculated by using turbine cycle energy balance.

Duty = (Heat Added MS + Heat Added HRH)
- 860 (Pgen + Pgen Losses + Heat Loss rad.)

Where:

.Duty	= Kcal/Hr
Heat Added MS	= Flow MS (hMS - hFW) Kcal/hr
Flow MS	= Kg/hr (Flow Main Steam)
hMS	= Kcal/Kg (Enthalpy of Main Steam)
hFW	= Kcal/Kg (Enthalpy of Feed water)
Heat Added HRH	= Flow HRH (hHRH - hCRH) Kcal/hr
Flow HRH	= Kg/hr (Flow Hot Reheat Steam)
Heat Loss rad	= 0.1% of Pgen (Radiation Losses) kW
Pgen	= kW (Gross Generator Output)
Pgen Losses*	= kW (Mech. Losses + Iron Losses + Stator Current Losses)

Note : Additional data other than indicated under item No. 4.7.2.4 required to be collected.

* Values to be taken from Generator Loss Curve

$$CCWF = \frac{Duty}{Cp\,(CCWTO - CCWTI) \times D} \quad m^3/hr$$

Where:

CCWF	= m^3/hr
Duty	= Kcal/hr
Cp	= 1 Kcal/Kg deg. C (Specific heat of water)
D	= 1000 Kg/cubic meter(Density of water)
CCWTO	= °C (Average C.W Outlet temp.)
CCWTI	= °C (Average C.W Inlet temp.)

4.7.2.9.3 IP-LP Turbine Expansion Line

CW flow can be calculated by using and assumed IP-LP turbine expansion line and the measured condensate flow. By plotting the expansion line on a Mollier diagram, the UEEP can be determined by first reading off the ELEP where the expansion line crosses the measured condenser pressure. Then using the turbogenerator exhaust loss curves, the UEEP can be determined.

The exhaust steam flow is calculated by subtracting all miscellaneous drain the flow into the condenser from the measured condensate flow.

The condenser duty is then the exhaust steam flow multiplied by the difference of the UEEP and the condensate enthalpy (determined from the steam tables at the condensate temperature).

$$\text{Duty} = \text{Exhaust steam flow} * (\text{UEEP} - h_{cond}) + \Sigma\{\text{Misc drain flow}_i * (\text{drain h} - h_{cond})\}$$

4.7.2.9.4 CCW Pump Curve

CW flow can be estimated from the CCW pump curve and the measured total head of the pump. This is probably the least accurate method, but it provides a check of the other methods. If one of the other method is used, and this method calculates a smaller number, there is probably a problem with the other method.

4.7.3 Feedwater Heater Performance Test

4.7.3.1 Purpose

The purpose of Feedwater heater tests is to assess the current performance and to provide data to assist in optimization of normal operation of the heater. Heater performance can be analyzed by calculating and monitoring Terminal Temperature difference (TTD), Drain Cooler Approach (DCA), the pressure drop on the feedwater side and the temperature rise across the heater. These parameters are trended over time for identification of abnormal changes in heater performance and to determine whether corrective action is required to maintain optimum performance.

4.7.3.2 Frequency

Time based, typically every 6 to 12 months, but the other triggers can take precedence:

- High TTD
- High DCA
- High drain temperature
- Low FW temperature
- Excessive tube bundle pressure drop

4.7.3.3 Unit Conditions

- Unit should be in operation at normal full feedwater flow at steady state condition.
- Ensure normal cascading of the heater drains as per the specified cycle conditions.
- Ensure venting of steam side and waterside to remove non-condensable gases.
- Operation of the feedwater heater shall be brought to the steady state condition prior to initiating the test run. It shall be kept at this condition through out the test run

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4.7.3.4 Data to be collected:

- Feedwater Inlet temperature to the heater
- Feedwater Inlet pressure to the heater
- Feedwater Outlet temperature from the heater
- Feedwater Outlet pressure from the heater
- Extraction steam pressure at turbine end
- Heater Shell pressure
- Extraction steam pressure at heater end
- Extraction steam temperature at heater end
- Drain / Drip Temperature
- Feed water differential pressure across the heater

4.7.3.5 Calculations

- Terminal Temperature Difference (TTD) = T_{sat} - T_{fwout}

  where T_{sat} - saturation temperature determined at heater shell pressure, deg C
  T_{fwout} - temperature of feedwater leaving the heater, deg C

- Drain Cooler Approach (DCA) = T_{drain} - T_{fwin}

  where T_{drain} - temperature of drain/drip leaving the heater, deg C
  T_{fwin} - temperature of feedwater entering the heater, deg C

- Temperature Rise (TR) = T_{fwout} -T_{fwin}

4.7.3.6 Analysis:

Four parameters TTD, DCA, temperature rise across the heater and pressure drop across the heater are trended. If the TTD values are higher than expected it could be because of improper heater level, inadequate heater venting, abnormal extraction steam conditions, waterbox partition plate leakage, tube leaks or just dirty tubes with deposits. TTD is a general indication of the amount of heat transfer through the heater, while DCA is an indicator of heater level and used primarily as a diagnostic tool for detecting tube pluggage, leaking tubes or a cracked baffle plate. Excessive deposits can result in a decreased temperature rise across the heater or in an increased tube bundle pressure drop.

4.7.3.7 Reference Documents

- ASME Performance Test Code 12.1 for closed feed water heaters
- HEI standards for closed feed water heaters
- For detailed test procedure refer Heat Rate Improvement Guide lines Vol. –II

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4.7.3.8 Attachments

Table 4.6 Example of Feedwater Heater Test Pre-test Check Sheet

HP FW Heater Performance Test Pre-Test Check Sheet

Unit: _____ Date: ____/____/____

Checks Made
By:_____

		Initials:
Auxiliary Steam Feed :	Isolated	_____
Both NRVs to 6A HP FWH :	Open	_____
Both NRVs to 6B HP FWH :	Open	_____
Both NRVs to 5A HP FWH :	Open	_____
Both NRVs to 5B HP FWH :	Open	_____
Both NRVs to Deaerator :	Open	_____
Both NRVs to LPH3:	Open	_____
HP Bypass Valves "A" and "B":	Closed	_____
LP Bypass Valves "A1", "A2", "B1", "B2":	Closed	_____
HPH 6A Drain Cooling Zone Steam Vent	Closed	_____
HPH 6B Drain Cooling Zone Steam Vent	Closed	_____
HPH 5A Drain Cooling Zone Steam Vent	Closed	_____
HPH 5B Drain Cooling Zone Steam Vent	Closed	_____
HPH 6A Condensing Zone Steam Vent	Open	_____
HPH 6B Condensing Zone Steam Vent	Open	_____
HPH 5A Condensing Zone Steam Vent	Open	_____
HPH 5B Condensing Zone Steam Vent	Open	_____
HPH 6A Shell Drain	Closed	_____
HPH 6B Shell Drain	Closed	_____
HPH 5A Shell Drain	Closed	_____
HPH 5B Shell Drain	Closed	_____
HPH 6A Shell Side Relief Valve	Closed	_____
HPH 6B Shell Side Relief Valve	Closed	_____
HPH 5A Shell Side Relief Valve	Closed	_____
HPH 5B Shell Side Relief Valve	Closed	_____
HPH 6A FW Ent Htr Line Drain	Closed	_____
HPH 6B FW Ent Htr Line Drain	Closed	_____
HPH 5A FW Ent Htr Line Drain	Closed	_____
HPH 5B FW Ent Htr Line Drain	Closed	_____
HPH 6A FW Lvg Htr Line Drain	Closed	_____
HPH 6B FW Lvg Htr Line Drain	Closed	_____
HPH 5A FW Lvg Htr Line Drain	Closed	_____
HPH 5B FW Lvg Htr Line Drain	Closed	_____
HPH 6A Drip to Condenser	Closed	_____
HPH 6B Drip to Condenser	Closed	_____
HPH 5A Drip to Condenser	Closed	_____
HPH 5B Drip to Condenser	Closed	_____
HPH #5 Extr. Drain to Condenser (6 total)	Closed	_____

HPH #6 Extr. Drain to Condenser (6 total) Closed _____

Location	DAS Set-Up elevation	ID	Node #	Channels	
Turbine Front Standard	15 m	_____	_____	_____	_____
Panel 437	0 m	_____	_____	_____	_____
Panel 441	0m	_____	_____	_____	_____
BFP	0m	_____	_____	_____	_____

Notes / Special Conditions:

Table 4.7 Example of Feedwater Heater Test Control Room Data Sheet

H. P. FW Heater Performance Test Control Room Data Sheet

Unit: _____ Date: ____/____/____ Readings Taken By:_____

Time		
Unit Load	gauge	MW
Turbine Speed	digital	rpm
Main Steam Flow	gauge	kg/h
Feedwater Flow	gauge	kg/h
HP Steam Chest D Inlet Temp	digital	C
HP Steam Chest D Inlet Press	digital	kg/cm^2
IP Steam Chest D Inlet Temp	digital	C
Condenser Vacuum "A"	gauge	mm Hg
Condenser Vacuum "B"	gauge	mm Hg
HP Exhaust Steam Pressure	dial	kg/cm^2
HP Exhaust Steam Temperature	dial	C
HP Heater 6 "A" Level	dial	m
HP Heater 6 "B" Level	dial	m
HP Heater 5 "A" Level	dial	m
HP Heater 5 "B" Level	dial	m

Notes:

4.7.4 Pump Performance Test

4.7.4.1 Purpose

BFP Performance tests are run to compare the current performance of the BFP to its 'expected' performance and also to compare its performance against similar pumps. The parameters to be determined include the total head developed by the pump, the balancing drum leak off flow and the efficiency of the 'train' (i.e. motor, hydraulic coupling and the pump) over the normal operating range. This information can be used to determine if any corrective maintenance is required immediately or will be required in the future.

4.7.4.2 Frequency

Time based, typically every 12 to 24 months, but the other triggers can take precedence:

- Before and after pump or hydraulic coupling overhauls
- High Balancing Drum Leak Off (BDLO) flow.
- Abnormal difference in the motor current on two pumps at the same flow.
- Abnormal difference in the flow of two pumps at the same motor current.
- Change in the flow/motor current ratio
- High power usage recorded during a Unit Auxiliary Power Consumption Test

4.7.4.3 Unit Conditions

Unit should be in operation at normal full load in steady state condition. Ensure that the recirculation valve is closed. Feed water flow to be maintained constant at the required level for the duration of the test run. Variations in feedwater flow shall not exceed +/- 2% Tests at different feedwater flows through the pump are conducted during a period when the frequency deviations are minimal.

4.7.4.4 Data to be collected:

- Pressure at main pump suction and discharge
- Pressure at booster pump suction and discharge (if applicable)
- Temperature at Pump(s) Suction and Discharge
- Static pressure, temperature and dP at the pump flow measurement location
- Static pressure, temperature and dP at BDLO flow measurement location
- Pump speed
- Power supplied to the motor

4.7.4.5 Calculations

BFP Discharge Volume Flow = FW Flow Nozzle Coefficient* SQRT((Feedwater Nozzle dP)* (Outlet specific Volume))

BDLO Volume Flow = BDLO Nozzle Coefficient* SQRT((BDLO Nozzle dP)*(Outlet specific Volume))

Outlet Specific Volume : from steam tables

4.7.4.6 Analysis

Balancing Drum Leak Off (BDLO) flow rate should be within permissible limits. If the flow is high, it is indicative of excessive internal clearances. If the gap between expected total head curve and actual total head (corrected to design speed) is large at low flow rates, it's indicative of increased internal clearances whereas, if the gap between expected total head curve and actual total head (corrected to design speed) is large at high flow rates, it's an indication of impeller wear or damage.

4.7.4.7 Reference Documents

- ASME Performance Test Code 8.2 on Centrifugal Pumps
- For detailed test procedure refer Heat Rate Improvement Guidelines for Indian power Plants Vol. -II

4.7.4.8 Attachments

Table 4.8 Example of Boiler FW Pump Performance Test Pre-Test Check Sheet

Unit : _____ Pump : _____
Date: _____

Checks Made
By:_____

		Initials:
Recirculation Valve	Closed	
Pump control	Manual	
Auto Valve	Closed	
BDLO Safety	Closed	
Warming Supply	Closed	
Pump Condition	Steady	

Notes / Special Conditions:

Table 4.9 Example of Boiler FW Pump Performance Test Control Room Data Sheet

Station : ..... Unit: ..... Test Date: .....

Boiler Feed Pump A / B / C

Sl.No	Description	Test readings				Remarks
		Units	Run 1 Time	Run 2 Time	Run 3 Time	
1	Unit Load	MW				
2	Turbine Speed	rpm				
3	Main Steam Flow	T/hr				
4	Feedwater Flow	T/hr				
5	BFP Suct Temp	C				
6	BFP Disch Temp	C				
7	BFP Diff Temp	C				
8	BFP Booster Suct Press	kg/cm2				
9	BFP Suct Press	kg/cm2				
10	BFP Disch Press	kg/cm2				
11	BFP Flow	m3/hr				
12	BFP Speed	rpm				
13	BFP Motor Current	Amps				
14	BFP Speed Cntl Position	%				
15	BFP Speed Cntl Position	rpm				

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Table 4.10 Example of Boiler FW Pump Performance Test DAS Data Sheet

Station : ..... **Unit: .....** **Test Date: .....**

Sl.No	Description	Test readings				Remarks
		Units	Run 1 Time	Run 2 Time	Run 3 Time	
1	D/A Storage Tank Temp	C				
2	BFP A Suction Temp	C				
3	BFP A Discharge Temp	C				
4	BFP B Suction Temp	C				
5	BFP B Discharge Temp	C				
6	BFP C Suction Temp	C				
7	BFP C Discharge Temp	C				
8	D/A Storage Tank Steam Press	kg/cm^2				
9	BFP Booster Pump A Suction Press	kg/cm^2				
10	BFP Booster Pump B Suction Press	kg/cm^2				
11	BFP Booster Pump C Suction Press	kg/cm^2				
12	BFP A Suction Press	kg/cm^2				
13	BFP B Suction Press	kg/cm^2				
14	BFP C Suction Press	kg/cm^2				
15	BFP A Discharge Press	kg/cm^2				
16	BFP B Discharge Press	kg/cm^2				
17	BFP C Discharge Press	kg/cm^2				
18	HP Htr Common FW Inlet Press	kg/cm^2				
19	BFP A Speed	rpm				
20	BFP B Speed	rpm				
21	BFP C Speed	rpm				
22	BFP A Flow	t/hr				
23	BFP B Flow	t/hr				
24	BFP C Flow	t/hr				
25	BLR FW In total Flow	t/hr				
26	Generator Load	MW				

4.7.5 Turbine Cycle Heat Rate Test

4.7.5.1 Purpose

The purpose of the Turbine Cycle Heat Rate test is to determine overall turbine cycle performance (both heat rate and capability) as well as component thermal performance and to provide a check of accuracy of critical station instruments. It is used as guidance in scheduling maintenance outages on the basis of performance trends. It is also used to evaluate major modifications of the turbine or turbine cycle, and changes in operating procedure.

4.7.5.2 Frequency

Time based, typically every 12-24 months, but the other triggers can take precedence:

- Before and after turbine overhauls
- Increase or decrease in steam flow at constant turbine control valve position
- Degradation in cylinder efficiency based on line calculations
- Abnormal first stage pressure
- High extraction temperatures
- High Condenser heat load
- Increase in 'unaccountables" heat rate deviation

4.7.5.3 Unit Condition

Test runs must be conducted over the normal operating load range from Valves Wide Open (VWO) to minimum load. In units with sequential valve control, test runs must be conducted at the valve points. In units with full throttling control, the operating range must be divided into equal increments based on the nominal load at VWO (Valve Wide Open) and minimum.

Other operating conditions for each test run include the following

- Unit off load control and steady state condition
- Main Steam and Reheat Steam temperatures at current expected value
- Boiler outlet O_2 at current expected value
- All feedwater heaters in service, normal levels and vent settings, normal drain cascading
- Auxiliary steam supply connection to other units, makeup to the hotwell and boiler blow down are isolated prior to the test.

4.7.5.4 Data to be collected:

- Main Steam pressure and temperature

- Cold Reheat Steam pressure and temperature
- Hot Reheat Steam pressure and temperature
- IPT exhaust pressure and temperature
- H P heater extraction pressure and temperature at turbine and heater
- FW pressure and temperatures entering and leaving HP heaters
- HP heater drain temperatures
- Feed Water pressure, temperature and flow orifice dP
- Reheater Spray flow orifice dP
- Condenser Vacuum
- Generator Load

4.7.5.5 Calculation:

Turbine Cycle Heat Rate (TCHR) is defined as the ratio of the net heat supplied to the cycle divided by the power output from the generator.

$$TCHR = \frac{Q_{ms}(h_{ms} - h_{fw}) + Q_{hrh}(h_{hrh} - h_{crh})}{P_{gen}}$$

where Qms - Main Steam flow kg/hr
Qhrh - Hot Reheat Steam flow kg/hr
hms - enthalpy of steam entering HP Turbine kcal / kg
hfw - enthalpy of feedwater leaving feedwater system kcal / kg
hhrh - enthalpy of steam entering IP turbine kcal / kg
hcrh - enthalpy of steam leaving HP turbine kcal / kg
Pgen - Electrical power measured at the generator terminals kWh

Both the main steam and reheat flows , are used in determining the rate of heat input to the cycle. Since reheat flow is difficult to measure accurately, the ASME PTC 6 recommends that all HP turbine extraction flows be calculated on the basis of heat balance on the heaters, and the reheat flow be calculated by subtracting these extraction flow.

4.7.5.6 Analysis

See Sections 4.7.1.6, 4.7.2.6, 4.7.3.6, and 4.7.4.6 for information on the individual component analysis. If the turbine cycle heat rate deteriorates, but the problem cannot be traced to any individual component, the cause is usually poor cycle isolation or LP turbine efficiency. The cycle should be checked for any high energy drains leaking through, heater drain pumps bypassed, etc.

4.7.5.7 Reference Document

- ASME Performance Test Code – 6, Steam turbines
- ASME Performance Test Code – 19.1, Measurement Uncertainty
- ASME Performance Test Code – 19.2, Pressure Measurement

- ASME Performance Test Code – 19.3, Fundamentals of Temperature Measurement

4.7.6 Primary Air Flow Calibration

4.7.6.1 Purpose

Calibration of primary airflow through the pulverizers is the first step in balancing fuel and air to the burners. It is important to ensure that each primary air flow instrument is indicating the actual primary air flow through the mill before proceeding with other pulverizer tests by calibrating the station flow indication against the flow measured in a grid using a standard pitot or a calibrated 'S' type pitot.

The data collected in this test may also be used to draw clean air curves.

4.7.6.2 Frequency

Time based, typically every two years, but the other triggers can take precedence:

- Evidence of coal setting in horizontal pipe runs
- Poor coal fineness
- Excessive mill spillage of coal
- High pressure drop across pulverizer

4.7.6.3 Unit Conditions

Primary air flow calibrations are normally done with the unit in service, so the air temperature during the test is the same as during normal operation. (Test can be conducted with the unit off-line, but on-line is preferable.) Steady conditions on the pulverizer and PA fans needs to be maintained during test period. Calibration should be done at minimum three different operating flows.

4.7.6.4 Data to Be Collected.

The control room data on Pulverizer to be tested should be logged on a control room log sheet to ensure that test has been conducted under stable conditions, and to compare the station flow to the actual (measured via the pitot and grid) flow. Readings should be taken every 10 minutes. See Vol. II for an example.

The air flow should be determined by dividing the duct into a grid, and measuring the temperature, static pressure and pitot (usually "S" type) differential in the centroid of each area. The number of points to be used should be determined according to Section 4.7.13. It is not very often that in a power plant requisite straight length would be available in ducting to a Pulverizer to measure air flow with an accuracy of +\- 2.0% which is

achievable with standard pitot. Velocity data profile obtained should be checked for standard deviation . In case the standard deviation is with in +\- 10%, data is suitable for averaging and computing average velocity which can be used for computing duct flows.

The barometric pressure should be recorded during the test period.

Duct area to be taken from engineering drawings for computation of flow.

Details of Primary sensor and conditions observed during unit shut down to be included for reference.

4.7.6.5 Calculations

The total pressure of a flowing air stream in a duct is the sum of the static pressure exerted on the side walls and the velocity pressure of the moving air. The difference between total pressure and static pressure is called velocity pressure which is used to determine the linear rate of air flow. Measure velocity pressures using pitot at each of the traverse point as per the cross section drawing . Calculate velocity at each of the point and than average them for final velocity value. The following formulae is used to process the data,

$$\text{Density } (\delta) = \frac{460 + 70\ ^{\circ}F}{460 + \ ^{\circ}F} \times \frac{Bp + Sp}{13.6 \quad 29.92\text{" Hg C}} \times 0.075\ \text{Lbs./ft}^3$$

$$\text{Velocity} = \frac{1095(Vh)^{1/2}}{(\delta)^{1/2}} \times (K)1\backslash 2$$

Average velocity = V1+V2+V3.....Vn

Volumetric Flow (Q)= Average Velocity x Duct Cross-sect. Area

Mass Flow (W) = Q x 60 min/hour x Density

Bp = Barometric Pressure ("Hg); Sp = Static Pressure ("WC)

K = Calibration Coefficient of the pitot (standard "S" type pitot is 0.84)

4.7.6.6 Analysis

If the indicated primary air flow is different by more than 5%, from the test flow, it should be recalibrated.

4.7.7 Pulverizer Clean Air Test

4.7.7.1 Purpose

The second test in a mill performance program is to check to see if each coal pipe is receiving the same amount of primary air. Clean airflow tests are also used to calibrate primary air flow measurement device wherever straight lengths are not available in PA ducting to the pulverizer.

4.7.7.2 Frequency

Time based, typically every year before the unit outage, but the other triggers can take precedence:

- Slag formation around certain burners of the same pulverizer.
- CO imbalance at the furnace exit
- Temperature imbalance at the furnace exit
- Large O_2 spread at AH inlet to maintain equal steam temperatures.
- High content of combustibles in fly ash
- Poor flame appearance

4.7.7.3 Unit Conditions

Clean air tests are done over the normal range of airflow through the mill, without feeding any coal, but preferably at the normal mill outlet temperature (the unit should be in service to provide hot primary air, but the mill being tested will be out of service).

Prior to conducting tests, details of work undertaken during the unit over haul should be recorded. Change of burner nozzles, fuel pipe orifice inspection and their replacement etc. should form part of detailed database to be maintained. Clean airflow tests in a running unit can be conducted when unit is operating at part load during off peak hours when more stable operating regime is likely to prevail. Calibration and checking of instruments associated with the specific pulverizer should be carried out before carrying out clean air flows tests. Such tests are run at a minimum of three different operating points.

4.7.7.4 Data to Be Collected

Control room data on Pulverizer to be tested should be logged on a control room log sheet to ensure that test has been conducted under stable conditions. Readings to be taken every 10 minutes. Control room log sheet and data sheet for recording velocity traverse data are annexed to Pulverizer test procedure No: CENPEEP/EFF/TP/201 forming part of Volume No.II.

The velocity in each coal/air pipe should be measured with a pitot (usually an "L" type) by traversing a minimum of two cross sections (measuring the same locations where coal is sampled for a fineness test, see Section 4.7.13.3). The traverse point should be covered in both the inward and outward movement of pitot. The air temperature is to be measured using K type of thermocouple.

Data on static pressure in fuel pipe and barometer pressure to be taken during the test period.

Fuel pipe area to be taken from engineering drawings for computation of flow.

4.7.7.5 Calculations

- Velocity in each fuel line is calculated to ascertain clean air balance. Clean air balance is expressed as a deviation from the mean velocity of all the pulverizers individual coal transport pipes. The following formulae process dirty air traverse data.

$$\text{Density } (\delta) = \frac{460 + 70\ ^\circ F}{460 + \ ^\circ F} \times \frac{Bp + \frac{Sp}{13.6}}{29.92\text{" Hg C}} \times 0.075\ \text{Lbs./ft}^3$$

$$\text{Velocity} = \frac{1095(Vh)^{1/2}}{(\delta)^{1/2}} \ ; \quad (Vh)^{1/2} = \frac{(Vh_1)^{1/2} + (Vh_2)^{1/2} + (Vh_3)^{1/2} + \ldots}{\text{no. of traverse points}}$$

$$\%\ \text{Deviation} = \frac{\text{Avg. Velocity} - \text{Velocity}}{\text{Avg. Velocity}} \times 100\%$$

Volumetric Flow (Q)= Velocity x Pipe Cross-sect. Area

Mass Flow (W) = Q x 60 min/hour x Density

Bp = Barometric Pressure ("Hg); Sp = Static Pressure ("WC)

K = Calibration Coefficient of the pitot (standard "L" type pitot is 1)

4.7.7.6 Analysis

The air flow through each coal pipe is measured and the flow through each individual pipe is compared to the average flow through all pipes. If each pipe is not within +/- 2% of the average, the coal pipe orifices should be adjusted or replaced. Database obtained can be used to plot clean air curves for the pulverizer.

Velocity traverse data obtained in each fuel pipes where ever adequate straight lengths are available should be plotted on a curve to ensure that profile across the cross section of pipe is regular. To improve the accuracy of measurement , traverse cross section could be more than two cross sections.

4.7.7.7 Reference

Test Procedure for Routine Performance Testing of Pulverizer Volume II of this document.

4.7.8 Pulverizer Dirty Air / Coal Flow Test

4.7.8.1 Purpose

The dirty air/coal flow test is usually the third mill performance test (run after obtaining satisfactory results from both the primary air calibration and clean air tests. This test is used to:

- Detect imbalance, if any, in the air and coal flows between the discharge pipes of a Pulverizer.
- Collect a representative, iso-kinetic, sample of pulverized coal from different pipes for determination of fineness fractions
- Cross-check the readings of the station instrumentation e.g. primary airflow through the mill, mill outlet temperature, coal flow through the feeder etc.

4.7.8.2 Frequency

Time based, typically after every pulverizer overhaul and before the unit outage, but the other triggers can take precedence:

- Slag formation around certain burners of the same pulverizer.
- CO imbalance at the furnace exit
- Temperature imbalance at the furnace exit
- Large O_2 spread at AH inlet to maintain equal steam temperatures.
- High content of combustibles in fly ash
- Poor flame appearance

4.7.8.3 Unit Conditions

Prior to conducting tests, details of work undertaken during the equipment over haul should be recorded. Change of burner nozzles, fuel pipe orifice inspection and their replacement etc. should form part of detailed database to be maintained.

Tests on a specific pulverizer should be done when the mill parameters outlet temperature and primary airflow are steady and mill is running around "nominal loading".

The definition of "nominal loading" must not be changed from test to test. Before the first test on a pulverizer, the test engineer should select that are as close to the maximum capacity, but, shall always be achievable.

4.7.8.4 Data to Be Collected

- A member of the test crew logs the mill parameters every fifteen minutes during the test to ensure the stability of test conditions.
- Two or more ball valves are installed on straight section of each mill discharge pipe as per ASME code depending on the upstream and downstream lengths available from the nearest bend or disturbance. (See Section 4.7.13.3)
- Dustless connectors are used with the ball valves to ensure dust free working.
- Conditions of mill internals with reference to running hours, overhauls, pipe orifice replacement etc. are noted for each mill to be tested.
- Traverse points on the pitot tube are marked on an equal area grid in accordance with ASME Performance Test Code for traversing circular ducts or pipes.
- Two equal lengths of tubing are cut to required length and then taped or bound together. One tube is marked on both ends to identify as the 'high pressure' line or 'impact' line. The second tube is used as the 'low pressure' line or 'static' line.
- A 5" to 10" vertical inclined manometer is set up on a level and stable work area. The tubing is connected to the high and low side taps on the manometer.
- Dirty pitot tube is inserted in each sampling port and traversed across the pipe to measure differential pressure at 12 pre-determined points as per ASME code. Based on these values, the sampler dP is obtained using formulae for iso-kinetic sampling.
- Static pressure and temperature are measured using static pressure probe. The following data should be recorded for each test.
    - Coal pipe designation
    - Individual velocity head for each traverse point (For 2 ports - 24 points)
    - Temperature and static pressure for each coal transport pipe
- After determination of the dirty air velocities in their associated coal pipes, iso-kinetic coal samples are extracted. The coal sampling probe is marked identical to the dirty air test probe.
- The sampler differential pressure obtained in step 8 will result in a velocity through the sampler tip that is equal to the velocity of the coal and air mixture through the coal transport pipe (i.e. iso-kinetic sampling).
- The power consumed by the mill during the test can be computed by connecting energy meters

4.7.8.5 Calculations

- Velocity in each fuel line is calculated to ascertain dirty air balance. Dirty air balance is expressed as a deviation from the mean velocity of all the pulverizers individual coal transport pipes. The following formulae process dirty air traverse data.

$$Bp + \underline{Sp}$$

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$$\text{Density } (\delta) = \frac{460 + 70\ ^{\circ}F}{460 + \ ^{\circ}F} \times \frac{13.6}{29.92''\ Hg\ C} \times 0.075\ Lbs./ft^3$$

$$\text{Velocity} = \frac{1095(Vh)^{1/2}}{(\delta)^{1/2}} \times K\ ; \quad (Vh)^{1/2} = \frac{(Vh_1)^{1/2} + (Vh_2)^{1/2} + (Vh_3)^{1/2} + \ldots}{\text{no. of traverse points}}$$

$$\%\ \text{Deviation} = \frac{\text{Avg. Velocity} - \text{Velocity}}{\text{Avg. Velocity}} \times 100\%$$

Volumetric Flow (Q)= Velocity x Pipe Cross-sect. Area

Mass Flow (W) = Q x 60 min/hour x Density

Bp = Barometric Pressure ("Hg); Sp = Static Pressure ("WC)
K – Calibration Coefficient of the pitot

- Iso-kinetic fuel sample is collected, weighed & analyzed for fineness data using four standard mesh screens.
- After completing the testing of all the coal transport pipes of a mill, following formulae are used to calculate the coal flow through each coal pipe.
- The performance data obtained can be put on a feedback format to circulate for necessary corrective action.

$$\text{Coal Flow} = \frac{\text{Sample Weight (gms)}}{\text{gms / kg}} \times \frac{60\ \text{min / hr}}{\text{Tc min}} \times \frac{\text{Pipe area} - m^2}{\text{Sample tip Area } m^2}$$

Tc- Time of sample collection per pipe in minutes

$$\text{Air to fuel ratio} = \frac{\text{Air Flow in kg per hour}}{\text{Coal Flow in kg per hour}}$$

- Gross Coal sample is collected from feeder inlet chute for determination of coal characteristics namely size of raw coal, moisture, HGI for applying corrections to the measured test data.

Table 4.11 Example of Dirty Pitot Survey- Summary Data (Mill X)

	UCB	Measured
Air Flow (T/hr)	40	60
Mill Outlet Temperature C	80	78
Coal Flow T/hr	--	23

Description	Corner				Mean	Desired
	1	2	3	4		
Velocity (m/s)	31.4	28.4	30.7	27.2	29.4	>18 m/s
Air Flow (T/hr)	15.9	14.6	15.6	13.9	15.0	-
Dev. From Mean%	6.0	-27	4.0	-7.3	--	<+/-5%
Mill Outlet Temp. (°C)	79.0	76.0	79.0	77.0	77.8	~85
Coal Flow (T/hr)	46.0	6.2	7.0	5.3	5.8	<+/-10%
A/F Ratio	22.0	22	23.0	20	22.0	1.8 to 2.5
%Retention (50mesh)	1.2	3.5	7.0	24	3.5	<1%
%Pass (200 mesh)	80.0	65.9	48.2	72.8	66.7	~70%

- Measured Mill outlet temperature matches with the control room value but the Mill operating PA flow differs by 20 t/hr.
- High +50 mesh retention could be ascribed to the high PA flow through the mill.

4.7.8.6 Analysis

Acceptable Values are from a dirty air test are:

- Pipe to Pipe Dirty Air Flow Variation +/- 5%

- Minimum Dirty Air Velocity 17-18 M/s
- A/F Ratio 1.8+

  (An A/F Ratio less than 2.5 results in good fineness)
- A/F Mixture minimum temperature 60 ° C
- Pipe to Pipe Fuel flow variation +/- 10.0%
- % Retention on 50 mesh < 1%
- % Through from –200mesh ~ 70 %

4.7.8.7 References

4.7.8.8 Attachments

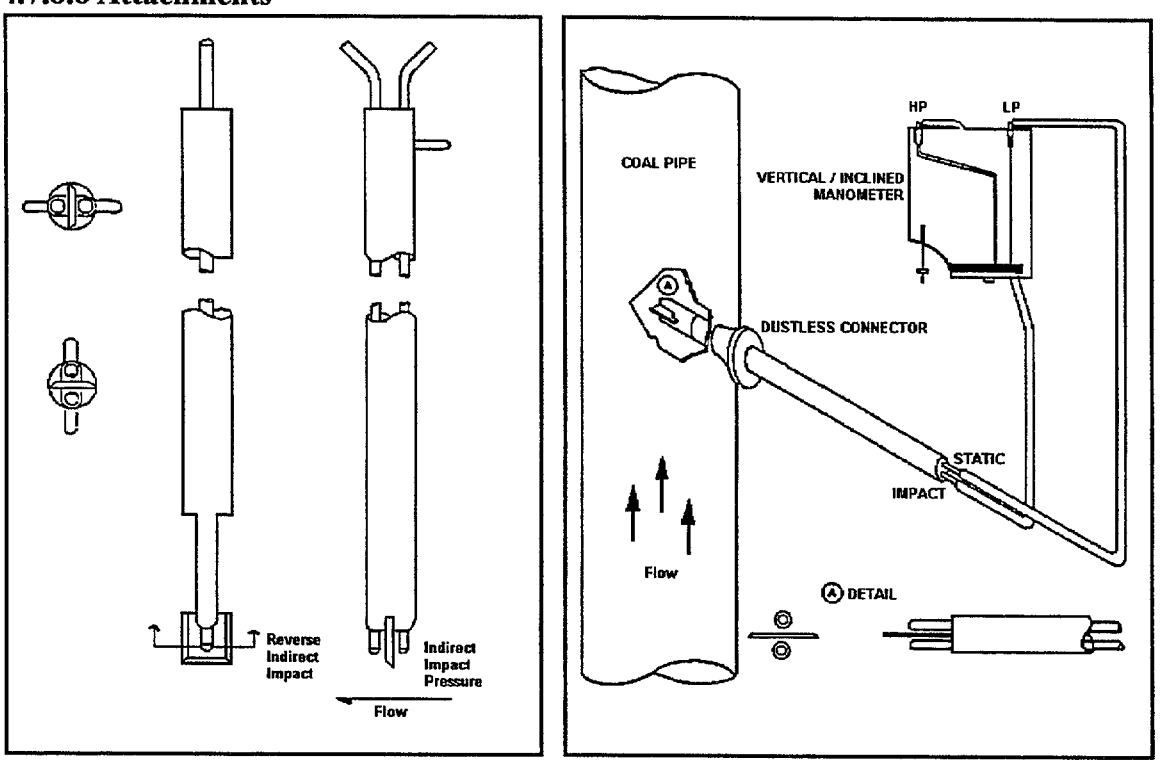

Figure 4.2 Dirty Air Probe

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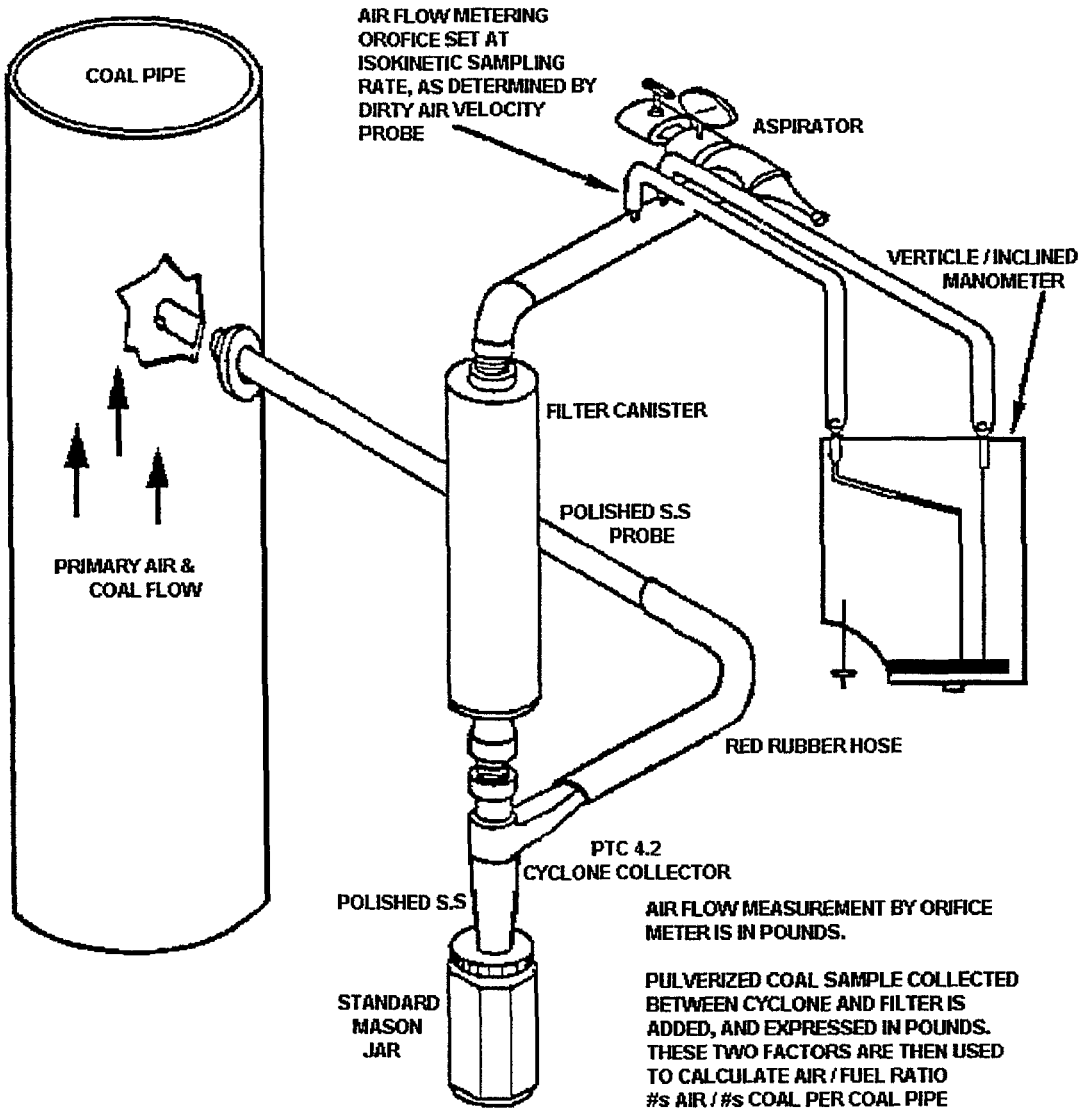

Figure 4.3 Pulverized Coal Sampling Kit

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4.7.9 Pulverizer Fineness Test

4.7.9.1 Purpose

To determine the "fineness of the coal produced by a pulverizer. The "fineness" is represented by two values, the percentage of coal that pass through a 200 mesh screen, and the percentage of coal that is retained on a 50 mesh screen.

4.7.9.2 Frequency

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Time based depending on the type of mill and coal conditions, typically every one-three months (depending on the mill type, and coal), but the other triggers can take precedence:

- High carbon in fly ash
- Slag formation around certain burners
- High CO at the furnace exit
- CO & temperature imbalance at the furnace exit
- Large O_2 spread at AH inlet to main equal steam temperatures.

4.7.9.3 Unit Conditions

Testing should be done when mill parameters like outlet temperature, primary airflow, are steady and mill is running around "nominal loading".

The definition of "nominal loading" must not be changed from test to test. The test engineer should pick conditions that are as close to the maximum capacity, but will always be achievable.

4.7.9.4 Data to Be Collected

A sample of pulverized coal is to be collected from each coal pipe of the mill using an isokinetic fuel sampler. While it is not recommended, some plants have gotten "repeatable" results using only a cyclone sampler.

4.7.9.5 Calculations

The sample from each coal pipe is screened separately and fineness determined using at least 50, 100 and 200 mesh standard screens (and preferably a fourth screen either 70 mesh or 140 mesh).

- If the sum of the weights from the 4 sieves and the bottom pan ($W_{50} + W_{70} + W_{100} + W_{200} + W_{BP}$) is not within 0.25 grams of the original weight (50 grams) too much sample was lost in handling, and the procedure must be repeated.
- Determine the % passing each screen, starting with the coarsest sieve:

  % passing No. 50 sieve = { 1.0 -
  (Weight remaining on No. 50 sieve / Total Weight)}*100

  % passing No. 70 sieve = % passing No. 50 sieve -
  (Weight remaining on No. 70 sieve / Total Weight)*100

  % passing No. 100 sieve = % passing No. 70 sieve -
  (Weight remaining on No. 100 sieve / Total Weight)*100

  % passing No. 200 sieve = % passing No. 100 sieve -
  (Weight remaining on No. 200 sieve / Total Weight)*100

- If the original sample is exactly 50 grams, the above equations can be written as:

  The % passing the 50 Mesh = $\%_{50} = 100 - (W_{50} * 2)$
  The % passing the 70 Mesh = $\%_{70} = \%_{50} - (W_{70} * 2)$
  The % passing the 100 Mesh = $\%_{100} = \%_{70} - (W_{100} * 2)$
  The % passing the 200 Mesh = $\%_{200} = \%_{100} - (W_{200} * 2)$

The results should be plotted on a Rosin & Rammler graph (see Figure 4.4). This is a log - log scale graph with the logarithm of the opening (which is related to the sieve designation) on the x-axis, and the logarithm of the logarithm of 100 divided by the percent remaining on the sieve on the y-axis. If the collection and weighting are not biased, the "R&R" plot of the 4 points should fall in a straight line. If the line is "dog-legged" the sample collection process should be repeated because something is biased (either too many or too few "fines" were collected (compared to the percentage in the coal pipe) normally due to the aspirator being set at too high or too low respectively).

4.7.9.6 Analysis

The fly ash and bottom ash carbon content (LOI) depends to a large extent on the fineness of the pulverized coal. If the ground coal is too coarse, LOI may increase, and the unit will have a greater tendency to slag. Improved fineness provides other operational benefits such as reduced boiler tube erosion due to its impact on ash particle momentum. On the other hand if the pulverized coal is too fine, mill power consumption increases, thereby affecting heat rate adversely. The additional mill maintenance and auxiliary power costs are unit specific depending on the number, type and capacity of the mills.

Acceptable Values are from a dirty air test are:

% Retention on 50 mesh	< 1%
% Through from –200mesh	~ 70 %

PF Fineness in the coal pipeline may be different at different corners at the same elevations.

Poor 50 mesh fineness is frequently an indication of a problem inside the mill, allowing coal to bypass the classifier (hole in the classifier skirt, etc.). Poor 200 mesh frequently requires an adjustment to the classifier.

4.7.9.7 Reference Documents

ASME PTC 4.2 Coal Pulverizers

4.7.9.8 Attachments

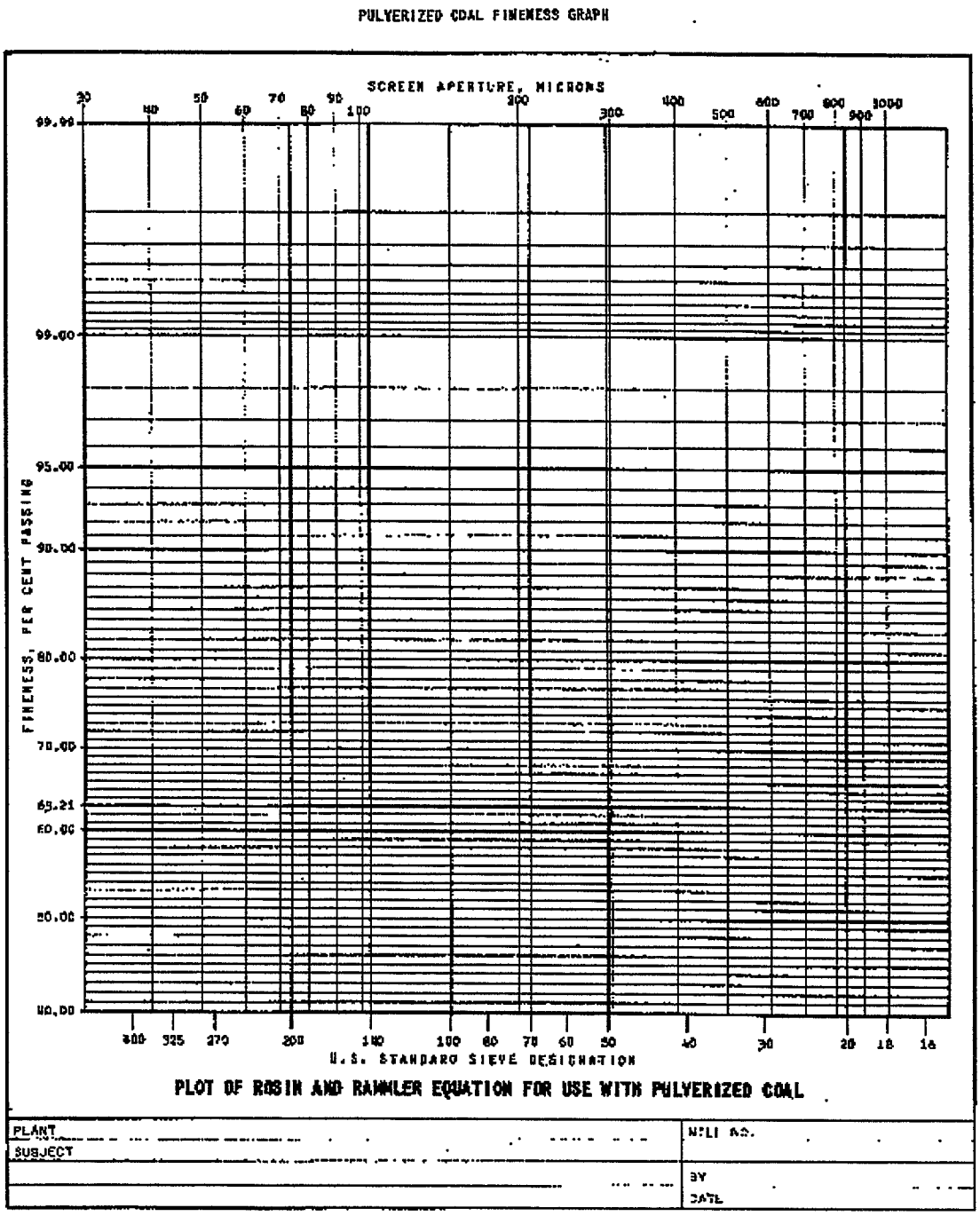


Figure 4.4 Rosin and Rammler Graph

4.7.10 Air Heater Performance Test

4.7.10.1 Purpose

Monitoring of air heater performance involves measurement of several critical parameter:

- To determine air heater leakage rate (the amount of air that leaks to the gas side) which affects station service (fan power) and affects the "indicated" exit gas temperature used in heat rate deviation calculations
- To determine air heater x-ratio (which is an indication of boiler air infiltration, and a driver of AH efficiency)
- To determine air heater gas side efficiency (how well the AH is transferring heat from the gas side to the air side)
- To validate the station instrument readings e.g. air and gas temperatures, oxygen, air and gas side pressure drops etc.

4.7.10.2 Frequency

- AH tests are done prior to unit outages to determine whether corrective action is required to maintain optimum air heater performance and provide guidance in determining materials, tools equipment, workers, cost estimates and scheduling.
- AH tests are done following outages to assess improvement in air heater performance on account of work done on the air heater.
- During normal operation air heater tests are done at least once in six months to identify abnormal changes in air heater leakage and efficiency or provide information to allow accounting for the contribution of air heater performance degradation to unit heat rate and capacity.

4.7.10.3 Unit Conditions

The unit conditions (including a pre-defined airflow, mill combination, primary and secondary flows etc.) should be held stable for at least an hour prior to the test. The bypass dampers and the economizer hopper gates are closed and no sootblowing is done during the test. Steam supply to air pre heating coils, if any, should be isolated.

4.7.10.4 Data to be collected

Data to be collected using specifically installed off line test instruments:

- AH inlet and outlet flue gas temperatures
- Primary air inlet and outlet temperatures
- Secondary air inlet and outlet temperatures
- Air and gas side pressure differentials
- AH inlet and outlet flue gas composition - O_2 and CO_2

Flue Gas sampling train at AH inlet and outlet locations consist of the following (see Figure 4.5 - 4.8, Section 4.7.10.8).

1) Multi-point sampling probes made of carbon steel pipes to form a representative-sampling grid at the specific cross sections.

2) High temperature resistant tubing to bring the gas sample from the probes to the bubble jar
3) A single point bubble jar, partially filled with water & sealed at the top.
4) A copper condenser coil immersed in an ice bucket.
5) A transparent desiccant column containing easily replaceable, CaSO4, silica gel or a similar desiccant
6) A transportable O_2 / CO_2 / CO analyzer as per test requirement
7) Vacuum Pump

Flue Gas samples are drawn by a vacuum pump from the test grid probes through high temperature resistant tubing into a bubble jar as shown above.

The purpose of the bubble jar is to clean (remove particulate matter), cool the sample (and remove some moisture), and in a multi-point bubble jar to mix the sample from multiple locations. It is easy to make one using a conical flask. The flask is sealed at the top with a rubber cork and one glass / copper tube is inserted in it. The flue gas sample is sucked in from the top (due to the vacuum created by the vacuum pump) and released in the water, while outlet is taken from the side of the flask using a flexible rubber tube.

The outlet tube from the bubble jar carries the cool and relatively clean gas sample to a copper coil condenser kept in an ice bath (a bucket filled with ice and water). The condenser condenses the most of the water vapor out of the gas sample. It has a coil for the gas to pass through while any condensed water gets collected at the bottom. The collected water can be removed after completion of each test by disconnecting the condenser tubing.

The sample from the condenser is routed to a desiccant column to remove any remaining water vapor that got through the condenser. The desiccant could be CaSO4 or Silica gel or similar regenerative type moisture absorbing anhydrous chemical. The purpose is to ensure that the gas sample is completely dry by the time it leaves the condenser. The column containing the desiccant is glass or plastic, transparent enough to observe the change in color of the desiccant on moisture absorption. Generally the desiccant is sent to labs for regeneration when a little over half the column has been exhausted. (the desiccant column should also have a filter in it to kept the desiccant clean, and to remove any particulate matter that went through the bubble jar.)

The outlet of desiccant jar is connected to a vacuum pump, which pulls the sample out of the duct, and through the preceding stream. The pump discharge is routed to the gas analyzers through flexible tubing.

Control room data is also noted down every 15 minutes in a specific log sheet developed for this test run to ensure stability of parameters, which have a bearing on air heater performance. Air heater tests check list and control room log sheets are appended to Test procedure on Air heaters forming part of Volume No.2 of this guideline.

The schematic in Section 4.7.10.8 depicts a typical flue gas side & an air side probe. The gas side probe has three (sometimes more) tubes of different lengths for sampling flue gas from the duct and one tube for carrying three (again sometimes more) thermocouple wires. The air side probe, used for measurement of primary and secondary air temperature measurements has a single tube for carrying the thermocouple wires.

The tube for carrying thermocouple wires for both gas & air side probes has two small holes (preferably elliptical for easy insertion of thermocouple wire) drilled in it, close to where the sampling tubes end. The intent is to measure flue gas temperature at the flue gas sampling point only. For the same, three thermocouple wires are inserted in the tube. The thermocouple junction ends protruding out of the holes are taped to the tubes to prevent wire insulation erosion due to fly ash. Also, the junction ends are grounded by tying them with a steel wire to the tube. This is to prevent interference by any static charges caused due to moisture in flue gas.

For flue gas sampling, high temperature resistant plastic tubes are connected to the sampling tube ends with ferrules & fittings. Good quality ferrules and fittings should only be used as the entire gas sampling system is under vacuum and the slightest air ingress can influence the results. *Under stable conditions, when very marginal carbon monoxide is present in the flue gas , sum of CO2% and O2% at any location in the flue system should be same.*

4.7.10.4.1 Temperature/Gas Sampling Probe and test port

Test ports at APH inlet and outlet in flue gas duct are made as per ASME PTC 19.10 (see also Section 4.7.13) to install a multi-point flue gas monitoring system for performance tests. The grid so installed provides a means of determining oxygen and carbon dioxide/monoxide profile for strategically placing the single point monitoring probes at a point that would provide the most representative sample of the weighted average of all the points. The following points need be noted while selecting the traverse locations.

a) The gas inlet traverse plane should be located as close as possible to the air heater.
b) The air inlet traverse plane should be located after any air heating coils and as close as possible to the air heater inlet. Since the entering air temperature is typically uniform, it may be possible to reduce the numbers of traverse points. A trial traverse can be made to provide data to support this decision.
c) The gas outlet traverse plane should be located as far downstream from the air heater as possible, to allow mixing of the flue gas to improve the accuracy of temperature and flue gas composition data. In no case outlet cross-section should be located downstream of common duct before ESP inlet.
d) The air outlet traverse plane should be located as far downstream from the air heater as possible to allow mixing of the flow to reduce the temperature stratification.
e) Flue gas temperature profile at air heater exit taken with 18 point grid is illustrated on the sketch given below to highlight the point that temperature profile after Tri- sector

air heater can have high variation on account of air heater leakage as well as different heat transfer conditions in primary and secondary sectors of a regenerative air heater.

4.7.10.5 Calculations

Air Heater Leakage is the weight of air passing from air side to gas side (assumed to leak directly from air side to gas side)

$$\text{Air Heater Leakage (\%)} = \frac{(\%\ CO2in - \%\ CO2out)}{\%\ CO2out} * 0.9 * 100$$

$$\text{If } O_2 \text{ analyzers are used AH Leakage (\%)} = \frac{(\%\ O2out - \%\ O2in)}{(20.95 - \%\ O2\ out)} * 100$$

where
% CO2 in - % of CO2 in flue gas entering the air heater
% CO2out - % of CO2 in flue gas exiting the air heater
% O2in - % of O2 in flue gas entering the air heater
% O2out - % of O2 in flue gas exiting the air heater

Note: The preferred method is to use CO_2 data.

Air Heater Gas side efficiency is defined as the ratio of gas temperature drop across the air heater corrected for no leakage to the temperature head.

$$\text{AH gas side efficiency (\%)} = \frac{\text{Gas Drop}}{\text{Temperature Head}} * 100$$

$$= \frac{Tg1 - Tg2}{Tg1 - Ta1} * 100$$

where
Tg1 - gas temperature entering air heater
Ta1 - air temperature entering air heater
Tg2 - gas temperature leaving air heater corrected to no leakage condition

Air Heater X- ratio is defined as the ratio of heat capacity of the air passing through the air heater to the heat capacity of the gas passing through the air heater.

$$\text{AH X - ratio} = \frac{Tg1 - Tg2}{Ta2 - Ta1} = \frac{Wa\ Cpa}{Wg\ Cpg}$$

where
Tg1, Tg2, Ta1 are as defined above
Ta2 - Air temperature leaving air heater

Cp - Mean specific heat of air or gas
W - Mass flow rate of air or gas

4.7.10.6 Analysis

Bi-sector regenerative air heaters with proper seal clearances should have leakage rates between 6% to 8% while the leakage rates for tri-sector air heaters should be between 10 - 12%. The leakage levels depend on the proportion of primary air to secondary air, differential pressure between the air and gas side of the air heater, the degree of air heater pluggage and the condition of the seals. Introduction of additional pulverizer and boiler operation at higher PA header pressure also results in high air heater leakage leading to:

a) Higher loading of fans, resulting in increased maintenance and auxiliary power requirements.
b) Increased flue gas volume, causing
    - Increased sensible heat loss
    - Increased particulate emissions from ESP
    - Unit operation at reduced loads due to insufficient margins in fans to cope up with additional flows

An increase in leakage usually indicate a problem with the seals (axial, radial, or circumferential) and a need to check and adjust or replace as necessary. Another problem (especially after an AH fire) is "warping" of the air heater.

If the x-ratio decreases, it is an indication of increased air flow bypassing the air heater. This is usually due to boiler (furnace or convection pass) air infiltration. Increased mill tempering air flow could also be the cause.

A change in the air heater gas side efficiency is most often a result of the following three events, listed in order of their relative effect on gas side efficiency:

- a change in the air or gas flow rate through the air heater (x-ratio)
- a change in the cleanliness of the air heater
- long-term corrosion of the air heater heat transfer surface

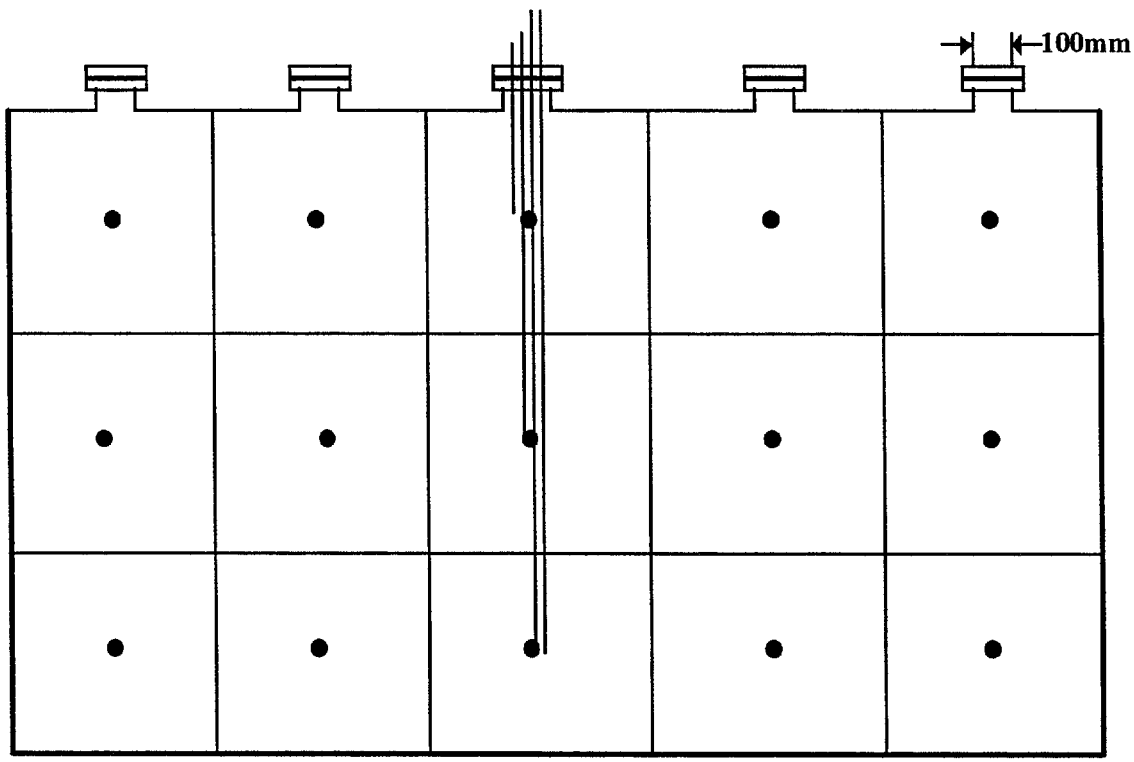
Also, air and gas flow pressure drops trended over time give an idea of degree of air heater pluggage.

4.7.10.7 References

- ASME PTC 4.3 Air Heaters
- ASME PTC 19.10 Flue and Exhaust Gas Analyses

4.7.10.8 Attachments

Sampling Ports in Flue Gas Ducts (Typical)



● **Sampling Point for Flue Gas Temperature & Composition**

Figure 4.5 Sampling Ports in Flue Gas Ducts (Typical)

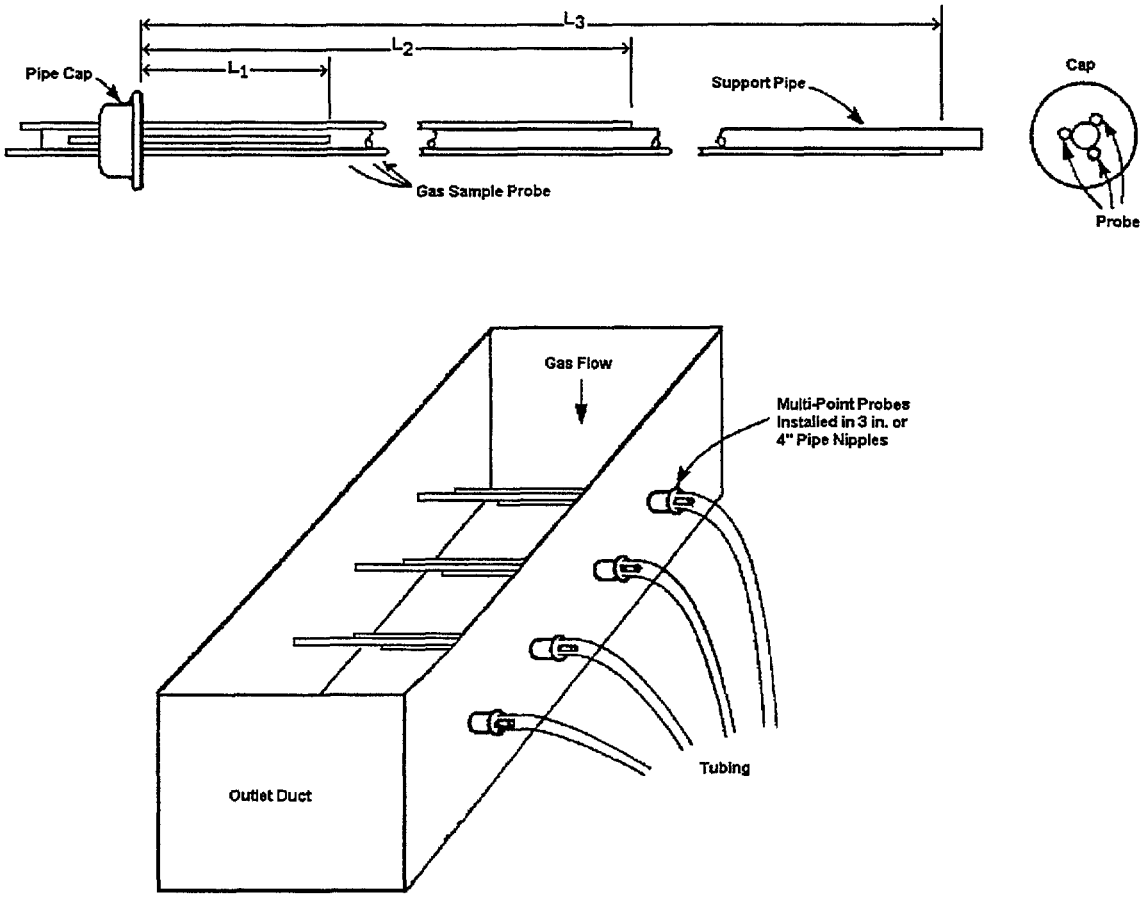


Figure 4.6 Example of Multi-Point Probes and Probe Configuration

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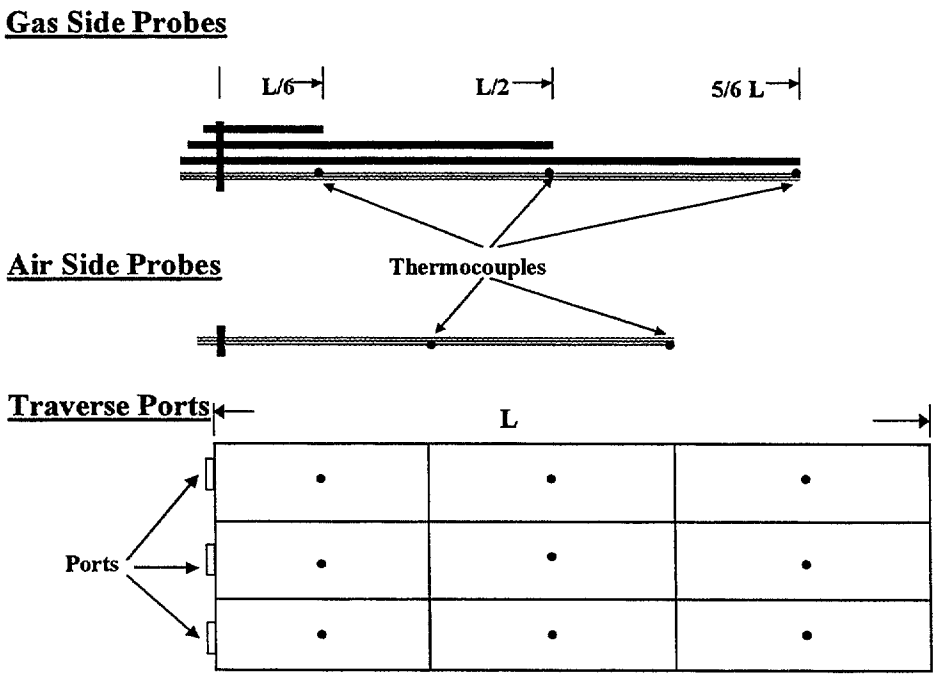

Figure 4.7 Schematics of Flue Gas Side and Air Side Probes and Traverse Ports

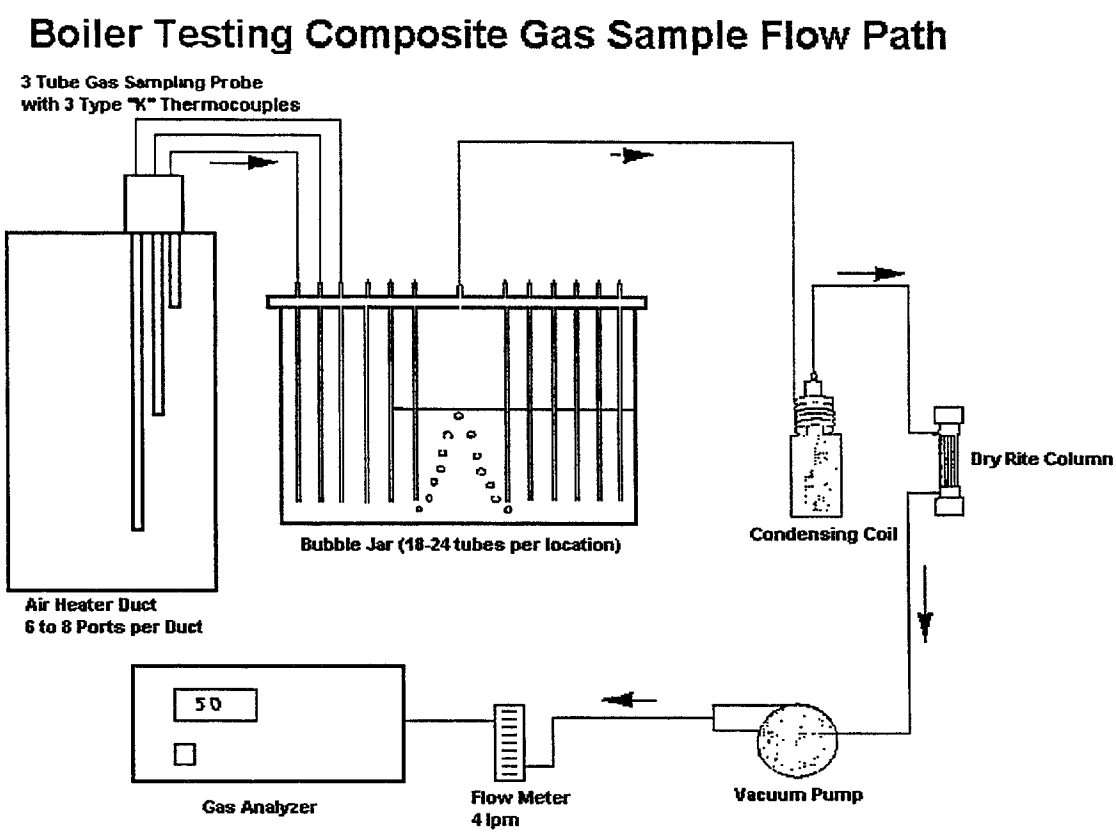

Figure 4.8 Schematics of a Typical Gas Sampling Station

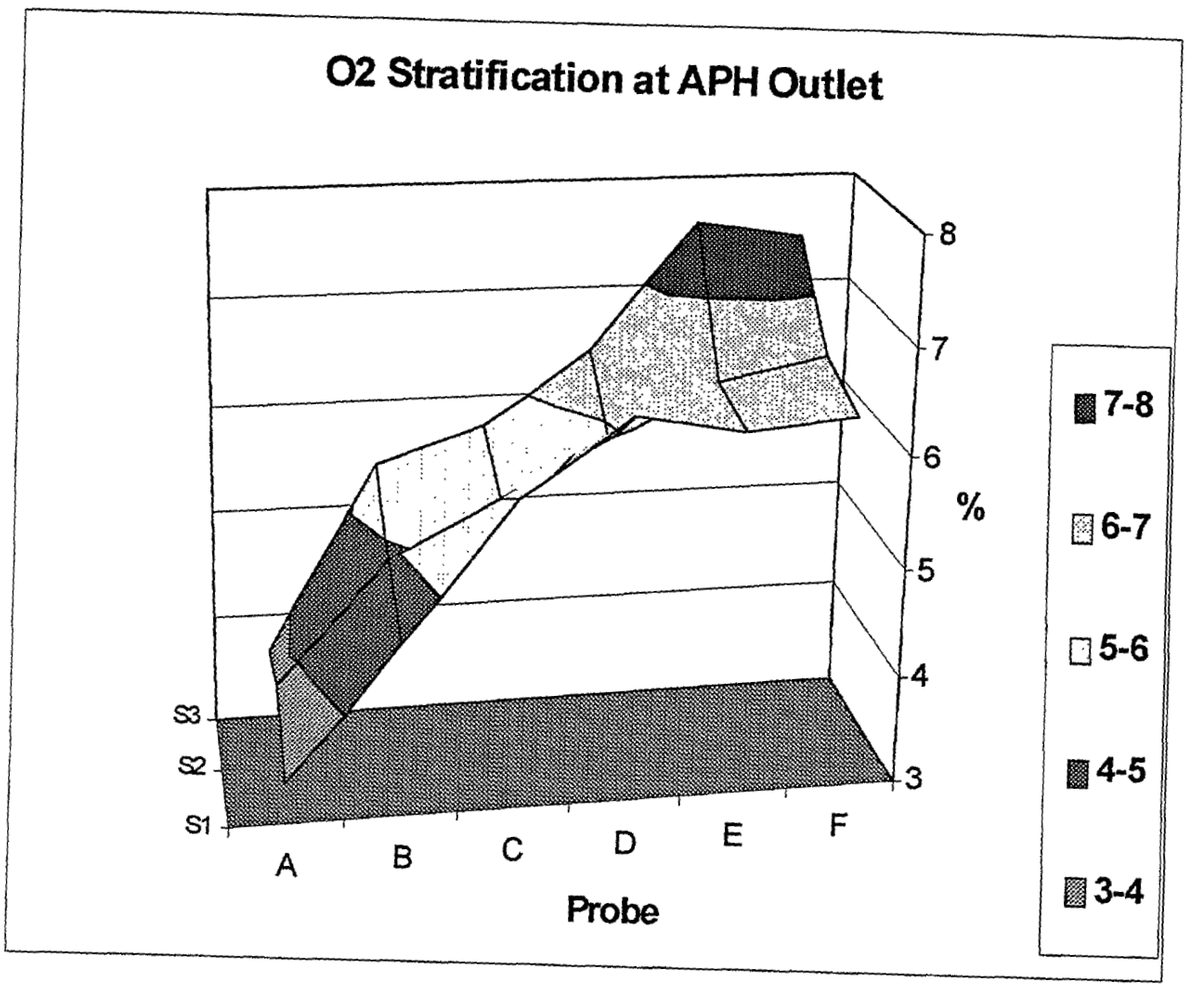

Figure 4.9 Example of O_2 Stratification in AH Outlet Duct

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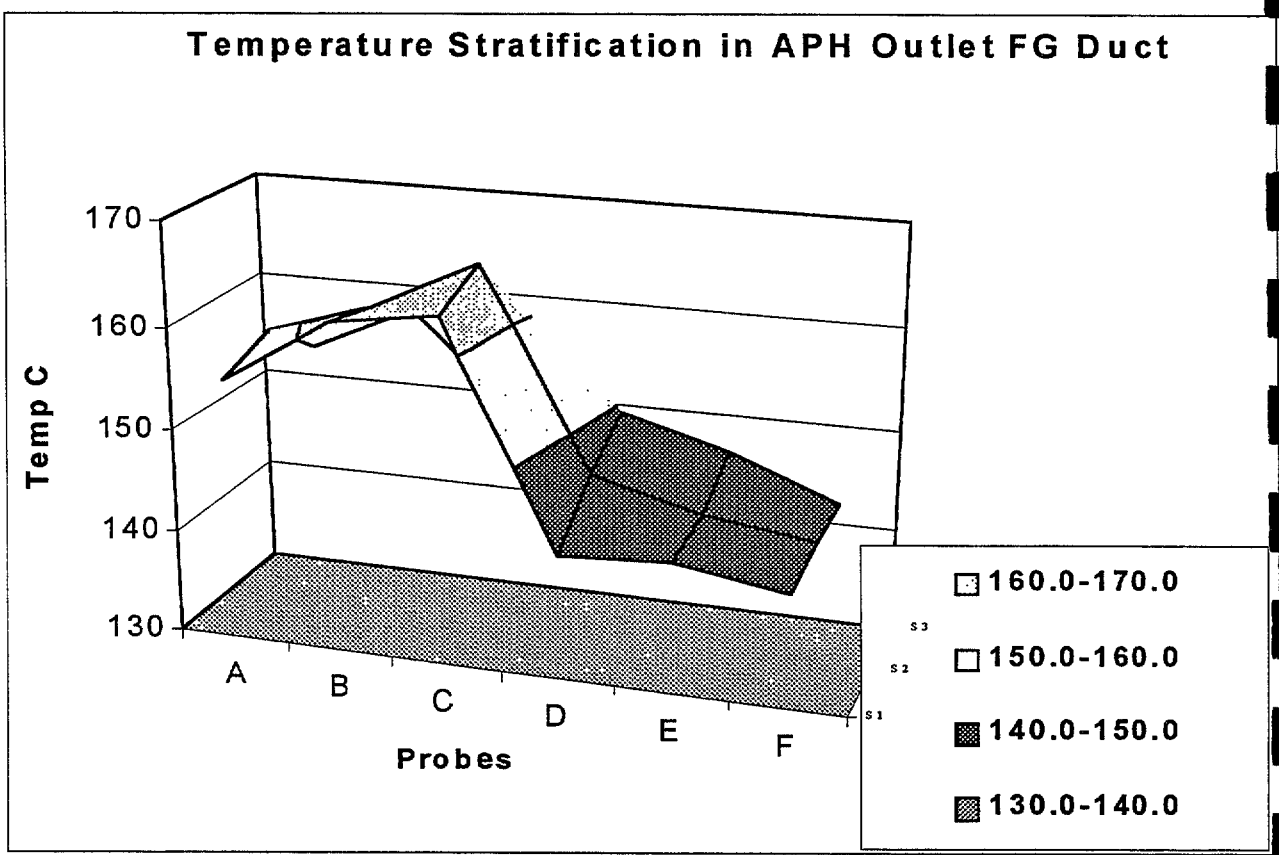

Figure 4.10 Example of Temperature Stratification in Tri-sector AH Outlet Duct

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Table 4.12 Typical Test Summary Sheet Obtained for a Tri-Sector Air Heater

200 MW Unit X				
Parameter		**Baseline**	**Increased O2**	**Four Mill Opn.**
Air Preheater Leakage	%	18.70	19.06	15.47
Air Preheater X-Ratio		0.68	0.73	0.79
APH Gas Side Efficiency	%	59.6	62.8	65.5
Inlet O2 - Measured	%	2.9	3.4	2.6
Inlet CO2	%	16.3	15.9	16.0
Outlet O2	%	5.7	6.3	5.4
Outlet CO2	%	13.5	13.1	13.7
Avg. Temp Gas In - Measured	C	338.2	344.2	339.4
Avg. Temp Gas In - Control Room	C	315.0	322.0	----
Avg. Temp Gas Out - Measured	C	140.0	132.7	123.4
Avg. Temp Gas Out - Control Room	C	122	117	----
Avg. Temp Pri Air In	C	42.0	38.5	39.0
Avg. Temp Pri Air Out	C	282	286	280
Avg. Temp Sec Air In	C	34.5	34.5	28.0
Avg. Temp Sec Air Out	C	310.2	310.8	287.6
Pri Air dp - Measured	mmWC	87	93	80
Sec Air dp - Measured	mmWC	68	80	85
Flue Gas dp - Measured	mmWC	90	95	85

200 MW Unit X				
Parameter		**Baseline**	**Increased O2**	**Four Mill Opn.**
Air Preheater Leakage	%	18.70	19.06	15.47
Air Preheater X-Ratio		0.68	0.73	0.79
APH Gas Side Efficiency	%	59.6	62.8	65.5
Inlet O2 - Measured	%	2.9	3.4	2.6
Inlet CO2	%	16.3	15.9	16.0
Outlet O2	%	5.7	6.3	5.4
Outlet CO2	%	13.5	13.1	13.7
Avg. Temp Gas In - Measured	C	338.2	344.2	339.4
Avg. Temp Gas In - Control Room	C	315.0	322.0	---
Avg. Temp Gas Out - Measured	C	140.0	132.7	123.4
Avg. Temp Gas Out - Control Room	C	122	117	---
Avg. Temp Pri Air In	C	42.0	38.5	39.0
Avg. Temp Pri Air Out	C	282	286	280
Avg. Temp Sec Air In	C	34.5	34.5	28.0
Avg. Temp Sec Air Out	C	310.2	310.8	287.6
Pri Air dp - Measured	mmWC	87	93	80
Sec Air dp - Measured	mmWC	68	80	85
Flue Gas dp - Measured	mmWC	90	95	85

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4.7.10.9 Flue gas composition using Multi-Point Bubble Jar vs. single point Bubble Jar

For extraction of flue gas sample from the duct, a grid of multi-point probes made of carbon steel pipes is made at the specific cross-sections at air heater inlet and outlet locations. Flue gas is pulled out by a vacuum pump from these test grid probes through high temperature resistant tubing into a bubble jar. The bubble jar is partly filled with water, which serves to cool and clean the flue gases sample. When a multi-point bubble jar is used, all the sampling points of the probes can be sampled simultaneously. Another option is to use a single point bubble jar wherein each point in the flue gas inlet and outlet ducts is sampled separately one by one. Both have some distinct advantages.

When a multi-point bubble jar or header is used, the sampling tubes from all the probes in a duct are connected to a single transparent bubble jar partly filled with water and a single representative sample is drawn and checked for its analysis. It is possible to regulate the flow of flue gas sample by throttling the needle valve on each sampling line by looking at the gas bubbling rate in the water, however, it is very difficult to maintain steady sampling rates for very large number of sampling points. Analyzer outputs can be directly fed in to a computer using a data scanning and logging system. Major advantage is the reduction in the time required for the test. With a multi-point bubble jar, an hour test run under stable conditions is adequate to establish an over all leakage.

Point by point sampling which enables developing a complete picture of flue gas composition profile at a cross section. Sampling tubes of all the probes mounted on the duct are connected through high temperature tubing to the bubble jar and the succeeding sampling train one by one. The analyzer readings are recorded manually. The degree of fluctuation observed in each constituent can be recorded to confirm stable conditions. Point to point sampling mode should be adopted for air heater testing. The range of variation of oxygen at air heater outlet is illustrated on the gas profile curve in Section 4.7.10.8. The temperature readings can be taken manually or collected using a data scanning system, wherein all the probe thermocouples are connected by extension cables to datascan boxes for analogue to digital conversion and onward transmission to a computer.

Sometimes a multi-point header connected to all the sampling points of one particular probe only with a single point bubble jar, is used. Once the readings stabilize, the header is disconnected and connected to the next probe. This saves time and can be adopted for regular testing after establishing a data base with point by point sampling.

Regardless of which system is used, it is critical to continually check the system for leaks (ambient air into the sampling system.).

4.7.11 Boiler Performance Optimization Test

4.7.11.1 Purpose

- To determine a set of operator controllable settings for optimum boiler performance which results in the best achievable heat rate.
- To validate station instruments' readings e.g. oxygen, fly ash combustible, air temperatures, exit gas temperatures etc.
- To generate a repeatable database for problem diagnosis and future reference
- To provide inputs to the maintenance engineer for scheduling maintenance activities and evaluating them after completion.

4.7.11.2 Frequency

Tests are conducted prior to or following major outages or whenever major maintenance work or modifications are carried out in the boiler to ensure that performance levels have been established during outage or to develop a work out a new expected level of performance.

4.7.11.3 Unit Conditions

After setting up the off line instruments and test equipment for assessing Boiler Furnace performance, a series of tests are run to understand the combustion regime operative. Any imbalance in combustion regime or temperature profile is corrected by adjusting air distribution, fine tuning mills parameters etc. After correcting these deviations which may be observed in any boiler, base line tests are run to establish current level of performance. A minimum of two test runs should be run to establish a repeatable database. Then a series of tests are conducted by changing only one variable in each test run with respect to base line test regime. Typically, the variables could be, boiler excess air levels / oxygen, mill primary air flows, mill outlet temperatures, auxiliary and fuel air damper positions, burner tilt position, windbox to furnace differential pressure etc. The effect of varying the operating conditions is analyzed with respect to base line performance and an operating regime with minimum losses is evolved. This regime is then checked for its efficiency by actually running the unit on the recommended regime.

Flue gas composition and temperature at furnace gooseneck are measured using a water-cooled probe called High Velocity Thermocouple. While it isn't required for calculating boiler efficiency, it provides excellent information about combustion completion and helps to diagnose problems in the furnace related to heat transfer and combustion imbalance in the furnace. (See Section 4.7.12).

Test runs are conducted at a repeatable level of unit operation so that these tests can be conducted at the same conditions later on and results compared. The following criterion apply to all the tests conducted,

- Furnace wall blowing and the air heater soot blowing should be completed at least an hour before testing begins. This is to ensure the same level of furnace cleanliness in all the test runs.
- Unit operation regime is set at least two hours prior to the commencement of test and conditions maintained steady prior to the test and throughout the test run.
- Main steam pressure and temperature and reheat steam temperature are normally set at design values.
- Steam coil air preheaters' steam supply is isolated and gas recirculation dampers if any, are tightly shut.
- No soot blowing or mill change over is done during the test.
- Economizer hopper de-ashing as well as bottom hopper de-ashing is done on completion of tests wherever system installed for this is used intermittently.

4.7.11.4 Data to be Collected:

- AH outlet and inlet for composition of flue gas
- AH inlet & outlet for temperature of flue gas
- Economizer outlet - for fly ash sampling
- Bottom ash sampling
- Coal sampling from all the running feeders
- Furnace gooseneck for furnace gas composition and temperature profile.
- Main steam temperature
- Hot reheat steam temperature
- Attemperation flows
- Air preheater steam flow (if applicable)

For sampling of flue gas and temperature measurement at AH inlet and outlet sections , a grid of multi point sampling probes as described in Section 4.7.10.8 and 4.7.13 is installed. A "representative" sample of the fly ash should be collected from the gas stream below the economize, but before any hoppers. Bottom ash samples are collected after completion of the test. The methodology would vary with the type of bottom ash collection system. Coal Samples are collected from all running feeders every 20-30 minutes for an 'as fired' sample.

4.7.11.4.1 High Volume Sampler

Various methods can be used for fly ash sampling, including iso-kinetic fly ash sampling, high volume fly ash sampling, and sampling from the air heater, economizer, or precipitator hopper. The preferred method is high volume fly ash sampling as it is simpler than the iso-kinetic method and more accurate than the hopper sampling method.

High Volume Sampler probe is used to recover a representative sample of the fly ash by traversing the flue gas duct at economizer outlet on both sides. The sample is collected, by drawing the gas sample into the collection canister by an aspirator (air ejector). Within

the canister the ash is separated from the gas by passing the sample through high temperature filtering paper. The fly ash sample is collected separately from left and right side and tested for unburned immediately after the test, using a Loss on Ignition - LOI machine and an analytical balance. For insertion of the probe, 100mm diameter-flanged openings are made at economizer outlet on both sides. The sampling location is just above the zirconia probe at economizer outlet. Depending on the duct configuration and resources available, the sample can collected by a single traverse (the probe is put in from one port and sample collected at a number of points) or a grid traverse.

4.7.11.5 Calculations

The calculations are based on ASME Simplified Efficiency Test Procedure that considers only the chemical heat in the fuel as the input and ignores other heat credits like sensible heat in fuel, boiler cooling water pumps' power etc. The efficiency is calculated using the heat loss method. Only the major losses or those that do not require collection of additional data other than detailed in the procedure are considered viz.

- Dry gas loss and its heat rate deviation
- Unburned carbon loss and its heat rate deviation
- Loss due to moisture in fuel and its heat rate deviation
- Loss due to moisture from hydrogen in fuel and its heat rate deviation
- Loss due to CO in flue gases and its heat rate deviation
- Loss due to moisture in air and its heat rate deviation
- Radiation loss and its heat rate deviation
- Unaccountable loss and its heat rate deviation
- Heat Rate deviation due to main steam temperature
- Heat rate deviation due to hot reheat steam temperature
- Heat rate deviation due to attemperation flows
- Heat rate deviation due to air preheater steam flow

The detailed Boiler Efficiency calculation procedure is included in Routine Boiler Efficiency testing procedure no. CENPEEP/EFF/TP/203 forming part of volume No.2 of this guidelines. The heat rate deviations for steam temperatures ad attemperation flows are from the turbogenerator thermal kit. Boiler performance calculations are carried out on line using computational software. Computational procedure involves feeding lot of manual input to on line data processor along with the data which is drawn from off line measurements system installed.

4.7.11.6 Analysis

There are only two operator controllable boiler losses 'Dry Gas Loss' and 'Loss due to Unburned Carbon', and the steam cycle losses are also controllable (steam temperatures, attemperation and APH flows). Dry gas loss is primarily a function of flue gas leaving air heater oxygen and temperature. So as to reduce this loss, both the parameters have to be controlled around optimal values. Exit gas temperatures can be reduced by improving heat

absorption and keeping heat exchange surfaces clean by regular sootblowing, while the weight of flue gas can be reduced by optimizing on the amount of excess air used.

Loss due to unburned carbon in ash is affected by several factors including imbalance in the fuel / air ratio, condition of burners, fineness of pulverized coal.

In the boiler optimization exercise, only one operating variable is varied at a time and its effect on the controllable losses is studied and compared. For an optimized regime the sum of the controllable losses (boiler and turbine) should be as low as possible.

Loss due to formation of carbon monoxide is also operator controllable in the sense that by ensuring complete combustion in the furnace, the operator can ensure low CO levels. The heat loss due to moisture in the coal is uncontrollable as such but can be reduced to some extent by judicious use of water sprays in coal yards and bunkers. The heat loss due to water in flue gas from hydrogen in the fuel is also uncontrollable. The heat loss due to moisture in air depends on ambient conditions. Radiation loss can be kept below the estimated value by maintaining proper insulation conditions. The radiation loss can be estimated using ABMA curve in Fig. 8 of PTC 4.1.

A typical summary sheet of boiler optimization test is shown in next three pages.

Table 4.13 Boiler Performance Tests - Summary Data

Test Number		**1**	**2**	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**
Unit Load	MW	202	204	180	207	203	204	206	206	205	205	205
MS Flow	t/hr	620	640	560	650	630	640	630	650	640	640	540
First stage Pressure	ksc	90	93	82.5	95	95	96	96	95	93.5	93.5	93
MS Pressure	ksc	129	130	132	134	134	135	135	136	135	135	138
MS Temperature	C	542	540	536	538	537	540	540	540	539	537	540
SH Attemperation Spray - Right	t/hr	5	7	4.5	2.2	14	1.4	16	13	17.8	12	15
SH Attemperation Spray - Left	t/hr	9	8.2	14	6.8	21	6.4	21.5	21	19.5	12	15
RH Steam Temperature	C	540	539	536	531	542	541	537	537	540	540	540
Total Air Flow	**t/hr**	**795**	**840**	**740**	**870**	**870**	**870**	**920**	**850**	**890**	**880**	**750**
Secondary Air Flow - Right	t/hr	230	270	250	285	285	290	310	285	300	300	305
Secondary Air Flow - Left	t/hr	215	250	235	270	265	260	290	265	275	275	285
Secondary Air Outlet Temp - Right - PCR	C	255	247	242	244	243	242	245	240	240	239	242
Secondary Air Outlet Temp - Right -Test Grid	C	---	---	---	284.2	291.2	300.6	289.7	289.6	289.8	290.1	280
Secondary Air Outlet Temp - Left - PCR	C	265	265	258	262	265	264	263	261	255	256	260
Secondary Air Outlet Temp - Left -Test Grid	C	---	---	---	296.7	300	300	301.7	300	298.1	297.8	292
Primary Air Outlet Temp - Right - PCR	C	275	272	265	270	272	270	270	266	268	268	268
Primary Air Outlet Temp - Right -Test Grid	C	---	---	---	284.2	288.7	287.7	287	287.1	283.7	284.8	279
Primary Air Outlet Temp - Left - PCR	C	280	280	275	277	280	278	280	276	275	275	275
Primary Air Outlet Temp - Left -Test Grid	C	---	---	---	293.2	297.6	296.5	298.3	297	293.7	295	287
Dry Bulb Temp	**C**	**25**	**27**	**29**	**28.5**	**32.5**	**30.7**	**31**	**33**	**32.8**	**32.5**	**31**
Wet Bulb Temp	C	17.5	17	17	15.3	17.5	17.5	18.2	18.5	18	18	18
PA Header Pressure	mmWC	750	740	730	760	770	770	770	770	740	750	760
Windbox Pressure Right	mmWC	105	95	90	100	95	115	105	100	100	95	90
Windbox Pressure Left	mmWC	105	80	85	100	90	110	105	95	95	95	95
Fuel Air Damper Position	%open	0	0	0	0	0	0	10	10	10	0	0
AA Damper Position	%open	0	0	0	0	0	0	0	0	0	10	10
Burner Tilt	%	0	-30	-10	-5	-5	-10	-20	-20	-24	-20	-15
PCR Flue Gas O2 at Econ Out - Right	%	1.0	1.2	1.5	2.1	1.9	1.5	2.1	1.7	2.2	2.0	2.1
O2 In Flue Gas From HVT Probe -Right	%	< 0.5	< 0.5	3.1	---	---	---	---	---	2.1	---	2.2
AH A Inlet O2 - Right	**%**	**2.4**	**2.7**	**3.9**	**3.8**	**3.7**	**3.5**	**4.6**	**3.1**	**4.2**	**4.0**	**4.3**

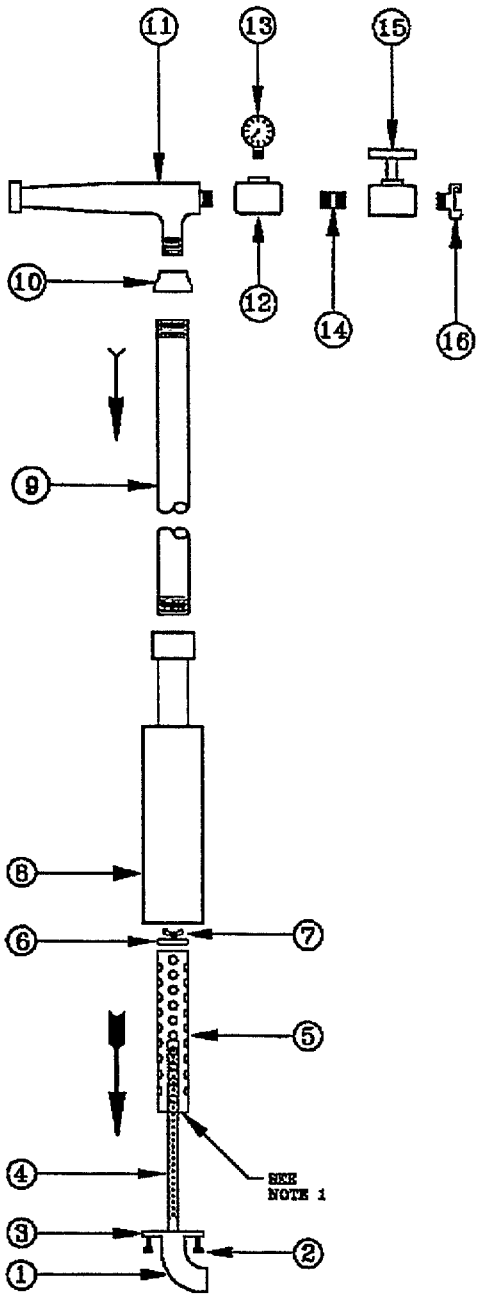
AH A Outlet O2 - Right	%	---	---	---	**7.3**	**7.0**	**7.0**	**7.4**	**6.6**	**7.2**	**7.1**	**7.5**
ID Fan A outlet O2 - Right		---	---	---	---	---	8.4	---	---	---	---	7.89
PCR Flue Gas O2 at Econ Out - Left	%	4.0	5.0	5.0	6.0	6.5	6.4	6.3	5.6	5.6	5.5	5.5
O2 In Flue Gas From HVT Probe - Left	%	1.9	Not Measured	1.5	---	---	---	---	---	0.7	1.3	2.7
AH B Inlet O2 - Left	%	**4.6**	**3.8**	**5.3**	**4.5**	**4.7**	**4.9**	**4.9**	**4.2**	**4.4**	**4.4**	**4.6**
AH B Outlet O2 - Left	%	---	---	---	**7.5**	**7.7**	**7.6**	**7.7**	**6.8**	**7.3**	**7.3**	**7.75**
ID Fan B outlet O2 - Left	%	---	---	---	---	---	8.6	---	---	---	---	8.04
Average AH Inlet O2	%	**3.5**	**3.3**	**4.6**	**4.1**	**4.2**	**4.2**	**4.8**	**3.7**	**4.3**	**4.2**	**4.5**
AH B Inlet CO2 - Left	%	14	14.8	13.5	14	14	---	---	---	---	---	13.7
AH B Outlet CO2 - Left	%	---	---	---	11.3	11.26	11.28	11.44	11.35	11.2	11.2	11.1
AH A Inlet CO2 - Right	%	15.6	15.2	14.6	14.6	---	---	---	---	---	---	---
AH A Outlet CO2 - Right	%	---	---	---	11.7	11.87	11.71	11.33	11.73	11.37	11.4	11.3
CO In Flue Gas From HVT Probe - Right	ppm	>3000	>3000	152	---	---	---	---	---	25	272	73
AH A Inlet CO - Right	ppm	>3001	>3001	56	186	34	210	0	17	0	0	0
AH A Outlet CO - Right	ppm	---	---	---	128	238	268	0	12	0	0	0
CO In Flue Gas From HVT Probe - Left	ppm	11	Not Measured	>3000	---	---	---	---	---	930	23	121
AH B Inlet CO - Left	ppm	53	739	28	0	6	9	0	212	11	0	0
AH B Outlet CO - Left	ppm	---	---	---	0	0	0	0	0	0	0	0
Avg Temp Of Gas At HVT Probe - Right	C	---	---	1155	---	---	---	---	---	1255	---	1212
Avg Temp Of Gas At AH A Inlet - Right PCR	C	330	333	320	331	337	335	339	333	331	333	320
Avg Temp Of Gas At AH A Inlet - Right Test	C	332.3	341.6	329.1	342.4	346.3	344.7	347.7	344.1	344.1	344.9	330.0
Avg Temp Of Gas At AH A Outlet - Right - PCR	C	149	150	149	148	152	151	150	152	149	150	150
Avg Temp Of Gas At AH A Outlet - Right - Test	C	---	---	---	141.1	145.7	143.5	141.7	143.3	141.3	143.3	139.5
Avg Temp Of Gas At HVT Probe - Left	C	1242	---	1322	---	---	---	---	---	1334	1310	1192
Avg Temp Of Gas At AH B Inlet - Left - PCR	C	328	332	320	332	340	340	342	340	340	340	332
Avg Temp Of Gas At AH B Inlet - Left - Test	C	334.9	346.1	330.3	343.8	346.8	346.5	350.3	346.0	344.6	345.5	340.0
Avg Temp Of Gas At AH B Outlet - Left - PCR	C	141	146	141	145	150	149	146	149	143	147	144
Avg Temp Of Gas At AH B Outlet - Left - Test	C	---	---	---	143.0	147.3	146.0	144.1	146.5	141.8	143.9	142.0
Avg FG In Temp at AH Inlet - Test	**C**	**333.6**	**343.9**	**329.7**	**343.1**	**346.6**	**345.6**	**349.0**	**345.1**	**344.4**	**345.2**	**335.0**
Avg FG Temp at AH Outlet - Test	**C**	---	---	---	**142.1**	**146.5**	**144.8**	**142.9**	**144.9**	**141.6**	**143.6**	**140.7**
Avg Weighted air Temp In AH	C	---	---	---	33.3	37.3	35.5	35.7	37.8	37.6	37.2	35.1
EGT Corrected to Design Inlet Air Temp	C	---	---	---	149.7	151.4	150.9	149.0	149.6	146.5	148.7	147.1

FD Fan A	kW	---	---	---	189.4	183.4	210.8	218.8	215.2	---	---	---
FD Fan B	kW	---	---	---	177.4	188.3	179.1	191.6	200	---	---	---
PA Fan A	kW	---	---	---	1439	1326.4	1262.4	1324	---	---	---	---
PA Fan B	kW	---	---	---	1185.6	1190.6	1236.6	1186.8	---	---	---	---
ID Fan A	kW	---	---	1120.8	1236.8	1084.2	1052.8	1032	1096.8	---	---	---
ID Fan B	kW	---	---	1028.2	1127.6	1106.8	1109.9	1090	1052	---	---	---
GCV as fired	kcal/kg	3920	4040	3880	3508	3611	3762	3498	3570	3617	3616	3616
Total Moisture	%	16.17	15.22	15.49	15.16	15.6	14.77	15.88	15.42	14.92	15.27	15.27
Ash	%	30.23	31.19	32.22	35.76	35.04	33.96	35.92	35.55	35.2	35.21	35.21
VM	%	23.1	23.44	23.83	25.64	22.48	22.67	22.71	22.61	26.53	22.59	22.59
FC	%	30.5	30.15	28.46	23.44	26.88	28.6	25.49	26.42	23.54	26.93	26.93
Average LOI of All Ash Streams	%	3.7	2.6	1.7	2.1	1.7	1.4	2.2	1.8	2.1	1.4	1.6
LOI From Hi Vol Right	%	3.7	2.3	0.7	1.1	1.0	0.7	1.3	1.0	0.8	0.6	0.28
LOI From Hi Vol Left	%	1.8	0.7	0.9	1.9	0.8	0.7	1.3	1.3	1.0	1.0	0.4
LOI From Bottom Ash - Right	%	7.7	7.0	7.0	5.3	5.3	4.9	5.2	4.0	7.2	3.6	5.2
LOI From Bottom Ash - Left	%	7.7	7.0	4.0	3.9	4.7	3.8	6.9	4.9	6.5	4.1	8.1
Dry Gas Loss	%	5.8	6.18	5.78	5.73	5.92	5.88	5.81	5.80	5.63	5.9	5.83
Combustible in Refuse Loss	%	2.4	1.6	1.2	1.8	1.4	1.1	1.9	1.5	1.7	1.1	1.3
Moisture in Fuel Loss	%	2.6	2.4	2.5	2.7	2.7	2.4	2.8	2.7	2.6	2.6	2.6
Hydrogen Loss	%	4.2	4.2	4.3	4.7	4.3	4.2	4.4	4.4	4.7	4.3	4.3
Moisture In Air Loss	%	0.08	0.08	0.08	0.06	0.07	0.08	0.09	0.08	0.08	0.08	0.08
CO Loss	%	1.48	1.45	0	0.03	0.03	0.02	0.00	0.00	0.00	0.00	0.00
Radiation and unaccounted loss	%	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7	1.7
Total Losses	%	18.2	17.7	15.5	16.7	16.1	15.4	16.7	16.1	16.3	15.8	15.8
Boiler Efficiency	%	81.75	82.29	84.46	83.31	83.88	84.56	83.26	83.89	83.66	84.25	84.15

4.7.11.7 References

- ASME PTC 4 - 1998 Fired Steam Generators
- ASME PTC 6 - 1996 Performance Test Code on Steam Turbines

4.7.11.8 Attachments



NOTES:
1 – FILTER PAPER TO BE INSERTED INTO RECOVERY SCREEN

1 – SAMPLER TIP
2 – 1/4" MACHINE SCREWS (4)
3 – CYLINDER FLANGE
4 – PRIMARY RECOVERY SCREEN
5 – RECOVERY SCREEN
6 – SCREEN CAP
7 – WINGNUT
8 – CYLINDER COVER
9 – 1 1/4" PIPE
10 – 1" X 1 1/4" REDUCER
11 – ASPIRATOR
12 – 1/4" X 3/4" X 3/4" TEE
13 – PRESSURE GAGE
14 – 3/4" CLOSE NIPPLE
15 – 3/4" NEEDLE VALVE
16 – CHICAGO FITTING

Figure 4.11 High Volume Sampler

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4.7.12 HVT Traverse Test

4.7.12.1 Purpose

High Velocity Thermocouple probe is used to measure the temperature and composition of gases leaving the furnace (or at any point in the convection pass). Normally traverses are conducted at furnace exit near the gooseneck.

HVT traverses are usually performed along with boiler optimization tests, or independently to check the amount of air infiltration in the convection pass or to determine the flue gas temperature, O_2 and CO levels or profiles entering the convection pass.

The probe is water-cooled and temperature measurement as high as 1320 °C. The probe has three concentric pipes, the innermost for flue gas sample flow, middle one for cooling water inlet flow and outer one for cooling water outlet flow. Uninterrupted water supply for cooling of HVT probe with water pressure > 5 kg/cm^2 is ensured by providing an additional booster pump if required.. This pump normally takes suction from a service water line or a fire water hydrant. The skin temperature is monitored with bare hands during actual traverse and should not increase more than 60°C. Furnace gas temperature is measured by a 'K' type thermocouple that runs through the length of innermost pipe. For measurement of temperature an air aspirator is used to suck flue gas from the furnace. The thermocouple tip at the end of the probe is shielded from direct radiation in the furnace using a 'radiant shield' so that a true furnace gas temperature is obtained at the sampling point. The furnace width is traversed every two feet on both sides and FG samples are drawn to the analyzers by a vacuum pump.

4.7.12.2 Frequency

HVT traverses are usually performed:

- Whenever a boiler optimization test is run
- Before boiler outages

Other triggers can take precedence:

- High carbon in fly ash
- Slag formation around certain burners
- High CO at the furnace exit
- CO and temperature imbalance at the furnace exit
- Large O_2 spread at AH inlet to main equal steam temperatures

4.7.12.3 Unit Condition

HVT traverses can be performed at almost any condition. However, to have repeatable conditions for periodic monitoring, the following should have a set value for each test:

- Burner tilts
- Fuel air, auxiliary air, overfire air, etc. damper positions
- Steam flow

- Air flow

4.7.12.4 Data to be Collected

The probe is capable of measuring flue gas temperature and extracting a gas sample, usually to be analyzed for O_2 and CO. For insertion of the probe, a suitable port hole is located on either side of the boiler near the gooseneck. If the same is not available, a LRSB - Long Retractable Soot Blower is removed on both the sides for the insertion of the probe.

4.7.12.5 Calculation

The O_2 at the furnace exit is compared to the O_2 at the economizer outlet/air heater inlet. Also the profile of the furnace CO, O_2 and temperature are plotted.

4.7.12.6 Analysis

If there is a considerable increase in the O_2 between the furnace outlet and the economizer outlet AH inlet, it indicates excessive air inleakage in the convection pass. In addition to the heat rate impact due to cold air bypassing the air heater, the operator will be running the unit with less O_2 in the furnace than it would appear to have. This can increase LOI and exit gas temperature.

If the CO levels are above 50-100, it indicates that combustion is not complete in the furnace. This could be due to air/fuel imbalances at one or more burners, insufficient O_2 overall, poor fineness, etc.

If the temperature/CO/O_2 profile is not relatively uniform at the furnace outlet, it indicates either an air/fuel imbalance at one or more burners or a problem with the tilts or dampers in one or more corners.

4.7.12.7 References

n/a

4.7.12.8 Attachments

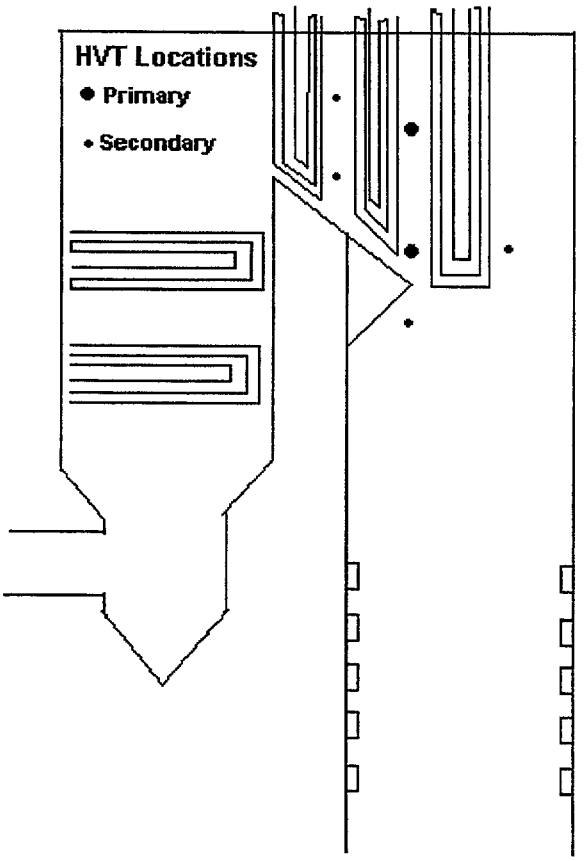

Figure 4.12 Sketch of Two Pass Boiler and Locations of HVT Probe

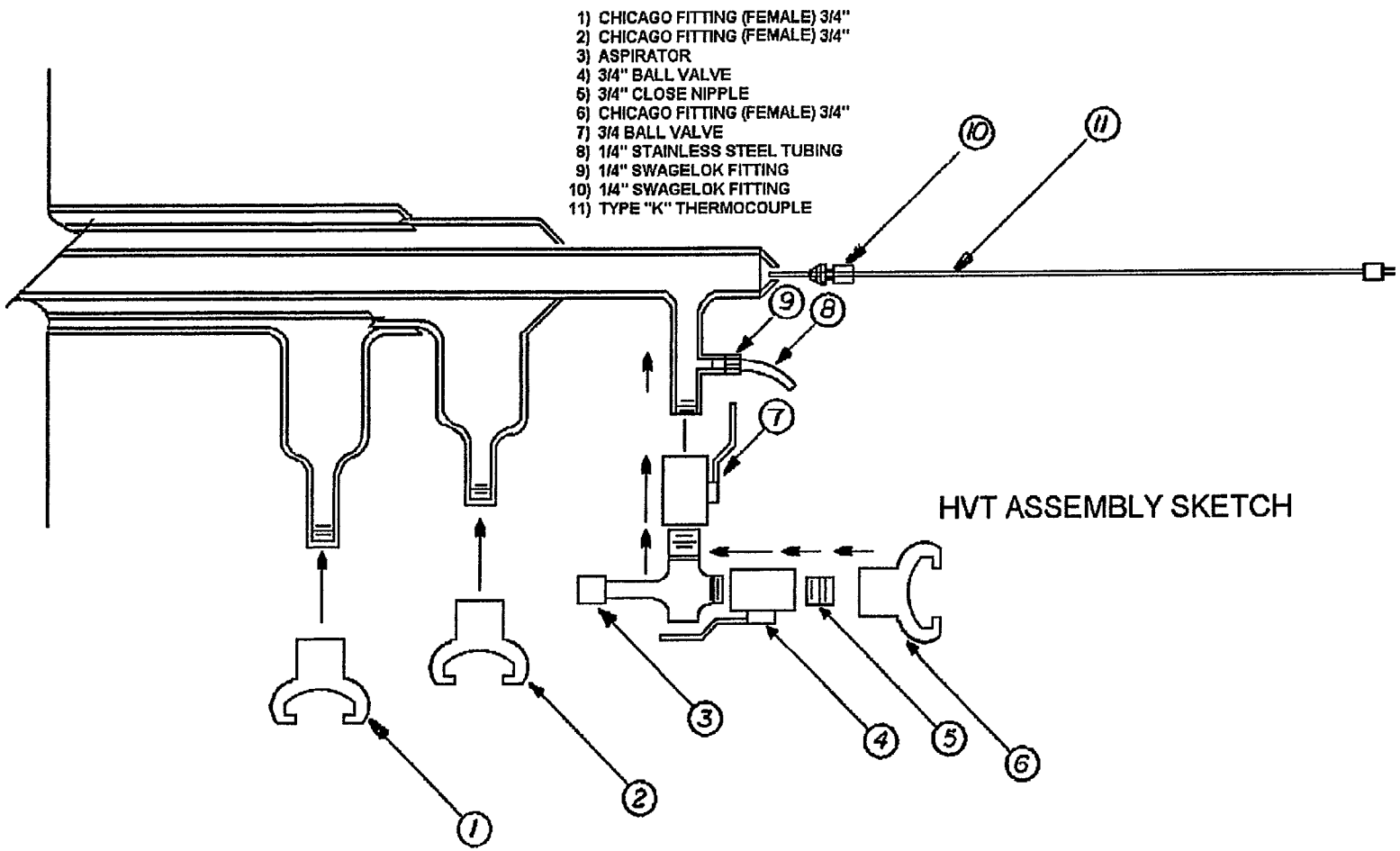

Figure 4.13 High Volume Sampler

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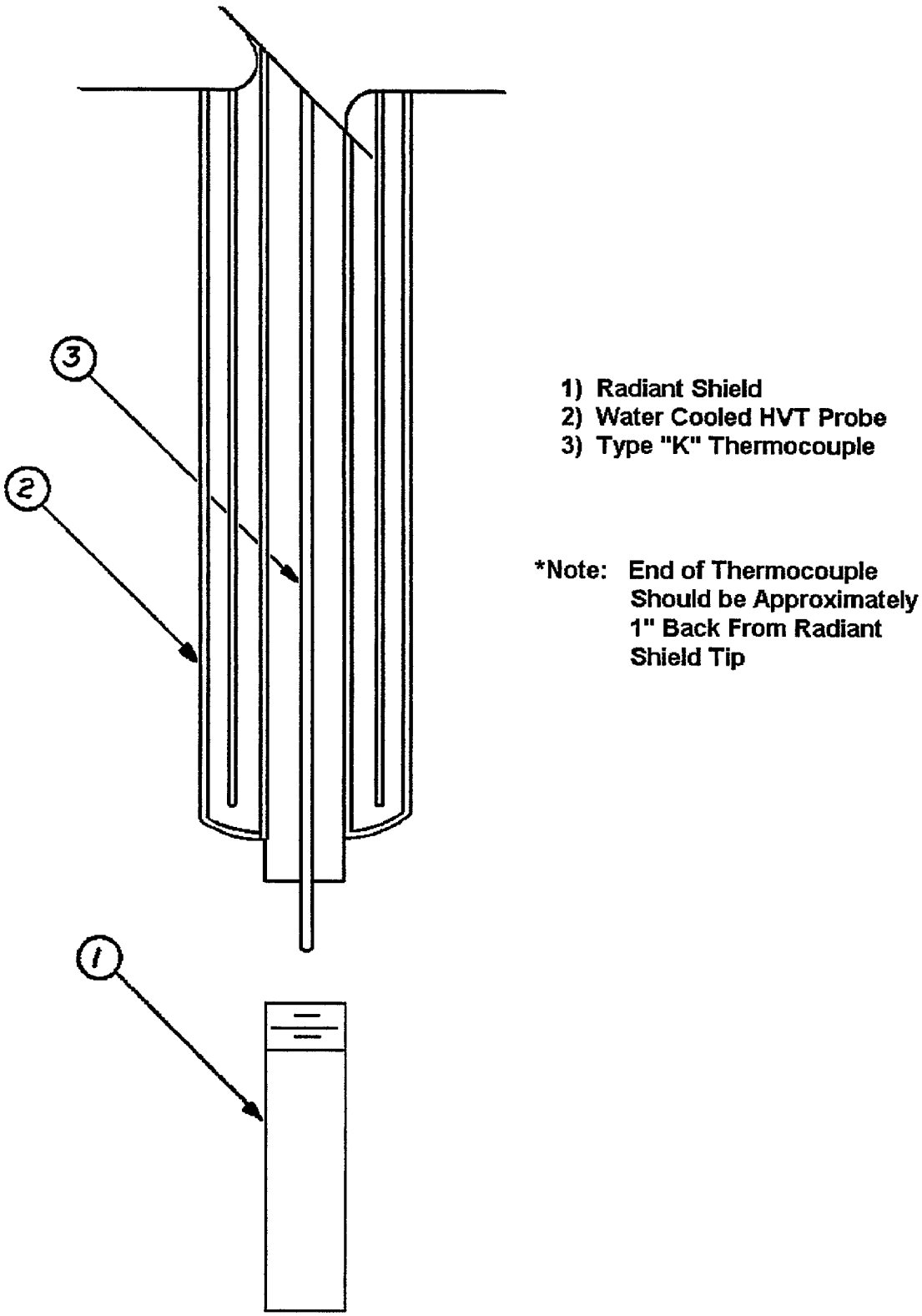

Figure 4.14 High Volume Sampler - Probe Details

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4.7.13 How to Determination of Number of Sample Points and Locations in Ducts

ASME PTC 19.10 sections 3.02.4 and 3.02.5 discuss in detail how to determine how many sample locations are required, and their locations. The sample plane should be a location where the velocity profile is as uniform as possible. If there isn't any data on the velocity profile, a velocity traverse should be made at the planned location prior to any testing (usually a s-type pitot is used for this). For exactness, the measurements at each sample point (temperature, gas composition, etc.) should be a weighted average based on the velocity at each point. In "routine tests" this is rarely done, but if the design of the duct work is such that the isn't a "good" test plane with relatively uniform velocities, this will have to be done.

In general, a sample plane that is at least eight (8) diameters downstream and two (2) diameters upstream of the nearest disturbance can be considered to have an acceptably uniform velocity profile. To determine the number of sample points that are required, refer to Figure 4.15

For rectangular ducts, the equivalent diameter is calculated by:

diameter = 2 x (length x width) / (length + width)

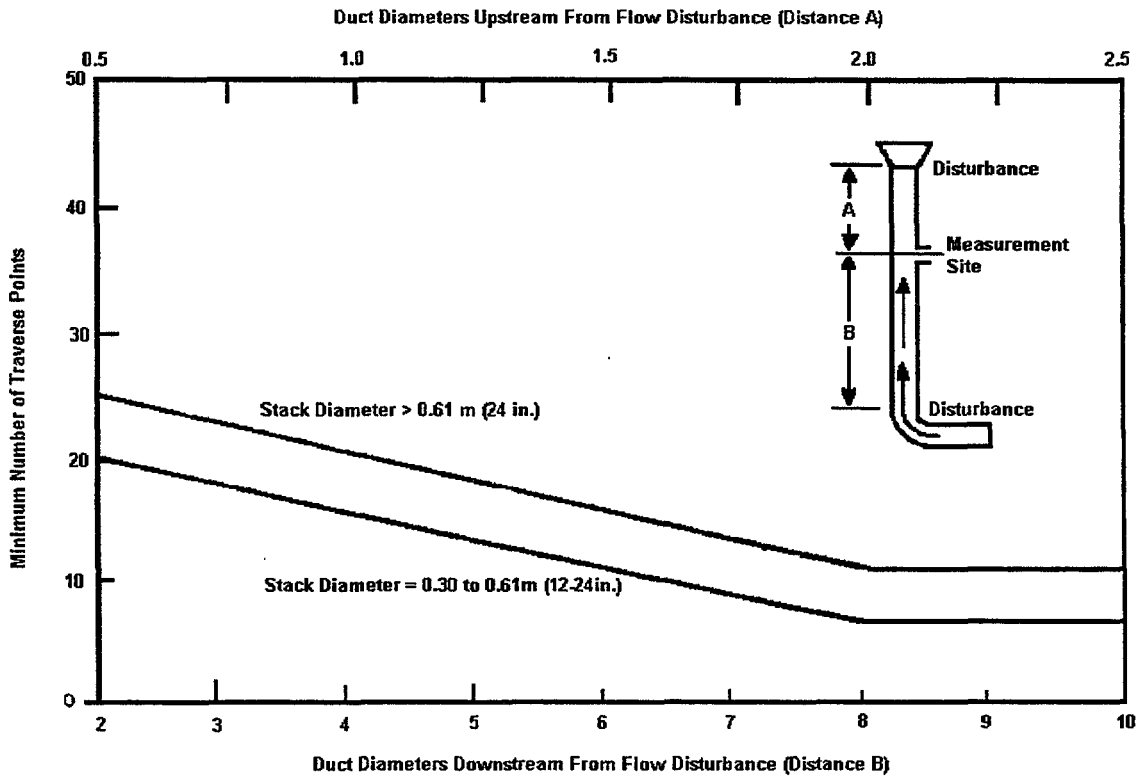

Figure 4.15 Number of Test Points vs. Duct Diameters from Disturbances

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Following this procedure may lead to a conservative number of sample points being used. Another helpful check is to continuously (not at discrete positions) measure a single component across the duct at several locations, and see how much stratification is present. Other considerations in the determine of the "best" location for the test plane are

A. Availability of services, power for the analyzers, pumps, and other equipment; compressed air; cooling water for water cooled probes; communications; etc.
B. Ease of access and safety for the installation and handling of the large probes
C. Distance from furnace or air preheater (on balanced draft units, the further from the boiler you are the more likely the flue gas will be diluted by air inleakage.
D. Ambient conditions, if outside, can the area be covered so testing can continue during inclement weather; is the lighting sufficient; dust; heat; etc.

4.7.13.1 Rectangular Ducts

In general, a large rectangular duct should be divided in several smaller areas, each of should be identical in size and shape. The shape (ratio of width to length) of the smaller areas should approximate the shape of the duct. The sample points should be in the center of these smaller areas. See Figure 4.16

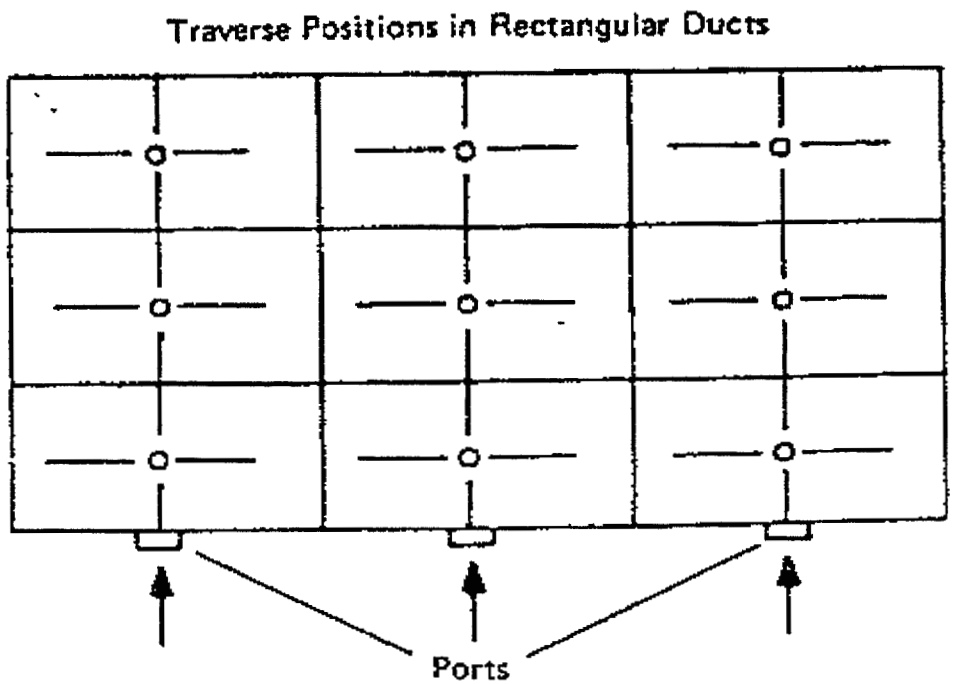

Figure 4.16 Measurement Locations for a Rectangular Duct
(flow is perpendicular to the page)

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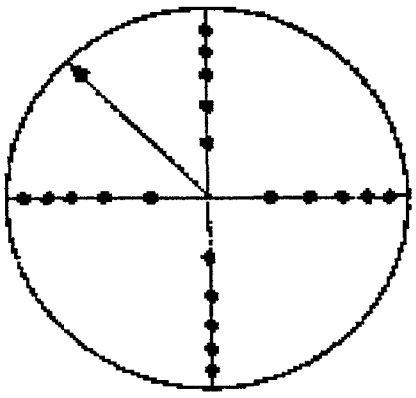
4.7.13.2 Circular Ducts

In general, a large circular duct should be divided in several annular areas by drawing concentric circles so that each annular section has the same area. The sample points are in the centers of the areas. Usually the pipe or duct is probed from two locations 90 degrees apart. See Figure4.17.



Cross Section of
Circular Gas Passage

NOTE: indicates location of sample point

$Rn = R\sqrt{(2Na - 1) / Nt}$

Rn = distance from sampling point to center of pipe
R = radius of pipe

Na = no. of sampling points counted from center as zero
Nt = total no. of sampling points on a diameter

Figure 4.17 Typical Divisions of a Circular Duct into Equal Areas

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4.7.13.3 Coal Pipes

When probing in a two phase flow stream (air and coal particles) more test points are usually required. Figure 4.18 shows typical divisions of coal pipes of various sizes, and Figure 4.19 shows how to determine the number of traverses that are required for a coal pipe with different distances up and down stream of a disturbance. (For pulverizer testing 2 traverses are usually sufficient if the sample plane is at least 10 diameters downstream and 5 diameters upstream of the nearest obstruction. Three traverses (60 degrees apart) are needed if the test plane is at least 5 diameters downstream and 2 diameters upstream of the nearest obstruction. Five traverses (30 degrees apart) are recommended if the closest obstruction is 2 diameters up stream or 1 diameter downstream.)

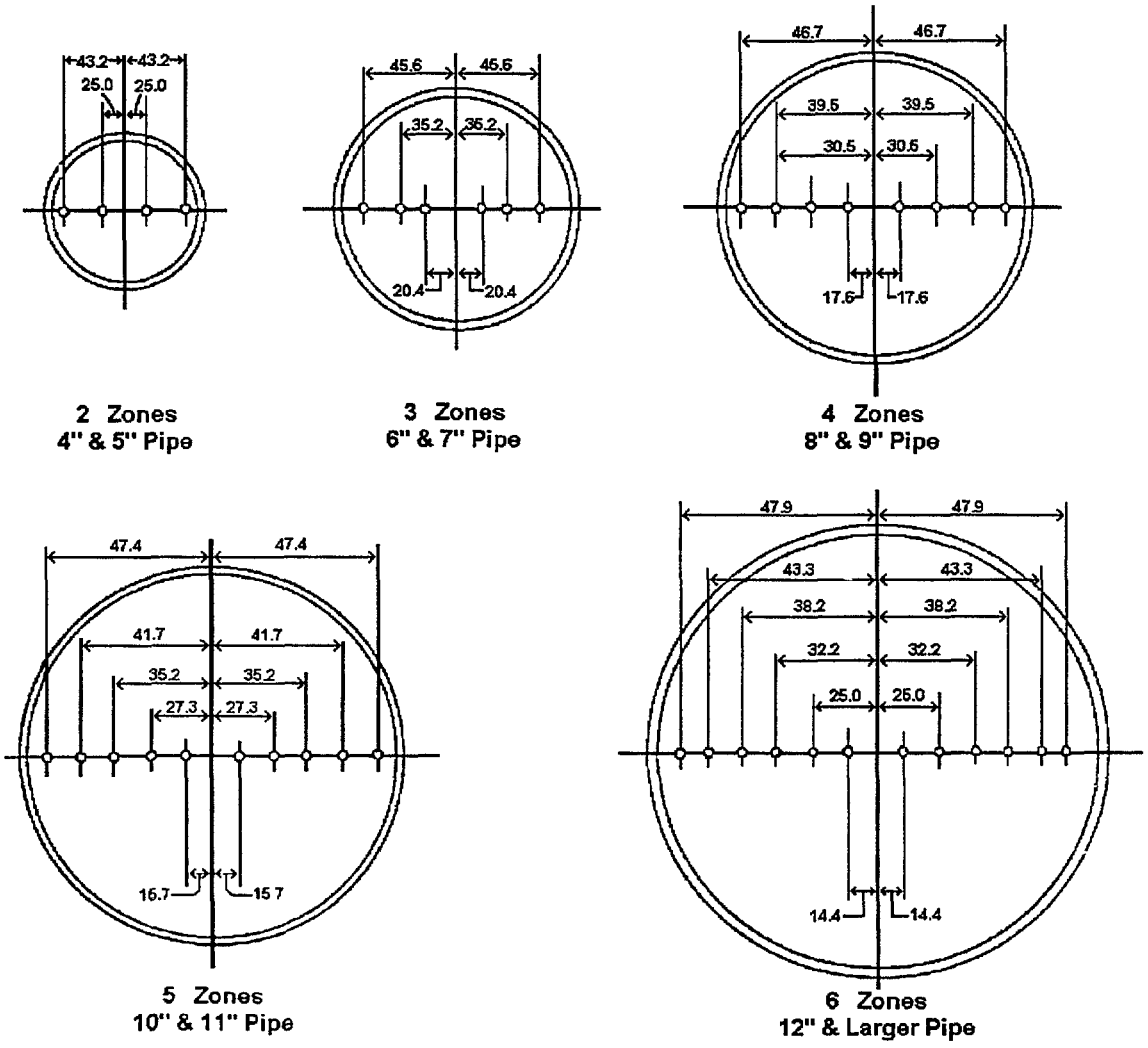

Figure 4.18 Sampling Points for Pulverized Coal

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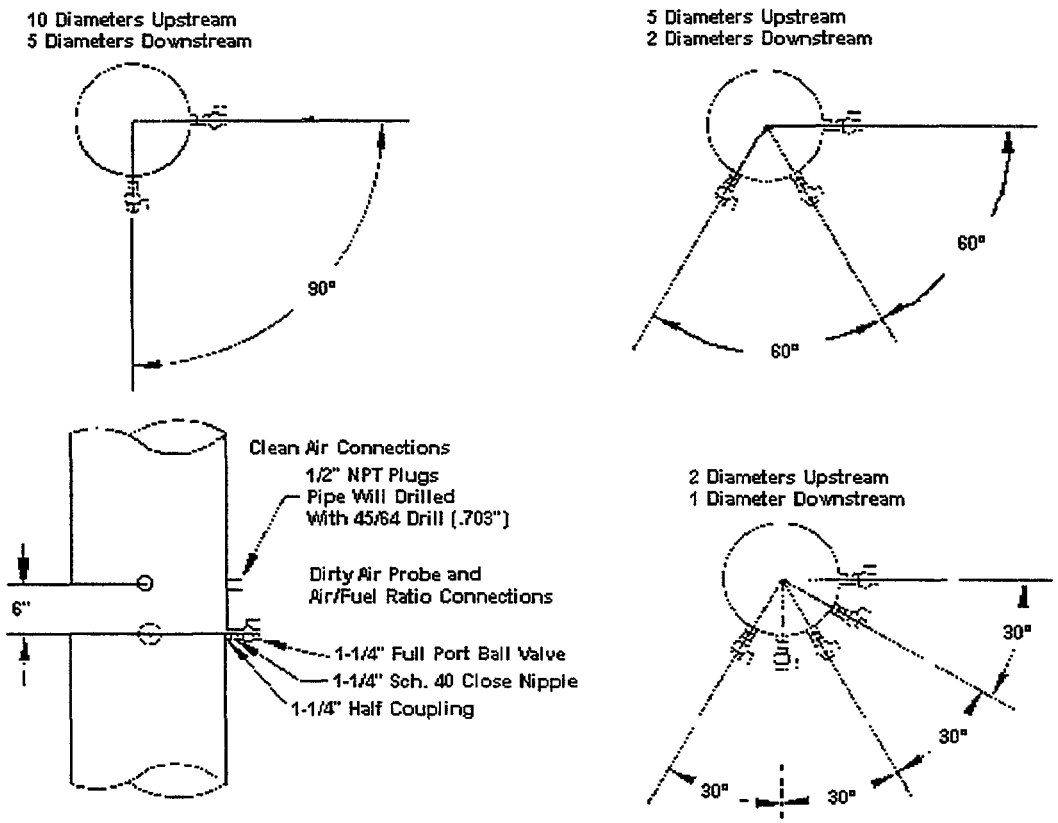

Figure 4.19 Number of Required Traverses of Coal Pipes

4.7.14 Unit Auxiliary Power Consumption Test

4.7.14.1 Purpose

Auxiliary power consumption tests are conducted to account for the power that is being used by all large auxiliaries, and to identify the equipment that is using excessive power. It is also used to determine the general cause of excessive power usage, equipment degradation or running at higher capacity.

It is important to keep a record of expected values/base values for the energy consumption for the unit as well as individual sections/auxiliaries for comparison.

4.7.14.2 Frequency

Time based, typically every year before and after the unit outage, but the other triggers can take precedence:

- High auxiliary power or station service
- Equipment capability becomes limited

4.7.14.3 Unit Conditions

Tests should be run at the maximum continuous rating feedwater flow, and a set boiler O_2. The allowable ranges of these two parameters should be specified on the pre-test checksheet.

The unit should be held steady for a sufficient period of time (typically two hours) to stabilize the parameters before the commencement of the test, The test duration is usually 6-8 hours.

4.7.14.4 Data to Be Collected

The energy consumption and MW readings for the following to be collected:

1. Station load – average MW reading for the period of test
2. Unit load – average MW reading for the period of test
3. Energy consumption for Common Station Services (CSS) for e.g. CT pumps, ash pumps, etc.
4. Integrated energy meter readings at beginning and end of test for all major unit equipment such as BFP, FDF, PAF, IDF, mills, CCWP, etc.
5. Feedwater flow (total and for each BFP)
6. Air flow (total and for each fan)
7. Fuel flow (total and for each mill)

The preferred accuracy for the power/energy meters to be used is + / - 0.2%. Arrangement for online hooking up of these meters is to be provided in the panel.

4.7.14.5 Calculations

Calculate total units Auxiliary Power Consumption considering the prorated common station services.

A. Prorated Common Station Service (PCSS)

PCSS = (Unit MW / Pant MW) x (CSS – unit load on CSS)
Total PCSS = Metered CSS dedicated to the unit only + PCSS

B. Total Station Service (TSS)

TSS = (Units Station Service + TPCSS)

C. Percentage of TSS to Unit load

$$\% \text{ TSS} = \frac{\text{T.S.S}}{\text{Unit Power}} \times 100$$

4.7.14.6 Analysis

Trend Total Station Service (TSS) plot
Plot TSS as percentage of unit generation
Look for deviations
In case of deviations perform the root cause analysis by:

- Observe deviations down the line from common bus to the individual auxiliary level
- Perform analysis on the deviated auxiliary
- Make action plan for corrective
- Repeat the survey after doing the needful to confirm the status.

4.7.14.7 References

TVA Test Procedure
CenPEEP Test Procedure No. CENPEEP/EFF/TP/40

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SECTION 5 HEAT RATE ACTION PLAN

5.1 Introduction

Sections 2, 3 and 4 discussed collecting data, and turning it into useful information. This section discusses how to take that information and develop an action plan to:

- Ensure that necessary and cost beneficial improvement activities are carried out.
- Set "target" values for parameters (such as boiler outlet O_2) and for their heat rate deviations for each unit.
- Predict future values for heat rate deviations and heat rates for strategic planning purposes for individual units, plants, and entire utility systems.

The lowest level of a heat rate action plan (HRAP) includes a list of all planned activities that will affect the heat rate of an individual unit. It includes activities that will increase or decrease the efficiency of the unit (such as condenser tube cleaning), normal deterioration (such as occurs to turbines), fuel changes, or modifications to equipment (such as adding additional air heater basket depth). Other information should be included with each activity identifying its schedule, the name of the person responsible for the activity, how much the activity will affect each heat rate parameter, etc. Figure 5.1, at the end of this section, shows an example of an HRAP.

Once an HRAP is developed for each unit at a plant, the resulting target values (heat rate deviations for each parameter and heat rate) can be "rolled-up" to plant targets. The plant targets can be further "rolled-up" to area or utility targets.

A heat rate action plan usually covers 3 to 7 years. However, it should not be a static document. It should be reviewed and revised periodically. The following list includes some of the reasons for revising the plan:

- Identifying and scheduling a new activity that will affect the heat rate
- Changing the schedule for an activity already in the plan (such as moving an outage that includes heat rate activities)
- If the same activity is planned for several units, after the completion of the first unit, the process indicator target and heat rate deviation target for the remaining units may need to be changed to reflect the values that were achieved.

5.2 Identify Activities To Be Included in the Heat Rate Action Plan

There are several types of activities that affect the heat rate of a unit.

5.2.1 Identify Activities That Will Correct Heat Rate Deviations

The first step in developing a Heat Rate Action Plan is to know where deviations exist between the current actual performance and the "achievable" performance, given the current plant configuration. (Section 3.3 discussed the various "baselines" that can be used. Section 3.3.5 described the recommended baseline or "expected" level, as the actual measured performance when the unit was new, adjusted for permanent configuration changes.) Wherever there are gaps between the actual and "expected" parameters values, there are heat rate deviations, which are opportunities for improvement. (See section 3.2 for a list of typical "parameters.") This does not necessarily mean that there is a cost effective action that should be taken.

The second step is to rank all known areas where heat rate deviations exist from the largest to the smallest, and calculate the cost of each. For example, one unit might have high condenser pressure as its largest heat rate deviation, followed by low HP turbine efficiency, followed by high boiler outlet O_2. For each heat rate deviation, the monthly or annual cost (in Rupees) is calculated. On the Example Plan (figure 5.1), the column labeled "BASE: 12MTD Jul 99" lists the heat rate deviations for the 8 parameters on this plan, ranked from the largest (141 Btu/kWh due to Exhaust Pressure to 49 Btu/kWh for Superheat Attemperation).

The third step is to determine the "root cause" of each significant heat rate deviation. Too often, instead of eliminating the root cause of problems, only symptoms are addressed. For example, a unit may have poor turbine efficiency, therefore the turbine is overhauled and the efficiency improves. However, if the root cause of the problem is poor water quality, the higher post outage efficiency will not last long. Another example would be a unit with a high drain cooler approach (DCA) temperature difference (drain temperature - FW temperature entering the heater). One way to reduce the DCA is to raise the water level in the heater. However, the problem could be due to fouled tubes and, while raising the level may improve the DCA, the FW outlet temperature will drop resulting in even poorer efficiency.

Determining the root cause of heat rate deviations is sometimes a complicated process. The Logic Trees in Appendix E are one resource that can be used to investigate deficiencies. Frequently, special purpose tests are required. Another helpful practice is to compare data from similar units if a deviation exists on one unit but not another. Having good quality data from the performance guarantee test time period is also valuable to compare to current conditions. Regardless of the method(s) that is/are used, it is critical that the <u>root cause</u> be determined.

The fourth step, taken after the root cause is determined, is to list what options are possible, calculate the cost and benefits of each, and determine the best course of action. For example if the efficiency of a turbine section is down, there are several options available, each option will cost a different amount, and each option will have different benefits. Replacing the rotor and inner cylinder is a very costly option, it will also show the largest gain, but usually is not economically justified. Another option would be to replace only the rotor, a third option would be to re-blade the existing rotor, a fourth option would be to repair the existing blades, etc. The last option is to do nothing, and sometimes it is the most economical option at the present time, but as conditions change (cost of fuel, power replacement cost, section efficiency) one of the other options may become the most economic.

Frequently, the plan will include a field titled "Activity Selected." If an engineering analysis has been performed, which identified the best course of action, but the activity is not scheduled, or is uneconomical at the present time, the activity can still be listed in the plan, but marked "No" under "Activity Selected" so it is not forgotten.

5.2.2 Identify Activities That Will Change the Expected Heat Rate

In addition to activities that are intended to correct heat rate deviations, there are also projects that change the "expected" heat rate. These projects permanently increase or decrease the thermal efficiency, therefore the expected level(s) should also be changed. Examples of such projects are:

- Replacing an existing turbine rotor with a new, higher efficiency design rotor.
- Adding additional air heater basket depth.
- Modifying the furnace or convection pass heat transfer surface area.
- Adding additional auxiliary equipment such as additional precipitator fields, coal crushers, etc.

A cautionary note: Too often, when looking for opportunities for heat rate improvements, there is a tendency to concentrate on large capital projects. While these types of activities should be considered, they should not be the primary focus of a heat rate action plan.

5.2.3 Identify Other Activities That Will Affect Heat Rate

The Heat Rate Action Plan should also list any other activities that will affect the heat rate of the unit. These types of activities are usually driven by considerations other than thermal efficiency, such as reliability, safety, fuel cost, etc. Examples of such activities are:

- Switching to a fuel with a significantly different moisture or hydrogen content due to price or availability.
- Decreasing steam temperatures or pressures due to boiler tube reliability issues.
- Increased planned outage time increasing "station service - idle" heat rate deviations.

5.2.4 Identify "Accounting" Changes

The last type of activity that is included in the heat rate action plan are activities that do not change the "actual" heat rate of the unit, but change the reported heat rate or the reported heat rate deviation for one or more parameters.

One type of accounting change is the result of correcting measurement errors. One example of this would be if a measurement error was found in the coal scales. Suppose the scales have been reading 2% low. The next year's reported heat rate would be expected to increase by 2%. (The

"true" heat rate of the unit would not increase, but the "reported" heat rate would increase.) Since each year's heat rate plan is based on historic data (called the "basis"), the heat rate action plan should show an increase under unaccountables. The amount of the increase (in kcal/kWh) in unaccountables due to correcting the measurement error would correspond to 2% of the basis period unit heat rate.

Another accounting change would be if a "new parameter" is being added to the parameters for which heat rate deviations are already being calculated. For example, a unit that did not measure the amount of steam used for auxiliaries during the basis period starts measuring this flow. From this measurement, it is determined that the amount of steam usage is different than the "expected" value, and a heat rate deviation is calculated. Since the heat rate deviation for this parameter historically was zero, the HRAP should show an increase in the heat rate deviation for this parameter, and a corresponding decrease in the "unaccountable" parameter.

5.3 Build the Heat Rate Action Plan

5.3.1 Set the Basis for the Plan

The first step in building a heat rate action plan for the upcoming year(s) is to set a basis. Usually the most current 12 month-to-date data is used to eliminate seasonal variations. (Sometimes, instead of using the most current 12 month-to-date data, the year end 12 month-to-date data is forecasted, and the next year's plan is built off that.) The "expected" heat rate and the heat rate deviations for all parameters are tabulated. If there are no activities that will affect any particular parameter, the target heat rate deviation for that parameter will be the same as the basis.

5.3.2 List Activities by Affected Parameter

Under each parameter for which heat rate deviations are calculated, each planned activity that will affect that parameter should be listed. For example, under "Exit Gas Temperature," the activities "Replace Air Heater Seals, and Set to Design Clearances," "Rebuild All IR Sootblowers," and/or "Change Pulverizer Classifier Setting from 1.5 to 3," might be listed.

Many activities may be listed under more than one parameter. For example, "Replace Air Heater Seals, and Set to Design Clearances" would probably affect both exit gas temperature and also unit auxiliary power, and should be listed under both.

> On the Example Plan, the activity "Boiler Sootblower Rehab Project" is listed under both Reheat Temperature and Superheat Attemperation because both will be affected by the activity.

The listed activities must be specific and verifiable. Activities with titles such as "Improve Sootblower Performance" or "Improve Performance of Operator Controllables" are too vague. How would someone know if it is being done?

In addition to the description of the activity, a measurable process indicator should be listed with its current value, the anticipated change or target value, and the method of verification. For example, for the activity "Replace Air Heater Seals, and Set to Design Clearances," the process indicator might be "Air Heater Leakage." The current value might be 20% and the target (post replacement) value might be 12%, and this would be verified by a post outage test. For the activity "Rebuild All Long Retractable Sootblowers," the process indicator might be "LR Sootblower Availability." The current value might be 50% and the target might be 85%, and this could be verified by the 6 month average availability after the last rebuild. For the activity "Change Pulverizer Classifier Setting from 1.5 to 3," the process indicator might be "200mesh Fineness." The current value might be 60% and the target value might be 72%, verified by the "average fineness from all tests conducted during the 6 month period after the end date".

Each activity must also list a schedule (as a minimum a start date and a completion date). Each activity should list a single individual who is "responsible" for ensuring that the activity is carried out, and the results verified.

Each activity should also list the projected impact it will have on the heat rate parameter. The normal sign convention is a negative value is an improvement to the heat rate, and a positive value indicates a projected increase in the heat rate. Usually, the projected impact is the effect the project will have on the 12 month-to-date heat rate, 12 months after the activity's completion. The impact to the 12 month-to-date heat rate for any period can then be calculated based on this value and the scheduled completion date.

Each activity should also indicate what "type" of activity it is. A good heat rate action plan should consist primarily of "I or Improvement/Degradation" activities. These are activities that affect the heat rate, such as the three described above. Most of these activities are intended to return the performance to the current "expected" levels. Also included in this type are activities that are not permanent degradations (such as temporarily reducing steam temperatures due to reliability concerns).

"E or Expected" activities are activities that will result in changes to the unit heat rate and changes to one or more parameter's expected level, such as replacing the HP turbine rotor with a new-high efficiency design or retubing feedwater heaters using a material with a poorer heat transmittance.

> On the Example Plan (figure 5.1), under the parameter "HP Turbine Efficiency" there is an activity to "Replace Rotor with Redesigned Rotor." There are two lines for this activity, the first line shows the heat rate deviations going to 0, restoring the performance to the current expected level, the second line is a type "E" activity, showing the expected heat rate and actual heat rate improving an additional 76 Btu/kWh.

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"A or Accounting" activities are activities that do not change the actual heat rate of the unit, just the bookkeeping. If the reported heat rate is projected to change due to correcting a problem with coal scales, this would be an "A" activity.

> On the Example Plan, there is an activity titled "Better Measurement of Air Heater Leakage" under Exit Gas Temperature (as a -15 Btu/kWh change) and under Unaccountables (as a +15 Btu/kWh change). The plant determined that the actual air heater leakage was 2 percentage points lower than the value used in the calculations of heat rate deviation for Exit Gas Temperature. The plant believes that the reported heat rate deviation for Exit Gas Temperature has been too high by 15 Btu/kWh. Therefore, the plan shows the heat rate deviation for Exit Gas Temperature decreasing by 15 Btu/kWh, and the heat rate deviation for Unaccountables increasing 15 Btu/kWh, the net effect on the total deviation and net unit heat rate is zero. The "target" parameter heat rate deviation for Exit Gas Temperature will be 15 Btu/kWh lower than the Basis, and the Unaccountable target will be 15 Btu/kWh higher.

"D or Degradation" is used under some parameters to account for normal deterioration of a parameter. Usually this is only used to account for turbine efficiency, which typically decreases at a 0.5-1.0% points per year.

> On the Example Plan, the parameters "HP Turbine Efficiency" and "IP Turbine Efficiency" show degradation rates of 9 Btu/kWh and 5 Btu/kWh respectively.

A last type of activity is an "S or Study." If a parameter has a significant heat rate deviation, but the appropriate corrective action has not been determined, a "study" should be initiated, and assigned to a "responsible person." Like the other types of activities it should include a schedule.

> On the Example Plan, under the parameter "Exit Gas Temperature," there is an activity titled "Replace Cold-End Baskets w/ High Eff. Design," and it is a type "S." The cost of the new baskets has been identified, but the heat rate impact (and therefore the benefit/cost) has not been determined. It has been assigned to "R. Helton" to complete the engineering analysis by 9/30/2000. Once the study is complete, if the project is economically advantageous and approved, it can be added to the plan as a type "I" activity.

Other information that is sometimes listed with each parameter is the activity cost (which may be broken down into Capital and O&M), rate of return or net present value, etc. This information is useful for deciding which projects should be approved and which may need to be delayed.

Another bit of information that is valuable to capture with each activity is either a document number or a "documentation" field where a more detailed description of the activity and the assumptions, calculations, etc., can be found.

A refinement that is sometimes added to the plan is to indicate whether the heat rate impact starts after the activity completion date (i.e., for a turbine overhaul) or if some of the benefits are gradually incurred during the activity (i.e., a project to rebuild the sootblowers one at a time while the unit is running might take 3 months, but some benefits would occur while the project is ongoing. The full benefits would not be realized until 12 months after the activity is complete).

5.3.3 Calculate the Target Heat Rate Deviations

After all activities have been identified, and the schedules and heat rate impacts listed, the 12 month-to-date heat rate impacts for each time period can be calculated. (Some plans calculate targets for each year only, other plans calculate targets for each month in the upcoming year and then financial year end targets for years after that.) For example, if the project's projected 12 month-to-date heat rate improvement is 12kcal/kWh, and is scheduled to be completed May of 2002, the June 2002 improvement will be 1 kcal/kWh, the July 2002 improvement will be 2 kcal/kWh, etc. The May 2003 and all future month's improvement will be 12 kcal/kWh.

After calculating the impact of each activity for each time period, the target heat rate deviation for each parameter is calculated by adding the basis parameter heat rate deviation to the heat rate impacts for each activity. For example, if the basis heat rate deviation for condenser pressure is 20 kcal/kWh, and there are two activities that have impacts of -2 and -4 kcal/kWh for the time period in question (i.e. Sept 2002), the target heat rate deviation would be 20-2-4 or 14 kcal/kWh.

In determining the target parameter heat rate deviations, only projects with type "I," type "D," and type "A" are considered. (Type "E" will change the expected heat rate, but not the parameter heat rate deviation.)

During the year, it is these targets to which the actual heat rate deviations should be compared. For example if the target is to reduce the heat rate deviation for condenser pressure to 14 kcal/kWh, this is the level of performance that the unit should obtain. Furthermore, for operator controllable parameters, the target parameter level can be determined by starting with the "expected" level, and adding the equivalent parameter value that corresponds to the heat rate deviation. Continuing with the condenser pressure example, if the target heat rate deviation is 14 kcal/kWh, this corresponds to about 7 mm of Hg above expected, so the target parameter value the operator should be obtaining would be 7 mm Hg above the "expected" value. Another example might be for Hot Reheat Steam Temperature at the Intercept Valve. The expected value might be 538 °C, and the target heat rate deviation might be 3 kcal/kWh. Since 3 kcal/kWh corresponds to about 6 °C, the target temperature would be 6 degrees below the expected parameter value, 538 °C - 6 °C or 532 °C.

5.3.4 Calculate the Target Heat Rates

After calculating the target heat rate deviations for each parameter for each time period, the target heat rate for the unit for each time period, can be determined. This is done by summing the basis expected heat rate with all target heat rate deviations, and with all type "E - Expected" activities (activities that will result in a change in the expected parameter level).

Normally the expected heat rate for the basis period is used as the expected heat rate for all future time periods. Since the expected heat rate for a period of time (i.e., a 12 month-to-date period) is a function of the unit loading pattern and the ambient conditions, the expected heat rate for the past 12 months is probably going to be different from the expected heat rate for the next 12 month period. However, accurately predicting future expected heat rates is very difficult. For this reason, the basis period's "expected" is usually used for all periods in the plan. There are some exceptions to this rule. For example, if a unit was continuously derated for some problem, such as having a string of feedwater heaters bypassed, when that problem is corrected, the average load on the unit will increase, and the expected heat rate will decrease. For situations such as this, the expected heat rate for one or more future periods should not be held constant at the level from the basis period, but should be adjusted.

5.3.5 Calculate Unit Group, Plant, and System Roll-ups

Once the plan is developed for each unit, roll-ups can be made to determine target heat rates for groups of units. For example if a plant is composed of both 200 and 500MW units, in addition to the target heat rate for each unit, a target heat rate for the 200 MW units and a target heat rate for the 500 MW units can be calculated by weight averaging the individual heat rates by the projected net generation for each unit. A target heat rate can be calculated in a similar fashion for the entire plant.

If each plant in a region or if all plants in a utility follow the same process, then a target heat rate for the region or system can be calculated. Another advantage of this process, is that it is quick and easy to see the effect on the unit, plant and system heat rate (and therefore fuel cost and production cost) of postponing an activity at any plant, or if additional money is allocated to do additional activities, etc.

5.4 Review the Plan

Building and maintaining the plan is only part of the process. It is vital to hold periodic reviews of the plan. Monthly, or at least quarterly, there should be a formal review of the plan by the plant and utility management. The review should ask and answer the questions addressed in Sections 5.4.1 through 5.4.6.

5.4.1 Are There any Parameters With Significant Heat Rate Deviations With No Corrective Activities?

The plan should be reviewed to ensure that all problem areas are being addressed. If there is a problem area with no planned activities to correct it, a study should be initiated and listed on the plan. Along with the name of the study, the plan should also list the schedule for the study and the person responsible for determining the root cause, possible courses of action, the benefits and costs of each course of action, and recommending the best option.

5.4.2 Are There any Adjustments That Should be Made ?

During the formal review, proposed changes to the plan can be considered, and if approved, the plan should be modified. Any proposed change should be considered carefully, as it will change the target parameter heat rate deviations, target parameter values, and target heat rates. Example of changes would be:

- New activities which have been identified and scheduled should be added to the plan.
- If the schedule for a activity has been changed, the dates in the plan should be changed.
- Activities that are listed on the plan, but are not "selected," should be reviewed to see if there have been any changes (fuel costs, labor costs, etc.) that would now justify selecting them for implementation.

5.4.3 Were Scheduled Activities Started On Time?

All activities that were scheduled to be started since the last review meeting should be checked to see if the activity was started on time. If the activity was not started on schedule, the person responsible should note the reason and what affect (if any) there is to the scheduled completion date.

If the people who are responsible for activities know that they are going to be held accountable for them, the likelihood of the activity being completed as scheduled increases dramatically.

5.4.4 Were Scheduled Activities Completed On Time?

All activities that were scheduled to be completed since the last review meeting should be checked to see if the activity was finished on time. If the activity was not finished on schedule, the person responsible should note the reason and the revised scheduled completion date.

5.4.5 Did the Process Indicator Achieve the Anticipated Result?

After each activity is completed, it should be checked to see if the anticipated change (or target) in the process indicator was met. If the anticipated change was not met, a study should be initiated to determine the reason:

- Was the engineering analysis faulty ?
- Was the activity not carried out as designed ?
- Were some other factors interfering ?

This step is especially important if the activity is scheduled to be replicated on other units. If the anticipated benefits were not achieved, the activity may need to be postponed or deleted from the plan(s) of other units.

5.4.6 Did the Heat Rate Deviation Show the Anticipated Change?

After each activity is completed, it should be checked to see if the anticipated change (or target) in the parameter(s) heat rate deviation(s) was met. If the activity achieved the anticipated change in the process indicator, the parameter(s) heat rate deviation(s) should have also changed as predicted, and this should be confirmed. There are two reasons this would not occur:

- Something else occurred that "masked" the anticipated change. For example, an activity might have been initiated to replace cold end air heater baskets. The process indicator was to improve gas side efficiency by 5 percentage points (decreasing the exit gas temperature by 16 °C) and the 12 month-to-date heat rate impact was 20 kcal/kWh. The process indicator could have been achieved, but the exit gas temperature may not have dropped as much as expected (if at all) because of new problems such as decreased convection pass cleanliness or increased convention pass air inleakage. Therefore, if the anticipated change in a parameter's heat rate deviation does not occur, the cause must be determined.
- There was a mistake in the engineering analysis. If this occurred, it should be found, and corrected. Otherwise future projects could make the same mistake.

5.5 Forecasting

In addition to using this procedure to set "targets," another use is to "forecast" future performance. There is a subtle difference between "targets" and "forecasts."

Targets are goals, usually set once each financial year. They should be aggressive, but achievable with effort. Targets are used during the year and at the year end to see how well the plant team is managing its resources. Targets are used primarily by the plant staff as a goal to be strived for, and with diligence, achieved.

A forecast is a more realistic expectation. Forecasts are used for strategic planning purposes (such as deciding how much fuel should be purchased, etc.) For example, the "target" heat rate for a plant might be 2500 kcal/kWh, that is an achievable value, but it will require that all activities are completed on time and that all activities achieve the anticipated results. Half -way

through the financial year, the same unit may have a "forecast" heat rate of 2575 kcal/kWh because some activities have not been completed or predicated results were not achieved.

One option is to make a copy of the heat rate action plan at the start of the year. One copy will be the target, and the other copy will be used for forecasting. The copy used for forecasting will be modified frequently. As schedules change for any reason, the dates in the forecast plan change. As projects are completed, the actual heat rate changes are entered in place of the anticipated change.

The copy used as the target is sometimes left unchanged during the year. Another option is to make limited changes for effects that are out of the control of the plant personnel, for example, when an outage is canceled/delayed by the utility management. Then, the schedule dates for activities associated with the outage would be changed. Another example would be if an approved activity was canceled by the corporate office.

Heat Rate Process Impro

PLANT: Marvil
UNIT: 1
Date: January 21, 2000

Enter Date for Base Case (Base 12MTD)
Jul-99

List Identified Activities below (If no activity is identified - state WHY)

	Yearly Improvement / Degradation Btu/kWh	Activity Selected Y/N	Type of Impact Trend Step	Activity Cost ($000)	Accounting Corr Change Expected Impacts HR Study	PROCESS INDICATOR VALUE (Temp, Press, eff, etc.) Basis	Change/ Target	Indicator	Units	Verification Method	Enter either the start and end date or the outage code: Outage Code	Start Date	End Date	Responsible person	Historica Deviati FY 98
EXHAUST PRESSURE															96
Yearly Degradation Rate	0														
Brush Cleaning Tubes - Feb, May, Aug	-36	Y	T	$20	I	60	10 C	Clean Factor	%	Monthly test		10/01/99	03/01/00	L Gant	
Repair Identified Air Leaks	-10	Y	T	$10	I	20	-15 C	Air Inleakage	SCFM	CMW Monthly Avg		11/01/99	02/01/00	T Bradford	
1B CCWP Rebuild	-12	Y	S	$50	I	48	8 C	CCWP 1B Flow	GPM	Pump Test		01/01/00	02/20/00	C Marvil	
EXIT GAS TEMPERATURE															Paramet 43
Yearly Degradation Rate	0														
Boiler Soot Blower Rehab Project	-3	Y	T	$50	I	50	90 T	SB Availability	%	12MTD value		10/01/00	09/30/01	E Kopetz	
Repair All Identified Boiler Air Leaks	-3	Y	S	$40	I	0.55	0.1 C	x ratio	-	Airheater test	x107	08/01/99	09/01/99	C Breeding	
Clean AH/Set Seals to Design Clearances	-10	Y	S	$10	I	55	65 T	Gas Side Eff	%	Airheater test	x108	11/13/01	12/16/01	G Hargraves	
Replace Cold-End Baskets w/ High Eff Design		N	S	$50	S							10/01/99	09/30/00	R Helton	
Better Measurement of Air Heater Leakage	-15	Y	S		A	15	13	AH Leakage	%	Airheater test		12/01/99	12/31/99	C McFall	
REHEAT TEMPERATURE															Paramet 13
Yearly Degradation Rate	0														
Repair Attemperator Block and Control Valves	-5	Y	S	$5	I	300	0 T	Time w/ Flow & Temp<997	hours	12MTD Operating Statistics	x109	05/12/03	06/29/03	S Bunnag	
Boiler Soot Blower Rehab Project	-4	Y	T	$50	I	50	90 T	SB Availability	%	12MTD value		10/01/00	09/30/01	E Kopetz	
HP TURBINE EFFICIENCY															Paramet 96
Yearly Degradation Rate	9														
Replace Rotor with Redesigned Rotor	-76	Y	S	$750	I	82	4 C	HP Turb Eff	%	Turbine Test	x107	08/01/99	09/01/99	T Gaither	
Replace Rotor with Redesigned Rotor	-76	Y	S	$2,000	E	86	4 C	HP Turb Eff	%	Turbine Test	x107	08/01/99	09/01/99	T Gaither	
IP TURBINE EFFICIENCY															Paramet 31
Yearly Degradation Rate	5														
IP Turbine Overhaul	-30	Y	S	$750	I	87	88.5 T	IP Turb Eff	%	Turbine Test	x107	08/01/99	09/01/99	T Gaither	
COMBUSTIBLES IN FLYASH															Paramet 23
Yearly Degradation Rate	0														
Repair All Identified Boiler Air Leaks	-5	Y	S	$40	I	0.55	0.1 C	x ratio	-	Airheater test	x107	08/01/99	09/01/99	C Breeding	
Adjust Classifiers Weekly to Maintain Proper Fineness Fineness	-5	Y	S	$3	I	66 / 99.0	72 / 99.5 T	200 mesh / 50 mesh fineness	% passing	Min from Weekly Fineness Test		01/01/00	04/20/00	J Workman	
Low-NOX Burner Retrofit / Reduced O_2	20	Y	S	$500	I	0.5	0.2	% Pts > Exp	% pts	Op. Statistics	x109	05/12/03	06/29/03	E Sikes	
UNACCOUNTABLE DEV.															Paramet 202
Yearly Degradation Rate	5														
LP Turbine Overhaul	-25	Y	S	$750	I	80	81.5 T	LPT Eff	%	Steam Path Audit	x107	08/01/99	09/01/99	T Gaither	
Better Measurement of Air Preheater Leakage	15	Y	S		A	15	13	AH Leakage	%	Airheater test		12/01/99	12/31/99	C McFall	
SUPERHEAT ATTEMPERATION															Paramet 46
Yearly Degradation Rate	0														
Repair Attemperator Block and Control Valves	-5	Y	S	$5	I	300	0 T	Time w/ Flow & Temp<1047	hours	12MTD Operating Statistics	x109	05/12/03	06/29/03	S Bunnag	
Boiler Soot Blower Rehab Project	-4	Y	T	$50	I	50	90 T	SB Availability	%	12MTD value		10/01/00	09/30/01	E Kopetz	
															Paramet

Projected Unit Performance Summary:

	Solution Total Cost ($000)	FY 98
Heat Rate Deviation with no Solution		551
Impacts to Heat Rate	$0	
Accounting Changes	$0	
Heat Rate Deviation with Solutions	$0	
Reported Expected Heat Rate Base Year		
Changes to Expected Levels	$0	
New Expected Heat Rate		
Projected Net Heat Rates	$0	
Total Study	$0	

Figure 5.1 Left Half of a Typical Heat Rate Action Plan

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vement Worksheet

Enter Outage Code and Outage Start and End Dates

Code		x107	x108	x109											
Start Date		08/01/99	11/13/01	05/12/03											
End Date		09/01/99	12/16/01	06/29/03											

112 MTD Heat Rate ous from Expected

Target 12 Month to Date Heat Rate Deviations

FY 99 Target	BASE : 12 MTD Jul 99	2000															
		Oct-99	Nov-99	Dec-99	Jan-00	Feb-00	Mar-00	Apr-00	May-00	Jun-00	Jul-00	Aug-00	Sep-00	2001	2002	2003	2004
51	141	141.0	141.0	141.0	141.0	141.0	141.0	141.0	141.0	141.0	141.0	141.0	141.0	141.0	141.0	141.0	141.0
		0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
		0.0	-0.5	-1.5	-2.9	-4.9	-7.3	-10.2	-13.6	-16.0	-19.0	-21.9	-24.8	-35.0	-35.0	-35.0	-35.0
		0.0	0.0	-0.2	-0.6	-1.3	-2.1	-2.9	-3.8	-4.6	-5.4	-6.3	-7.1	-10.0	-10.0	-10.0	-10.0
		0.0	0.0	0.0	0.0	0.0	-1.0	-2.0	-3.0	-4.0	-5.0	-6.0	-7.0	-12.0	-12.0	-12.0	-12.0
er HR dev from ex		141.0	140.5	139.3	137.5	134.9	130.6	125.9	120.6	116.4	111.6	106.9	102.1	84.0	84.0	84.0	84.0
26	36	36.0	36.0	36.0	36.0	36.0	36.0	36.0	36.0	36.0	36.0	36.0	36.0	36.0	36.0	36.0	36.0
		0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
		0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	-1.4	-3.0	-3.0	-3.0
		-0.3	-0.5	-0.8	-1.0	-1.3	-1.5	-1.8	-2.0	-2.3	-2.5	-2.8	-3.0	-3.0	-3.0	-3.0	-3.0
		0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	-7.5	-10.0	-10.0
		0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
		0.0	0.0	0.0	-1.3	-2.5	-3.8	-5.0	-6.3	-7.5	-8.8	-10.0	-11.3	-15.0	-15.0	-15.0	-15.0
er HR dev from ex		35.8	35.5	35.3	33.8	32.3	30.8	29.3	27.8	26.3	24.8	23.3	21.8	16.6	7.5	5.0	5.0
10	27	27.0	27.0	27.0	27.0	27.0	27.0	27.0	27.0	27.0	27.0	27.0	27.0	27.0	27.0	27.0	27.0
		0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
		0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	-1.3	-5.0
		0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	-1.8	-4.0	-4.0	-4.0
er HR dev from ex		27.0	27.0	27.0	27.0	27.0	27.0	27.0	27.0	27.0	27.0	27.0	27.0	25.2	23.0	21.8	18.0
46	73	73.0	73.0	73.0	73.0	73.0	73.0	73.0	73.0	73.0	73.0	73.0	73.0	73.0	73.0	73.0	73.0
		2.3	3.0	3.8	4.5	5.3	6.0	6.8	7.5	8.3	9.0	9.8	10.5	13.5	28.5	37.5	46.5
		-6.3	-12.7	-19.0	-25.3	-31.7	-38.0	-44.3	-50.7	-57.0	-63.3	-69.7	-76.0	-76.0	-76.0	-76.0	-76.0
		-6.3	-12.7	-19.0	-25.3	-31.7	-38.0	-44.3	-50.7	-57.0	-63.3	-69.7	-76.0	-76.0	-76.0	-76.0	-76.0
		0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
er HR dev from ex		68.9	63.3	57.8	52.2	46.6	41.0	35.4	29.8	24.3	18.7	13.1	7.5	16.5	25.5	34.5	43.5
6	43	43.0	43.0	43.0	43.0	43.0	43.0	43.0	43.0	43.0	43.0	43.0	43.0	43.0	43.0	43.0	43.0
		1.3	1.7	2.1	2.5	2.9	3.3	3.8	4.2	4.6	5.0	5.4	5.8	10.8	15.8	20.8	25.8
		-2.5	-5.0	-7.5	-10.0	-12.5	-15.0	-17.5	-20.0	-22.5	-25.0	-27.5	-30.0	-30.0	-30.0	-30.0	-30.0
er HR dev from ex		41.8	39.7	37.6	35.5	33.4	31.3	29.3	27.2	25.1	23.0	20.9	18.8	23.8	28.8	33.8	38.8
18	33	33.0	33.0	33.0	33.0	33.0	33.0	33.0	33.0	33.0	33.0	33.0	33.0	33.0	33.0	33.0	33.0
		0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
		-0.4	-0.8	-1.3	-1.7	-2.1	-2.5	-2.9	-3.3	-3.8	-4.2	-4.6	-5.0	-5.0	-5.0	-5.0	-5.0
		0.0	0.0	0.0	0.0	0.0	0.0	0.0	-0.4	-0.8	-1.3	-1.7	-2.1	-5.0	-5.0	-5.0	-5.0
		0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	5.0	20.0
		0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
er HR dev from ex		32.6	32.2	31.8	31.3	30.9	30.5	30.1	29.3	28.4	27.6	26.8	25.9	23.0	23.0	28.0	43.0
174	53	53.0	53.0	53.0	53.0	53.0	53.0	53.0	53.0	53.0	53.0	53.0	53.0	53.0	53.0	53.0	53.0
		1.3	1.7	2.1	2.5	2.9	3.3	3.8	4.2	4.6	5.0	5.4	5.8	10.8	15.8	20.8	25.8
		-2.1	-4.2	-6.3	-8.3	-10.4	-12.5	-14.6	-16.7	-18.8	-20.8	-22.9	-25.0	-25.0	-25.0	-25.0	-25.0
		0.0	0.0	0.0	1.3	2.5	3.8	5.0	6.3	7.5	8.8	10.0	11.3	15.0	15.0	15.0	15.0
		0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
er HR dev from ex		52.2	50.5	48.8	48.4	48.0	47.6	47.2	46.8	46.3	45.9	45.5	45.1	53.8	58.8	63.8	68.8
38	49	49.0	49.0	49.0	49.0	49.0	49.0	49.0	49.0	49.0	49.0	49.0	49.0	49.0	49.0	49.0	49.0
		0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
		0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	-1.3	-5.0
		0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	-1.8	-4.0	-4.0	-4.0
er HR dev from ex		49.0	49.0	49.0	49.0	49.0	49.0	49.0	49.0	49.0	49.0	49.0	49.0	47.2	45.0	43.8	40.0

FY 99 Target	BASE : 12 MTD Jul 99	2000															
		Oct-99	Nov-99	Dec-99	Jan-00	Feb-00	Mar-00	Apr-00	May-00	Jun-00	Jul-00	Aug-00	Sep-00	Sep-01	Sep-02	Sep-03	Sep-04
370	455	460	461	463	465	466	468	469	471	472	474	476	477	496	515	534	553
		-12	-24	-36	-50	-64	-80	-96	-113	-130	-146	-163	-180	-206	-220	-220	-212
		0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
		448	438	427	415	402	388	373	357	343	328	312	297	290	296	315	341
	9241	9241	9241	9241	9241	9241	9241	9241	9241	9241	9241	9241	9241	9241	9241	9241	9241
		-6	-13	-19	-25	-32	-38	-44	-51	-57	-63	-70	-76	-76	-76	-76	-76
		9235	9228	9222	9216	9209	9203	9197	9190	9184	9178	9171	9165	9165	9165	9165	9165
	9696	9683	9666	9649	9630	9611	9591	9570	9548	9527	9505	9484	9462	9455	9461	9480	9506

Figure 5.1 (continued) Right Half of a Typical Heat Rate Action Plan

SECTION 6 TOOLS

6.1 Introduction

This section discusses the "tools" that should be readily available to the person(s) responsible for monitoring the thermal performance of a unit. Many of the necessary tools have already been discussed in detail in other sections of this guidelines, such as detailed test procedures for major pieces of equipment, qualifications cards for ensuring that test personnel are trained and capable of properly using test equipment, generic heat rate logic trees (Appendix E), calculation procedures for determining the impacts of various parameters on unit performance, etc. This section discusses additional tools that should be at the disposal of the "performance engineer." Some of this information is contained in "Technical Diaries or Plant Detail Books" that are developed for a particular project, but for detailed analysis of thermal performance, additional data must be available.

6.2 Comprehensive Thermal Kit

One of the most valuable sources of information on the turbine cycle is the "thermal kit." It is usually supplied by the turbo-generator supplier. It is a collection of curves primarily, with some data, that describes the operating characteristics of the turbo-generator.

6.2.1 Heat Rate/Load Correction Curves

The turbine supplier usually provides a group of curves that describes the effect on the turbine cycle heat rate and gross load, of changes in various "external parameters" or boundary conditions, such as main steam temperature and pressure, hot reheat steam temperature at the intercept valves, pressure drop through the reheater, and condenser pressure. Occasionally, other curves will be provided for attemperation flows, feedwater heater TTDs and DCAs, etc. These curves are usually plotted with the parameter value on the x-axis, and percent change in heat rate (or gross load) on the y-axis. At the design parameter value, the curve(s) will cross the x-axis (where the percent change in heat rate is zero). Sometimes, for a parameter, multiple curves will be given, with each curve for a specified steam flow, or for a specified control valve opening.

6.2.2 Generator Loss Curves

Another curve (or family of curves) that is given is the generator loss curves. These curves represent the amount of energy loss in the generator. The turbine produces a certain torque on the shaft, and the generator converts 98-99% of that energy to electrical energy. The other 1-2% is "generator losses", which includes both mechanical and electrical losses. The amount of loss is given in different formats. The y-axis is almost always the generator losses in kW. Sometimes the x-axis is the generator output (real power) in kW (or MW) and a family of curves will be

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given for different power factors (the higher the power factor, the larger the loss). Other times the x-axis will be apparent power in kVA (or MVA). Other variations include :

- Sometimes the fixed losses are included in the curve, and sometimes it is tabulated.
- Since the losses also vary with the hydrogen pressure, sometimes a family of curves is given for various hydrogen pressures, sometimes an additional curve is given with a correction factor as a function of the hydrogen pressure.

6.2.3 Exhaust Loss Curves

When the steam passes through the last row of rotating blades in the LP turbine, it has a very high velocity (kinetic energy). As it turns down and slows, this kinetic energy is converted to an increase in enthalpy. The enthalpy of the steam leaving the last row of blades is called the Expansion Line End Point (ELEP). The enthalpy of the steam that is condensed in the condenser is the Used Energy End Point (UEEP). The difference between these two is the exhaust loss. (It also includes losses due to friction, and for very low velocities a rotational loss.)

These curves are usually plotted one of two ways. First, as a single curve as a function of the velocity of the steam (the velocity must be calculated using the mass flow, and the specific volume of the steam and the annular area). The second type of plot is of a family of curves versus exhaust flow, with each curve for a different condenser pressure.

6.2.4 Mollier Diagram Showing Turbine Expansion Lines

Another drawing that should be included in the thermal kit is a large Mollier diagram (enthalpy versus entropy plot) showing the anticipated turbine expansion lines for the entire turbine (HP and IP-LP). There should be multiple curves for several main steam flows. (Note that the end of each curve is the ELEP, not the UEEP.)

6.2.5 Curves of Stage Pressure Versus Flow to the Following Group

Other data that is usually provided is the relationship between the pressure ahead of a turbine stationary blade and the flow through that section. This relationship should be given for the first stage, and for each extraction point. If the relationship is given as a curve, it will have flow through the section on one axis and either pressure or flow divided by the square root of pressure divided by specific volume $\{Q / \sqrt{(p/v)}\}$.

Sometimes, just the values of $\{Q / \sqrt{(p/v)}\}$ are tabulated for the first stage and for each extraction point.

6.2.6 Turbine Packing Leakoff Curves or Constants

In order to determine the flow from turbine gland leakages and from valve stem leakages, a similar relationship as described above is usually given. For each gland or valve stem leakage, a "packing constant" is usually given, again where the leakage flow is calculated by multiplying the constant by the square root of the pressure divided by the specific volume $Q = C * \sqrt{(p/v)}$. Occasionally, instead of specifying packing constants, these flows will be given as a curve as a function of the pressure ahead of the leakage.

6.3 Plant Technical Information

In addition to the technical information provided in the thermal kit by the turbo-generator provider, there are other specifications and technical data that should be collected, made readily available, and kept current as equipment is replaced or refurbished. Preferably, the items listed below, along with the information from the thermal kit, should be collected into a single notebook.

6.3.1 Heat Balance Diagrams

Usually, several heat balance diagrams are provided for a range of steam flows and condenser pressures. Occasionally, some additional heat balance diagrams are also provided to some special conditions, such as the top heater out of service, or over pressure. Additionally, if a thermodynamic model is built for the plant, additional diagrams will be generated. All these should be kept together.

6.3.2 Flow Diagrams/Piping and Instrumentation Diagrams (P&ID)

The performance engineer should have a full set of flow or P&ID drawings showing all steam, water, air and flue gas streams. These drawings should include pipe sizes, locations of station instruments and test points.

One set should be marked up showing which valves/traps/etc. are in each cycle isolation checklist. (Section 12.4 describes breaking down the potential cycle isolation problems into 2-4 categories, depending on the potential severity.)

Another set of drawings should be marked up showing potential sources for condenser air inleakage. These potential inleakage locations would be marked one of two ways: locations that are always under vacuum, and locations that are under vacuum only at reduced load.

6.3.3 Pump and Fan Curves

In order to evaluate the performance of large pumps and fans, the curves of head, power and efficiency versus flow should be provided, along with supplemental data such as the speed(s) for which the curve(s) were drawn, the temperature, pressure and density for the fluid, impeller size, etc. It is preferable to have curves based on tests, but that is not always possible, especially for large pumps. The pumps and fans for which curves should be available include:

Pumps	Fans
Boiler Feedwater	Forced Draft
Boiler Feedwater Booster	Primary Air
Condensate Extraction	Gas Recirculation
Condenser Circulating Water	Induced Draft
Boiler Circulating Water	
Heater Drain	
Bottom Ash	
Fly Ash	
Raw Water	

Some boiler feedwater pumps have a balancing drum leakoff that includes an orifice for measuring the leakoff flow. If the pumps have this design, there should be a curve provided showing the relationship between the differential pressure and the flow rate.

6.3.4 Primary Flow Elements (Nozzle/Orifice) Specification Sheets

In order to properly convert the differential pressure across a nozzle or orifice to a flow rate the following characteristics are required:

- Pipe internal diameter (ID) at nozzle (cold condition)
- Pipe material (and, therefore, coefficient of thermal expansion)
- Flow element type (ISA nozzle, long radius nozzle, standard orifice plate, square edge orifice, etc.)
- Flow element (Nozzle or Orifice) ID
- Flow element material (and therefore coefficient of thermal expansion)
- Type of pressure taps (for orifices) such as corner, flange or D & D/2 taps

This information should be collected and tabulated in one location for all flow elements such as:

- Total feedwater flow
- Feedwater flow through individual pumps
- Condensate flow
- Attemperation flows
- Makeup flows
- Main steam flow
- Reheat steam flow

- IP turbine cooling steam flow
- Heater drains
- etc.

6.3.5 Specification Sheets and Drawings on Heat Exchangers

Each heat exchanger in the plant (especially condensers, feedwater heaters, external drain coolers, gland steam condensers) should have a detailed specification sheet, with the following information:

- Tube material(s)
- Tube actual and effective tube length
- Tube ID or wall thickness
- Number of passes
- Heat transfer rate(s) (for feedwater heaters a rate should be specified for each zone, desuperheating, condensing and drain cooling)
- Effective surface area(s)
- Design conditions (temperature, pressure and flow rates) of each stream in and out
- Design Performance (TTDs, DCA, temperature rises, LMTDs, etc.)
- Pressure drop on the water side ands in each zone on the shell side

Drawings showing the "tube map" should be provided, as well as sectional drawings showing the locations of baffles, shrouds, vents, sight glasses and normal water level marks.

6.3.6 CCW System Hydraulic Gradient Curves

In the design of the condenser circulating water system, curves have to be developed to show the hydraulic grade (the sum of the elevational energy and the pressure energy of the CCW) of the fluid flowing through the system. The hydraulic gradient curve is usually plotted over a sectional drawing of the CCW system, starting at the intake structure and ending at the outfall. Normally, several curves are drawn for various combinations of the number of pumps in service and the elevation of the water source at the intake structure. These curves are invaluable in resolving flow problems in CCW systems.

6.3.7 Water Leg Measurements

Most local pressure gauges and pressure transmitters do not read the true pressure of the steam/water. Instead, they read slightly higher, due to the water that fills the impulse tubing between the pipe and the instrument. For normal operation, the difference usually is not significant, but for high accuracy measurements, it can be. In order to be able to correct for these "water legs," a table should be maintained that lists either:

- the elevation of the pipe taps (and the floor elevation), or
- the distance from the pipe tap to the floor, or
- the distances from the pipe to the local gauge/transmitter and the elevation of the local pressure gauge.

At some plants, the gauges/transmitter are calibrated to take into account the water leg. Whether the calibration includes the water leg or not should be identified, so that test measurements, which almost always must be corrected for water legs, can be compared to the station readings.

6.3.8 Boiler and Air Heater Guaranteed Performance Sheets

The performance engineer should have detailed "predicted" performance sheets on the boiler/air heater. This would include data such as air and gas, temperatures and pressures, entering and leaving the air heater (and air preheater if provided); flue gas and steam/water, temperatures and pressures in and out of each section of the boiler (economizer, waterwall, secondary superheater, reheater, and primary superheater), excess air requirement, proximate and ultimate analysis, grindability, ash analysis, reducing and oxidizing ash fusion temperatures of the coal, boiler efficiency loss calculations, number of mills in service, air temperature entering mills, coal-air temperature leaving mills, mill fineness, material, surface area and tube spacing details of each section, etc.

6.3.9 Control System Logic Diagrams/Controller Set Points

A set of drawings should be maintained and available for the performance engineer showing the logic diagrams and set points for the unit controls. This data is especially critical as more equipment is being controlled by complex logic.

6.4 Performance Guarantee and Other Tests Reports

The previous information has all been "design" data. While it is useful, it does not represent the actual initial condition and operating characteristics of the unit. That is why it is imperative that the performance guarantee tests should do much more than just settle contractual issues. It is the best opportunity to collect high quality data on the unit, to which future operating data and performance will be compared.

Also, any time a test is run on the unit, a test report with not only a summary of the results and recommendations, but also containing all data, should be issued and kept. In addition, a computer database can be made for storing and trending the results of "routine" performance tests, such as turbine efficiency, condenser performance, auxiliary power usage, air heater leakage, pulverizer fineness, etc.

6.5 Historical Operating Statistics

In addition to high quality test data and results, it is advantageous to retain operating data from the station instruments.

6.5.1 Record of all Control Room, DAS, etc. Data from Initial Operation

A valuable resource for future years is to record data from every station instrument, at various loads, immediately after the unit achieves steady operation. This includes all points on the DAS (if applicable), as well as local gauges.

One use of such data is when there are any very slow, gradual changes in the performance of the unit, that may not be noticed. However, when current data is compared to the data when the unit was new, any gradual changes in performance will become apparent.

6.5.2 Retention of Key Indicators

With the advent of low cost computers and storage media, it is cost effective to collect and retain large amounts of operating data. A detailed data storage strategy should be developed. For example some critical DAS data might be kept complete (every value from every scan) for 6 months, then reduced to hourly average/maximum/minimum, and these three hourly values retained for 2 years, then only daily averages retained after that. Other data might be reduced to daily averages each day, and those might be retained for a few months only.

The key performance indicators (see Section 2) should be considered along with the requirements of other departments, and an appropriate data storage plan developed.

6.5.3 Historical Load Patterns

If important indicators are stored as described in section 6.5.2, then many types of analyses are possible. One such type is the generation of a load pattern. This shows how much time a unit (or group of similar units) spends in various load ranges. This information is required for determining the economic benefits for many potential heat rate improvement projects. For some projects, the benefits may only occur when the unit is operating in some narrow load range, or the benefit may vary with the load.

6.6 Maintenance Data

Along with operating data, maintenance data is very useful to the performance engineer.

6.6.1 Steam Path Audits

A very valuable tool to the performance engineer is a steam path audit of the turbine's steam path. These audits are usually performed when the turbine is first opened, before any other work is done. It involves measuring seal clearances, surface roughness, amount of erosion, mechanical damage, etc. This information is then fed into a computer program that estimates the heat rate and load penalty of each non-design seal clearance as well as any other defects. From this information, it is possible to minimize the cost of the turbine overhaul by only performing the work that is economical. For example, a small rub to a spill strip may cost more to replace, than the heat rate and load impact.

Once the decision is made as to which repairs are to be made, that information can be entered, and the predicted section efficiency and load capability determined.

6.6.2 Boiler Inspection Reports

Each time the boiler is inspected (wall thickness readings taken, tube samples cut out, etc.) the reports on what was found, recommendations, etc., should be published and made available to the performance engineer.

6.6.3 Air Heater Inspection Reports

The performance engineer should have access to the as-found and as-left seal clearances of the air heater, for comparison to the results of leakage tests.

6.6.4 History of Cycle Isolation Problems

With the large number of valves that can contribute to cycle isolation problems, having a database of which valves have caused problems in the past is useful to help determine where to look first, or which valves should be monitored continuously (because they frequently leak) and which may only require periodic monitoring. Also, if the database includes the type of valve, then there may be some correlation between the valve type (or manufacturer) and frequency of leaks.

6.6.5 Heat Exchanger Tube Pluggage History

The performance engineer should have a record of the number and location of tubes plugged in all heat exchangers.

6.7 Build Thermodynamic Model of the Plant

With the advent of computers and special purpose software, the tedious and time consuming manual effort required to make a "heat balance calculation" has been eliminated. In the past to determine the effect on the cycle (temperatures, pressures, flow, generator output, heat rate, etc.) of special conditions (such as bypassing feedwater heaters) or potential modifications (such as removing some superheat surface area) some "simplified" method usually would be used, which might or might not result in an answer that matched the true effect. The alternative to using some approximation was to go through the laborious process of generating a new heat balance diagram. The "simplified" method certainly would not show all the effects on other components. For example, if a "rule of thumb" was used to estimate the heat rate impact of bypassing the lowest pressure HP FW heater, it would not determine details such as what will be the pressure in the other heaters, and how much drain flow will they each have, etc. To get this information, a full heat balance calculation is required.

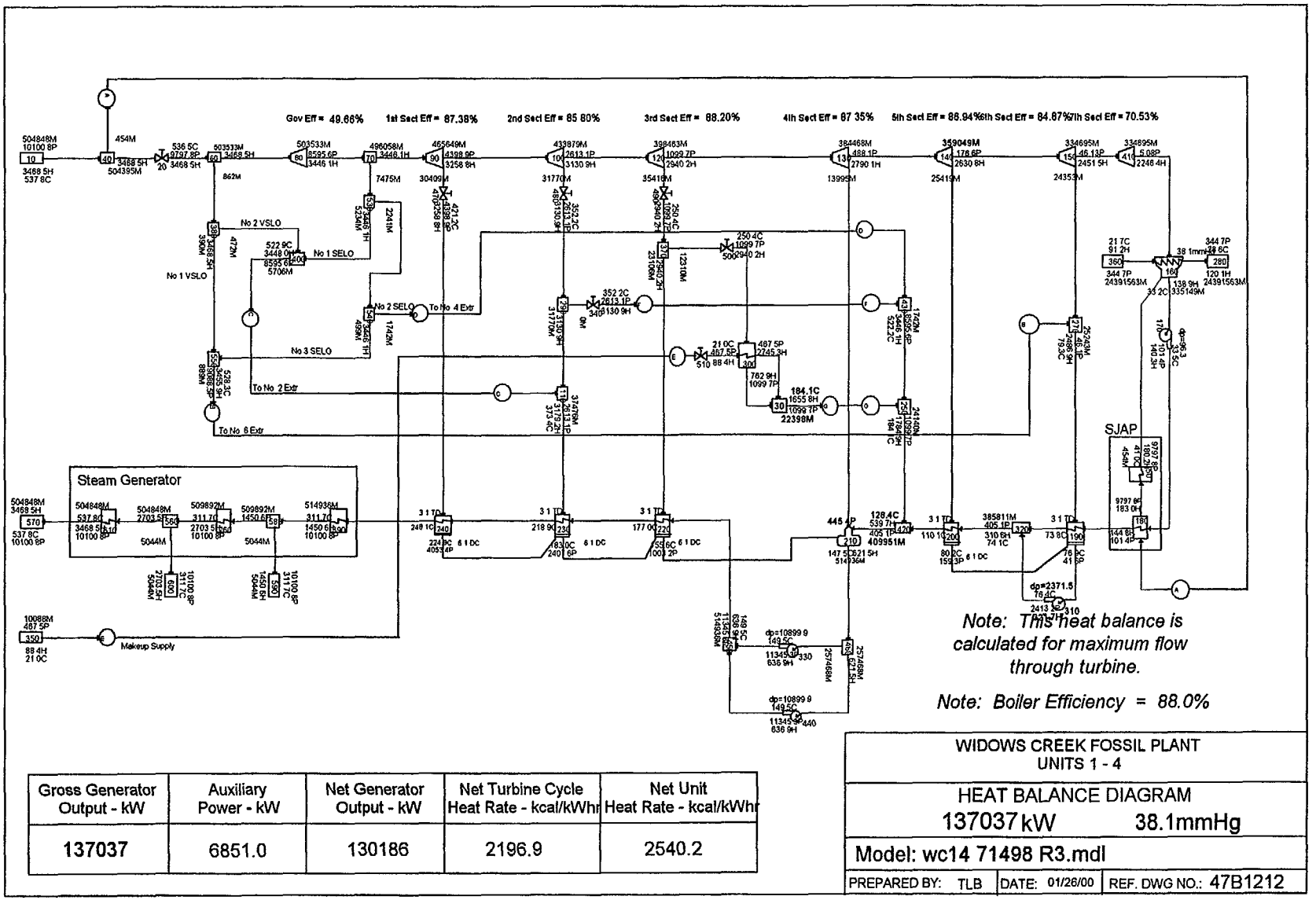
However, in recent years, several vendors have developed and offer computer programs for performing heat balance calculations. These programs, when used properly, will give accurate results in a matter of seconds. Some of the categories of uses include:

- Generating heat rate correction factors for various parameters such as attemperation flows, makeup flow, auxiliary steam usage, final feedwater heater temperature, etc.
- Confirming the accuracy of heat rate correction factors provided by vendors, (i.e. main steam temperature and pressure, hot reheat temperature, reheater pressure drop, condenser pressure, etc.).
- Analyzing test data (correcting results to some contract or design boundary conditions, estimating LP turbine performance, etc.).
- Determining the impacts of abnormal operating conditions, such as feedwater heaters out of service, leaking high energy drains, running two SJAEs instead of one, etc.
- Determining the impacts of equipment degradation, such as poor turbine efficiency, subcooling in the condenser hotwell, high feedwater heater TTDs or DCAs, etc.
- Determining the impacts of potential equipment modifications, such as retubing feedwater heaters or condensers with a different material, adding or removing surface in the superheater or reheater, changing the source of the superheater spray water from BFP outlet to final feedwater, etc.
- Determining the impacts of potential equipment modifications such as changing from full pressure to variable pressure, changing set points on controls, etc.

The first step is to "build" a model of the cycle (usually boiler and turbine models are developed independently, they can be linked later). This is done by first drawing a schematic diagram of the cycle, showing each turbine section, heat exchanger, points where the flow stream splits and where two or more flow streams mix, pump, etc. Next the "components" are connected by flow streams. Then detailed information about the physical attributes and operating characteristics of each component and flow stream is entered, such as the number of tubes and tube material in the condenser, the pressure drop in extraction piping, the pressure versus flow to the following group constant of each turbine section, etc. Assembling this information is not always an easy or straightforward task but, fortunately, it only has to be done once.

The second step, after the model has been drawn, and the component attributes and characteristics entered, is to "fine tune" the model by making minor adjustment to the inputs, so the output of the model (temperatures and pressures of all flow streams, generator output, heat rate, etc.) match some "baseline" data (which could be the design diagram, performance guarantee data, or current operating data).

Third, after the model is built and verified, it can be used as described above.



Gross Generator Output - kW	Auxiliary Power - kW	Net Generator Output - kW	Net Turbine Cycle Heat Rate - kcal/kWhr	Net Unit Heat Rate - kcal/kWhr
137037	6851.0	130186	2196.9	2540.2

Figure 6.1 Example of the Output From a Thermodynamic Model for a Non-Reheat Turbine Cycle

SECTION 7 MANPOWER AND TRAINING

7.1 Introduction

This section discusses the manpower requirements and training needed to initiate and maintain a heat rate improvement program. The guidelines given here are very general. The requirements will vary from utility to utility and from plant to plant. Some utilities have a large and technically strong "central staff", with minimal engineering support assigned to a specific plant. Other utilities have a minimal number of technically orientated "central support staff," and a large engineering organization at each plant. Some utilities do most engineering work with their own staff, while others make extensive use of OEMs or other contractors. Any of these approaches can be used, and there is not one that is superior to another for all circumstances. However, this section examines some of the functions that are required for a heat rate improvement program, the type of personnel that are normally assigned those responsibilities, and the training the personnel will require.

7.2 Plant Personnel

Each power plant has different staffing requirements. This section list some of the types of personnel and the work that is typically performed at each plant.

7.2.1 Statistician

Every heat rate improvement program requires a vast amount of information. Without accurate, timely data, guesses are made instead of informed decisions. Before computers became common place in power plants, the manual collection, filtering, storage and retrieval of data was an enormous task, and therefore much valuable data was not maintained. Computers have greatly improved these processes. Today much more data can be collected and maintained with fewer people. However, even the best systems are not fully automated. There is still the need for human intervention.

- Some data still must be logged manually (even if it is eventually transferred to a computer).
- Questionable data (that is outside the normal band) must still be checked to see if it is correct and should be saved, or if it should be discarded.
- Missing data must sometimes be "estimated" so calculations can be performed.
- Calculations must be manually performed or computer programs run and the results checked for reasonableness.
- Long term data storage and retrieval requires some manual work.

At some plants these activities are carried out by one or more personnel whose primary responsibility is the maintenance of these statistics. (This is not to say that all departments are not involved in the collection of data, only that all data is eventually supplied to one organization.) At other plants, these responsibilities are spread out over various persons (and sometimes various departments), and the collection and maintenance of this data is only part of their responsibilities. Either method can be successful. However, when data collection/maintenance is a secondary function, frequently, it is not given the attention that it requires. When this happens, critical information can be lost, or worse it may be corrupted. This can lead to wrong conclusions and actions.

Another advantage of having an individual or small organization responsible for all data is that relationships between bits of information can be seen and used to validate data. For example, if one organization keeps up with the availability statistics, and another organization keeps up with equipment tag out records, then the first organization might not know about all the equipment worked on during an outage or derating.

7.2.2 Technician

A job category that is often missing from Indian utilities is that of a "technician." This classification covers *skilled* labor that performs many of the heat rate improvement related activities, such as:

- Setting up test instruments, and collecting data.
- Condenser air inleakage leak detection.
- Coal and ash sampling.
- Coal analysis (proximate, ultimate, ash fusion, grindability, fineness, etc.).
- Cycle isolation surveys.
- Infrared scans (boiler casing, etc.).
- Vibration data collection.
- Oil sampling and analysis.

Most of the Indian plants that were visited in this project have some laboratory personnel performing some of these functions, but frequently some of these tasks are performed by unskilled labor under the direction of an engineer or manager. There are problems using unskilled labor, even under the direction of experienced engineers:

- Many tasks, such as pulverizer testing, require close coordination between personnel, and precise movements of equipment in order to get accurate results. Obtaining repeatable results from this type of work requires a highly skilled crew, who are familiar with the plant equipment, familiar with the test equipment, and can work together as a cohesive team.
- There are safety concerns, as persons who are not familiar with the equipment are likely to get hurt.

A better solution, that would result in quicker tests and more accurate results, would be to set up a group of personnel, and train them to perform these tasks.

7.2.3 Engineer

At some plants, one or more engineers is designated as the "Heat Rate" or "Performance" engineer. At other plants, all engineers are theoretically responsible for heat rate activities in their areas of responsibility. Either of these approaches can work. Some of the heat rate related functions of the plant engineers include:

- Reviewing the unit performance each day, including the control charts/trend charts of the "primary process indicators" (see Section 2), and where deviations are noted, assisting with determining the root cause and best course of action to restore optimum performance.
- Reviewing the heat rate losses, comparing current operation to both "target values" (for how well is the current operation) and also to "expected values"(looking for opportunities for long term improvements)
- Test Program. Usually the engineers are responsible for the plant's test program including developing detailed test procedures, determining test schedules (when various performance tests such as condenser performance or pulverizer fineness should be done), ensuring that the tests are carried out properly, reviewing results, and recommending corrective actions as necessary.
- Heat Rate Action Plan (HRAP). Engineers usually are responsible for much of the HRAP (See Section 5). They recommend activities to be placed in the "Heat Rate Improvement Plan", monitor progress of approved activities, review results to see if the desired results are obtained (and if not, why not).

7.2.4 Maintenance Responsibilities

There are several activities that must be carried out by the maintenance department. This includes both routine activities, that are done on a schedule, and activities that are performed on an as-needed basis. The level of staffing in the maintenance department should be sufficient to perform these activities.

Some of the routine activities that are required, but often get dropped due to time/manpower constraints include :

7.2.4.1 Calibration of Coal Weighing Equipment

The equipment used to measure the quantity of fuel burned must be regularly maintained. This could be batch coal scales, volumetric feeders, gravimetric feeders, or belt scales. All of these devices require periodic maintenance and adjustments to keep the indicated weights accurate.

7.2.4.2 Instrument Calibration

A frequent cause of heat rate losses is inaccurate instrumentation. Full loop calibrations should be performed on a frequent basis for critical measurements, such as throttle temperature and boiler outlet O_2.

7.2.4.3 Pulverizer Maintenance

Another common area where heat rate losses occur is in the fuel supply system. Too often, personnel take the attitude that if the pulverizer is running and grinding coal, all is well. However, if the skirt has holes, the grinding pressure is not correct, fuel line orifices are worn, etc., there will be efficiency losses. A detailed preventative maintenance program that includes performance issues should be in place at every plant.

7.2.4.3 Condenser Maintenance

One piece of equipment that is often overlooked is the steam condenser. Because of this, it frequently is the source of the highest heat rate deviations. Most condensers should be cleaned several times during the year, not just once a year during outages. With the proper equipment, (which is relatively inexpensive), condenser tube cleanliness can be maintained at a high level, improving not only heat rate, but also generation.

7.3 Corporate Departments

There are several heat rate related activities that are typically handled, at least partly, by the corporate staff.

7.3.1 Corporate Heat Rate Group

In most corporate offices, there is a staff whose primary function is overall administration of all heat rate improvement activities. This includes both assisting the plants' staff with their efficiency improvement activities, and monitoring the performance of all units in the system while looking for potential improvements.

One function of this group is to set minimum standards (each plant may expand these minimum requirements as they see fit) for various activities, such as procedures for collection, filtering, reduction, storage and retrieval of data. For example, they would set a corporate standard for the minimum ash sampling frequency, or develop a list of the minimum points that should be control charted or trend charted (see Section 2). This group would also set the minimum information that should be included in each unit's Heat Rate Action Plan, and minimum tracking requirements.

A second typical responsibility of this staff is assisting the plants' staff with developing and maintaining the expected level curves, heat rate correction curves, software, etc., necessary for

calculating heat rate deviations (see Section 3). From the utility perspective, it is important that all units be held to similar "expected" levels and that heat rate deviations be calculated in a uniform manner. If this is done, then comparisons between dissimilar units can be made, and resources (manpower and money) can be applied where the most benefits will result. Another result of having all units' heat rate deviations calculated in a similar manner, is that not only can individual units' losses be "rolled-up" to plant values, but they can also be rolled-up to system values. These system roll-ups are a responsibility of the corporate staff. Then questions such as "Which unit in the system has the most potential for improving the system heat rate (and reducing the system cost) from a high pressure turbine overhaul?" or "How much money can the system save if the system's heat rate deviation due to high condenser pressure is reduced by half?" Unless every unit in the utility is using "expected levels" developed the same way, these types of comparisons are not possible, or will not be correct.

Another responsibility that is usually assigned to the corporate staff is to maintain thermodynamic models of each cycle in the utility (turbine and in some cases boiler). This is not to say that the plant staff cannot build or use these models, only that the corporate staff usually has one or two "experts" in this field who assist the plant's personnel. Due to the complexity of these models, unless a person works with them frequently, it is difficult to stay proficient, and most individual plants do not need to work with the models often enough to maintain a high level of skill.

A very important function is to assist the plants' staff with analyzing the trend/control charts and heat rate deviations. When problems arise, both groups work together to determine the root cause of the problem, and to determine the best solution. Once a course of action is identified, that activity is added to the Heat Rate Action Plan" (see Section 5), along with the schedule for implementation and other related information.

Another function is to help disseminate information between the plants in the utility. If one plant has a problem, the corporate staff should be aware if similar problems have occurred at other plants, and what actions were taken there. Also, if one plant tries something new (equipment, operating procedure, maintenance procedure, etc.) the corporate staff should make the other plants aware of the results where they could be applicable. One way they sometimes do this is to organize short workshops/conferences where the personnel involved in heat rate improvement from each plant can meet and exchange ideas.

7.3.2 Corporate Test Group

Section 4 discusses the details of various types of performance tests that can be used not only to determine equipment condition, but also to help diagnose the cause of deterioration. Some utilities do all their performance testing with a corporate test staff. Other utilities perform all performance testing with the plant staff. Many utilities have both, a corporate test staff for large and contract tests, and each plant having some personnel and equipment for running smaller performance tests.

There are three reasons for having some testing capability at each plant. First, having a group of people at each plant that are trained in performance testing enables the plant to be able to run most types of tests at any time. Second, it allows the test schedule to quickly change as unit conditions change. (For example, if the corporate test crew is at a unit to run a turbine test, but the control valves cannot be fully opened for several days, they cannot move to another type of test as easily, if at all, as the plant crew.) Third, if all necessary testing were performed by the corporate crew, the cost (travel time, meals, lodging, etc.) becomes excessive. However, there are also drawbacks to splitting the responsibility for testing. First, it can be difficult to maintain a qualified test crew at each plant. At any plant, certain tests may not be run very often. Also, the plant personnel running performance tests may have other duties as well, therefore, these people may not be as experienced as a full time corporate test crew would be. This can lead to poor quality tests. The second drawback is that some equipment, which can be expensive to purchase and maintain, must be obtained for each plant, and since many tests are not run frequently, the equipment is under utilized. Each utility weighs these factors and decides how to best obtain the necessary test results.

Major tests, such as full turbine cycle heat rate or boiler optimization tests, where a large number of instruments are required, are almost always carried out by the corporate test crew. As a minimum, a turbine cycle heat rate test should be done before (approximately one year) and immediately after outages where the turbines are inspected. By testing one year before the outage, the results of the test can be used to determine the work scope. Boiler optimization tests are usually carried out any time there is a major change to the boiler cycle. Typical activities that would trigger these tests include changes to the boiler heat transfer area, significant fuel changes, etc.

The corporate group also has the capability of running smaller tests, such as enthalpy drop turbine efficiency, clean and dirty air tests, fan tests, etc. Even at utilities where each plant has a test group, the corporate group is sometimes called in to assist with special tests, train plant personnel, confirm results, or to catch-up if the plant group gets behind.

Another function of the corporate test group is to stay aware of new developments in test equipment and procedures, not only for themselves, but also to pass on this information to the plants' staff.

This group would include both engineers and test technicians. Some test groups also include instrument technicians to assist in maintaining/calibrating test equipment and installation work. All personnel should have completed the qualification cards for the equipment they use and for the types of tests they run. (see Appendix G)

A side benefit of a corporate test group is that they are frequently used as a training unit for new engineers. There are several reasons for temporarily assigning new engineers to this group:

- It provides training in several systems (boiler, turbine, condensate, feedwater, etc.)
- It provides training not only on the mechanical side of the systems, but also on the controls and instrumentation and power measurements.

- It provides training at several plants, where the new engineer can be exposed to various designs, control systems, operating practices, etc., and meet personnel at several plants for future contacts.
- The new engineers are made aware of the importance of thermal efficiency.

7.3.3 Corporate Laboratory

Most utilities have a large amount of test equipment, coal analysis equipment, etc. Most plants have personnel who can maintain and calibrate most equipment, but usually some equipment requires special attention, such as high accuracy test equipment. Also, it is not economically feasible to purchase some infrequently used, special purpose equipment for each plant. Some utilities contract these special laboratory services. Other utilities have an internal laboratory. Advantages of an in-house laboratory include better quality control and reduced turnaround time. These advantages must be weighed against the initial cost of obtaining the equipment and skilled personnel. This laboratory may perform one or more of the following functions:

- Check of new equipment to ensure that it meets specifications (such as test equipment)
- Calibration and / or repair of measuring and test equipment
    - Pressure, temperature, flow, mass and electrical
    - Chemical meters (pH, Conductivity, etc.)
- Oil Analysis
    - Fuel oils
    - Insulating oils
    - Lubricating oils
- Coal / ash analysis
    - Proximate
    - Ultimate
    - Ash chemistry
    - Ash fusion
- Metallurgical
    - Failure analysis
    - Metallography
    - Mechanical tests
    - Chemical tests

7.3.4 Corporate Controls Tuning Group

Another specialized skill is the ability to "tune" modern control systems. With advanced digital control systems, the control of a single parameter (such as steam temperature) may be a function of more than twenty inputs. Tuning these highly complex and intertwined control loops so that the unit operates smoothly and efficiently at all loads and during load changes requires a great amount of knowledge and skill (usually learned through trial and error over several years of on-

the-job-training). Because this function is not required frequently at an individual plant, it is usually assigned to a small corporate group, who specialize in this area, or it is contracted out.

7.4 Training

Training is a never ending activity. This is especially true of technical fields where new equipment, techniques and programs are continually being developed. Because of this, and the fact that power plants are such complex systems, no one is ever fully trained and there is always more to learn in any field. The training that a person needs depends on the work that he is expected to perform. There is some overlap, in that many training programs apply to several different job categories. For example, since heat rate is everyone's responsibility, all plant personnel should have some basic training on heat rate. Other training is specific to particular jobs. For example, the skills and need in-depth knowledge to properly tune a distributed control system is required by only a few personnel, and only the test crew needs to be fully trained in the proper use of a high velocity thermocouple (HVT) probe. (It is probably advantageous that many persons have some cursory knowledge of these activities, but everyone cannot be fully trained to perform all functions.)

There are several sources for training. There are vendors available to teach all the courses discussed here. However, in order to keep the cost to a minimum, frequently these courses are taught by the utility's own personnel. One advantage of this is that the course can be made specific to the equipment and procedures at each plant. Another advantage is that this allows more flexibility in the scheduling. A potential drawback is frequently the persons who are most knowledgeable in an area, are not trained to be teachers. However, usually with some training and practice, a utility's expert can be utilized to effectively pass on their expertise. A third method of instruction that is gaining in popularity is computer based training (CBT). In this method, lessons are available on a computer. Frequently, the lesson includes tests and feedback of results, and even remedial lessons are available. An advantage of this is that the student can take these courses when he has available time. A drawback is that "live" instructors and classes usually generate more interest and exchange of ideas than CBT.

The following sections (7.4.1 – 7.4.6) describe some of the heat rate related training typically provided to power plant personnel.

7.4.1 Plant Systems Overview

Frequently, as part of all new employee's orientation, there is a short training course on each of the systems in a power plant (i.e., feedwater, coal handling, power transmission, etc.). Included in this course are:

- Basics of fluid flow, thermodynamics, heat transfer, AC power
- Turbines, pumps, fans, valves,
- Condensers/circulating water system

- Condensate system
- Feedwater system
- Water chemistry and treatment
- Steam generation
- Instrumentation & control systems
- Electrical power generation
- Electrical power distribution

This type of course is taught different ways. At some plants, this course is taught all at once over a week or more, at other plants, it may be taught one hour a day for several months. Some utilities have much of this material available on computer based training (CBT) where employees can take lessons as their schedule allows.

7.4.2 Basic Thermal Performance

Individuals whose job functions can directly affect the efficiency of the unit are frequently given additional training on heat rate fundamentals, with emphasis on how heat rate is measured, and how it is affected (positively and negatively). This course usually places more emphasis on practical aspects of heat rate improvement with lots of examples of how operators, maintenance personnel, equipment modifications, changes to control loops, etc., can affect the efficiency of the unit, with a minimal amount of basic mathematics and theory. This course is strongly recommended, not only for engineers, but also for operations personnel and maintenance supervisors. Included in this course are:

- What is heat rate, and how is it measured.
- How are losses measured, and an explanation of expected levels and target levels.
- "Rules of thumb" - costs of typical heat rate losses
- Turbine cycle
    - Turbine performance
    - Condenser performance
    - Feedwater heater performance
    - Cycle isolation and auxiliary steam usage
- Boiler cycle
    - Pulverizers
    - Air heaters and air preheaters
    - Factors affecting combustion and heat transfer
- Turbine & boiler Interactions
    - FWH out of service affecting steam flow through reheater
    - Final FW temperature affect on firing
    - Steam extraction for air preheating coils
    - Attemperation
- Auxiliary Power Usage

7.4.3 Advanced Thermal Performance

A third course on advanced thermal performance is recommended for personnel who are deeply involved in heat rate improvement, but after they have had some experience in this field. This course is similar to the basic course, but more in depth. It includes:

* Emphasis and examples on how to work through logic or structure trees to determine the root cause of performance deterioration
* Boiler optimization testing
* Details of how heat balances are generated (manually and computer programs).
* Details on other calculations (how heat rate correction factors are developed, how boiler losses are calculated, pump affinity laws, etc.)
* Examples of interaction of various effects (i.e., how damage in the turbine can affect final feedwater temperature), etc.

7.4.4 Turbine & Boiler Modeling

A course that is very valuable is on thermodynamic modeling. One of the most useful tools available to help performance engineers is computer software for modeling the turbine and or boiler cycle of a power plant. This allows the engineer to quickly analyze test data, generate heat rate correction factors, and evaluate proposed equipment or controls modifications, and other "what if" analysis. These courses are usually taught by representatives from the software vendor, although there are third party vendors.

7.4.5 Equipment / Systems

A useful source of continuing education is advanced courses and workshops on various equipment and systems. Courses on turbines, pumps, valves, condensers, feedwater heaters, pulverizers, air heaters, boiler optimization, etc., are available from several sources. Some manufactures offer courses on their equipment, which go into detail on operation and maintenance practices. Universities, retired utility and OEM personnel, engineering societies such as ASME, and other third parties, also offer valuable courses. These courses are very useful for helping engineers gain a better, more in-depth knowledge of specific areas. They also help keep personnel aware of new equipment and techniques.

7.4.6 Testing

For those persons involved in performance testing, specialized training is required covering several areas including:

- The use of equipment (such as pitot tubes, coal sampling equipment, helium leak detection, etc.)

- How to set-up both the equipment and the unit for each type of test.
- Calculation procedures (which may be manual or computer based)
- How to interpret the results, and recommend corrective actions.

This training can include both class room and field work. In addition, the use of "Qualification Cards" ensures that the personnel handling equipment and running tests are trained. Appendix G lists some of the qualification cards and includes an example of an "Equipment" card and a "Test" card. Equipment cards are specific to a piece of equipment, and frequently are specific to a particular manufacturer and, in some cases, even a particular model. (Because the procedure for handling different models can vary, a qualification card is needed for each model.)

Before a person is "signed-off" on an "Equipment" card, he must demonstrate his ability to properly use the equipment (including any routine calibration and maintenance that it requires), and must have demonstrated his knowledge of the safety and technical aspects of the equipment (under what conditions the equipment can be used, potential problems, limitations, etc.).

Before a person is "signed-off" on a "Test" card, he must demonstrate his ability to lead a crew in running that test. This includes properly setting-up all necessary equipment, setting the unit conditions, collecting the data, calculating the results, analyzing the results, and recommending corrective action.

SECTION 8 HEAT RATE SURVEYS AND AUDITS

8.1 Introduction

Frequently, an organization is asked to go into an unfamiliar plant to:

- Evaluate the current cycle thermal performance and to determine potential areas for heat rate improvements.
- Be an impartial third party observer of a performance guarantee test or similar activity.
- Assist with implementing a thermal performance related program (such as mill testing and performance improvement or condenser tube cleaning).

Another function that is sometimes useful is to "audit" the current state of at plant's "heat rate program." Frequently this is done by an external organization (personnel from the same utility but other plants, or outside consultants).

The functions described above are similar in some ways, yet very different in others. The primary function of a survey is to collect basic information from an unfamiliar plant in preparation for performing (or assisting with) more detailed work later. The primary function of an audit is to find procedures/practices that are exemplary (and should be replicated at other locations) or inadequate (and opportunities for improvement). Because there is significant overlap in the data collection activities, which is the focus of this section, both are mentioned here.

Due to the various designs, equipment configurations, fuel characteristics, etc., going into an unfamiliar plant can be a very difficult undertaking. In order to minimize the amount of time that is taken up in face-to-face meetings with the plant's personnel, a data/drawing request and questionnaire can first be sent to the plant. The plant would be requested to collect certain drawings, reports, and to provide short answers to the questions. Supplying this data in advance of the first site visit saves all parties considerable time.

This pre-trip questionnaire is not intended to be extremely detailed and complete. Instead, it is intended to get basic information on:

- the physical configuration and attributes of the plant.
- the type and detail of data that is available.
- known problem areas.

In many cases, the answers provided to the initial questionnaire will lead to more detailed and pertinent questions to be asked later, usually in a discussion with the personnel most familiar with each area. Most of the initial questions should be able to be answered by either circling one item from a list, or by short (1-6) word phrases. Some questions may

require a detailed answer, and those may be answered in writing, or be held until the site visit, where the issue can be discussed in detail.

The questions have been divided in to three categories:

- Equipment characteristics
- Operation and maintenance
- Heat rate tracking and analysis

The first category could be deleted if the personnel are familiar with the unit.

If the organization has been asked to assist with implementing a specific program (such as mill testing and performance improvement or condenser tube cleaning), then the irrelevant reports/drawings and questions from these lists should be deleted.

8.2 Reports/Drawings

The following list of drawings and reports is very general, and would be useful for most surveys or audits. If a specific activity is planned, then additional information on that specific area would probably be needed. (i.e., if testing of boiler feedwater pumps was planned, then additional drawings on the pumps, information on the feedwater control system, etc., would be requested).

1. Heat Balance Diagrams

2. Flow Diagrams or P&IDs
   - 2.1. Boiler - water/steam side
   - 2.2. Boiler/air heater - air/flue gas side
   - 2.3. Pulverizer - coal and air flow
   - 2.4. Main steam/cold and hot reheat
   - 2.5. Extractions
   - 2.6. Feedwater/BFPs
   - 2.7. Condensate
   - 2.8. Heater drains
   - 2.9. Condenser circulating water system
   - 2.10. Miscellaneous drains (turbine drains, boiler drains, etc.)

3. Specification Sheets
   - 3.1. Boiler guaranteed performance
   - 3.2. Condenser performance specifications
   - 3.3. Feedwater heater specifications

4. Routine Data Collected:
   - 4.1. DAS point list

4.2. Operator log sheets(s) for one day

5. Heat Rate Deviation Reports
   5.1. Most recent weekly or monthly
   5.2. Most recent 12 month to date
   5.3. 12 month to date from same period as above, but previous year

6. Performance Guarantee Test Reports

7. Latest Routine Test Reports

8. Outage Reports:
   8.1. Last boiler outage
   8.2. Last turbine overhaul
   8.3. Renovation and Modernization outage (if any)

8.3 Questionnaire - Equipment Characteristics

1. Unit Characteristics:
   1.1. Design unit gross and net capability, current maximum dependable gross and net capability, and if there is a difference, what is the reason(s)
   1.2. Major equipment changes have been made since initial commercial operation
   1.3. Unit's normal minimum load

2. Turbine:
   2.1. Type of turbine (reaction/impulse)
   2.2. Type of turbine valve control (throttle or nozzle governed)
      2.2.1. If nozzle governed, number of valves
   2.3. Type of load control (Full pressure, hybrid variable pressure, full variable pressure, dual pressure)
      2.3.1. If hybrid variable pressure & throttle governed, what is the fixed valve position?
      2.3.2. If hybrid variable pressure & nozzle governed, at which valve point is the cross-over from full to variable pressure?
   2.4. Steam Conditions:
      2.4.1. Design and normal main steam pressure
      2.4.2. Design and normal main steam temperature
      2.4.3. Design and normal hot reheat steam temperature (at turbine)

3. Boiler:
   3.1. Pulverized coal, cyclone, oil, or gas fired
   3.2. Drum or once through
   3.3. Economizer (Y/N)

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3.4. Single wall, opposed wall , or tangential fired
3.5. Single furnace or dual furnace
    3.5.1. If dual furnace - is one furnace "superheat" and the other "reheat"?
3.6. Forced or balanced draft
3.7. Design coal (from guaranteed performance sheet) proximate analysis, HHV and ash fusion temperatures and typical current coal proximate analysis, HHV, and ash fusion temperatures
3.8. Steam temperature control (which design & which are currently used for both SH & RH):
    3.8.1. Attemperation (Y or N and if yes, the source) :
        3.8.1.1. Reheat
        3.8.1.2. Superheat
    3.8.2. Tilts (Y/N)
    3.8.3. Gas Recirculation / Tempering (Y/N)
    3.8.4. Gas biasing dampers (Y/N)
    3.8.5. Firing bias - furnace to furnace (Y/N)
3.9. Overfire Air Dampers (Y/N)
3.10. Underfire or "AA" dampers (Y/N)
3.11. Design and normal furnace exit gas temperature
3.12. Medium (steam/air) and drive (electric/air) is used for the sootblowers

4. Air Heater/Air Preheater:
    4.1. Tubular or regenerative air heater
    4.2. Bi-sector or tri-sector air heater
    4.3. Steam coil air preheater (APH) (Y/N)?
        4.3.1. If yes, when is it used?
        4.3.2. If yes, what is the design and normal air outlet temperature?
        4.3.3. If yes, what is the normal source of the steam?
    4.4. Performance:
        4.4.1. Design and normal leakage
        4.4.2. Design and normal gas inlet temperature
        4.4.3. Design and normal no leakage exit gas temperature, corrected to design inlet air temperature
        4.4.4. Design and normal x-ratio

5. Pulverizer:
    5.1. Type (bowl, ball & race, or ball and tube)
    5.2. Pressured or suction mills
    5.3. Design and current capacity (t/h)
    5.4. Performance:
        5.4.1. Fineness % passing 200 mesh sieve (normal range)
        5.4.2. Fineness % retained on 50 mesh sieve (normal range)
        5.4.3. Design and normal mill outlet temp

6. Feedwater Heaters:

6.1. Number of strings:
    6.1.1. HP
    6.1.2. LP
6.2. What is the number of heaters per string:
    6.2.1. HP
    6.2.2. LP
6.3. Design and normal final feedwater heater outlet temp

7. BFP:
    7.1. Drive (motor w/frequency control, motor w/ hydraulic coupling, turbine)
    7.2. Do the pumps have a measurable balancing drum leakoff flow?

8. Fans:
    8.1. Does each mill have its own primary air fan?
    8.2. Method of control (motor w/frequency control, motor w/ damper, motor w/inlet guide vane, turbine)

9. Condenser:
    9.1. Number of passes
    9.2. Source of cooling water
        9.2.1. If open system, what is the annual change in the elevation of the source?
        9.2.2. If open system, does the outfall flow in an open channel (Y/N)?
            9.2.2.1. If an open channel, what is the elevation of the outfall with respect to the elevation at the pump inlet?
    9.3. Ball cleaning system (Y/N)
        9.3.1. If yes, how often are balls circulated?
        9.3.2. If yes, what type of balls are used?
    9.4. Backwashing capability (Y/N)
    9.5. Unitized pumps or plant common CCW system
    9.6. Design and normal velocity through the tubes
    9.7. Type of air removal equipment (steam jet, vacuum pump)
    9.8. How and how often are condenser tubes cleaned?
    9.9. Is there a low level alarm on the waterbox level?
    9.10. How much (°C) is the air/steam mixture going to the vacuum pumps/SJAE subcooled?

8.4 Questionnaire - Operation and Maintenance

1. Instruments and Controls:
    1.1. Which of the following control loops are normally in automatic, and (where applicable) what is the set point or manually controlled operating point:
        1.1.1. Furnace draft
        1.1.2. Drum level/FW flow

1.1.3. Steam temperature
1.1.3.1. Tilts
1.1.3.2. Attemperation
1.1.3.3. Others
1.1.4. Primary air header pressure
1.1.5. Mill outlet temperature
1.1.6. Primary air flow
1.1.7. Windbox pressure
1.1.8. Combustion control
1.1.8.1. O_2 trim
1.1.9. Feedwater heater levels
1.1.10. Coordinated master control

1.2. How many O_2 sensors are installed on the unit, and what is their availability?

1.3. How many CO sensors are installed on the unit, where are they located, and what is their availability?

1.4. How is condenser pressure/vacuum measured:
1.4.1. Type of instrument(s)
1.4.2. Pressure tap method (Basket tips/wall tap/pipe inside condenser)

1.5. Steam temperature sensors (main steam and hot reheat at IV)
1.5.1. Type (t/c, rtd, etc.)
1.5.2. Are multiple readings available and compared?

1.6. Instrument calibration:
1.6.1. Pressure:
1.6.1.1. Frequency
1.6.1.2. Method (full loop, wiring only, sensor only, number of calibration points)
1.6.2. Temperature:
1.6.2.1. Frequency
1.6.2.2. Method
1.6.3. O_2:
1.6.3.1. Frequency
1.6.3.2. Method
1.6.4. Coal burned scales/feeder:
1.6.4.1. Frequency
1.6.4.2. Method

1.7. Operating data collected:
1.7.1. DAS
1.7.2. Hand-held data collectors w/data downloaded to a computer
1.7.3. Manual readings:
1.7.3.1. Entered into computer
1.7.3.2. Hardcopy only

2. Auxiliary Equipment:
2.1. How many mills are run at:

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2.1.1. Full load
2.1.2. Normal minimum load

2.2. How many feedwater pumps are run at:
2.2.1. Full load
2.2.2. Normal minimum load

2.3. How many FDF/IDF/PAFs are run at:
2.3.1. Full load
2.3.2. Normal minimum load

2.4. How many condensate extraction pumps are run at:
2.4.1. Full load
2.4.2. Normal minimum load

2.5. How many vacuum pumps/SJAEs are run at:
2.5.1. Full load
2.5.2. Normal minimum load

3. How Is Proper Cycle Isolation Ensured:
3.1. High energy drains
3.2. Heater emergency drains to DA or condenser
3.3. LP heater drains that usually are pumped forward
3.4. HP heater bypass valve(s) holding
3.5. Attemperator valves leaking through
3.6. Other

4. Control Charting/Trend Charting of Primary Process Indicators:
4.1. What indicators are graphed?
4.1.1. How are indicators filtered?
4.2. Is there any incentive program for minimizing out of control/out of specification indicators?
4.3. How often are they reviewed by management?

5. Condenser :
5.1. Normal condenser air-inleakage?
5.2. What triggers a manual tube sheet cleaning or backwashing?
5.3. Does the number of running CCWPs vary, and if so, what is the criteria for deciding how many to run?

6. Turbine Overhauls:
6.1. Frequency
6.2. Steam path audits
6.3. Typical findings (solid particle erosion, seal rubs, deposits (and material), etc.)

7. Boiler
7.1. Normal Air Heater Inlet O_2 (Full Load and Minimum Load)

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7.2. Dampers, how are they operated (fixed position, fixed position as a function of load, regulated to hold some parameter value, etc.):
   7.2.1. Fuel air
   7.2.2. Secondary air
   7.2.3. Overfire air
   7.2.4. Underfire air
   7.2.5. Primary air
7.3. Are the mills biased (firing rate)?
7.4. Sootblowers (wall, convection pass, air heater):
   7.4.1. Availability
   7.4.2. Blowing schedule / triggers
7.5. Does the unit have any problems with slagging/fouling:
   7.5.1. Around burners
   7.5.2. Waterwalls
   7.5.3. Convection pass
   7.5.4. Air heater
7.6. Condition of ash removal equipment:
   7.6.1. Economizer hoppers
   7.6.2. Air heater hoppers
7.7. For coal fired units, under what conditions (i.e., mill combination, unit load, etc.) is oil support used?

8. Pulverizers:
   8.1. How are the tempering dampers checked to ensure that they are not leaking?
   8.2. Is the primary air flow constant, or does it vary with mill loading?
   8.3. How often are orifices in coal pipes replaced?

9. Feedwater Heaters:
   9.1. Method of maintaining/setting water level (fixed level, controlled by DCA, controlled by TTD, etc.)
   9.2. Any heaters bypassed frequently or have excess number of tubes plugged?
   9.3. Any control valve bypass valves routinely open?

10. Are there any control, capacity or maintenance concerns with:
   10.1. Boiler feedwater pumps
   10.2. Induced draft fans
   10.3. Forced draft fans
   10.4. Primary air fans
   10.5. Pulverizers
   10.6. Heater drain pumps
   10.7. Control air
   10.8. Station air
   10.9. Sootblowing air
   10.10. Air heaters

10.11. Condenser air removal equipment
10.12. Condensate extraction pumps
10.13. Circulating water pumps
10.14. Other

8.5 Questionnaire - Heat Rate Tracking and Analysis

1. Heat Rate and Deviation Analysis:
   - 1.1. Heat rate measurement:
     - 1.1.1. Fuel sampled and analyzed:
       - 1.1.1.1. Where is the fuel sampled?
       - 1.1.1.2. What equipment is used to sample (auto sampler, manual full belt cut from a stopped belt, manual scoop from a moving belt, etc.)?
       - 1.1.1.3. How often is the fuel sampled?
       - 1.1.1.4. Where is the fuel analyzed?
       - 1.1.1.5. What procedures are in place to insure that the coal sample is not contaminated or dried before the analysis is performed?
     - 1.1.2. How is the quantity of "fuel burned", determined?
     - 1.1.3. How is station power (ash handling, water plant, coal handling etc.) prorated?
   - 1.2. Heat rate deviation calculation:
     - 1 2 1. For which parameters are heat rate deviations normally calculated?
     - 1.2.2 What is the standard frequency for calculations?
     - 1.2.3. Are calculations based on average during the time period or single point data?
     - 1.2.4. How were expected values determined?
     - 1.2.5. What type of roll-ups are made (i.e., weekly, monthly, 12MTD, etc.)?
     - 1.2.6. What is the standard report format and distribution list?
   - 1.3. Real time monitors:
     - 1.3.1. Does the operator have a real time:
       - 1.3.1.1. Controllable loss indication?
         - 1.3.1.1.1. If yes, which parameters?
       - 1.3.1.2. Heat rate indication?
         - 1.3.1.2.1. If yes, what is the method?
         - 1.3.1.2.2. If yes, what data (if any) must be "assumed"?
       - 1.3.1.3. Boiler cleanliness monitor?
       - 1.3.1.4. Condenser performance monitor?
       - 1.3.1.5. Cycle isolation monitor?
       - 1.3.1.6. Net load or total auxiliary energy (load) indication?
     - 1.3.2. If there is a real time heat rate or controllable loss monitor, how were the target values determined?

1.3.3. If there is a real time heat rate or controllable loss monitor, is there any incentive program for minimizing heat rate or parameter deviations?

1.3.4. Is the monitor linked to any fault trees, etc., to expedite analysis of the problem?

1.3.5. How is this information used by management?

1.3.6. If there is no real time monitor, does the operator have a sheet with "rules-of-thumb" listing the heat rate impact and cost of deviations in controllable parameters?

1.4. Fault/logic trees:

1.4.1. Specific to the unit(s)?

1.4.2. Include "target" values for verification of potential cause?

2. Formal Heat Rate Meetings or Teams Working on Heat Rate Issues:

2.1. Frequency

2.2. Minutes taken (Y/N)

3. Is There a Written Action Plan for Heat Rate Improvements?

3.1. Does it include:

3.1.1. Specific actions?

3.1.2. Responsible person?

3.1.3. Schedule?

3.1.4. Measurable indicator?

3.2. Are periodic reviews held asking:

3.2.1. Are schedules being met?

3.2.2. Are anticipated results being obtained?

3.2.3. Are there any parameters with large deviations with no listed activities in the action plan?

4. Are Physical Inventories Conducted on Coal Piles?

4.1. If yes, what is the frequency?

4.2. If yes, how do the physical inventories compare to the "book" quantity (as percent of coal burned since the previous inventory)?

5. Combustibles in Ash Sampling (Fly Ash and Bottom Ash):

5.1. Where are they collected?

5.2. How are they collected?

5.3. Frequency?

6. Performance Testing:

6.1. How often is each test conducted in a five year period (with station instruments and with test instruments):

6.1.1. Turbine cycle heat rate tests

6.1.2. HP / IP enthalpy drop test

6.1.3. Condenser performance

6.1.3.1. How is CCW flow determined?
6.1.4. HP feedwater heater performance
6.1.5. LP feedwater heaters performance
6.1.6. Boiler feedwater pump
6.1.7. Boiler efficiency or optimization
6.1.8. Coal fineness
6.1.8.1. How is the coal sampled?
6.1.9. Clean air (check for balanced air flow in all coal pipes of a mill)
6.1.10. Dirty air flow/coal flow (check for balanced air and coal flow)
6.1.11. Primary air flow calibration
6.1.12. Air heater leakage
6.1.13. Air heater x-ratio
6.1.14. Furnace exit HVT
6.1.15. Auxiliary power usage

6.2. Are detailed unit specific test procedures available for each test?
6.3. How are personnel using and conducting performance tests, trained and certified?
6.4. How are test results tracked/trended?

8.6 Unit Walkdown

The following list are some of the items to look for during an initial "unit walkdown" in preparation of heat rate surveys or audits.

1. Boiler/Air heater:
   1.1. Lagging/insulation condition
   1.2. Air Ingress points:
      1.2.1. Inspection doors open
      1.2.2. Sootblower packing
   1.3. "Sparklers" entering convection pass
   1.4. Slag
   1.5. Flame appearance:
      1.5.1. Color
      1.5.2. Location
   1.6. Bottom ash appearance
   1.7. Tilts/air dampers in same position in each corner

2. Pulverizers:
   2.1. Spilling large amount of coal

3. Condenser:
   3.1. Condition (hot or cold) of drains entering the condenser
   3.2. Discharge valve position
   3.3. Waterbox sight glasses

3.4. Temperature of air/steam removal pipe
3.5. Differential pressure

4. Feedwater Heaters:
   4.1. Levels
   4.2. Local sight glasses:
      4.2.1. In service and readable
      4.2.2. Proper levels

5. If performance tests have not been run in the past, are test ports available?

6. Steam Leaks:
   6.1. Number
   6.2. Severity

7. Condition of pump seals:
   7.1. Boiler feedwater
   7.2. Condensate extraction?

8. Unit control room:
   8.1. Control loops in manual
   8.2. Do most indications look reliable?
   8.3. Are indications stable, equal (where two or more sides or equipment)?
      8.3.1. Steam temperatures
      8.3.2. Main steam pressure
      8.3.3. Main steam flow
      8.3.4. Draft
      8.3.5. Mill outlet temperature
      8.3.6. Exit gas temperatures
      8.3.7. Windbox pressure
      8.3.8. Primary air flows and pressures

9. General appearance/housekeeping of unit

SECTION 9 NEW PLANT SPECIFICATIONS

9.1 Introduction

While improvements can and should be made to existing Rankine cycle power plants, it is easier to include the items that either directly improve the cycle efficiency or indirectly improve cycle efficiency (by enabling special measurements, tests, etc.) in the original design of the plant. There are many things that can be done only when the plant is being built. Other items can be retrofitted, but are much easier and cheaper to install during the initial construction. This section includes items that should be considered in the design and specifications for a new plant. All items may not apply to all plants, and all items may not be economically feasible at all plants. However, many of the items listed here can be included, at minimal incremental cost, and will result in improved efficiency for the life of the plant.

9.2 Plant Equipment - Boiler

In the boiler area, there are several items that are not included in the "standard" specifications, that, if included, will enable the unit to operate efficiently, or will enable better monitoring of the unit condition.

9.2.1 Boiler Cleanliness Monitor.

There are several Boiler Cleanliness Monitors commercially available today. These systems use temperature and pressure sensors on the water/steam side and temperature sensors on the convection pass gas side of the boiler to determine the cleanliness of different sections of the boiler. This information can then be used to determine which section(s) of the boiler should be cleaned (with sootblowers). The efficiency is improved in two ways. First, by only using the sootblowers that are necessary, when they are necessary, the usage of steam and/or air is minimized. Second, by maintaining high cleanliness and therefore heat transfer rates in all sections, the flue gas exit gas temperature is minimized. An additional benefit of this system, is that by reducing the amount of sootblowing, sootblower erosion is reduced, thus reducing maintenance.

9.2.2 Block Valves at Attemperation Valves.

Historically, one leading cause of reduced efficiency is leaking attemperator valves, especially reheat attemperation valves. Reheat attemperation is sometimes required to maintain proper steam temperatures. However, a leaking attemperator valve causes efficiency losses both from the water that bypasses the high pressure heaters and HP turbine, and also by reducing the reheat steam temperature. These valves see very severe service, and are prone to leak. One solution is to install air operated block valves that work with the control system. The block valves are used

to provide tight shutoff, leaving the control valves to regulate flow (when it is needed). When attemperation is needed, first the block valve opens fully, then the control valve is allowed to open as required. After the control valve goes shut, the block valve is shut automatically. Two additional options that should be considered are the use of multi-stage or drag valves to better handle the large pressure drop, or supplying the attemperator with water from an intermediate stage of the boiler feedwater pump, so the pressure drop across the valve is not as high.

9.2.3 Furnace O_2 and CO Monitors

A relatively new development available today is probes for measuring oxygen (O_2) and carbon monoxide (CO), at elevated temperatures, such as those encountered at the furnace arch. For several decades, probes for measuring O_2 at the air heater (AH) inlet have been in common use, and CO monitors before and/or after the AH have been installed in recent years. However, for proper combustion, it would be better to know the concentration of these two gases as close to the burners as possible. There are two problems with measurements at the AH. First, because almost all boilers today are built as balanced draft, air inleakage in the convection pass makes O_2 readings at the AH unreliable as an indication of conditions in the furnace. Second, if the furnace is operating properly, there should be little or no CO entering the convection pass, but due to the elevated temperatures at that point, CO that is present there can "burnout" to CO_2 and not be seen at the air preheater. There is ongoing research by several organizations attempting to develop a suitable instruments to measure these gases at each burner, however there currently are instruments that can be permanently installed at the furnace arch to measure the concentrations of these gases.

Monitoring the O_2 level at the furnace exit should not be considered as an alternative to the O_2 measurement at the air heater inlet. Because the flue gas at the furnace exit can be stratified, readings can be misleading at this location under certain conditions. For this reason, monitoring the furnace exit O_2 should be a supplement to the air heater inlet measurement.

9.2.4 Increased Mill Outlet Temperature

Another area where the boiler efficiency can be increased is by increasing the mill outlet temperature. There are two indirect improvements. First, the pulverization of the coal is accomplished more efficiently at higher temperatures. Second, the coal is less likely to stick together at higher temperatures. A direct improvement results from reducing the quantity of tempering air, that bypasses the air heater, thereby increasing boiler efficiency. New plants should be designed to efficiently handle a wide range of fuels, including a wide range of moisture and volatile contents.

9.2.5 Automatic Coal Samplers

In order to know the "bottom line", what the heat rate of a unit is, it is imperative to know the heating value of the fuel. Because this can vary considerably (due to blending different fuels, etc.) the fuel going to the bunkers should be sampled continuously. This can be done manually or with automatic samplers. With the manual method the coal that is collected is not a true "sample" but is a "specimen" because the requirements for collecting a sample manually (that is not biased in any way), are never fully followed. Therefore to collect a true representative sample, it is necessary to use an automatic sampler that meets the requirements of ASTM D2234.

9.2.6 Gravimetric Coal Feeders

The second input to the equation for calculating heat rate is the amount of fuel burned. The best way to measure coal is to use gravimetric feeders. These should always be specified for new plants. A bypass chute should be provided to allow calibration by collecting and weighing the same coal that is run through the feeder.

9.3 Plant Equipment - Turbine Cycle

There are several areas where the efficiency of the turbine cycle can be improved, either by improving base efficiency or by minimizing losses that usually occur, or by early detection of losses, so that corrective actions can be carried out immediately, minimizing any adverse effects.

Sections 9.3.1 through 9.3.6 describe items that can be used to increase the efficiency of units directly. Items 9.3.7 and 9.3.8 are items which could be used to increase the efficiency of units by minimizing losses. Item 9.3.9 can be used to increase the efficiency of units by quickly locating losses.

9.3.1 High Pressure Turbines with Extraction

Normally the highest pressure extraction to a feedwater heater is from the exhaust of the high pressure turbine. By specifying an HP turbine with an intermediate extraction, the top feedwater heater operates at a higher pressure, and the final feedwater temperature entering the boiler is increased, increasing the cycle efficiency.

9.3.2 Condenser Tube Cleaning System

At many plants, the biggest heat rate deviation is due to high condenser pressure, frequently due to fouling on the water side of the tubes. This fouling can be removed manually, but it requires either an outage or a derating. With an on-line cleaning system, the tubes can be kept very clean without any lost generation. Another important benefit, in addition to reduced fuel cost and reduced emissions, is additional load is also generated. Where a condenser tube cleaning system is to be installed, the inlet water box must be supplied with debris-free water. Usually additional

attention is given to the design of the debris removal system where on-line tube cleaning systems are to be installed.

9.3.3 Closed Conduit Condenser Circulating Water (CCW) System.

Many open (i.e. non-closed loop) CCW systems are designed with an open channel at the discharge where the warm cooling water drains back to the water supply (lake, river, canal) by gravity. With this design, as the water level of the river/lake/canal (and therefore at the CCW pumps) drops, the required pump total head increases (because the suction lift increases), and the operating point on the CCW pump curves moves back, so that the pump supplies less flow. This reduced flow causes the condenser pressure to increase, and the heat rate to increase. Where the CCW system is designed as a closed conduit from the intake structure to a submerged outfall, as the water level drops, the absolute pressure at any point drops, but the total pressure drop through the system remains constant, so the flow remains constant.

9.3.4 Remove Pumps from CCW Inlet Tunnel.

Plants are frequently designed with service water or other pumps taking suction from the condenser circulating water inlet tunnels. This often causes problems in two ways. First, during unit outages, it may not be possible to stop all CCW pumps on that unit, if other pumps (such as raw water pumps, or fire protection, etc.) that take suction from this tunnel are required to be in service. Running a CCW pump to supply water to a small pump or two is a huge waste. Second, under certain conditions, the pressure in the tunnel may drop, causing the auxiliary pump to cavitate or lose suction. To prevent this, common practice is to throttle the condenser discharge valves, to reduce the flow and raise the CCW pump discharge (and therefore tunnel) pressure. Again, this is a waste, because for good heat rate, the condenser needs to have as high a velocity of water as possible (without exceeding the velocity where erosion can occur) flowing through the tubes. If pumps are placed where they take suction from the inlet tunnel, they should be designed to be able to operate with a low (even negative) suction lift. Also, there should be redundant pumps on each unit (or additional suction piping so each raw water pump can take suction from more than one unit) so all CCW pumps on a unit can be shut down during an outage.

9.3.5 Enhanced Surface Condenser Tubes

Almost all condenser tubing is smooth. However, spirally indented tubes are commercially available, and have been in limited use for at least 18 years. The wall thickness and outside diameter (OD) remain unchanged, but the turbulence from the ridges increases the inside heat transfer and reduces the condensate film on the tube outside diameter. These improvements in heat transfer reduce the condenser pressure and improve the unit heat rate. There are three drawbacks that must be considered. First, the fouling rates are usually higher, requiring more frequent cleaning (which if combined with a automatic tube cleaning system would be eliminated). Second, enhanced tubing requires more tube supports (for new designs the tube sheet spacing is

usually decreased slightly, for existing condensers, tube stakes are sometimes required). Third, the enhanced tubing has a higher pressure drop per foot of length than smooth tubing. Therefore for the same flow, enhanced tubing will require more pump power, or for a given pump, enhanced tubing will require slightly more power and the flow will be reduced slightly.

9.3.6 Feedwater Heaters with Negative Terminal Temperature Difference (TTD)

Since the steam supplied to the high pressure feedwater heaters is superheated, the temperature of the feedwater leaving the heater can be hotter than the saturation temperature inside the heater. By increasing the size of the heater (especially the desuperheating zone), the heater TTD can be as low as –2 °C. Over the life of the plant, this can result in significant savings in fuel and emissions.

9.3.7 Non-Condensing Drive Turbines

Large units typically have major auxiliaries (boiler feedwater pumps and sometimes forced draft fans) driven by steam turbines. Usually, these turbines exhaust to a condenser (either the main condenser or an auxiliary condenser). A more efficient design is to use a larger drive turbine with a higher pressure exhaust, that supplies steam to a low pressure feedwater heater. In this design, none of the energy in the steam supplied to the drive turbine is lost.

9.3.8 Turbine Coatings for Solid Particle Erosion Protection (SPE)

One of the biggest thermal performance losses, and one that is difficult to recover, is turbine efficiency. One common cause of reduced turbine efficiency is solid particle erosion. This can be minimized by maintaining proper water chemistry, but it cannot be eliminated. Another way to minimize SPE is to install erosion resistant coating on the turbine parts (especially in the front of the HP and IP turbines).

9.3.9 Improved Turbine Seals.

Another cause of poor turbine efficiency is damage to seals. There are several new types of seals available that prevent damage. One type of seal is a retractable seal, where the seals are retracted away from the shaft during startup (where the chance of damage is highest) by a spring. As the unit is loaded, steam pressure overcomes the spring force, and the seal is pushed down into place. Other types of seals are also available that have "sacrificial" surfaces that keep the "knife-edge" seals from being damaged.

9.3.10 Cycle Isolation Detection.

A common problem at almost all power plants is cycle isolation. Internal leaks (those that are not seen or cause increased makeup, but typically leak into the condenser) are especially difficult to locate. One technique that is in use at many plants is to install thermocouples at the isolation valves, and monitor the temperature. If the line is hot, the valve is leaking. By daily or continuously monitoring these lines, any leaks are immediately seen, and can be repaired at the earliest opportunity. Also, after a startup, it is not uncommon for one of the startup drains to be missed, and left open. A 25mm drain from a main steam line to the condenser is a very expensive error that can go undetected for days if it is not monitored.

The following is a list of high energy drains that are continuously monitored via thermocouples with an indication in the unit control room, at one 210 MW unit:

- Drip from HP Heater #6 to HP Flashbox
- Drip from HP Heater #5 to HP Flashbox
- DA Storage Tank Overflow to LP Flashbox
- Drip from LP Heater #3 to LP Flashbox
- Evacuation Line from HP Heater #6 to HP Flashbox
- Evacuation Line from HP Heater #5 to HP Flashbox
- Left LP Bypass Warm-up Line Drain to HP Flashbox
- Right LP Bypass Warm-up Line Drain to HP Flashbox
- Drain from Up-stream of NRV for Extr. Line to HP Heater #6 to HP Flashbox
- Drain from Down-stream of NRV for Extr. Line to HP Heater #6 to HP Flashbox
- HP Heater #5 Extr. Line Drain to Flashbox
- CRH to DA Extr. Line Drain to HP Flashbox
- CRH to DA Extr. Line Drain Bypass to HP Flashbox
- Left HRH Steam Supply Line Strainer Drain to HP Flashbox
- Right HRH Steam Supply Line Strainer Drain to HP Flashbox
- Left CRH Steam Supply Line NRV Drain to HP Flashbox
- Right CRH Steam Supply Line NRV Drain to HP Flashbox
- Left MS Supply Line Strainer Drain to HP Flashbox
- Right MS Supply Line Strainer Drain to HP Flashbox
- Condensate Spray Drain to HP Flashbox
- Condensate Spray Drain Bypass to HP Flashbox
- CRH to DA Pegging Line Drain to HP Flashbox
- CRH to DA Pegging Line Drain Bypass to HP Flashbox
- BFP A Safety Line Drain to Condenser
- BFP B Safety Line Drain to Condenser
- BFP C Safety Line Drain to Condenser
- Wet Steam Washing Drain Lines to HP Flashbox
- CRH Turbine Gland Sealing Steam Supply Line Drain to HP Flashbox

9.4 Plant Equipment - Instrumentation & Controls

9.4.1 Variable Pressure/Sliding Pressure Operation

Any unit designed today should be designed to run in both full pressure and in a variable pressure mode. There are three types of variable pressure operation. In full variable pressure operation, the turbine control valves are left in a fixed position (either wide open or slightly closed), and the load on the unit is controlled by varying the boiler pressure. This mode of operation is most often used with throttle governed (full arc admission) turbines. In sliding pressure mode, from full load the load is decreased by closing the control valves, to some valve point. Below that load, the load is further reduced by reducing steam pressure. This mode of operation is applicable to nozzle governed (sequential arc) turbines. The third type of variable pressure, dual pressure, is similar to full variable pressure, in that the turbine control valves are left wide open. However, instead of running reduced pressure from the BFP to the turbine, full pressure is maintained through the high pressure heaters and waterwalls, and the pressure is reduced in the superheater (between the primary and secondary) using special valves. This type of operation is applicable for units that require full pressure on the water cooled circuits (i.e. supercritical pressure units, cyclone fired units, etc.) (See Section 12.3.)

The primary advantage of variable pressure operation is improved heat rate. This is the result of eliminating the main steam temperature drop that occurs with the pressure drop across the control valves. The cold reheat steam temperature is also hotter, resulting in reduced reheat duty. In full variable pressure, there is an additional advantage of reduced pump power, since the feedwater is not pumped to full pressure, then reduced through the valves. There are other advantages such as constant steam/metal temperatures in the turbine over the load range, additional available energy in the cold reheat, etc.

Depending on the type of boiler and turbine, the unit (boiler metals, controls, etc.) should be designed to operate in the appropriate variable pressure mode.

9.4.2 Variable Speed Drives for Major Auxiliaries

Large equipment such as forced draft fans, primary air fans, induced draft fans, motor driven boiler feedwater pumps, and condensate pumps require substantial amounts of auxiliary power. At reduced loads there are large losses associated with either guide vanes, dampers, recirculation valves or hydraulic couplings that are used for control. These losses are often present at full load, as the auxiliary equipment is usually slightly "oversized." To eliminate these losses (large at low loads and minimal at full load), equipment, whenever possible, should be controlled with frequency control variable speed drives. There are three additional advantages to the use of variable speed drives. First, is the "soft start" capability, where the motor is not subjected to large starting currents. Second, is the elimination of the maintenance and control problems associated with vanes and dampers. Third, with India's power system frequency fluctuations, the fans and pumps can be controlled better with variable speed drives than with dampers, vanes, etc. because the speed of the pump/fan is independent of the system frequency.

A second option is to use two speed motors. While they are not as efficient as variable speed drives, they are more efficient than single speed motors.

9.4.3 Better Quality Instrumentation at Critical Locations

In order to operate power plants efficiently, the operators must have reliable and accurate information on the unit. Small errors in sensors can result in large "unaccountable" heat rate deviations. For example, if the unit is "expected" to operate at 538°C steam temperature at the turbine stop valves, but the instrumentation is indicating 3°C too higher than the actual temperature, the unit will operate with the steam temperature 3°C too low, resulting in a heat rate deviation around 10 kJ/kWh or 2.5 kcal/kWh. For high temperature thermocouples, 3°C drift is not unusual. Even if just a few key instruments are in error, the unit could have a large heat rate deviation. Figures 9.1 and 9.2 shows the uncertainties of various type so temperature and pressure instruments.

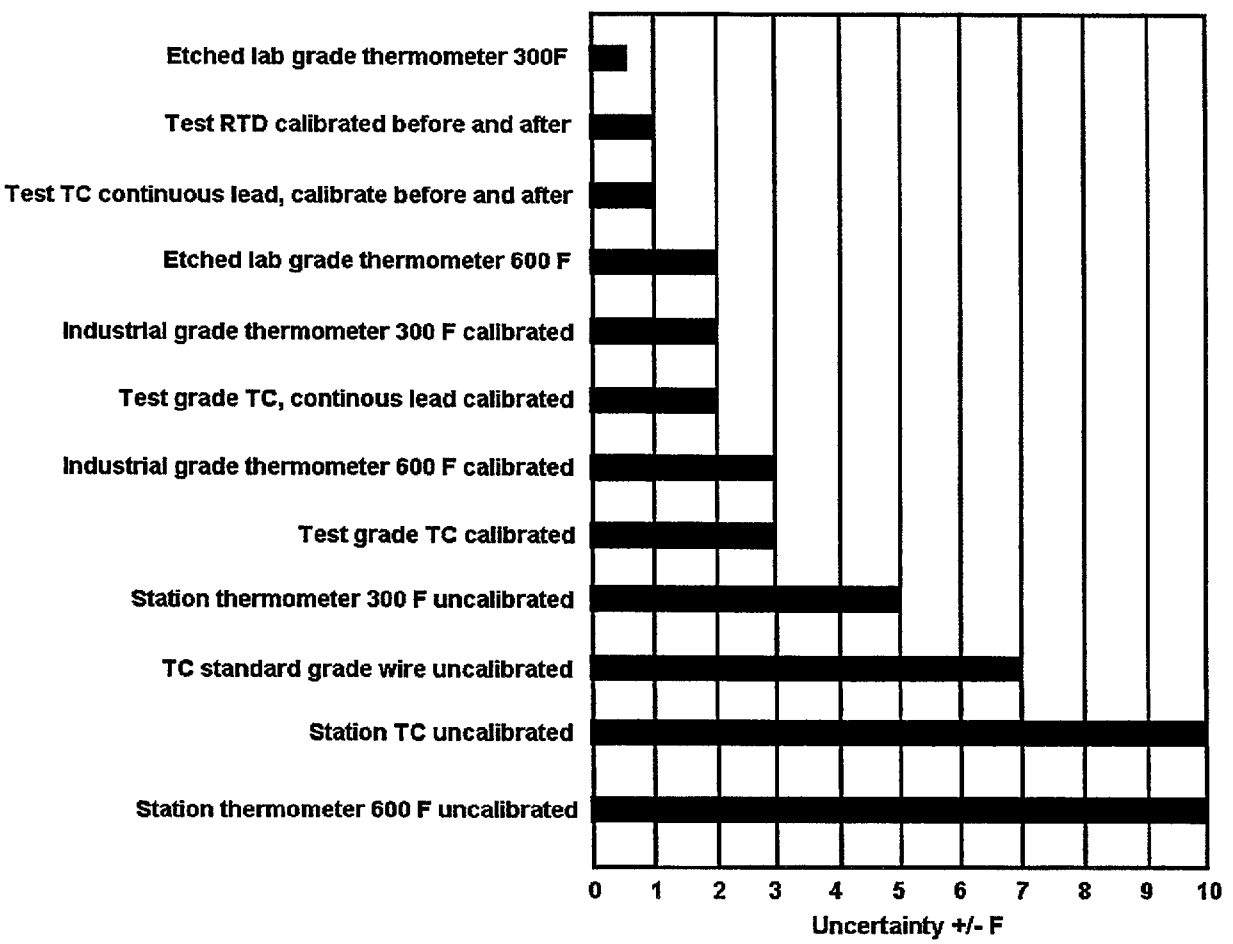

Figure 9.1 Temperature Measurement Device Uncertainties

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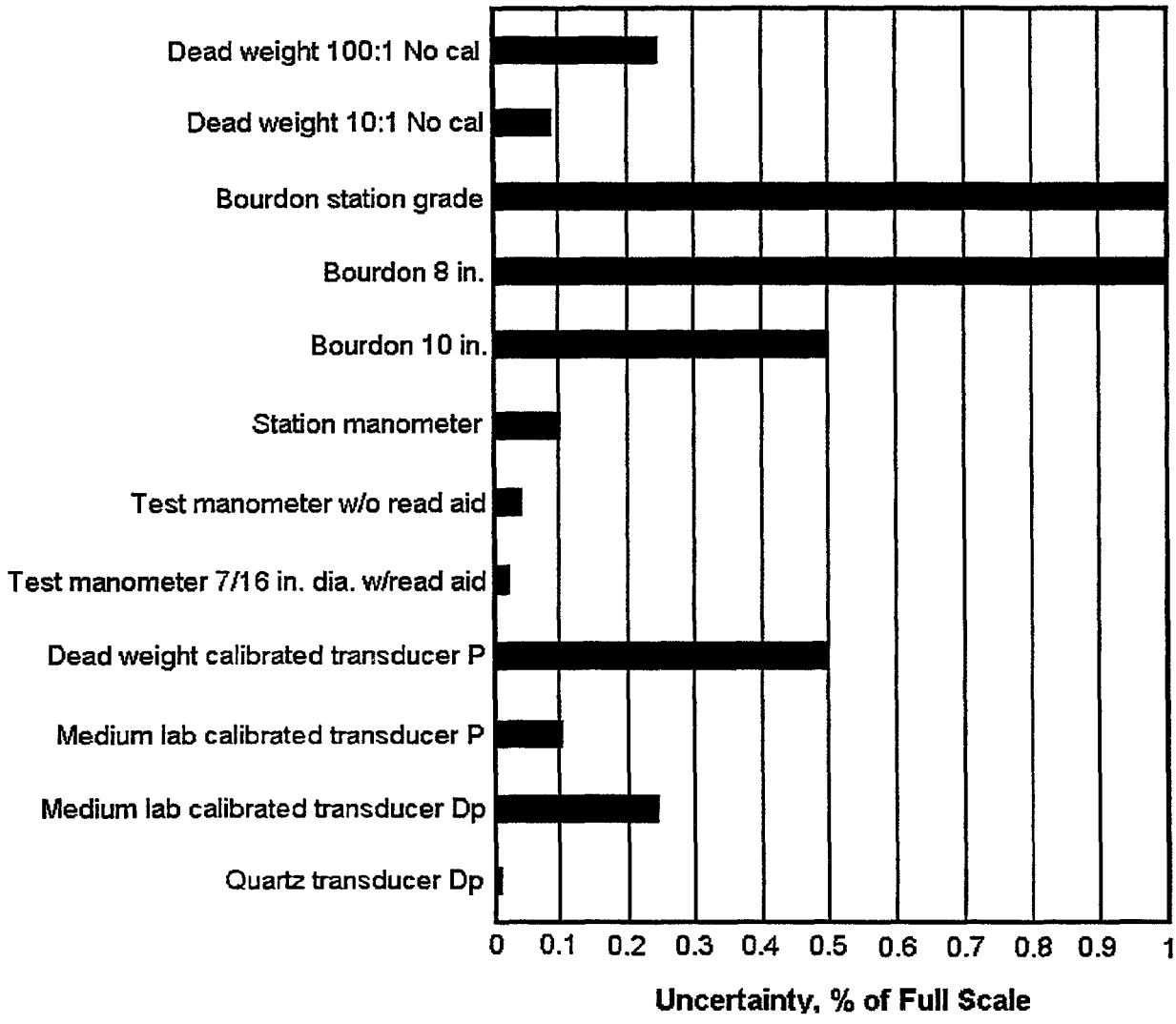

Figure 9.2 Pressure Measurement Device Uncertainties

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Some places where better quality instrumentation is justified are:

Use RTDs instead of thermocouples. Since thermocouples tend to drift over time and there can be errors introduced at junctions, RTDs are a much better choice for critical measurements such as main steam and hot reheat.

For the air heater exit temperature measurement, several thermocouples should be installed due to the size and stratification in the duct. The thermocouples should be installed a few meters downstream of the air heater to allow some mixing.

High accuracy pressure transmitters. Many manufacturers offer pressure transmitters with accuracy's better than 0.1%. Many older power plants were designed with pressure gauges, with a readability of 1 bar, which for main steam pressure is equal to a heat rate deviation of 5 kJ/kWh or 1.2 kcal/kWh. Later, when electronic pressure transmitters were introduced, they generally had accuracies of only 0.25%, which is little better than the readability of the old gauges. For most locations this is sufficient, but for critical locations, the more accurate transmitters should be used. Examples of such locations are main steam pressure, feedwater flows, attemperation flows, makeup flow, and condenser pressure. Most locations would use gauge pressure transmitters, but locations such as the condenser, would use absolute pressure transmitters. An advantage of using an absolute pressure transmitter for condenser pressure over a vacuum gauge, is that no additional instrument has to be used, or calculations have to be made, to correct for variations in barometric pressure.

Each unit should have a real time indication of its auxiliary power use (especially unit auxiliary power and total auxiliary power including prorated station service), as well as the resulting *net* generation. This includes monitoring not only the unit auxiliary power, but also the common and start and emergency buses, and properly prorating them. Too often, emphasis is placed on gross power output of the generator, when the amount of generation that is supplied to the grid (net load) is of much more importance. Major auxiliaries (mills, FDFs, PAFs, IDFs, BFPs, CCWPs, etc.) should have their own power transducer.

A last sensor that is often over looked is the O_2 measurement. Because they are installed in large ducts where the flue gas is stratified, multiple sensors (2 or 3) should be installed in each duct. This will enable the operator to better know what the true average O_2 is in the duct. Also, if one sensor goes out, the operator would still have one or more sensors in that duct until the faulty sensor can be repaired. An option on most permanently installed O_2 probes, is to have an automatic calibration feature. This option allows the sensor to be periodically calibrated without any manual intervention, by passing calibration gas over the sensor.

9.4.4 Condenser Monitors

Because the condenser is usually one of the locations of large heat rate deviations, it deserves additional instrumentation to monitor its performance. In addition to the high accuracy pressure transmitter mentioned above, some indication of flow through each waterbox should be provided. The absolute accuracy of this measurement is not as important as the repeatability. One frequent cause of reduced condenser performance is reduced CCW flow, but most plants do not have any indication of critical parameter. The most common method of providing this indication is to provide a pressure differential measurement somewhere (such as at the waterbox to outlet pipe transition, or between the inner side and outer side of an elbow, etc.) that can be related to flow (i.e. Flow = C * $\sqrt{dP}$). The flow constant "C" can be determined periodically from a dye dilution test, or turbine flow meter traverse. Some plants have tried to use permanently installed annubars or turbine flow meters but, due to fouling, their repeatability has not been good.

Another important indication is the level of water in the outlet waterbox. If the waterbox is not maintained full, the effective surface is reduced, leading to high condenser pressure. Sight glasses and float switch alarms are easy to provide.

Some critical pressure measurements that affect condenser performance that should be monitored are the pressure drop through each waterbox, the pressure drop across the traveling water screens, and the pressure drop across the coarse bar screens in front of the CCW pumps.

One last critical condenser performance related measurement that is often ignored is the quantity of air inleakage. As a minimum there should be a rotameter at the air removal equipment (vacuum pump or steam jet). In recent years, several companies have offered anemometers for continuously monitoring the amount of air inleakage.

9.4.5 Real Time Controllable Loss Monitor

Many plants today have been designed with, or have been retrofitted with, real-time heat rate or controllable loss monitors. While these systems are installed at many locations, at most, they are not used. There are several reasons for this.

1. The quality/reliability of the instrumentation is inadequate, so the system does not give meaningful information. This is especially true of systems that attempt to calculate heat rate by doing extensive calculations requiring dozens of instruments to be working and in calibration at all times.
2. The "design" or "reference" or "expected" values that the parameters are being compared to are unrealistic.
3. Critical parameters are not monitored. For example, in many installations, auxiliary power is not included. Without knowing the auxiliary power consumption, the operator cannot make informed decisions about the effect of removing equipment from service.
4. There is no guidance on what to do when a parameter's deviation is high. A few systems are tied to a fault tree, with values (both actual and expected) for secondary levels filled in. For example, if the exit gas temperature is too high, the fault tree for this parameter is displayed, with the actual and expected inlet air temperature displayed, the actual and expected gas temperature entering the APH displayed, the actual and expected x-ratio displayed, etc. This helps the operator decide what the root cause of the problem is and what action to take. Another method is to tie the loss monitor to a neural net control system that can actually control the unit.
5. Variations in database design, calculation procedures, etc. between plants (and at some plants, all units do not have the same systems). If a utility can pick one design and use it at all plants, it makes maintenance of the system much easier.

These systems are valuable tools, but they must be installed with care to be sure that they will be used and the full benefits realized.

9.4.6 Data Acquisition System with Enhanced Capabilities

Most units built in recent years have had some type of data acquisition system installed. Some features that are very useful, but not always present, are:

- Capability to generate control charts
- Data validation routines
- Advanced graphics (trend charting, mimics, etc.)
- Capability to communicate with other systems, networks, etc.

9.5 Plant Equipment - Laboratory and Test

Along with the power plant itself, additional facilities are usually supplied such as workshops, water analysis laboratory, office space, etc. This section discusses the laboratory and major test equipment that should be included at each plant site that would be used in performance monitoring and testing.

9.5.1 Coal Analysis Equipment

Each coal fired power plant should have a coal laboratory with facilities for running analyses on coal. This includes equipment for determining the surface moisture (Air Drying Oven), higher heating value (Calorimeter), proximate analysis including inherent moisture, volatiles, and ash, % sulfur (Sulfur Analyzer), grindability (Hargrove Grindability Tester), and fineness. Related equipment that is required to perform these analysis would include balances, crushers, pulverizers, riffles, mixing wheels, Sieve Shaker (with sieves between 50 and 200 mesh).

While not normally required at each plant, it is highly desirable to have equipment for running ash fusion tests and an elemental analyzer for running ultimate analysis (carbon, hydrogen, nitrogen, sulfur, and oxygen).

In order to determine the efficiency of the power plant, the coal analysis, including the higher heating value, must be determined. It is important to determine the actual heat rate frequently, perhaps as often as daily. For this reason, and the fact that many power plants receive fuel from more than one source, it is important that the fuel be sampled and analyzed frequently. Therefore, the equipment used to run proximate analysis and heating values should be capable of running multiple analysis every day.

9.5.2 Test Equipment

Each power plant should also have a set of test equipment for running routine performance tests on major pieces of equipment (see Section 4.5). Large utilities frequently have a central test crew to run large tests, such as turbine cycle heat tests or boiler optimization tests. However, even if

large tests are carried out by an outside organization, it is worthwhile to have an in-house test capability so that condensers, turbines, pumps, pulverizers, air heaters, etc. can be tested whenever an indicator (see Section 2 Monitoring Primary Process Indicators) shows a need.

9.5.2.1 Temperature Measurement

For turbine cycle testing, a plant should have a variety of resistance temperature detectors (RTDs). Historically, thermocouples have been used because of their low cost and high reliability. RTDs, while much more accurate, have not been used widely (until recently) due to their higher cost and poorer reliability. In recent years, the reliability of RTDs has improved greatly and, while still not as durable as a thermocouple, the higher accuracy has made the higher cost acceptable. A plant with 200 to 500 MW unit(s) probably needs 30-50 RTDs of various lengths, from 250mm to 500mm.

For condenser testing, even high accuracy RTDs are not sufficient for performance tests, where temperatures must be measured to at least 0.1 °C accuracy. For these tests, partial immersion thermometers, long enough to be read without having to remove them from the thermowell are required. A plant with 200 to 500 MW unit(s) probably needs 15-30 thermometers.

For boiler testing, the temperature in large ducts will need to be measured at several locations. Here, accuracy is not as critical as in turbine testing, so thermocouples are usually used. The thermocouples are placed in tubing, which can be inserted into the ducts. Sometimes the thermocouples are made from thermocouple extension wire, and sometimes stainless steel jacketed thermocouples are used.

9.5.2.2 Pressure Measurement

For performance testing, a plant should have a variety of pressure sensors. Typically a plant with 200 - 500 MW unit(s) will have a variety of high accuracy (better than 0.075%) electronic pressure transmitters. There should be gauge pressure transmitters of several ranges between 0-500 mm H_2O and 0-300 bar. (The highest pressure normally measured in performance testing is the BFP discharge.) There should be several differential pressure transmitters, of various ranges. Low dP transmitters are required for measuring the pressure drop across nozzles and orifices, across the condenser waterbox, draft losses, etc. Higher dP transmitters are needed for measuring the pressure drop across feedwater heaters, etc. A few absolute pressure transmitters are also required for measuring barometric pressure, condenser pressure, etc. A typical plant might need to have 35-70 pressure transmitters to cover the required ranges and tests.

While most tests are run using pressure transmitters, manometers (both u-tube and incline) are used in some performance tests, and a plant should have a variety of these.

9.5.2.3 Flow Measurement

For measuring air flows, a plant should have several pitot tubes available. There should be some "L" type pitots available for measuring clean air flows in small ducts or pipes (such as is done in a

pulverizer clean air tests, etc.). For measuring air or gas flows in large ducts (or in small ducts with large concentrations of particulates) "S" types pitots should be available.

9.5.2.4 Flue Gas Analysis

For performance testing, the gases that are measured are carbon dioxide (CO_2), oxygen (O_2), and carbon monoxide (CO). Accurate, reliable, and fast gas analyzers are necessary to analyze for these gases. For many years, the use of an Orsat was the standard, and Orsats are still used today for checking single point measurements. However, when running a test, an Orsat is not sufficient to perform the number of analyses necessary in a reasonable amount of time. Various types of electronic analyzers are available. Chemical cell analyzers are frequently used, but they must be used with caution as the cell must be replaced periodically as it exhausts. More accurate analyzers that use a paramagnetic sensor for O_2, and infrared sensors for CO_2 and CO are better choices. The number of instruments required at a plant will vary depending on the level of testing that is planned. If a plant intends to run boiler optimization tests, 4-8 devices may be required, each device performing two analyses (at the HVT locations the analyzer should check O_2 and CO, at the APH locations the analyzers should check for O_2 and CO_2). If a plant only intends to run APH and other small tests, 2 to 4 devices may be sufficient.

9.5.2.5 Power Meters

Several performance tests require that the instantaneous power or integrated energy produced by the unit, or consumed by auxiliary equipment, be measured. This requires high accuracy (at least 0.1%) power meters, that can measure the power in all three phases. For most routine tests, 2 or 3 such instruments are sufficient. If boiler optimization tests or auxiliary power surveys are to be done, several additional instruments will be required

9.5.3 Miscellaneous

Several miscellaneous instruments/equipment should be available at every power plant. These include:

1. High velocity thermocouple probe (HVT)
2. High volume fly ash samplers (HVS)
3. Dirty -air test kits
4. Helium leak detection equipment
5. Strobe lights
6. Psychrometers
7. Set of equipment for cleaning condenser tubes via brush/scrapers

9.6 Test Ports

A common step in determining the performance of equipment is to run a test. Frequently, the required test ports, used to install sensors for measuring temperature, pressure, flow, gas analysis, etc., are either not existing, or not properly installed. In some cases they can be retrofitted. However, in most retrofits, the port(s) will not be in the best location, and the test results may or may not be reliable. For this reason, and because it is always easier and cheaper to install these ports during initial construction, these ports should be included in the original design. Another error that is sometime found is that the ports are installed, but in a location that is not accessible with the unit running. Particular attention should be made to this point when locating test ports. Also, impulse piping from the root valve to the terminal point should be permanently installed, so it can be used at any time. It is highly desirable to run all these lines to a common location to facilitate easy instrument installation for routine tests, and so differential pressure can be installed without requiring long temporary tubing runs.

Due to the large duct size, and stratification that exist in air and gas ducts, measurements in the boiler area are difficult at best. ASME's Performance Test Code 19.10 on Flue and Exhaust Gas Analyses specifies the number of measurements required to be taken at a location, depending on the number of lengths of straight duct upstream and downstream of the test plane (see Section 4.7.13). It is extremely unusual to find a power plant that meets the strict requirements of this code. Unfortunately, the ductwork at many plants is designed with no consideration of the testing requirements. This can make accurate determination of boiler performance (including the air heater) impossible.

In addition, the requirements for straight sections of duct up and down stream of the test location, there are other requirements including electrical power supply (for gas analyzers, vacuum pumps, etc.), station service air (for aspirators), raw water (for cooling as with the HVT), lighting, walkways (large enough for handling large, heavy, and hot probes). If these are not provided with the initial design, they usually can be supplied later (either temporarily or permanently), but at a higher cost and if temporary facilities are required to be set up each time a test is run, the test probably will not be conducted as often as required.

Some places where test ports are required, that should be considered during design of the plant are:

1. Ports at the air preheater gas inlet and gas outlet for measuring temperatures, for sampling the flue gas, and for collecting fly ash samples.
2. A permanently installed ash sampling device should be considered for the air heater inlet/economizer outlet location to be the normal ash sampling location.
3. Ports at the air preheater air inlet and air outlet for measuring temperatures.
4. Ports at the economizer outlet for extracting flyash samples
5. Ports at the furnace arch, and at various positions through the convection pass for inserting a HVT probe.
6. Ports for calibrating all air flow measurements (total primary air, secondary air, and air flow to each individual mill).
7. Ports in a vertical run of each burner pipe for measuring clean air flow (1/2" nipple w/ plug), and for dirty air test and coal fineness sampling (1-1/2" nipple w/ full port ball

valve). The number installed should be as per ASME PTC 4.2, based on the number of straight pipe diameters up and down stream of the test plane. It is preferable to find a long vertical section so only two ports have to be installed in each pipe.

8. Ports in the coal chutes supplying each coal feeder for sampling raw coal before the pulverizers.

A recent development is temperature and gas analysis mapping of the furnace zone. While these systems can be permanently installed, the cost makes this technology more applicable to use as a periodic test. These systems require several test ports in either the furnace or convection pass or both, preferably crossing the area in several directions and in multiple planes. Sometimes these systems can be fitted on existing openings, but if the boiler is initially designed for these, installation is much easier and cost effective.

In the turbine cycle, problems are also found with either missing test ports, or improperly installed test ports. Some guidelines for turbine cycle test ports include:

1. There should be redundant test ports for both temperature and pressure measurements in critical locations, such as main steam, hot reheat steam, CCW inlet and outlet, final feedwater, etc. A common problem is that "test" ports are either not installed or are at some time used for permanent instrumentation. When this is done, in order to run a test, the station instrument(s) must be removed, which may not be possible. If it is removed, then the test measurement cannot be used to check the calibration of the station instrument.
2. In turbine extraction lines, there should be temperature and pressure ports at both the turbine (as close as possible) and at the feedwater heater (again as close as possible). However, where there is both an extraction and a crossover/crossunder to another turbine section (or for cold reheat), the test points (temperature and pressure) in the extraction line are not a substitute for temperature and pressure measurements in the crossover/crossunder/cold reheat piping.
3. Where possible, locate thermowells just downstream of a bend or elbow. This will mix the fluid, so the temperature at the well will be representative. Also the thermowells should be positioned so that gravity will not "pull" the sensor out of the well (i.e., place them on top or at a slight angle from vertical in horizontal pipes, and place the well in a 45 degree angle in vertical pipes).
4. For pressures that are above atmospheric pressure, the sense lines must slope continuously down to the location where the test instrument will be located.
5. For pressures that are below atmospheric pressure, the sense lines must slope continuously up to the location where the test instrument will be located.
6. Flow elements (nozzles, orifices, etc.) should be installed, with appropriate pressure taps for measuring the flow thorough individual feedwater pumps, balancing drum leakoff, turbine cooling steam flow, makeup flow, heater drain flow(s) flow to auxiliary equipment, etc. Where there is a station instrument for measuring any of these, there should be an additional set of taps so a test sensor can be installed in parallel. The plant must receive full information on all installed nozzles and orifices (pipe and nozzle material and ID; type of taps, maximum flow, maximum dP, design

pressure and temperature, etc.). This information should be engraved on the orifice "T", and on the nozzle OD, and it is desirable to have it engraved on a metal plate that is mounted on a column or local board near the device.

7. Test ports and ports for station instruments should be installed close to each other, to enable the reading from each to be compared.
8. Each end or pipe at the exhaust of a turbine section should have test temperature and pressure ports.
9. Pressure taps, as well as thermowells, should be installed between each feedwater heater, in addition to those for entering and leaving the string.
10. Boiler feedwater pumps with balancing drum leakoffs should have an orifice with dP taps for measuring the leakoff flow.

For electrical power/energy measurement, the generator output and major auxiliaries (FD, PA, & ID Fans, BF Pumps, CCW Pumps, Mills, etc.) should have simple "plug" connections to the secondary side of the CTs and PTs to allow connection and removal of test instrumentation with the unit/equipment on-line. These systems should be designed to measure all three phases.

9.7 Performance Guarantee Tests

The Performance Guarantee (PG or Acceptance Test) is the best time to collect information on the actual performance of the unit. Because the PG test is so important, it is critical that its scope be expanded to include more that just the required information to determine if contractual guarantees have been met. There are two alternatives to running an expanded series of tests. The first is to use design data. However, the design performance of various parts of the unit may or may not be reached. In some instances, the design performance may be exceeded. For these reasons, the actual performance measured during the PG tests should be the basis for the unit, and not the design performance. The second alternative is to run a second group of tests, independent of the PG tests. This can be done, but it is much more costly than expanding the scope of the PG tests.

Another question that should be considered is who should run the test. Historically, the manufacturer has run tests. However, while that is one option, it is not the only option. Large utilities may want to develop the expertise to perform these tests themselves, then they have the equipment and knowledge to perform other tests. A third option is to use an independent third party.

The tests should be run over the entire operating range. At each condition, at least two tests should be conducted. If the two tests do not closely agree, the cause of the discrepancy should be determined, and a third test run. For turbine cycle tests, the performance should be measured over the entire operating range of the unit. If the turbine is operated in nozzle governed (sequential arc) mode, a minimum would be to test at each valve point, including valves wide open (VWO). If the turbine is operated in throttle governed (full arc) mode, it should be tested at VWO, minimum load, and at least two intermediate points. Boiler performance tests should be conducted at full load, minimum load, and at least two intermediate points. Tests on pumps, fans,

etc. should also be run over their entire operating range. During both the turbine cycle and the boiler tests, the unit auxiliary power usage should be recorded.

Typically boiler performance tests run by the manufacturers are done only to ensure that some contractual level of performance can be reached, usually at only one load. For contractual purposes, these test may still be done, however these are not sufficient to meet the utility's needs. There are several reasons for this. First, the manufacturer may run the performance test at a condition that the utility would not normally allow (due to slagging, mills plugging, operating certain control loops manually or locally, operating less equipment than the utility would normally run, etc.). Second, the manufacturer may run the test, and if the guarantee is met, nothing additional is done, even though the best possible efficiency may not have been reached. Third, as was stated above, the utility needs to know the performance of the boiler over the entire operating range.

A better approach to the boiler testing is to first set whatever constraints the utility believes are required. For example, if certain dampers are manually controlled, and the utility's operating procedures are to leave them in a fixed position, then the tests should be run with them in the fixed position, over the load range. Another example, if the utility sets a minimum O_2 to prevent the formation of slag, the performance tests should not be run below that O_2 level. A third typical type of constraint is the equipment in service. The utility's operating procedures may require a certain number of mills to be in service at each load. The performance tests should be run with that number of mills in service.

Once any constraints are set, several series of optimization tests should be conducted. Each series of tests would cover one load. The purpose of each optimization test series is to determine the boiler settings that results in the best performance, while still meeting other constraints (environmental, reliability, etc.). See Section 4 for more information on boiler optimization testing. Typically, "optimization tests" are not as rigorous as "PG tests", so once the optimum conditions for a load are determined, the "code-level" tests at that load can be run.

A common question that should be addressed in the contract is what is to be done with the primary flow element after the tests are run. The utility may want to consider requesting that they retain the element, and that it be removed and properly laid up after the test is complete, so it could be reused at some later test.

One final consideration for the boiler tests. Usually many special probes must be constructed for sampling the gas and measuring the temperatures in the ducts. Since probes will be needed by the utility for periodic testing, the utility should request that all probes used in the PG tests should be turned over to the utility at the end of testing.

Another important consideration of the PG tests is the content of the final report. In addition to knowing if the equipment met guarantee, the utility should require:

1. A complete listing of the each measured parameter, for every test. This includes every temperature, pressure, differential pressure (for flows), gas analysis, etc. A table should be provided with the parameter name, the value, and the units for the value.
2. A complete listing of each calculated parameter, including intermediate calculated values. This includes all flow rates, TTDs, DCA, turbine section efficiencies, pump heads, etc. An example of an "intermediate calculated value" would be the calculated flow rates such as turbine extractions, determined from a measured flow (such as condensate flow) and an energy balance around a feedwater heater. Another "intermediate calculated" parameter would be the "APH no-leakage exit gas temperature corrected to the reference air entering temperature." This value is an intermediate step in determining the boiler efficiency.
   Note: much of this data is also typically placed on a heat balance diagram. Preferably two heat balance diagrams should be made for each test, one with the actual data, and the other with "corrected to contract conditions" data.
3. A complete list of all "constants" required for calculations, such as data on flow nozzles (pipe & nozzle IDs, etc.).
4. A copy of all curves required for calculations, such as turbine exhaust loss curves, generator loss curves, heat rate and load correction to heat rate curves, flow nozzle/orifice calibration curves, etc.
5. A list of all "assumed" values. Typically, some minor flows, etc. are not measured. All parameters not actually measured should be listed, with their value or curve given.
6. An example of the calculations should be provided, showing all calculations for one test.
7. Copies of the pre test as-left and post test as-found instrument calibrations should be part of the report.

This information is extremely valuable in allowing the utility to make additional calculations, determine baseline performance (See Section 3), set specification/target values for trend charts (see Section 5), etc.

SECTION 10 CARBON DIOXIDE EMISSIONS

10.1 Introduction

One important result of heat rate improvements that is often overlooked, is that an improvement in heat rate reduces the amount of CO_2 produced per unit of generation. This reduction is of national and global importance.

There are three methods to determine CO_2 emissions. First is a direct measurement of the flue gas quantity and CO_2 or O_2 concentration, the second method is based on an ultimate fuel analysis, and the third is based on the quantity of heat input to the boiler.

10.2 Direct Measurement of Stack Gas

In the direct measurement method, flue gas CO_2 concentration is measured using in-situ sensors (within the flue gas stream) or via extractive systems (a flue gas sample is extracted from the stack or duct and analyzed using a bench type or portable analyzer). In-situ type sensors cannot be absolutely calibrated, while the sensors in the extractive method can be calibrated with standard calibration bottles. Therefore, the extractive method is generally preferred. A draw back to the extractive system is that the analyzers need a moisture free, dust free sample. In order to provide the analyzer with an appropriate sample, a conditioning system must also be in service. Conditioning systems either remove moisture and dust by condensation and filtration, or dilute the sample so that the sample dew point is above analyzer conditions. In either case, the sampling system and CO_2 analyzer need a formal preventive maintenance program.

Another group of sensors is used to measure the volume flow rate of the flue gas. Typically, a grid of hot wire anemometers is used for this measurement. The volumetric flow measurement is on a "wet" basis, meaning the total volumetric flow, including water vapor, is measured. Depending on the method for measuring the CO_2 concentration, it can be measured on a "wet" basis or on a "dry" basis. If the dry method is used (where the water vapor has been removed from the sample before it is analyzed), the measurement must be corrected by multiplying the dry concentration by $(100-\%H_2O)/100$.

The CO_2 mass emission rate can be calculated from the equation:

$$CO_2 \text{ (US tons/h)} = K * C_H * Q_H$$

where $K = 5.7 * 10^{-7}$ (U.S. tons CO_2/scf)/%CO_2
C_H = CO_2 concentration % by volume, wet basis
Q_H = Volumetric Flow Rate scf/hour

$$CO_2 \text{ (metric tons/h)} = K * C_H * Q_H$$

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where K = $1.8 * 10^{-5}$ (metric tons CO_2/sm^3)/%CO_2
C_H = CO_2 concentration % by volume, wet basis
Q_H = Volumetric Flow Rate sm^3/hour

If the CO_2 is not measured, it can be approximated from a measured O_2. (Note, that with either the CO_2 or the O_2 method, the gas that is analyzed must be sampled at the same location that the flow rate is measured.) The equations to convert from O_2 to CO_2 are:

If the O_2 is on a dry basis:

$$CO_2 = 100 * (F_C/F) * ((20.9-O_2)/20.9)$$

If the O_2 is on a wet basis:

$$CO_2 = [100/20.9] * [F_C/F] * [20.9*((100-\%H_2O)/100)-O_2]$$

$\%H_2O$ = Moisture content of the flue gas in percent
F_C = Carbon-based "F" factor in standard $m^3/10^6$kcal
F_C can be calculated from the equation below if the higher heating value and carbon content of the fuel are known

$$F_C = 36.1 * 10^3 * UltC / HHV$$

F = Dry basis "F" factor
F can be calculated from the equation below if the higher heating value and ultimate analysis of the fuel are known

$$F = 0.1123*10^6 * (3.64*UltH + 1.53*UltC + 0.57*UltS + 0.14*UltN - 0.46*UltO) / HHV$$

F and F_C can also be approximated from the following table:

Table 10.1 U.S. EPA 40CFR Part 75 Appendix F F and F_C Factors

Fuel	F ($sm^3/10^6$kcal)	F_C ($sm^3/10^6$kcal)
Anthracite Coal	1135	221
Bituminous & Subbituminous Coal	1099	202
Lignite	1108	215
Oil	1033	160
Natural Gas	979	117

One significant advantage of this method is that the heat input to the unit can be estimated from these measurements. This heat input, along with the generation, can be used to calculate the heat rate of the unit. This is called the "F" factor method. There are four equations for the heat input, depending on the analyzed gas, and the basis (wet or dry):

CO_2 dry: $HI = Q * [((100-\%H20)/100) / (100 * F_C)] * [\%CO_2/100]$

CO_2 wet: $HI = [Q / F_C] * [\%CO_2/100]$

O_2 wet: $HI = [Q / F_C] * [(20.9/100) * (100-\%H_2O) - \%O_2] / 20.9$

O_2 dry: $HI = Q * [(100-\%H_20) / (100*F_C)] * [(20.9 - \%O_2)/20.9]$

where HI = Heat input to the boiler in 10^6kcal/h
Q = Volumetric flow rate in standard m^3/h

In addition to the problem of getting a representative sample of the flue gas (maintaining a leak free system that is usually under vacuum, stratification of the gas due to duct leaks, etc.), there are potential problems with the flow measurement. Enough measurement points must be used to get the true flow as the flow profile is usually non-uniform. Also, the direction of the flow is often not parallel to the duct wall, but is swirling. This must be taken into account to measure the true flow rate. Usually, this involves using a directional pitot tube (such as a 3 or 5 hole Fechheimer) to calibrate the hot wire anemometer(s).

10.3 Direct Measurement of Boiler Input

The second method utilizes the ultimate fuel analysis, specifically the carbon content of the fuel along with the mass of fuel burned. This requires an intensive, continuous sampling of the fuel, followed by an ultimate analysis of the fuel, as well as a method for measuring the quantity of fuel (i.e., scales for coal units). When 1 kilogram of carbon burns, 3.664 (44.01/12.01) kilograms of CO_2 is produced. The equation to calculate CO_2 emissions is then:

Kilograms of CO_2 = (Kilograms of Fuel Burned) * (% carbon in as-fired fuel/100) * (3.664 kilograms of CO_2/kilogram carbon)

This calculation assumes that all the carbon in the fuel is burned to CO_2. When firing coal, some of the carbon is not burned and ends up as carbon in the ash. The amount of unburned carbon in the ash is normally small enough that it can be ignored without appreciably affecting the final result. Some of the carbon is not completely burned, but forms CO. The amount of CO is typically very small (on the order of 100 ppm), and will not affect the accuracy of the calculations.

10.4 US DOE's Fuel Emissions Factor

The third method is to use US DOE's fuel emissions factor (Reference 2). This table lists approximate conversion factors for millions of metric tons of CO_2 per quadrillion Btu of heat input, for different fuels. To use this method, the heat input to the boiler (coal burned

and HHV or heat rate and generation), and the fuel type must be known. The equation to calculate CO_2 emissions is then:

English Units

A. Tons of CO_2 = (Pounds of Fuel Burned) * (HHV of fuel Btu/pound) * (Table 10.2 factor 10^6 Tons $CO_2/10^{15}$ Btu)

B. Tons of CO_2 = (Generation kWh) * (Heat Rate Btu/kWh) * (Table 10.2 factor 10^6 Tons $CO_2/10^{15}$ Btu)

Metric Units

A. Tons of CO_2 = (kilograms of Fuel Burned) * (HHV of fuel kcal/kilogram) * (Table 10.2 factor 10^6 Tons $CO_2/10^{15}$ kcal)

B. Tons of CO_2 = (Generation kWh) * (Heat Rate kcal/kWh) * (Table 10.2 factor 10^6 Tons $CO_2/10^{15}$ kcal)

Table 10.2 U.S. DOE Table 11 in Emissions of Greenhouse Gases in the United States 1985-1990. DOE/EIA-0573

Fuel Type	Million Short Tons CO_2 per 10^{15} Btu	Million Metric Tons CO_2 per 10^{15} kcal
Distillate Fuel	79.9	288
Residual Fuel	86.6	312
Petroleum Coke	109.2	393
Special Naphtha	77.7	280
Anthracite Coal	112.5	405
Bituminous Coal	101.5	365
Subbituminous Coal	105.0	378
Lignite	106.5	383
Natural Gas	58.2	210

10.5 Examples

Given:

Heat Rate = 2520 kcal/kWh Generation = 6.7 10^9 kWh
HHV of coal = 6668 kcal/kg Coal Burned = 2.532 10^9 kilograms
% Carbon in as-fired fuel from Ultimate Analysis = 66.4%
Bituminous fuel

Calculations:

Bituminous fuel ⇒ from Table 10.2 ⇒ 365 10^6 tons CO_2 / 10^{15} kcal

10.5.1 Direct Measurement of Boiler Input

kilograms of CO_2 = (Kilograms of Fuel Burned) * (% carbon in as-fired fuel/100) * (3.664 kilograms of CO_2/kilograms carbon)

kilograms of CO_2 = (2.532 10^9 kilograms) * (66.4/100) * (3.664 kilograms of CO_2/kilograms carbon)

kilograms of CO_2 = 6.16 10^9 kilograms = 6.16 10^6 tons of CO_2

10.5.2 US DOE's Fuel Emissions Factor (Using Coal Burned and HHV)

Tons of CO_2 = (Kilograms of Fuel Burned) * (HHV of fuel kcal/kilogram) * (U.S. DOE factor 10^6 tons CO_2/10^{15} kcal)

Tons of CO_2 = (2.532 10^9 kilograms) * (6668 kcal/kg) * (365 10^6 tons CO_2 / 10^{15} kcal)

Tons of CO_2 = 6.16 10^6 tons of CO_2

10.5.3 US DOE's Fuel Emissions Factor (Using Heat Rate and Generation)

Tons of CO_2 = (Generation kWh) * (Heat Rate kcal/kWh) * (U.S. DOE factor 10^6 tons CO_2/10^{15} kcal)

Tons of CO_2 = (6.7 10^9 kWh) * (2520 kcal/kWh) * (365 10^6 tons CO_2 / 10^{15} kcal)

Tons of CO_2 = 6.16 10^6 tons of CO_2

10.6 Conclusions & References

For immediate implementation of CO_2 emissions tracking, the third method, detailed in Section 10.4, could be used. Since the type of fuel is known, and most Indian plants have a relatively good measurement of the amount of fuel burned, estimation of CO_2 emissions using U.S. DOE's Fuel Emissions Factor could be started immediately. For future installations, installation of stack sensors to use the direct measurement method should be considered.

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United States Federal Register, Volume 64, Number 101
40 CFR Parts 72 & 75 Acid Rain Program; Continuous Emission Monitoring Rule Revisions; Final Rule

40 CFR Part 60 Appendix - Test Methods
Method 1 - Sample and velocity traverses for stationary sources

Method 2 - Determination of stack gas velocity and volumetric flowrate (type S pitot tube)

Method 3 - Gas analysis for carbon dioxide, oxygen, excess air, and dry molecular weight

Method 4 - Determination of moisture content in stack gases

United States Department of Energy, Energy Information Administration. 1993. Table 11 in Emissions of Greenhouse Gases in the United States 1985-1990. DOE/EIA-0573

SECTION 11 ECONOMIC DISPATCH OF MULTIPLE UNITS

11.1 Introduction & History

Economic Dispatch (also known as Merit Order Dispatch) is the process by which each unit is loaded, so as to minimize the total cost of power production while providing the required power to the system. This process can be applied to two or more units at a single plant, to two or more units in a region, or to dozens of units at multiple sites. The result of economic dispatch is that the total cost of providing the required amount of electricity is minimized. The cost of production at any individual unit may not be minimized, but the total cost of the required energy will be minimized.

Prior to 1930, there were primarily two methods used to dispatch units in a system:

1. "Base Load Method" where the most efficient unit is loaded to its maximum capability, then the second most efficient unit is loaded to its maximum capacity, etc.
2. "Best Point Loading" where units are successively loaded to their lowest heat rate point beginning with the most efficient unit, and working down to the least efficient unit, etc.

These methods result in the more efficient units operating at a low cost, but the less efficient units are left producing power at a very high cost. The net result on the overall system is a production cost that is higher than it should be.

It was recognized by 1930 that the "Equal Incremental Cost Method" yielded the most economic results. The idea was for the next increment in load to be picked up by the unit whose incremental cost was the lowest; it was recognized that the net effect would be an equalizing of incremental costs. By 1943, the incremental cost characteristics were represented by straight line segments, so that the equal incremental criterion was conveniently applied.

The only cost that is considered is the variable cost of power production. The variable cost is made up of two components, fuel cost and variable "operations and maintenance" cost.

The fuel cost is the cost of the required energy to meet an assigned load. It is usually derived from the heat rate curve for each unit. Multiplying the heat rate curve by the unit output gives an Input versus Output curve (energy input rate versus net output). The derivative of the input versus output curve is the curve of incremental heat rate (kJ/kWh or kcal/kWh) versus net output (kW). Multiplying this curve by the fuel cost results in an incremental fuel cost curve.

While the total rupee amount of the variable operations and maintenance cost (O&M) for a specified period varies with the amount of generation, its increment value in Rs/MWh is considered constant over the entire operating range (not a function of load). It includes the O&M cost of pulverizer parts and labor, fuel handling, demineralized water production, etc. These costs vary with the amount of generation. Examples of fixed O&M costs that would not be included

are fire protection equipment, certain personnel costs (that are going to be working regardless of the load on the units), potable water, etc. These costs do not vary with the amount of generation, and therefore do not contribute any increment of cost when the load is changed.

Another category of cost that is not included is any sunk cost, such as the capital cost of constructing the plant. That cost is "sunk" or spent, and does not vary with the amount of generation. Therefore it is not a part of the incremental cost curve, and is irrelevant in economic dispatch.

11.2 Proof

The proof of the principle that the minimum combined heat input (or cost) for a given combined output is obtained when two units operating in parallel are operating at outputs which correspond to the same incremental heat rate value (or incremental cost value), if the input/output equation $I(x)$, dI/dx, and d^2I/dx^2 are continuous and $d^2I/dx^2>0$ is as follows:

If $I_a(X_a)$ and $I_b(X_b)$ are the required cost for unit A at output X_a and unit B at output X_b, respectively, then

$$I_c(X_c) = I_a(X_a) + I_b(X_b) \qquad (1)$$

and

$$X_c = X_a + X_b \qquad (2)$$

where I_c = The combined cost requirement (fuel and variable O&M) for the two unit system.
X_c = The combined output for the two unit system

Let X_d represent the system demand, then

$$X_c = X_d \text{ a given constant} \qquad (3)$$

Therefore,

$$X_a + X_b = \text{Constant} \qquad (4)$$

Differentiating (4) with respect to X_a yields

$$1 + dX_b/dX_a = 0 \qquad (5)$$

or

$$dX_b/dX_a = -1 \qquad (6)$$

Differentiating (1) with respect to X_a yields

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$$dI_c/dX_a = dI_a/dX_a + dI_b/dX_a$$
$$= dI_a/dX_a + dI_b/dX_b * dX_b/dX_a \qquad (7)$$

Substituting (6) into (7) gives

$$dI_c/dX_a = dI_a/dX_a - dI_b/dX_b \qquad (8)$$

Since I(x), dI/dx, and d^2I/dx^2 are continuous,

$$dI_c/dX_a = 0 \qquad (9)$$

or

$$dI_a/dX_a - dI_b/dX_b = 0 \qquad (10)$$

or

$$dI_a/dX_a = dI_b/dX_b \qquad (11)$$

will yield a minimum or maximum I_c.
Differentiating (8) with respect to X_a yields

$$d^2I_c/dX_a^2 = d/dX_a\ (dI_a/dX_a - dI_b/dX_b) \qquad (12)$$

$$= d^2I_a/dX_a^2 - d/dX_b\ (dI_b/dX_b) * dX_b/dX_a$$

$$= d^2I_a/dX_a^2 - d^2I_b/dX_b^2\ (-1)$$

$$= d^2I_a/dX_a^2 + d^2I_b/dX_b^2 \qquad (13)$$

Since d^2I_a/dX_a^2 and d^2I_b/dX_b^2 are both greater than zero,

$$d^2I_c/dX_a^2 > 0 \qquad (14)$$

Therefore, $dI_c/dX_a = 0$ yields a minimum I_c.
In other words, when $dI_a/dX_a = dI_b/dX_b$, I_c is a minimum value.

11.3 Mathematical Expressions

The first line of the following table shows three possible forms of Input versus Output curve, I(x). The first is a 1st order (a straight line), next is a 2nd order curve, and last is a 3rd order curve. The second line of the table shows each Input versus Output equation divided by the Output, which results in the equation for the heat rate, I/x. The third line shows the first derivative of the heat

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rate curve with respect to the output, dI/dx, which is the incremental heat rate curve. The fourth and fifth lines show the second derivative of the Input versus Output curves and the conditions that are required to make them positive.

	Input/Output Polynomials		
	1st Order	2nd Order	3rd Order
Input vs. Output	$I(x)=C_1x+C_0$	$I(x)=C_2x^2+C1x+C_0$	$I(x)=C_3x^3+C_2x^2+C1x+C_0$
Heat Rate	$I/x=C_1+C_0/x$	$I/x=C_2x+C_1+C_0/x$	$I/x=C_3x^2+C_2x+C_1+C_0/x$
Incremental Heat Rate	$dI/dx=C_1$	$dI/dx=2C_2x+C_1$	$dI/dx=3C_3x^2+2C_2x+C_1$
d^2I/dx^2	$d^2I/dx^2=0$	$d^2I/dx^2=2C_2$	$d^2I/dx^2=3C_3x+2C_2$
$d^2I/dx^2>0$	N/A	$C_2>0$	$C_2/C_3<0$

In Section 11.2, it was shown that if the equation of input versus output I(x), incremental heat rate dI/dx, and the second derivative of the equation of input versus output $d^2 I/dx^2$, are continuous and $d^2I/dx^2>0$, the minimum combined cost for a given combined output is obtained when the units operating in parallel are operating at outputs which correspond to the same incremental heat rate value.

In order to apply this principle, the incremental cost must always increase as the output increases. Therefore, the sum of the incremental fuel cost curve (the incremental heat rate multiplied by the fuel cost) and the incremental O&M cost must always increase as the output increases. If the incremental O&M cost is a fixed number (as is typical), then the incremental heat rate must always increase as the output increases.

The slope discontinuities at valve points and the non-convex nature along valve loops of a sequential valve machine will often conflict with the above principle imposed by the incremental dispatch model.

The economic dispatch computer model requires that an incremental cost curve be represented by a series of linear segments. Therefore, in order to have multiple line segments for the incremental heat rate, the input/output equation must be at least 3rd order. If the unit has a sequential arc turbine, usually one straight line is drawn between successive valve points. If the unit has a full arc turbine, the curve is divided into straight segments to get the best fit. The Equation of the Line Segment Between i^{th} and $(i^{th}+1)$ valve points is as follows:

$$(dI/dx - dI/dx_{at\ x=Xi}) / (x - X_i) = (dI/dx_{at\ x=Xi+1} - dI/dx_{at\ x=Xi}) / (X_{i+1} - X_i)$$

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For a third order input versus output polynomial, an incremental heat rate linear segment between X_i and X_{i+1} is:

$$[dI/dx - (3C_3X_i^2 + 2C_2X_i + C_1)] / (x - X_i) = [3C_3(X_{i+1}^2 - X_i^2) + 2C_2(X_{i+1} - X_i)] / (X_{i+1} - X_i)$$

or

$$dI/dx = mx + b$$

$$\text{where } m = [3C_3(X_{i+1}^2 - X_i^2) + 2C_2(X_{i+1} - X_i)] / (X_{i+1} - X_i)$$

$$b = (3C_3X_i^2 + 2C_2X_i + C_1) - m X_i$$

11.4 System Development

The steps required to set up the mathematical equations are:

1. Gather acceptance test results (only available at valve points)
2. Establish input vs. output curve
3. Develop incremental line segments between valve points

11.4.1 Input vs. Output Curve

Plot gross power supplied to the boiler (10^9J/hr) versus net electrical power produced (MW) preferably based on acceptance test data (if available). For a full arc admission turbine, it is an almost linear smooth curve. For a sequential valve turbine, it has a loop between valve points as shown in Figure 11.1 due to the throttle loss.

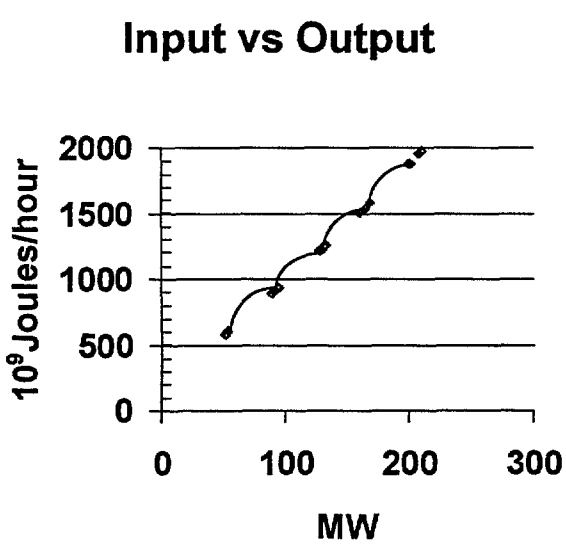

Figure 11.1 True Input vs. Output Curve

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Owing to the non-convex nature of the input-output, an accurate performance checkout model will often conflict with the requirements imposed by the incremental dispatch model.

I(x) continuous
dI/dx continuous
$d^2I/dx^2>0$

Due to constraints imposed by the incremental dispatch algorithm, a 3rd order polynomial is fit to the data at each valve point.

$$y = C_3x^3 + C_2x^2 + C_1x + C_0$$

$$I(x) = 2.24E\text{-}5x^3 - 1.33E\text{-}3x^2 + 7.81x + 187.7$$

$$dI/dx = 6.72E\text{-}5x^2 - 2.66x + 7.81$$

$d^2I/dx^2 = 1.34E\text{-}4x - 2.66$ and $d^2I/dx^2>0$ when x>min load

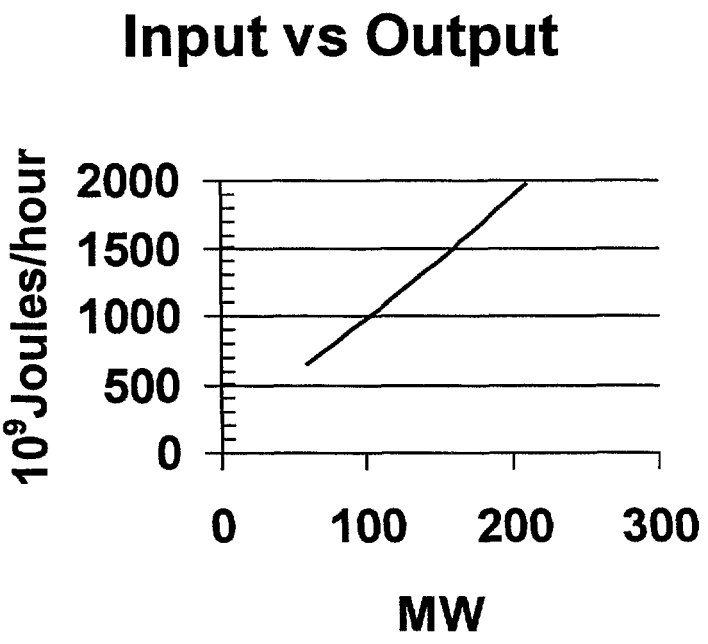

Figure 11.2 Smoothed Input vs. Output Curve

11.4.2 Develop Incremental Line Segments

Determine the equation for each line segment.

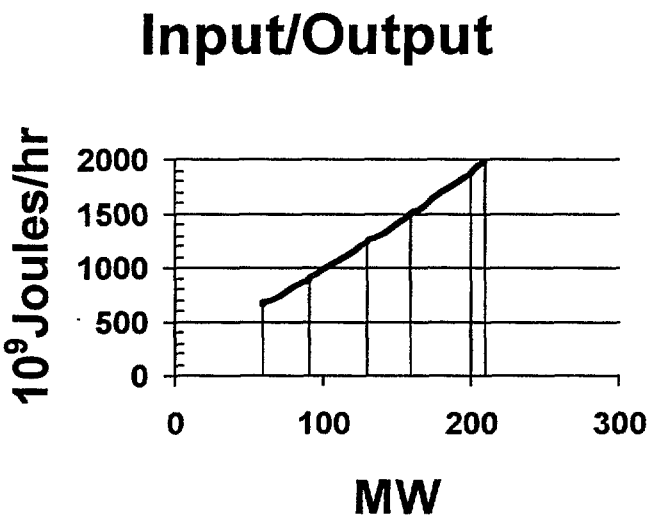

Figure 11.3 Segmented Input vs. Output Curve

$dI/dx_{x=x2 \text{ to } x1} = (dI/dx_{x=x2} - dI/dx_{x=x1}) * (x - x1) / (x2 - x1) + dI/dx_{x=x1}$

$dI/dx_{x=90 \text{ to } 50} = \{ (8110 - 7850) * (x - 50) / (90 - 50) \} + 7850$

$dI/dx_{x=90 \text{ to } 50} = 6.50x + 7525$

11.4.3 Incremental Heat Rate Curves

The curve below is a typical incremental heat rate curve for a unit in its original (performance guarantee) condition.

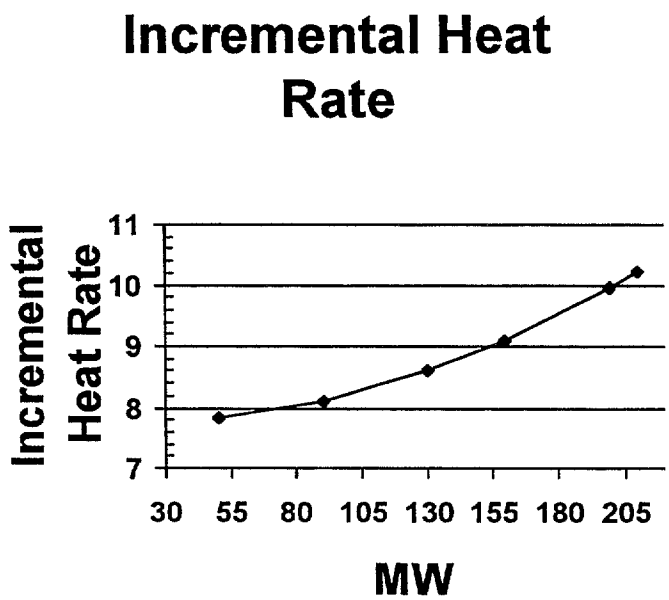

Figure 11.4 Incremental Heat Rate Curve

11.5 Incremental Heat Rate Adjustment Factors

Because the heat rate of units changes over time, the incremental heat rate (and incremental cost) curves must be updated periodically. One method of doing this would be to run a full heat rate test on each unit annually or perhaps every 6 months. Although this is a very expensive and time-consuming practice, there are several advantages to doing this.

1. Accurate, detailed information is obtained on the unit
2. The current position and shape of the actual performance of the unit, over the load range, is obtained.

A second procedure would be to have a real-time, continuous indication of not only the current heat rate of the unit, but an indication of what the heat rate curve is over the load range.

What is frequently done is to establish the original performance of the unit over the load range. Once this performance curve is made, it is left unchanged. Then an incremental heat rate adjustment factor is periodically calculated that can be multiplied by the reference curve to generate a curve that is representative of the current operation of the unit. The incremental heat rate adjustment factor is the ratio of actual heat rate to reference heat rate. Typically, this periodic calculation is done using full load data (actual heat rate and the reference heat rate at full load), and the resulting adjustment factor is assumed constant over the load range. However, adjustment factors could be calculated at several different loads, and instead of having a single factor, a curve of the incremental heat rate adjustment factor versus load could be developed.

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If:

$$\text{Expected Heat Rate} = \text{Reference Heat Rate} * (1 + D_1) * (1 + D_2) * \ldots * (1 + D_m) \quad (1)$$

where D_m = The fractional effect on heat rate due to m^{th} design change, and

$$\text{Actual Heat Rate} = \text{Expected Heat Rate} * (1 + O_1) * (1 + O_2) * \ldots * (1 + O_n) \quad (2)$$

where O_n = The fractional effect on heat rate due to n^{th} operational deviation from expected level, then, substituting (1) into (2) yields

$$\text{Actual Heat Rate} = \text{Reference Heat Rate} * (1 + D_1) * (1 + D_2) * \ldots * (1 + D_m) * (1 + O_1) * (1 + O_2) * \ldots * (1 + O_n) \quad (3)$$

$$= \text{Reference Heat Rate} * (1 + P) \quad (4)$$

Therefore,

$$1 + P = \text{Actual Heat Rate} / \text{Reference Heat Rate} \quad (5)$$

Also,

The incremental heat Rate Adjustment Factor

$$= \text{Actual Heat Rate} / \text{Reference Heat Rate} = 1+P \quad (5)$$

It can be easily proved that

$$\text{Actual Incremental Heat Rate} / \text{Reference Incremental Heat Rate} = 1+P \quad (6)$$

For the convenience of the dispatching data management, the reference heat rate curves (for each unit) are stored in the economic dispatching model as fixed bases. The incremental heat rate adjustment factors (one for each unit) are updated periodically, typically quarterly. They are then used to update the "input versus output" curve, the heat rate curve, the incremental heat rate curve and, ultimately the incremental cost curves, and entered into the dispatch algorithm.

Usually the incremental heat rate adjustment factor is calculated at full load. For example, if a 200 MW unit measures its actual heat rate daily, then the daily heat rates and reference heat rates would be calculated each day. For each day when the load was continuously high (between 185-200 MW for example), the actual heat rate would be divided by the reference heat rate to calculate an incremental heat rate adjustment factor for that day. Those incremental heat rate adjustment factors are averaged to determine the incremental heat rate adjustment factor to be used in the dispatch algorithm for the next period.

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For example, if the incremental heat rate adjustment factor is 1.15, then the original reference curve and the curve representing the current incremental heat rate would look like Figure 11.5. The adjusted curve would be used for dispatch.

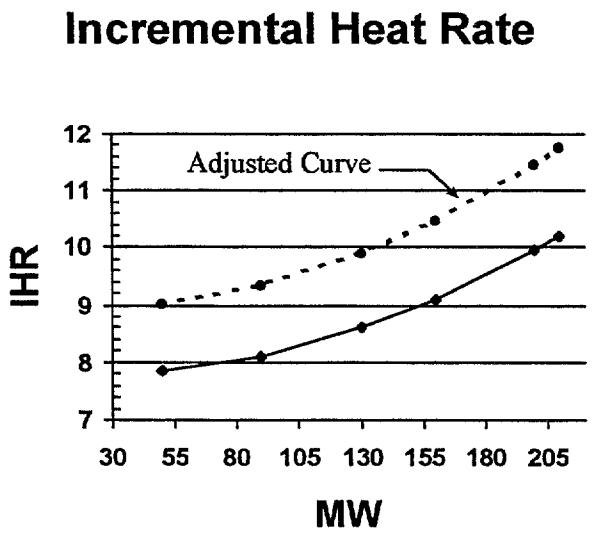

Figure 11.5 Incremental Heat Rate Curves, "Reference" and "Adjusted" for Actual Condition

11.6 Example

The following is an example of dispatching based on the "Equal Incremental Method", comparing two unit's incremental heat rate curves (see Figure 11.6).

The units would be dispatched so that their combined output would meet the demand while having the same incremental rate.

If the demand was 190 MW, the outputs of "Unit A" and "Unit B" at the same incremental rate of 8.55 would be 60 MW and 130 MW, respectively.

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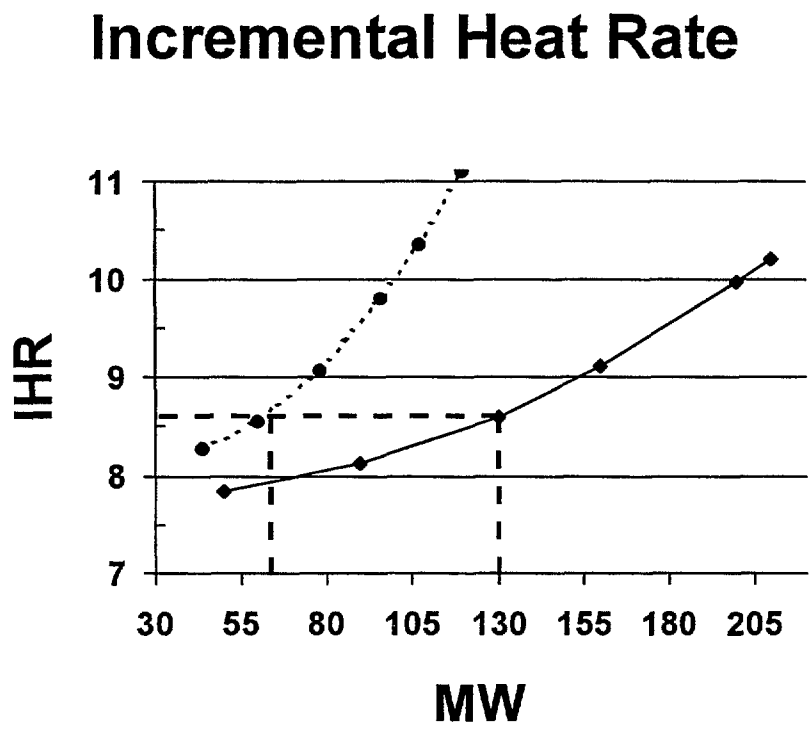

Figure 11.6 Incremental Heat Rate Curves, Unit A and Unit B

11.7 Conclusions

Units are dispatched based on the "Equal Incremental Cost Method."

The Incremental Cost Curves are derived from the incremental heat rate curve, and the variable O&M cost.

The minimum cost for a given combined output is obtained when the incremental costs of each generating unit are equal.

Even though the "Input versus Output" curve may follow a convex shape, the dispatch model forces the valve loops to fit in at least squares form with a positive slope for each segment.

The incremental cost curve is the sum of the "incremental heat rate curve" plus the incremental "variable operation and maintenance" costs.

Incremental curves should be updated periodically to better reflect the unit's current condition.

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11.8 References

Economy Loading of Power Plants and Electric Systems, by Max J. Steinberg and Theodore H. Smith, published by John Wiley & Sons, Inc.

Optimal Power Dispatch - A Comprehensive Survey, by H. H. Happ, published in the IEEE Transactions on Power Apparatus and Systems, Vol. PAS-96, no. 3, May/June 1977.

SECTION 12 TYPICAL AREAS FOR HEAT RATE IMPROVEMENTS

12.1 Introduction

This section reviews some areas where heat rate improvements are possible at many plants. It should not be used in place of reviewing all heat rate deviations, investigating the problem to determine the root cause, and implementing the appropriate solution. However, these eleven areas are typical opportunities for improving efficiency, reducing emissions, reducing maintenance, and obtaining other additional benefits. Not all plants have problems in each of these areas, but in many plants, these problems are commonly encountered.

Some of these problems show up as "unaccountable" heat rate deviations, which are not readily apparent, therefore, they often go unnoticed (such as cycle isolation). Other potential improvement areas are overlooked because the true "expected" performance level is not defined (such as condenser performance or boiler outlet O_2).

Most of these improvements require little, and in some cases such as variable pressure operation, no expense (material or manpower). Usually, these items can be implemented and improvements realized in a short amount of time.

12.2 Improved Condenser Cleanliness

In almost all plants, there are opportunities for increased thermal efficiency is by increasing the cleanliness of the condenser. Even on units that have a closed loop condenser circulating water system, with treated water, over time, deposits (organic, inorganic, or both) will form on the internal diameter of the condenser tubes. The deposit or "fouling" does not have to be very thick, it may not even be apparent to the eye, for it to "insulate" the tubes. The additional resistance to heat transfer causes the condenser pressure (and steam temperature) to increase, thereby increasing the temperature differential, so that the heat of condensation can flow through the tube, to the circulating water. This increase in condenser pressure increases the heat rate and decreases unit load. If the unit is not restricted by steam flow, coal flow, air flow, etc., the firing rate may be increased to maintain the unit load, but at an additional heat rate penalty. Other times, especially in the summer, if the steam flow cannot be increased, the unit output will decrease.

Many plants clean their condensers only once a year, during annual outages. For almost all units, this is insufficient, and results in higher production costs (and emissions) than necessary. A cost versus benefit analysis should be done, comparing the cost of cleaning a condenser to the heat rate improvement that will result, to find the optimum cleaning cycle that minimizes the total cost (cost of cleaning and cost due to the heat rate penalty from dirty tubes).

This approach requires that the "true expected" condenser pressure be known for a unit. This is not a constant ("design") value. The "true expected" condenser pressure is frequently below the "design" pressure, and sometimes above. It varies with the heat load on the condenser (primarily

influenced by the unit load), the circulating water inlet temperature, and the circulating water flow rate. Once the "true expected" condenser pressure is known, it can be compared to the actual condenser pressure, and using the turbogenerator's condenser pressure versus heat rate correction curve, the heat rate deviation and cost can be calculated.

Since poor condenser cleanliness is not the only cause of high condenser pressure, the cleanliness of the tubes should be measured periodically. (Otherwise, the tubes might be cleaned when the cause of the high condenser pressure is another problem, such as high air inleakage, low CCW flow, low waterbox level, etc.) This normally requires a condenser test or, on some units, cleanliness may be calculated continuously via readings from station instruments. Either way, with periodic tests or continuous measurements, an estimate of the "fouling rate" of the condenser can be established, and the proper decision on when and how often to clean can be made.

It is not unusual for units in the 200 to 500 MW range to require multiple cleanings each year. Since fouling has a greater impact in the summer (because for a given change in condenser pressure, the change in heat rate is larger at higher condenser pressure, which occurs in the summer) more cleanings will be required at the start and during summer, than in winter.

With inexpensive equipment, plant personnel (or a service provider) should be able to clean 5000 tubes in a waterbox in one 8 hour shift. This makes on-line cleaning of one waterbox, during the night when demand is reduced, very practical and cost effective.

On-line cleaning requires that the load be reduced so one waterbox can be removed from service. Another requirement is that the inlet valves (and in some designs, the outlet valves) are functional, and while they do not have to be watertight, the leakage should be small.

For on-line cleaning (and in many cases, cleaning during annual outage) brushes or scrapers are "shot" through the tubes with water only or water and air pressure. These brushes usually have nylon bristles that can remove organic and light inorganic fouling. For harder scales, metal scrapers are used. Frequently, an objection is raised to the use of metal scrapers, especially when used to clean copper alloys. However, properly used, metal scrapers will not reduce the life of the tubes. Some points to remember when cleaning condenser tubes:

1. Use the correct size of brush or scraper. The brushes should fit tight in the tubes, otherwise that will not remove any material. Some units may require more than one size cleaner, as the condensing zone and air removal tubes may have different internal diameters.
2. The brushes and scrapers have a relatively short life, 5-10 passes, after that they will be worn out enough that they should be discarded. If the cleanliness factor does not improve after a cleaning, frequently the root cause is found to be the use of worn brushes or scrapers.
3. Use the proper brush or scraper. On many units, brushes will be sufficient, but some units may occasionally need metal scrapers (for example, some plants use nylon brushes twice a year and once a year they use metal scrapers). There are different types of brushes and scrapers as well.

4. Do not leave any brushes or scrapers in a tube, as this will contribute to corrosion and eventually a tube failure, as will any foreign material that sticks in a tube. To help ensure that no cleaning devices are left in the tubes, count how many cleaners are supplied to the cleaning crew, and how many they return.
5. For fastest cleanings, first, place a cleaner in every tube in the waterbox, then have two guns shooting cleaners through the tubes in the upper half, and two guns working in the lower half.

Through a regular cleaning program, the average condenser pressure over a year can be reduced, especially in the summer when it is most important. HP jet cleaning may still be required during annual outages, but frequently, if the tubes are brushed and/or scraped during the year, nothing else may be required. Chemical cleanings (sometimes followed by shooting brushes or scrapers or by HP jet) may be required initially to remove thick/hard scale that have accumulated over time, that other methods alone will not remove. Once the thick/hard scale is removed, the tubes usually can be kept clean by a regular program of brush or scraper cleanings during the year.

12.3 Variable Pressure Operation

Many power plants built in recent years are designed to be able to operate in full variable pressure mode. These units have throttle governed (full arc admission) turbines. Also, other units that were not specifically designed for full or hybrid variable pressure mode, can operate in one of these modes. Some units in India, including units that were not specifically designed for variable pressure operation, are currently operating in a variable pressure mode.

In normal Full Pressure (FP) (also known as Constant Pressure or (CP)) operation, the main steam pressure ahead of the turbine stop and control valves is maintained at its "design" value (i.e. 130 kg/cm^2 or 150 kg/cm^2 or 170 kg/cm^2, etc.) At full load, the control valves are "close" to being full open, and there is some relatively small pressure drop across the control valves. Along with the drop in pressure, there is also a drop in temperature, since the enthalpy of the steam is unchanged. At low loads, the pressure and temperature drop across the valves can be very large. The result is that the steam pressure and temperature entering the turbine is much below the "design" conditions.

Instead of controlling the unit in this manner, there are three main alternatives, each of which can have some minor variations. Each of these are called variable pressure or sliding pressure operation.

In Full Variable Pressure (FVP) mode, the control valves are left in the full open (valves wide open or VWO) or a fixed, slightly closed position, and the unit load is changed by changing the throttle pressure. At full load, the throttle pressure will be close to the "design" value, at minimum load, the throttle pressure may be only 50% of the design value. This mode is usually adopted for sub-critical units with throttle governed or full arc admission turbines (the turbine valves all open and close simultaneously). This mode will result in an improvement in heat rate over the entire load range of the unit, with larger improvements at lower loads.

Hybrid Variable Pressure (HVP) is usually used on nozzle governed or sequential arc turbines, where the control valves open (and close) one at a time. In this mode, the load is dropped in the normal manner down to some load where one or more valves are completely shut, and the other valve(s) are either wide open, or just beginning to close. To reduce load further, the valves are left in this position, and the throttle pressure is reduced. At loads above this change-over point, there is no change in heat rate. Below the change over point the heat rate improvement increases as the load decreases. The heat rate improvements in this mode are not as large as with full variable pressure.

Hybrid variable pressure is also run on throttle governed (full-arc) turbines. On these units, instead of operating with the valves fixed wide open, the valves are fixed at a point close to wide open, around 85%, and the load on the unit is controlled by varying the throttle pressure. The gains are not as high as with full variable pressure, but it shows significant improvements over full pressure operation, and there is some valve movement available (both open and closed) for grid emergencies.

The third type of variable pressure is called dual pressure. It is applied at supercritical units, cyclone units, and other units that require full pressure in the water cooled circuits for proper circulation and cooling. The design pressure is maintained through the waterwalls and primary superheater, but instead of operating the turbine control valves to control the unit load, "throttling division valves", between the primary and secondary superheater are used. In this manner, the steam temperature at the inlet to the turbine is maintained at its design value. Generally, in this mode, the heat rate improves over the entire load range, with larger improvements at lower loads, but the improvements are not as large as with full variable pressure.

If a unit was designed for variable pressure, it should be run in that mode, as the heat rate and therefore operating cost and emissions will decrease. The most common objection to variable pressure operation is that the unit will not change load as rapidly as in full pressure mode. This is usually true; however, unless the system frequency is required to be very steady (<0.01 Hz variation), this is not a problem.

As with any major change to a unit (fuel changes, boiler surface changes, etc.) potential problem areas should be identified and considered. Below are two such lists for moving from full pressure to variable pressure operation. On most units, most of these potential concerns will not cause problems, but the two lists do provide some guidance for areas to consider before switching to a mode of variable pressure operation.

The first list is of potential "reliability concerns" that are sometimes raised when considering a type of variable pressure operation on older units that were not designed specifically for that mode of operation.

1. Increased thermal cycling of drum.
2. Warping of drum due to increase in differential expansion.
3. Exceeding allowable stress in fixed connections to drum due to warping.

4. Increased damage to drum internals.
5. Increased thermal cycling of primary superheater inlet header.
6. Increased fatigue damage to primary superheat tubes going to end of inlet and outlet header due to flexing caused by relative expansion of boiler walls and roof.
7. Increased creep fatigue damage to primary superheat tubes caused by increased temperatures.
8. Exceeding allowable stress due to increased outlet temperature in primary superheat outlet header.
9. Increased of thermal cycling of secondary superheat inlet header due to increased attemperation.
10. Increased waterwall tube failure by overheating (DNB) in areas of high heat flux.
11. Increased waterwall tube failure by overheating (DNB) due to reduced circulation.
12. Increased waterwall tube failure by overheating in sloped tubes due to steam aggregation.
13. Increased waterwall tube failure by overheating during emergency shutdown.
14. Increased waterwall tube failure by overheating due to flow imbalance.
15. Exceeding allowable stress in economizer by overheating due to flow imbalance caused by steaming.
16. Exceeding allowable stress due to increased economizer outlet temperature.
17. Increased stresses on the casing connections to water cooled parts due to changing saturation temperature.
18. With steam driven boiler feedwater pumps, with close coupled booster pumps, at the lower speed, will the booster pump supply enough net positive suction head for the BFP.
19. Increased air heater cold end corrosion due to potentially lower air heater exit gas temperatures.

The following is a list of potential operational concerns that are sometimes raised when considering a type of variable pressure operation on older units that perhaps were not designed specifically for that mode of operation:

1. Limits for firing rate will be dependent on limiting component allowable temperature rate of change.
2. Limits for drum temperature rate of change will vary as a function of saturation temperature.
3. Decreases due to over-firing limits on allowed temperature changes.
4. Increased superheat attemperation required due to increased steam temperatures.
5. Increased reheat attemperation required due to increased steam temperatures.
6. Increases boiler swell effect.
7. Decreased water quality by loosening deposits due to temperature changes in saturated circuits.

12.4 Improved Cycle Isolation

A common problem found at many plants is improper cycle isolation. This includes those steam and water leaks that can be seen, but more importantly, it includes those "internal" leaks that do not cause increased makeup. These leaks include drains to the condenser, recirculation lines, attemperation lines, heater bypass lines, heater drains, etc., that should be closed and sealed, but frequently are inadvertently left open or leak through.

The first area of cycle isolation is one that is usually overlooked. It is high energy drains (mostly startup drains) that leak through to the condenser (perhaps first passing though a drain or flash tank). Because these leaks are not visible, and do not cause a derating unless they get very bad, they are easily overlooked. This category of drains includes both boiler (i.e., superheater header drains, reheat header drains, etc.) and turbine cycle drains (i.e., stop valve above and below seat drains, turbine loop line drains, traps on extraction piping, etc.). Leaks through these valves cause large heat losses, increase condenser pressure, and can cause load reductions.

The second category of cycle isolation includes feedwater heater drains that should cascade to the next lower heater but, instead are dumped to the deaerator or condenser; low pressure feedwater heater drains that should be pumped forward, but are dumped to the condenser; boiler feedwater pump minimum flow recirculation valves that leak through; attemperation valves that leak through; and other flows that are occasionally needed during operation. Frequently, these valves are automated, and because they are sometimes opened during operation, if they leak through, they are considered insignificant.

Both of these categories of cycle isolation problems are extremely detrimental to the thermal performance of a unit. Unfortunately, they are also frequently missed when looking for improvement opportunities. Every plant should have a program to maintain tight cycle isolation of the units. If there hasn't been a concentrated program to do this in the past, it may take some effort to get the unit in a relatively "tight" condition, then the routine cycle isolation program should, relatively easily, be able to maintain a "tight" cycle by correcting problems quickly.

The first part of a cycle isolation program is to know exactly which valves are leaking. In some plants, there is a haphazard approach to this. Occasionally, someone will look over part of the unit, and if they find something, it gets reported. While this approach will find some problems, it probably will not find them all, and it may not find big problems until they have significantly impacted the heat for a long time.

A better approach is to first review the plant drawings, making a list of all valves, and traps that can leak through. Second, divide these valves and traps into 2-4 categories, depending on the relative magnitude of the heat rate (and in some cases load) impact if it leaks. The big question that always comes up is, "How much flow is passing ?" If this is known, using thermal modeling software, the heat rate impact can be determined. However, coming up with an estimate of the flow is guess work at best. There are techniques for measuring the temperature on the pipe at two locations, and with an estimate of the pipe's insulation, calculating the flow rate. Other methods include some analysis of the acoustic signal that can be measured upstream and

downstream of the valve, and personnel with a lot of experience with locating leaks can usually give an estimate of the size of the leak. A third approach is to install orifices with pressure taps for "measuring" the flow. Unfortunately, often these methods are more time consuming than repairing the leaking valve, and the results are not much better than a guess.

Once the valves/traps are divided into categories, each category should have a methodology and frequency for checking the valves in that category. For the top priority category, which would include main steam, hot reheat steam, secondary superheat inlet headers, etc., it is recommended that a permanent, "continuous" monitoring system be installed. Normally thermocouples are tack welded to the OD of the pipe. The thermocouple extension wire is run to the control room where the temperature can be displayed via a multi-position switch, or perhaps connected to the DAS. Because leaking, high energy drains are so detrimental to heat rate, and because of their difficult location, they sometimes are overlooked after a startup, they should be monitored at least daily.

For the second category valves, monthly checks may be sufficient. For still lower categories, quarterly, and the lowest category may only be checked before annual outages.

There are several methods available for periodic checking of these lower priority valves. The first is similar to the continuous monitoring of category 1 valves, but instead of permanently attaching thermocouples, a portable, handheld reader with a thermocouple attached is used to check the temperatures upstream and downstream of valves. Another method that is gaining popularity is acoustic monitoring.

Once the leaking valves are identified, they must be repaired. These leaking valves, especially category 1 valves, should be on the top of the short shutdown list, so that they can be repaired at the first available opportunity.

12.5 Control Loops in Functioning in Automatic

Present day power plants are very complex systems. Because of this, they are designed with most control loops having automatic control in addition to manual control stations. Under most circumstances, manual control is possible. However, in most cases, manual control cannot approach the effectiveness of a properly tuned automatic control system. With manual control, temperature swings, pressure swings, etc. are normal. These swings are not only inefficient, but they are detrimental to the equipment. Therefore, emphasis should be placed on maintaining all control loops in automatic control, and to "tune" the controls whenever there are significant changes to the equipment or the fuel.

Some of the more important control loops that should be maintained in automatic mode are:

- Furnace draft
- Drum level/FW flow
- Deaerator and hotwell level
- Superheat and reheat steam temperature (including attemperation and burner tilts)

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- Primary air header pressure
- Mill outlet temperature
- Primary air flow
- Windbox pressure
- Combustion control with O_2 trim
- Feedwater heater levels
- Coordinated master control

With these loops tuned and in automatic operation, the unit should run efficiently, even during load changes, or other disturbances.

12.6 Balancing Primary Air Flows to Each Burner

In order to have proper combustion, a certain ratio of primary air to fuel should be maintained at each burner. If this ratio is not maintained, slagging, excessive NOx and CO formation, uneven convection pass metal temperatures, coal settling in pipes, an increase in unburned carbon in ash, reducing atmospheres, and other detrimental impacts can occur.

The first step toward obtaining this proper and equal air-fuel ratio at each burner is to balance the "clean" air flows in each coal pipe. This should be done for each pulverizer after any major pulverizer work is performed or if coal pipes or burner maintenance is performed. At the same time this test is being carried out, the mill primary air flow calibration can be checked by comparing the sum of the air flow through each pipe to the indicated primary air flow.

Section 4.7.6 lists the details of when and how to perform this test. The general procedure is to run a mill with no coal and measure the air flow in each pipe. This should be done over the air flow range of the mill. It should (if possible) be run with the mill outlet temperature as close to the normal set point as possible. This requires that the unit be on-line. These tests can also be run during outages, but preferably, they should be run with the unit on-line so the air temperature is at the normal mill outlet temperature.

For each mill, the sum of the air flow through each pipe can be compared to the station primary air flow indication for the mill, and adjustments made to the station indications as necessary. Second, for each mill, at each loading, the air flow in all pipes is averaged, and the flow in each pipe is compared to the average. The goal is to have the maximum deviation from any one pipe to be less than ±2%.

If the air flows are not within ±2% of the average, the orifices in the coal pipes need to be adjusted. This is somewhat a trial and error process, but it will pay dividends in the end. Also, as experience is gained, the trial and error process of adjusting orifices becomes easier.

12.7 Reduced Furnace O_2 Levels

The two largest thermodynamic losses in a power plant are from the heat rejected in the condenser, and the heat that goes out the stack. Typically five percent or more of the energy that is put into the boiler is lost out the stack. This loss can be considered in two parts, the energy level (temperature) of the gas, and the quantity (flow rate) of gas. The flow rate is primarily set by the O_2 level at the economizer outlet/air heater inlet.

The "required" O_2 level varies from unit to unit, and at any unit the "required" O_2 level varies with time due to changes in the fuel supplied, boiler load, tightness of the furnace and casing, and other factors. At many plants, the amount of air supplied to the furnace is greater than what is necessary for efficient, reliable, and safe operation. A very generic "rule-of-thumb" is that 2-3% O_2 should be available at the *furnace* exit (however, some units can run with less than 2%, and some units require more than 3%). In order for a plant to determine the O_2 level it needs, a series of optimization tests can be run (see Section 4.7.11 for details on boiler optimization testing), varying various parameters. It is not uncommon to find at units that are relatively tight (little air in-leakage), that the O_2 (measured at the economizer outlet) can be reduced one percentage point or more from the normal setting. This reduction in O_2 does not adversely affect the operation of the unit, but does significantly improve the thermal performance of the unit. Even small reductions in O_2 (1/4 to 1/2 percentage points) can result in noticeable improvements in heat rate.

This does not imply that all units can reduce the O_2 levels in their furnaces, only that at many units, there are opportunities for large heat rate improvements, with only the minor costs for some testing. Another benefit is that the optimization testing often finds other areas where efficiency improvements can be made.

Frequently, arguments are made against reducing O_2 based on instrument problems. These should be addressed by ensuring that the on-line O_2 monitors are representative of the average O_2 level in the duct. This may require multiple sensors, or ensuring that the location of the on-line sensor gives representative values at various conditions. Another instrument that can aid in reducing O_2 is a CO monitor. A third method to help provide assurances that O_2 is not being lowered too much is to perform HVT traverses, measuring the furnace outlet O_2. If the unit operates most efficiently at very low levels of O_2 (less than 2%), monitoring the waterwall tubes for wastage, due to operating in a potentially reducing atmosphere, should be considered.

12.8 Reduced Station Service

Another area for heat rate improvement that is often overlooked is auxiliary power. At many plants, the emphasis is on *gross generation*, the amount of power leaving the generator. Instead of looking at gross generation, what should receive the most attention is the *net generation*, or the amount of power that leaves the plant. The difference between these two is the station service or auxiliary power, the power consumed by equipment in the plant.

One reason that the emphasis is put in the wrong place is that the total unit auxiliary power is usually the combination of two unit boards and prorated values from various other boards (i.e. coal handling, intake structure, etc.). In most plants, the equipment to sum and display in real

time just the unit boards is not provided (and even fewer units have provisions to prorate the common auxiliary power). Where this is the case, integrators are usually read periodically and the net generation calculated hourly or each shift or day, but this is after the fact. Therefore, one way to improve the heat rate is to provide a real-time indication of net generation and auxiliary power (at least the sum of the unit board's auxiliary power if not the total unit auxiliary power), as is done with other critical parameters such as condenser pressure, boiler outlet O_2, etc.

Another way to reduce station use is to set-up procedures to ensure that equipment that is not required at lower loads is idled. One of the first pieces of equipment that should be idled as load is dropped is pulverizers. Another savings from removing a mill from service as the load is reduced is that, by keeping the minimum number of mills in service, the air flows though the mills is maintained high enough to prevent coal from settling out. If an excessive number of mills are in service, each may be backed down to a point where the air velocity is not sufficient to keep the coal in suspension. At some plants, ID fans and/or BF pumps can be removed from service at low loads.

Another savings in station service that must be weighed against a potential increase in condenser pressure is, at low inlet water temperatures during the winter, a circulating water pump may be able to be taken out of service without increasing the condenser pressure. When this should be done will depend on the flow thorough the condenser, the inlet temperature, and the heat load on the condenser.

Another frequent cause of high station service is some equipment is required to be in service (that should not be) due to poor performance. For example, some units should be capable of running full load with one condensate pump, but because of wear, etc., to both pumps, neither pump can carry full load alone. So at full load both pumps are run. This also sometimes occurs with boiler feedwater pumps and pulverizers. Another example is where a unit's heat rate has deteriorated, causing the feedwater flow and air flow to be increased, to maintain the same load. Then at some loads, additional pumps/fans/etc. must be in service to maintain that load.

Another method to reduce auxiliary power usage is to periodically test equipment and, where there are redundant pieces of equipment, to run the more efficient pieces. For example, if the unit has three boiler feedwater pumps, and one is much less efficient than the other two, it should be held in standby until it can be overhauled.

12.9 Key Instrument Calibration

Section 9, New Plant Specifications, discusses the need to have high accuracy instrumentation at key locations. These "key instruments " fall into two categories. The first category includes instruments that the operator and/or control system uses to control the unit, and also have large impacts on heat rate. Example of instruments that fall into this category are:

1. Initial steam temperature and pressure
2. Hot reheat steam temperature

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3. Boiler outlet O_2
4. Exit gas temperature
5. Mill outlet temperatures
6. Primary air flow rates
7. Waterbox differential, etc.

The second category includes instruments that measure conditions that are not directly controlled by the operator, but these instruments are used to monitor the actual performance of the unit. Instruments in this category include:

1. Final feedwater temperature
2. Attemperation flows
3. Makeup flow
4. Condenser pressure
5. Station service, etc.

In Appendix D, Heat Rate Parameters - Effects, Causes, and Corrections, it is shown that, on average, a 1°C deviation in initial steam temperature, causes a 2.7 kJ/kWh (0.64 kcal/kWh) deviation in unit net heat rate. Individual thermocouples frequently deviate 2°C from the "standard" at high temperatures. Also, over time, and with temperature cycles, thermocouples will drift. This alone can cause significant errors in the indicated temperature. If the *indicated* temperature is higher than the *true* temperature, then the thermal performance of the unit will decrease. Errors in pressure readings, and flue gas O_2 are also frequently the cause of lost efficiency. What makes these losses especially unfortunate, is that:

1. They are easily avoidable.
2. If they are not detected and corrected by a proper calibration program, the heat rate losses they should measure end up in the "unaccountable" category.

How can these losses be avoided, or at least be detected quickly and corrected? One action that can be taken is to install better accuracy instrumentation in critical locations (i.e., RTDs instead of thermocouples; higher accuracy pressure transmitters, etc.). A second action is to install multiple sensors at critical locations. For example, install two or three RTDs in each initial steam pipe at the turbine.

Another vital step is to have a calibration program. This will enable any instrument errors that occur to be detected and corrected.

The first part of the "key instrument calibration" program is to make a list of the key instruments. The two lists above are a start. For each unit, someone should consider each instrument and ask these two questions:

1. If this instrument is in error, can it adversely affect the efficiency of the unit?
2. If this instrument is in error, will it affect the calculated heat rate or the calculated heat rate deviations for any parameter(s)?

If the answer to either is "yes", it should be considered a critical instrument. (This is not to say that other instruments are not critical for *reliable* and *safe* operation of the unit, only that these are critical for *efficient* operation of the unit.)

Second, for each instrument, or type of instrument, a written, detailed, calibration procedure should be made. "Instrument calibration" means different things to different people. To some, it means that various signals were introduced at the sensor (i.e. 4-20 mA or 0-20 mV, etc.), were read in the control room in engineering units (i.e., kg/cm^2 or °C, etc.), and were checked. To some it means the sensor was removed, brought to a shop, and bench tested. Neither of these two procedures, by itself, is a calibration. If these two parts are put together properly, the entire "loop" would be calibrated. However, too often, only one part of the job is done. A full loop calibration should always be performed. Also, if the bench check of a thermocouple shows a 2°C error at 550°C, a 1°C error at 450°C, a ½°C error at 350°C, and minimal error at lower temperatures (as is common for thermocouples), can the equipment that is reading it (a data acquisition system, recorder, LCD indicator, etc.) accept a calibration curve, or will it only use the standard tables? If the system reading the sensor cannot "use" the calibration curve, at what "deviation from standard tables" should a thermocouple be discarded? This also must be addressed in the calibration procedure. If the calibration curve or data can be entered into the DAS, this step must be part of the calibration procedure.

A last part of "key instrument calibrations" is setting a calibration frequency for each critical instrument. This will depend on several factors:

1. The potential heat rate penalty (and therefore fuel cost) of any error.
2. Are there redundant sensors at that location that can be used as a check?
3. Are periodic tests run using high accuracy sensors, that measure this point in parallel, that can be used as a check of the station instrument's accuracy?
4. Is there any calculated data (from other sensors) that can be used as a check?
5. Can this sensor be calibrated with the unit on-line (can it be temporarily removed with the unit on-line)?
6. What is the cost of the calibration?

Once the key instrument calibration program is set up, and running, it should be monitored periodically. Are some sensors being tested more often than is required? If they are showing negligible changes, perhaps the time between calibrations can be extended. Are some sensors showing a drift at every check? If so, are they being checked often enough, or perhaps a different type of sensor should be used at that location?

By setting up a proper calibration program, heat rate losses due to these instrument inaccuracies should be minimized.

12.10 Improved Coal Fineness

Proper fineness is essential for good combustion, as well as preventing slagging and delayed combustion, etc. When looking at fineness, it is important to consider both the high end and low end, or both the amount passing (or retained) on the 200 mesh and the 50 mesh screens.

Before analyzing the results of a fineness test, the sample should be checked to ensure that the sample was collected properly (that the sample is not biased). To do this, at least three sieves (50, 100, and 200 mesh) and preferably four sieves (50, 70, 100, 200 or 50, 100, 140, and 200) should be used and the % passing each sieve determined. These results are then plotted on a "Rosin and Rammler" diagram. If the three or four points do not fall in a straight line, the sample is biased, and should be discarded. The sample collection procedure should be checked and another sample collected. Isokinetic sampling may be required to obtain a representative sample. Also, all coal pipes from a mill should be sampled.

Some mills may have good 200 mesh fineness but have 1-2% remaining on a 50 mesh screen. Most of this very coarse coal will not burn, and will end up in the bottom ash. It also frequently causes slagging problems around the burners.

Other mills may have good 50 mesh fineness, but poor 200 mesh fineness. This will frequently contribute to high combustibles in the fly ash.

Whenever the topic of "What should the fineness be?" is brought up, there is always a lively discussion. This will vary from unit to unit, due to mill type, furnace configuration, coal properties, etc. With high ash and abrasive coal (resulting in a high wear rate) the fineness may be reduced slightly to extend the life of pulverizer grinding components and pipe materials. Where slagging and/or high combustibles in ash is a problem, the fineness may be held at a high level. Some rough "rules of thumb" are that the amount of coal passing through the 200 mesh screen should be in the range of 70-75%. The amount of coal retained on a 50 mesh screen should be no more than 0.1 to 0.5%

In addition to the problems listed above, poor fineness causes problems with coal distribution to the burners. If the coal is not properly ground, the distribution to the burners may not be even, resulting an air-fuel imbalances at one or more burners.

A secondary problem resulting from poor fineness is often additional excess air is supplied to the boiler to attempt to compensate for the poor fineness. This may partially compensate for the fineness, but it comes at the price of higher dry gas loss and higher auxiliary power usage.

Some important factors to look at include:

1. Classifier setting in the proper position.
2. Ensure that the "skirt" does not have any holes which would allow coal to bypass the classifier.
3. Throat clearances are proper and uniform around the bowl.
4. Maintaining proper and equal grinding pressures.
5. Proper air velocities through the mill.

6. Mill air inleakage (on suction mills).

12.11 Locating and Eliminating Sources of Condenser Air In-leakage

A common problem at many plants is high condenser air-inleakage. This can be detrimental not only to heat rate, but also to the water chemistry and therefore the reliability of the unit. (High dissolved oxygen in the condensate/feedwater is a major contributor to boiler tube leaks. If hydrazine is fed in the unit to remove the oxygen, large amount of ammonia can be formed, attacking the condenser tubes.)

The thermal performance can be adversely affected due to "air blanketing" the condenser tubes, effectively reducing the heat transfer surface area. If the air in the condenser covers some of the tubes, the steam cannot get to the tube surface to be condensed. In extreme cases, this can be seen on some units by a large difference in outlet water temperature at different positions in the outlet water pipe. On other units, thermocouples have been positioned in the ends of certain tubes (especially in and around the air removal zone), and the lead wires run out of the waterbox. If the water leaving any particular tube is close to the inlet water temperature, it is not picking up any heat. At higher loads this would be due to air blanketing. (At low loads, not all the heat transfer area is required, therefore some tubes may have a low temperature rise.)

A second effect on the thermal performance can result if there is insufficient air removal capacity (air ejectors or vacuum pumps), the condenser pressure will rise until the capacity of the removal equipment is equal to the inleakage. Vacuum pumps and steam jet air ejectors have a head versus volumetric flow curve like any other pump. The "head" is the difference between the condenser pressure and the ambient pressure. As the condenser pressure drops (condenser vacuum rises) the head rises, and the capacity of the vacuum pump/SJAE decreases. In Figure 12.1, if the amount of air ingress is Q_1, and the vacuum is at V_1, the pump(s) are capable of handling that flow at that vacuum. If the air inleakage increases to Q_2, the pump(s) are still capable of removing that quantity at the vacuum V_1. However, if the quantity of air inleakage increases to Q_3, the pumps cannot remove that quantity at the original head (vacuum) and the vacuum will decrease to V_3, at which point the pumps can handle the additional flow.

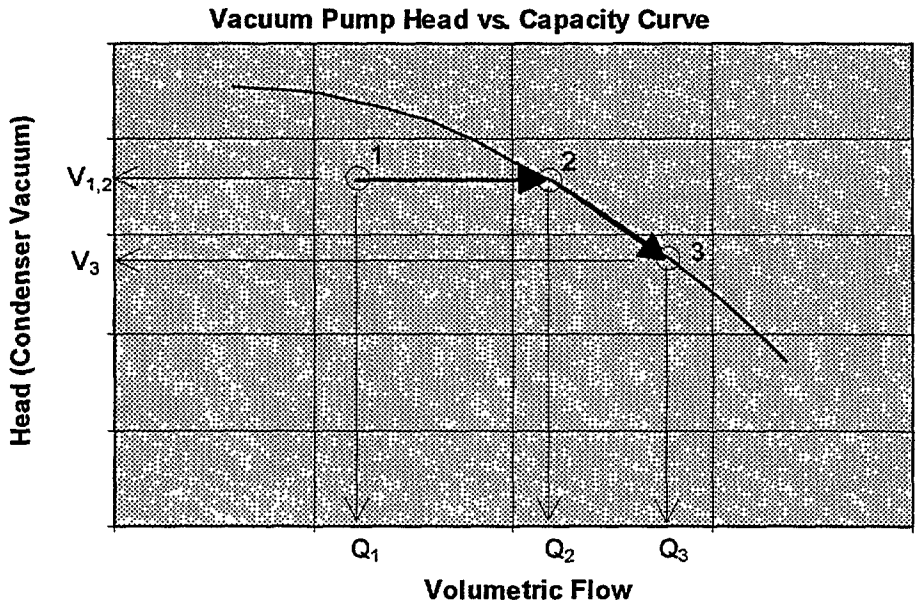

Figure 12.1 Affect of Excessive Condenser Air Inleakage on Vacuum

Another point to remember when looking at the capacity of the vacuum equipment is the effect of steam entrained with the air. If the steam/air mixture being drawn off the air removal equipment is not subcooled (the usual rule of thumb is it should be subcooled at least 4-5 °C), then the removal equipment will be handling mostly steam.

If air inleakage is a problem, locating the source(s) can be difficult. If the unit is off-line, most condensers can be hydrostatically tested by filling the steam side with water. Then the unit is walked down looking for locations where water is leaking out.

If the unit is on-line, a tracer gas is used. A sensor that is capable of detecting small quantities of the tracer gas is placed at the exhaust of the air removal equipment (a small vacuum pump, cooler and desiccant is used to remove and dry a sample of the exhaust gas before it is passed over the sensor). Then small amounts of the tracer gas are sprayed at potential leaks (LP turbine shaft seals, LPT horizontal joint, turbine to condenser joint, hotwell sight glasses, condensate extraction pump seals, etc.). If the sensor detects the gas, there is a leak at that point. If no leak is detected, the person spraying the helium moves on to the next potential leak location. The severity of the leak can be inferred from the time lag between the spray and the detection, the amount of gas detected, and the time required for the tracer gas to clear the system. For tracer gas, first Freon was used, until it was learned that it damaged the earth's Ozone layer. Next helium was used, and now both helium and sulfur hexafluoride (SF_6) are used.

Each plant should have this equipment available so that whenever the condenser air inleakage increases, a search for the source can be initiated. Often the leaks are in valve packings that can be repaired by simply tightening the packing. Some leaks can be temporarily repaired with duct tape or a sealant, until the unit comes off line, when permanent repairs can be made. Other leaks

must be well marked and documented so they can be relocated when the unit is down, and then repaired.

To help assist the search, a set of Flow or P&ID drawings should be marked up showing the potential locations where condenser air in-leakage could occur. It is preferable to mark the potential locations as one of two types. Some locations are always under vacuum when the unit is above its minimum load. Other locations are only under vacuum below some set load. If the air inleakage is only high at low loads, then the locations that are always under vacuum do not need to be checked. By marking the drawing in this manner, the time required to check the unit can be decreased.

Another helpful way to assist the team searching for leaks, is to make a notebook, divided by floors and further divided into areas, and to take pictures of the potential in-leakage locations.

12.12 Complete Mill Performance Program

Section 12.6 Balancing Primary Air Flow to Each Burner and Section 12.10 Improved Coal Fineness, are two key steps to improving the performance of a coal fired boiler. However, these steps are only part of a complete mill performance program.

Section 12.6 discussed one method to calibrate the primary air flow measurements of each mill, by summing the air flows through each burner pipe (which were measured to determine if the pipe to pipe flows were balanced), and comparing the total to the station flow indication. Another method to calibrate the station flow reading, is to probe the duct, near the station airfoil or pitot, with an "L" or "S" type pitot. The test plane should be in a relatively long straight duct, and new number of test points should approach the number that would be required by ASME PTC 19.10 Figure 3 (based on the number of equivalent diameters up and down stream to the nearest disturbance). One advantage to this method is that less measurements are required than are required by traversing, through two ports 90° apart in each coal pipe leaving the pulverizer. Another advantage of this method is that it measures the same air flow that the station instrument sees, where the flow in the coal pipe also includes tempering air, includes infiltration (for a suction mill) and excludes losses (for a pressurized mill). Maintaining an accurate indication of primary air flow is critical for good combustion and efficiency, therefore these calibrations should be a routine activity.

Sub-sections 6 and 10 also discussed clean air tests to balance the amount of air flow through each coal pipe. Normally, if the air flows are balanced when there is no coal flow, and the mill's fineness is good, the "dirty" air flow will also be balanced. Dirty air flow is the flow rate of the air, when it is also carrying pulverized coal. This can be measured with a special type of pitot tube that is designed to measure the air flow with out plugging quickly.

After running a dirty air test, the average velocity in each coal pipe can be measured, and an iso-kinetic coal sample collected from the pipe, and the flow rate of the coal can be determined (see Section 4.7.8). With this procedure, the air flow, coal flow, and air to fuel ratio can all be

determined. Just as maintaining an accurate indication of primary air flow is critical for good combustion and efficiency, so is maintaining proper air and equal distribution of air and coal in each pipe. Therefore these tests should also be a routine activity.

APPENDIX A ABBREVIATIONS

1.	Acc	Accountable
2.	Act	Actual
3.	Act-Des	Actual versus Design
4.	Act-Exp	Actual versus Expected
5.	AH	Air Heater
6.	AHAOT	Air Heater Air Outlet Temperature (°C)
7.	AHE	Air Heater Efficiency (%)
8.	AHGIT	Air Heater Gas Inlet Temperature (°C)
9.	AHL	Air Heater Leakage (%)
10.	AHO	Air Heater Outlet
11.	AHOO2	Flue gas O_2 at the Air Heater Outlet, dry basis (If the O_2 is measured on a wet basis, it can be corrected by O_2 dry = O_2 wet / (1-Moisture Fraction), where the moisture fraction for coal fired units is usually between 0.07 and 0.09)
12.	AirT	Air Temperature Entering the Boiler (If Air Preheating Coils are used, it is the temperature leaving the coils, otherwise the ambient temperature is used) (°C)
13.	AmbT	Ambient Air Temperature (°C)
14.	APH	Air Preheater
15.	ASME	American Society of Mechanical Engineers
16.	ASTM	American Society for Testing and Materials
17.	AuxDuty	Heat Rejected in Auxiliary Condensers (kcal/h)
18.	BDP	Boiler Differential Pressure (kg/cm^2)
19.	BE	Boiler Efficiency (%)
20.	BFBP	Boiler Feedwater Booster Pump
21.	BFP	Boiler Feedwater Pump
22.	BFPT	Boiler Feedwater Pump Turbine
23.	BLComb	Combustible in Ash Loss, percentage point change in Boiler Efficiency, due to unburned carbon in the ash (% points)
24.	BLDG	Dry Gas Loss, percentage point change in Boiler Efficiency, due to quantity of and energy in, the flue gas (% pts.)
25.	BLH2	Hydrogen Loss, percentage point change in Boiler Efficiency, due to formation of water from hydrogen in the fuel (% points)
26.	BLMA	Moisture in Air Loss, percentage point change in Boiler Efficiency, due to heating the moisture in the air (% points)
27.	BLMC	Moisture in Coal Loss, percentage point change in Boiler Efficiency, due to heating the moisture in the fuel (% points)
28.	BLMisc	Miscellaneous Boiler Losses, such as radiation loss, sensible heat in dust, etc. (%)

29.	BLPyr	Pulverizer Pyrite Loss, percentage point change in Boiler Efficiency, due to quantity of and energy in, the rejected pyrites (% points)
30.	BO	Boiler Outlet, the exit plane between the economizer outlet and the air heater inlet.
31.	BOO2	Flue gas oxygen (O_2) at boiler outlet (%), dry basis (If the O_2 is, measured on a wet basis, it can be corrected by O_2 dry = O_2 wet / (1-Moisture Fraction), where the moisture fraction for coal fired units is usually between 0.07 and 0.09)
32.	BO2CO2	At the boiler Outlet, the sum of the O_2 and CO_2 concentrations in the Flue Gas. (%)
33.	Cb	Carbon Burned (kilograms per kilogram of as-fired fuel)
34.	CBT	Computer Based Training
35.	CCW	Condenser Circulating Water
36.	CCWF	Condenser Circulating Water Flow Rate (kg/h)
37.	CCWP	Condenser Circulating Water Pump(s)
38.	CCWTI	Condenser Circulating Water Inlet Temperature (°C)
39.	CCWTO	Condenser Circulating Water Outlet Temperature (°C)
40.	CCWTR	Condenser Circulating Water Temperature Rise (°C)
41.	CenPEEP	Centre for Power Efficiency and Environmental Protection
42.	CEP	Condensate Extraction Pump
43.	CF	Condenser Cleanliness Factor (%)
44.	CP	Constant Pressure
45.	CombBA	Combustibles in Bottom Ash (%)
46.	CombFA	Combustibles in Fly Ash (%)
47.	CombTA	Combustibles in Ash, usually weighted averaged from CombBA & CombFA (%)
48.	CorAmb	Corrected for Ambient Conditions (Boiler Inlet Air and CCW Inlet Water Temperatures)
49.	CorAmbT	Corrected for Ambient Air Temperature
50.	CorDCP	Corrected to Design Condenser Pressure
51.	CorECP	Corrected to Expected Condenser Pressure
52.	CorRAT	Corrected for Reference Inlet Air Temperature
53.	CorRCP	Corrected to Reference Condenser Pressure
54.	CorSF	Corrected for Startup Fuel
55.	CorTNL	Corrected for Leakage (or corrected to no leakage)
56.	CP	Condenser Pressure (mm Hg abs.)
57.	Cp	Heat Capacity (kcal/kg)
58.	CSArea	Surface Area of Condenser (m^2)
59.	CT	Current Transformer
60.	DA	Deaerator
61.	DAS	Data Acquisition System
62.	DCA	Drain Cooler Approach (°C)

63.	Des	Design
64.	DNB	Departure from Nucleate Boiling
65.	DOE	(United States) Department of Energy
66.	Duty	Rate of Heat Transferred to the Condenser Circulating Water (kcal/h)
67.	Econ	Economizer
68.	EGT	Air Heater Measured Exit Gas Temperature (°C)
69.	ELEP	Expansion Line End Point
70.	EPA	(United States) Environmental Protection Agency
71.	EST	Exhaust Steam Temperature in the Condenser (°C)
72.	Exp	Expected
73.	Exp+BOO2	All parameters at expected values, except boiler outlet O_2 which is at its actual value.
74.	Exp+CombTA	All parameters at expected values, except combustibles in ash which is at its actual value.
75.	Exp-Des	Expected versus Design
76.	Exp+EGT	All parameters at expected values, except exit gas temperature which is at its actual value.
77.	Exp+FuelM	All parameters at expected values, except moisture in fuel which is at its actual value.
78.	Exp-Ref	Expected versus Reference
79.	Exp+UltH	All parameters at expected values, except hydrogen in fuel which is at its actual value.
80.	Exp+Wre	All parameters at expected values, except pulverizer rejects which is at its actual value.
81.	FC	Fuel Cost (Rs/kcal)
82.	FDF	Forced Draft Fan
83.	FDFT	Forced Draft Fan Turbine
84.	FFWT	Final Feedwater Temperature (°C)
85.	FG	Flue Gas
86.	FGH	Enthalpy of the moisture in the Flue Gas (kJ/kg)
87.	FP	Full Pressure
88.	FS	Total Quantity of Fuel Supplied to Unit (kg)
89.	FuelA	Ash, As-Fired, from a Proximate or Ultimate Analysis (fraction)
90.	FuelFC	Fixed Carbon, As-Fired, from a Proximate Analysis (fraction)
91.	FuelTM	Total Moisture, As-Fired, from Proximate or Ultimate Analysis (fraction)
92.	FuelVM	Volatile Mater, As-Fired, from a Proximate Analysis (fraction)
93.		
94.	FVP	Full Variable Pressure
95.	FW	Feedwater
96.	FWF	Feedwater Flow (to the economizer) (kg/h)
97.	FWH	Feedwater Heater

98.	GG	Gross Generation (MWh)
99.	GLoad	Average Gross Load for the period (MW)
100.	GLoss	Generator Losses, both electrical and mechanical (kW)
101.	GRF	Gas Recirculation Fan
102.	GTCHR	Gross Turbine Cycle Heat Rate (kcal/kWh)
103.	GUHR	Gross Unit Heat Rate (kcal/kWh)
104.	h	Specific Enthalpy, the amount of energy in a substance per unit mass (kJ/kg or kcal/kg)
105.	Hdp	Heating value of Total Dry Refuse (kcal/kg)
106.	HEI	Heat Exchange Institute
107.	HEIHTC	HEI's Heat Transfer Coefficient for a clean tube (kcal/h/m^2/C)
108.	HEITCF	HEI's Heat Transfer Coefficient Correction Factor for Inlet Water Temperature
109.	HHV	Higher Heating Value of the Fuel (kcal/kg)
110.	HICold	Heat supplied before synchronization during a cold startup (kcal)
111.	HIHot	Heat supplied before synchronization during a hot startup (kcal)
112.	HOU	Hours of Use (h)
113.	HP	High Pressure
114.	HPT	High Pressure Turbine
115.	HPTEff	High Pressure Turbine Efficiency (%)
116.	Hr	Heating Value of Pulverizer Rejects (kcal/kg)
117.	HRAP	Heat Rate Action Plan
118.	HRCF	Heat Rate Correction Factor, the % change in heat rate (%)
119.	HRCFR	The ratio of two heat rate multipliers, i.e. (1+HRCFx/100)/(1+HRCFy/100)
120.	HRD	Heat Rate Deviation (kcal/kWh)
121.	HRHIV	Hot Reheat Steam Temperature at inlet to Intercept Valve (IP turbine) (°C)
122.	HRIP	Heat Rate Improvement Program
123.	HTC	Heat Transfer Coefficient (kcal/h/m^2/C)
124.	HTran	Heat Transmittance, the Rate of Heat Transfer per unit area (kcal/h/m^2)
125.	HtrBP	High Pressure Feedwater Heater(s) Bypassed
126.	HVP	Hybrid Variable Pressure
127.	HVS	High Volume Sampler
128.	HVT	High Velocity Thermocouple (Probe)
129.	ID	Inside Diameter
130.	IDF	Induced Draft Fan
131.	IK	Short Retractable (Wall) Sootblower
132.	IPT	Intermediate Pressure Turbine
133.	IPTEff	Intermediate Pressure Turbine Efficiency (%)
134.	IR	Long Retractable (Convection Pass) Sootblower

135.	ISA	Instrument Society of America
136.	ISO	International Organization for Standardization
137.	IV	Intercept Valve
138.	kcal	Kilocalorie
139.	kJ	KiloJoule
140.	kVA	KiloVolt-Ampere
141.	kW	Kilowatt
142.	kWh	Kilowatt-hour
143.	LMTD	Logarithmic Mean Temperature Difference (°C)
144.	LPT	Low Pressure Turbine(s)
145.	LTSH	Low Temperature Superheater (also known as primary superheater (PSH)
146.	MCR	Maximum Continuous Rating
147.	MDBFP	Motor Driven Boiler Feedwater Pump
148.	Mg	Kilograms of Moisture in the Flue Gas per Kilogram of As-fired Fuel
149.	MSF	Main Steam Flow to the HP turbine (kg/h)
150.	MU	Makeup Water (% of feedwater flow)
151.	MVA	MegaVolt-Ampere
152.	MW	MegaWatt
153.	MWH	MegaWatt-hour
154.	NLoad	Net Unit Load (MW)
155.	NoCold	Number of Cold Startups during the Period
156.	NoHot	Number of Hot Startups during the Period
157.	NTPC	National Thermal Power Corporation
158.	NUHR	Net Unit Heat Rate (kcal/kWh)
159.	O&M	Operations and Maintenance (i.e., O&M costs)
160.	OD	Outside Diameter
161.	OEM	Original Equipment Manufacturer
162.	P & ID	Piping and Instrumentation Diagrams
163.	PAF	Primary Air Fan
164.	PG	Performance Guarantee
165.	PH	Period Hours, the number of hours (or fractional hours) in the time period being studied (h)
166.	Pmg	Partial Pressure of the Moisture in the Flue Gas
167.	PSH	Primary Superheater (also known as low temperature superheater, LTSH)
168.	PSSI	Prorated Station Service when the unit is off-line (MW)
169.	PSSO	Prorated Station Service when the unit is on-line (MW)

170.	PT	Potential Transformer
171.	PTC	(ASME's) Performance Test Code(s)
172.	PumpQ	Heat Added to the Turbine Cycle by the BFPs (kcal/h)
173.	Ref	Reference
174.	Ref+BOO2	All parameters at reference values, except boiler outlet O_2 which is at its expected value.
175.	Ref+CombTA	All parameters at reference values, except combustibles in ash which is at its expected value.
176.	Ref-Des	Reference versus Design
177.	Ref+EGT	All parameters at reference values, except exit gas temperature which is at its expected value.
178.	Ref+FuelTM	All parameters at reference values, except moisture in fuel which is at its expected value.
179.	Ref+UltH	All parameters at reference values, except hydrogen in fuel which is at its expected value.
180.	Ref+Wre	All parameters at reference values, except pulverizer rejects which is at its expected value.
181.	RH	Reheat or Reheater
182.	RHA	Reheat Attemperation (% of feedwater flow)
183.	Rs	Rupees
184.	RTD	Resistance Temperature Detector
185.	S1	Season 1 (i.e., Winter)
186.	S2	Season 2 (i.e., Monsoon)
187.	S3	Season 3 (i.e., Summer)
188.	Sat	Saturation condition where both liquid and vapor can exist (as opposed to the compressed liquid state or the superheated vapor state)
189.	SCAPH	Steam Coil Air Preheater
190.	SCAPHF	Steam Flow Supplied to Air Preheating Coils (kg/h)
191.	scf	Standard Cubic Feet
192.	SEB	State Electricity Board
193.	SH	Superheat or Superheater
194.	SHA	Superheat Attemperation (% of feedwater flow)
195.	SJAE	Steam Jet Air Ejector(s)
196.	sm^3	Standard Cubic Meter (25 °C temperature and 760 mmHg pressure)
197.	SPC	Statistical Process Control
198.	SPE	Solid Particle Erosion
199.	SQRT	Square Root
200.	SSH	Secondary Superheater, also known as finishing superheater
201.	SSI	Station Service - Idle (MW)
202.	SSO	Station Service - Operating (MW)

203.	t/h	tons (metric) per hour
204.	TAAHO	Total Air at the Air Heater Outlet (%)
205.	TABO	Total Air at the Boiler Outlet (%)
206.	TDBFP	Turbine Driven Boiler Feedwater Pump
207.	ThAir	The kilograms of air theoretically required for complete combustion of one kilogram of fuel.
208.	Tot	Total
209.	TP	Throttle (or Initial Steam) Pressure (kg/cm^2)
210.	TT	Throttle (or Initial Steam) Temperature (°C)
211.	TTD	Terminal Temperature Difference (°C)
212.	TubeBWG	Condenser Tube Wall Thickness (BWG)
213.	TubeID	Internal Diameter of Condenser Tubes (mm)
214.	TubeMat	Condenser Tube Material
215.	TubeMGC	HEI's Condenser Tube Material & Gauge Correction Factor (Table 3)
216.	TubeNo	Number of Condenser Tubes
217.	TubeOD	External Diameter of Condenser Tubes (mm)
218.	TubeU	HEI's Uncorrected Heat Transfer Coefficient for condenser tubes
219.	TubeWV	Velocity of Water Flowing through the Condenser Tubes (m/s)
220.	TVA	(United States) Tennessee Valley Authority
221.	UAP	Unit Auxiliary Power (on-line) (MW)
222.	UEEP	Used Line End Point
223.	UltC	Carbon Content of Fuel, from Ultimate Analysis (fraction)
224.	UltH	Hydrogen Content of Fuel, from Ultimate Analysis (fraction)
225.	UltN	Nitrogen Content of Fuel, from Ultimate Analysis (fraction)
226.	UltO	Oxygen Content of Fuel, from Ultimate Analysis (fraction)
227.	UltS	Sulfur Content of Fuel, from Ultimate Analysis (fraction)
228.	UnAcc	Unaccountable
229.	USAID	United States Agency for International Development
230.	VWO	Valves Wide Open (all turbine control valves are in the full open position)
231.	Wa	Kilograms of Total Dry Air Supplied per Kilogram of As-fired Fuel
232.	Wdp	Kilograms of Total Dry Refuse per Kilogram of As-fired Fuel
233.	Wg	Kilograms of Dry Gas Produced per Kilogram of As-fired Fuel
234.	Wma	Moisture in Inlet air to the Boiler (kg/kg dry air)
235.	Wre	Pulverizer Reject Rate (kilograms of rejects/kilogram of as-fired fuel)
236.	WW	Waterwall(s)
237.	12MTD	12 Month to Date

For Example:

$CCWF_{Exp}$ is the expected (Exp) condenser circulating water flow rate (CCWF)

$HRCF_{CP,\ Act\text{-}Exp}$ is the heat rate correction factor (HRCF) for condenser pressure (CP) due to the difference between the actual and expected pressures (Act-Exp)

$GUHR_{Act,\ CorSF,\ CorECP}$ is the actual (Act) gross unit heat rate (GUHR), corrected for startup fuel usage (CorSF) and also corrected for expected condenser pressure (CorECP)

APPENDIX B DEFINITIONS

Actual	The measured (or calculated) value for a parameter (such as throttle temperature) during the period in question. (See also "Expected" and "Reference")
Air Heater	Equipment used to heat combustion air by transferring heat from the flue gas. Most air heaters are regenerative or tubular.
Air Heater Efficiency	The ratio of the drop in flue gas temperature to the temperature head across the air heater. The temperature drop of the flue gas (flue gas entering AH temperature - flue gas leaving AH, no leakage, temperature) divided by the temperature difference between the flue gas temperature entering the AH and the air temperature entering the AH.
Air Heater Leakage	Quantity of wet air leaking from the air side of an air heater divided by the quantity of wet air entering the air heater, multiplied by 100.
Air Preheater	Equipment used to preheat combustion air before it reaches the air heater. Most air heaters use steam coils, using steam from the turbine cycle.
Boiler Differential Pressure	The pressure drop on the water/steam side of the boiler. For once through units, it is usually measured between economizer inlet and primary superheater outlet. For forced circulation, drum units it is measured across the circulating water pumps.
Boiler Outlet	The exit plane of the boiler, or the entrance plane of the air heater.
Bottom Ash	The portion of ash in the fuel that drops out the bottom of the furnace. Normally, it also includes some combustible material. See Fly Ash.
Cold Reheat	Steam at the inlet to the boiler's reheat section, measured just before the reheat inlet header. (The temperature and pressure of this steam is usually slightly lower than the "HP Exhaust" steam).

Condenser Duty	The amount of heat transferred to the condenser circulating water, mostly from condensing the low pressure turbine exhaust steam, but also from miscellaneous drains.
Condenser Pressure	The static pressure of the LP turbine exhaust steam, measured just above the tube bundle. In order to measure the static pressure in the presence of the high velocity steam, ASME basket tips should be used.
Dual Pressure	A mode of variable pressure in which the water-cooled circuits are maintained at the design pressure, and throttling division valves are used to reduce the pressure (and to control unit load with the turbine valves wide open) between the primary and secondary superheater.
Exit Gas Temperature (EGT)	Temperature of the flue gas leaving the air heater. If there is no qualifier, it is assumed to be the actual or measured temperature.
EGT-No Leakage	The flue gas temperature at the AH exit, that would have been measured if there had been no leakage of incoming air to the exiting flue gas via the AH seals. The leakage is normally assumed to have occurred at the cold end of the AH.
EGT-No Leakage, Corr. RAT	The flue gas temperature at the AH exit, that would have been measured if there had been no leakage, and corrected for the difference between the actual and reference inlet air temperature.
Expected	The value for a parameter (such as throttle temperature) that should be obtainable during the period in question, with the present plant configuration. It may not be achievable due to equipment degradation but, with maintenance (including identical replacement of equipment), it could be obtained. (See also "Actual" and "Reference")
Final Feedwater	Feedwater entering the boiler (if the boiler includes an economizer, entering the economizer).
Fly Ash	The portion of ash in the fuel that is carried by the flue gas through the convection pass. Some of it drops out into hoppers at the economizer outlet, the APH outlet, and the remainder is collected in the precipitator or baghouse.

	Normally, it also includes some combustible material. See Bottom Ash.
Full Pressure Operation	Also known as Constant Pressure or (CP) operation, the unit operating mode in which the load of the unit is controlled by opening and closing the turbine control valves, leaving the steam pressure ahead of the stop valves at the design value.
Full Variable Pressure	An operating mode in which the unit load is controlled by varying the steam pressure at ahead of the stop valves while leaving the turbine control valves wide open. (FVP)
Furnace Exit	The exit plane of the furnace is defined differently by different manufacturers. To some it is the horizontal plane at the nose arch or goose neck. Other manufacturers define it as the vertical plane (or horizontal plane for tower boilers) in front of the first tube bundle whose tube spacing is 18 inches (457 mm) or less.
Generation	The electrical energy (or work) produced by a generating unit, usually measured in kWh or MWh. (See also "Load".)
Gross Generation	The electrical energy produced by all generators on a unit.
Net Generation	The energy that a unit is able to supply to the distribution system. It is the gross generation minus the unit's auxiliary energy use, including (i.e., BFP, FDF, etc.) and station service energy (coal handling, service water, etc.). At plants with multiple units, the station service energy must be prorated between the units.
Heat Rate	The reciprocal of the efficiency. It is usually expressed as the heat input divided by the electrical power output, with units of kcal/kWh, or kJ/kWh or Btu/kWh.
Actual Heat Rate	The true heat rate of a unit, usually calculated from the amount of fuel burned during the period, the higher heating value of the fuel, and the generation for the period. The integrated readings of fuel burned and generation are (i.e. daily, weekly, or monthly) used to calculate the actual heat rate. Actual heat rate includes fuel burned during startups, and the negative net generation during off-line periods.

Expected Heat Rate	The heat rate the unit is capable of obtaining based on the present design configuration (but the unit may not be capable of this heat rate with the present condition of that equipment). Calculated by adjusting the effects of permanent modifications to the reference heat rate. (Revised whenever permanent modifications are made to the unit.) Expected heat rate is almost always a net unit heat rate.
Gross Turbine Cycle	The heat rate calculated using the heat input to the turbine cycle and the "Gross" electrical generation.
Gross Unit	The heat rate calculated using the heat input to the boiler and the "Gross" electrical generation.
Net Unit	The heat rate calculated using the heat input to the boiler and the "Net" electrical generation.
Reference	The heat rate the unit is capable of obtaining based on the initial design configuration. Generally derived from the initial turbogenerator and boiler performance guarantee or acceptance test results.
Heat Rate Parameter	A measured or calculated (from measurements) item such as throttle temperature, condenser pressure, or HP turbine efficiency, for which a heat rate deviation can be calculated relatively easily, from its deviation from some design/reference/expected value.
HP Exhaust	Steam at the exit of the high pressure turbine section. (The temperature and pressure of this steam is usually slightly higher than the "Cold Reheat" steam).
Hot Reheat	Steam at the exit of the reheater, measured at the boiler. (The temperature and pressure of this steam is usually slightly higher than the "Intercept Valve" steam).
HRH at Intercept Valve	Steam at the entrance to the intercept valves before the intermediate pressure turbine. (The temperature and pressure of this steam is usually slightly lower than the "Hot Reheat" steam).
Hybrid Variable Pressure	An operating mode of the unit. For nozzle governed (sequential arc) turbines, the load is reduced in the normal manner by closing one or more turbine control valves. Then to further reduce load, the steam pressure ahead of the stop valves is reduced. For throttle governed (full arc) turbines,

	the control valves are fixed at some point, such as 85% open, and the unit load is controlled over the entire range by the pressure ahead of the stop valves. (HVP)
Incremental Cost	The additional cost required per hour for the next kW (Rs/kWh) at a given net output.
Incremental Heat Rate	The additional energy required per hour for the next kW at a given net output(kJ/kWh or kcal/kWh).
Input versus Output	The total required hourly energy input (kJ/h or kcal/h) for a given net output (kW).
Load	The instantaneous electrical power produced, usually measured in kW or MW. (See also "Generation".)
Gross Load	The instantaneous electrical power produced by all generators on a unit.
Net Load	The instantaneous electrical power that a unit is able to supply to the distribution system. It is the gross generation minus the unit's auxiliary energy use, including (i.e. BFP, FDF, etc.) and station service energy (coal handling, service water, etc.). At plants with multiple units, the station service energy must be prorated between the units.
Makeup	The high quality water that must be supplied to a unit to "makeup" for losses from the cycle, such as steam sootblowers, boiler blowdown, vents, leaks, etc. This does not include water supplied to the unit for hydrostatic tests, chemical cleanings, etc.
Primary Process Indicator	Operating data, such as pressure drop across the condenser waterbox or mill outlet temperature. These are the basic measurements taken on the plant. These are distinguished from "heat rate parameters" in that while they can affect the thermal performance of the unit, their effect (on heat rate) is difficult to quantify, therefore it is not routinely determined.
Pyrites	Material that is rejected from the pulverizer. Primarily hard material such as rocks or sand, however some coal is usually "spilled" along with the undesirable material
Reference	The value for a parameter (such as throttle temperature) that was obtainable during the period of time for which the

	"Reference Net Heat Rate" curves were developed. (See also "Actual" and "Reference.")
Station Service Power	The power fed through the station service transformers and used by common equipment such as coal handling, ash handling, etc. For plants with multiple units, it must be prorated between the units and added to the unit auxiliary power, then this total subtracted from the gross power to get the unit net power
Throttle	Steam at the entrance to the main stop valves before the high pressure turbine, also known as "main steam" or "initial steam."
Unit Auxiliary Power	The power fed through the unit auxiliary transformers and used by unit equipment such as BFPs, FDFs, IDFs, mills, etc. It must be added to the prorated station service power, then this total subtracted from the gross power to get the unit net power

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APPENDIX C LIST OF HEAT RATE RELATED ITEMS RECOMMENDED FOR A TECHNICAL LIBRARY

C.1 Test Codes

American Society of Mechanical Engineers Performance Test Codes (PTCs):
ASME - United Engineering Center
345 East 47th Street
New York, NY 10017

a)	PTC 1	General Instructions
b)	PTC 2	Definitions and Values
c)	PTC 4.1	Steam Generating Units
d)	PTC 4.2	Coal Pulverizers
e)	PTC 4.3	Air Heaters
f)	PTC 6	Steam Turbines
g)	PTC 8.2	Centrifugal Pumps
h)	PTC 11	Fans
i)	PTC 12.1	Closed Feedwater Heaters
j)	PTC 12.2	Steam Condensing Apparatus
k)	PTC 12.3	Deaerators
l)	PTC 19.1	Measurement Uncertainty
m)	PTC 19.2	Pressure Measurement
n)	PTC 19.3	Temperature Measurement
o)	PTC 19.6	Electrical Measurement in Power Circuits
p)	PTC 19.10	Flue and Exhaust Gas Analysis
q)	PTC PM-1993	Performance Monitoring Guidelines for Steam Power Plants

American Society for Testing and Materials
1916 Race Street
Philadelphia, PA 19103

ASTM D 197-87	Standard Test Method for Sampling and Fineness Test of Pulverized Coal
ASTM D 409-93a	Standard Test Method for Grindability of Coal by the Hardgrove-Machine Method
ASTM D 2013-86	Standard Method of Preparing Coal Samples for Analysis
ASTM D 2234-98	Standard Practice for Collection of a Gross Sample of Coal
ASTM D 4749-87	Standard Test Method for Performing the Sieve Analysis of Coal and Designating Coal Size
ASTM E 11-95	Standard Specification for Wire Cloth and Sieves for Testing Purposes

ISO 5167-1 Measurement of Fluid Flow by Means of Pressure Differential Devices - Part 1: Orifice Plates, Nozzles and Venturi Tubes inserted in circular cross-section Conduits Running Full.

Cooling Tower Institute CTI Code Tower, Standard Specifications, Acceptance Test Code for Water-Cooling Towers.
3003 Yale Street, Houston TX 77018

"AMCA Fan Application Manual" published by Air Movement and Control Association Inc. 30 West University Drive Arlington Heights, Illinois 60004
Part 1 - Fans and Systems
Part 3 - Field Performance Measurements

C.2 Books

"Combustion" by Combustion Engineering
"Centrifugal Pump Clinic, Second Edition" by Igor J. Karassik, published by Marcel Dekker, Inc.
"Evaluating and Improving Steam Turbine Performance" by K. C. Cotton, published by Cotton Fact Inc., 216 Sugar Hill Road, Rexford, NY 12148
"Fan Engineering" by Robert Jorgensen, published by Buffalo Forge Company
"Flow of Fluids through Valves, Fittings" , Crane Technical Bulletin 410
"Handbook for Electrical Metering", by the Edison Electric Institute
"Power Plant Performance" by A.B. Gill, published by Butterworths
"Power Plant Theory and Design" by Philip J. Potter, published by John Wiley & Sons,
"Pump Handbook" edited by Karassik, Krutzsch, Fraser and Messina, published by McGraw-Hill Book Company
"Standards for Surface Steam Condensers" by the Heat Exchange Institute (HEI) 1300 Sumner Avenue, Cleveland, Ohio 44115-2851
"Standards for Closed Feedwater Heaters" by the Heat Exchanger Institute (HEI)
"Steam/Its Generation and Use" by Babcock and Wilcox
"Steam Turbines and Their Cycles" by Kenneth Salisbury, published by Robert E. Krieger Publishing Co.

C.3 Magazines

"Power"	ph (609) 426-5667	email : udieditor@aol.com web site: www.powermag.com
"Power Engineering"	ph (918) 835-3161	email : pe@pennwell.com
"Electric Light & Power"	ph (918) 831-9456	email : kelly@pennwell.com
"I&CS - Instrumentation & Control Systems"		ph (610) 964-4405

C.4 Company Newsletters

Scientech News -	Monthly Newsletter of Scientech Inc., the company that has the PEPSE and PMAX computer programs web site : www.scientech.com email : snewsletter@scientech.com
Bullet-IN CONCO	Newsletter of CONCO Systems, Inc. A company that supplies equipment for and performs condenser tube cleanings, leak detection, etc. email : 76165.1540@compuserve.com
no name	Newsletter of Power Technologies Inc., Company that provides training, supplies technical expertise in several areas including generation, transmission, distribution, etc. web site : www.pti-us.com email : edpro@pti-us.com
Lehigh Energy Update	Newsletter of the Lehigh University Energy Research Center. The center does research for EPRI and others, primarily in the combustion area web site : www.lehigh.edu/~inenr/inenr.htm
no name	Newsletter of Storm Technologies Inc., A company that supplies HVT probes, Dirty-Air probes, etc. They also run tests themselves. email : dstorm@concentric.net

C.5 EPRI Documents

Electric Power Research Institute:
Electric Power Research Institute Distribution Center
207 Coggins Drive,
P.o. Box 23205
Pleasant Hill, CA 94523

Ph (510)934 - 4212

1.	TR-111047	Sep. 1998	Proceedings: 1998 EPRI Heat-Rate Improvement Conference
2.	TR - 109546	Jul. 1998	Heat Rate Reference Manual
3.	TR-108110	Aug. 1997	Evaluation of Heat Rate Discrepancies from CEMs
4.	TR-106529	May 1996	Proceedings: 1996 EPRI Heat-Rate Improvement Conference

5.	TR-102098	Mar. 1993	Proceedings: 1992 EPRI Heat-Rate Improvement Conference
6.	GS-7329	Jun. 1991	Heat Rate Demonstration Project Salem Harbor Station Unit 4
7.	TR-100901	Jul. 1991	Proceedings: 1991 EPRI Heat-Rate Improvement Conference
8.	GS-7295	Jul. 1991	Demonstration of EPRI Heat-Rate Improvement0 Guidelines
9.	GS-6985	Sep. 1990	Proceedings: 1989 EPRI Heat-Rate Improvement Conference
10.	GS-6635	Jan. 1990	Proceedings: 1988 EPRI Heat-Rate Improvement Conference
11.	GS-6725	Dec. 1989	Fossil Plant Retrofits for Improved Heat Rate
12.	CS-4554	May 1986	Heat Rate Improvement Guidelines for Existing Fossil Plant
13.	TR-106781	Aug. 1996	Proceedings: Condenser Technology Conference
14.	GS-7349	Aug. 1991	Proceedings: Condenser Technology Conference
15.	CS-6014	Sep. 1988	Condenser Leak-Detection Guidelines Using Sulfur Hexafluoride as a Tracer Gas
16.	CS-5235	July. 1987	Recommended Practices for Operating & Maintaining Steam Surface Condensers
17.	TR-108483	July. 1997	Proceedings: Cooling Tower Technology Conference
18.	TR-104867	Feb. 1995	Proceedings: Cooling Tower Technology Conference
19.	GS-6370	1989	Cooling Tower Performance Prediction & Improvement
20.	CS/EL-4415		Power Plant Performance Monitoring & Improvement
	a) Volume 1	Feb. 1986	Boiler Optimization
	b) Volume 2:	Feb. 1986	Incremental Heat Rate Sensitivity Analysis
	c) Volume 3	Feb. 1986	Power Plant Performance Instrumentation Systems
	d) Volume 4:	Dec. 1987	Boiler Optimization
	e) Volume 5	Dec. 1987	Turbine Cycle Performance Analysis
	f) Volume 6:	May 1989	Economic Dispatch Optimization
	g) Volume 7:	Feb. 1990	Operation of Regenerative Air Preheater
	h) Volume 8:	Mar. 1990	Characteristics Needed for Dispatch
21.	GFS-6203	Mar. 1989	Heat Exchanger Tube Coatings & Liners
22.	GS-6393	Apr. 1990	Development & Application of the Coal Quality Impact Model CQIM
23.	GS-6788	Apr. 1990	Steam Turbine Maintenance & Repair Technology
24.	TR-111316	Sep. 1998	Proceedings: Second Annual EPRI Workshop on Power Plant Optimization
25.	TR-111046	Dec. 1998	Achieving NOx Compliance at Least Cost: A Guideline for Selecting the Optimum Combination of NOx Controls for Coal Fired Boilers
26.	TR-102401	June 1993	Guidelines for Chemical Cleaning of Fossil-Fired Steam-Generating Equipment

27.	TR-108292	May 1997	Instrument Calibration & Signal Validation Workshop
28.	TR-102052	Mar. 1993	Tuning Guidelines for Utility Fossil Process Control
29.	GS-6772	Mar. 1990	Variable Pressure Operation: An Assessment
30.	TR-111046	Aug. 1998	Proceedings: 1998 Feedwater Heater Technology Symposium
31.	GS-6935	Aug. 1990	Feedwater Heaters Maintenance & Repair Technology
32.	TR-101692	Dec. 1992	Proceedings: Symposium on Coal Pulverizers
33.	TR-101698	Jan. 1993	O&M Guidelines for Draft Fans

APPENDIX D HEAT RATE PARAMETERS - EFFECTS, CAUSES & CORRECTIONS

1. High Pressure Turbine Efficiency

   A. Effect on Heat Rate (per percentage point):
      1. US Utility Average: 19.8 kJ/kWh or 4.74 kcal/kWh
      2. US Utility Range: 18.9-22.68 kJ/kWh or 4.51-5.42 kcal/kWh
      3. Plant-Specific Value: Industry data can be used as an estimate. More specific value can be obtained from heat balance code.

   B. Possible Causes of Deviation
      1. Erosion of nozzle blocks
      2. Erosion of turbine blades
      3. Deposits on nozzles or blades
      4. Broken turbine blades
      5. N2 packing leak (HP and IP turbine are in the same shell)
      6. Excess gland packing leaks
      7. Strip Seal Leakage
      8. Malfunctioning Control Valve

   C. Possible Corrections
      1. Operator Controllable
         a) None
      2. Maintenance Correctable
         a) Repair or replace nozzle block
         b) Repair or replace turbine blades
         c) Clean turbine blades
         d) Replace gland packing
         e) Replace turbine seal strips

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2. Intermediate Pressure Turbine Efficiency

A. Effect on Heat Rate (per percentage point):
   1. US Utility Average: 15.3 kJ/kWh or 3.66 kcal/kWh
   2. US Utility Range: 11.6-20.2 kJ/kWh or 2.77-4.82 kcal/kWh
   3. Plant-Specific Value: Industry data can be used as an estimate. More unit specific value can be obtained from heat balance code analysis.

B. Possible Causes of Deviation
   1. Erosion turbine blades
   2. Deposits on turbine blades
   3. Reheater bypass valve leakage
   4. Excess Gland Seal leakage
   5. Strip seal leaks

C. Possible Corrections
   1. Operator Controllable
      a) None
   2. Maintenance Correctable
      a) Repair or replace turbine blades
      b) Repair leaking reheater bypass valve
      c) Repair strip seal
      d) Repair gland seals

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3. Main Steam (Throttle) Pressure

A. Effect on Heat Rate (per bar)
   1. US Utility Average: 0.37 kJ/kWh or 0.088 kcal/kWh
   2. US Utility Range: 0.03-0.69 kJ/kWh or 0.008-0.16 kcal/kWh
   3. Plant-Specific Value: Effects on heat rate of deviation in throttle pressure is normally supplied with the unit thermal kit or using the method of Chapter x.

B. Possible Causes of Deviation
   a) Feedwater flow too low (once-through units)
   b) Firing rate inadequate

C. Possible Corrections
   1. Operator Controllable
      a) Increase feedwater flow
      b) Increase firing rate

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4. Main Steam (Throttle) Temperature

A. Effects on Heat Rate (per °C)
   1. US Utility Average: 2.7 kJ/kWh or 0.64 kcal/kWh
   2. US Utility Range: 1.3-3.2 kJ/kWh or 0.32-0.77 kcal/kWh
   3. Plant-Specific Value: Effect on heat rate of deviation in main steam temperature is normally provided with the unit thermal kit or using the method of Chapter x.

B. Possible Causes of Deviation
   1. Superheater spray control problems
   2. Superheater spray valve leakage
   3. Fouling of the superheater (low temperature)
   4. Fouling of the boiler waterwall (high temperature)
   5. High excess air
   6. Burner tilts mispositioned
   7. Gas tempering flow inadequate
   8. Bypass dampers mispositioned
   9. Temperature control setting calibration drift
   10. Superheater tube leaks
   11. Incorrect amount of superheater heat transfer surface.

C. Possible Corrections
   1 Operator Controllable:
      a) Blow soot
      b) Adjust burner tilts
      c) Adjust bypass damper settings
      d) Control excess air
      e) Manually control superheater spray flow
   2. Maintenance Correctable
      a) Calibrate temperature control setpoint
      b) Repair superheater spray control valve
      c) Clean boiler waterwalls
      d) Clean superheater platens
      e) Repair superheater tube leaks
      f) Add or remove superheater heat transfer surface

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5. Reheat Temperature

A. Effects on Heat Rate (per °C):
   1. US Utility Average: 2.5 kJ/kWh or 0.59 kcal/kWh
   2. US Utility Range: 1.7-3.6 kJ/kWh or 0.41-0.86 kcal/kWh
   3. 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 Chapter x.

B. Possible Causes of Deviation
   1. Reheat attemperation control problems
   2. Reheat attemperation control valve leakage
   3. Fouling of the reheater (low temperature)
   4. Fouling of the boiler waterwall (high temperature)
   5. Fouling of the superheater
   6. High excess air
   7. Burner tilts mispositioned
   8. Gas tempering flow inadequate
   9. Bypass dampers mispositioned
   10. Reheater tube leaks
   11. Incorrect amount of reheater heat transfer surface

C. Possible Corrections
   1. Operator Controllable
      a) Blow soot
      b) Adjust burner tilts
      c) Adjust bypass damper settings
      d) Adjust attemperating air flow damper
      e) Control excess air
      f) Manually control reheat spray flow
   2. Maintenance Correctable
      a) Repair superheater spray control valve
      b) Clean boiler waterwalls
      c) Clean superheater platens
      d) Clean reheater platens
      e) Repair reheater tube leaks
      f) Add or remove reheater heat transfer surface

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6. Superheater Attemperation

A. Effect on Heat Rate (for 10,000 kg/h flow rate):
   1. US Utility Average: 1.18 kJ/kWh or 0.28 kcal/kWh
   2. US Utility Range: 0.62-1.48 kJ/kWh or 0.15-0.35 kcal/kWh
   3. Plant-Specific Value: Industry data can be used as an estimate. More specific value can be obtained from turbogenerator vendor's thermal kit or heat balance code.

B. Possible Causes of Deviation
   1. Improperly adjusted control setpoint
   2. Leaking spray control valve
   3. Broken spray nozzle
   4. Fouling of boiler waterwalls
   5. High levels of excess air
   6. Improperly set gas attemperation
   7. Improperly set gas bypass dampers

C. Possible Corrections
   1. Operator Controllable
      a) Blow waterwall soot
      b) Reduce excess air to proper levels
      c) Adjust gas attemperation
      d) Adjust gas bypass dampers
   2. Maintenance Correctable
      a) Repair spray valves
      b) Calibrate temperature controls
      c) Replace spray nozzle

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7. Reheat Attemperation

A. Effect on Heat Rate (for 10,000 kg/h flow rate):
   1. US Utility Average: 10.3 kJ/kWh or 2.46 kcal/kWh
   2. US Utility Range: 4.8-17.5 kJ/kWh or 1.1-4.19 kcal/kWh
   3. Plant-Specific Value: Industry data can be used as an estimate. More specific value can be obtained from turbogenerator vendor's thermal kit or heat balance code.

B. Possible Causes of Deviation
   1. Fouled waterwalls
   2. High levels of excess air
   3. Fouled superheater sections
   4. Improperly set gas bypass dampers
   5. Improperly adjusted temperature setpoint
   6. Leaking spray control valve
   7. Broken spray nozzle

C. Possible Corrections
   1. Operator Controllable
      a) Adjust gas bypass dampers
      b) Adjust excess air to proper levels
      c) Soot blow waterwalls
      d) Soot blow superheater sections
   2. Maintenance Correctable
      a) Repair spray control valve
      b) Replace spray nozzle
      c) Calibrate temperature control setpoint

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8. Excess Air (O_2)

A. Effect on Heat Rate (per percent O_2)
   1. US Utility Average: 31.0 kJ/kWh or 7.41 kcal/kWh
   2. US Utility Range: 19.0-38.0 kJ/kWh or 4.5-9.1 kcal/kWh
   3. Plant-Specific Value: Can be estimated from the following formula:

$$\text{HR Deviation} = \text{NUHR} * [\{\text{BE}/(\text{BE} - .26 * O_2\text{DEV})\} - 1]$$

where

NUHR	=	Net unit heat rate
BE	=	Expected boiler efficiency
O_2DEV	=	Actual - expected O_2 measurement

B. Possible Causes of Deviation
   1. Fuel/air flow control problems
   2. Change in mill fineness
   3. Boiler casing leaks
   4. Air heater leaks
   5. Hot precipitator leaks
   6. Malfunctioning burner(s)
   7. FD fan inlet vanes mispositioned
   8. Burner registers mispositioned

C. Possible Corrections
   1. Operator Controllable
      a) Adjust FD fan inlet vanes
      b) Adjust FD fan speed (variable speed)
   2. Maintenance Correctable
      a) Adjust burner registers
      b) Clean or repair burners
      c) Repair air leaks
      d) Calibrate fuel/air flow controls
      e) Adjust pulverizer classifier vanes
      f) Replace pulverizer grinding wheels, balls, or rings

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9. Exit Gas Temperature

A. Effect on Heat Rate (per °C):

1. US Utility Average: 5.1 kJ/kWh or 1.2 kcal/kWh
2. US Utility Range: 4.0-8.0 kJ/kWh or 0.95-1.91 kcal/kWh
3. Plant-Specific Value: Can be estimated from the following formulas:

Exit gas temperature must be corrected for air heater leakage and for combustion air inlet temperature. The correct exit temperature can be calculated using the following equations:

$$\text{AH leakage (\%)} = 90 * \frac{(O_2 \text{ in} - O_2 \text{ out})}{(O_2 \text{ out} - 21)}$$

or

$$\text{AH leakage (\%)} = 90 * \frac{CO_2 \text{ in} - CO_2 \text{ out}}{CO_2 \text{ out}}$$

where all gas measurements are made at the entrance and exit of the air preheater.

$$TG\ (NL) = [\text{AH leak (\%)} * .24 * (TG - TA)]/24.2 + TG$$

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.

$$\text{HR Deviation} = NUHR * [\{BE/(BE - 2.0 * TGDEV)\} - 1]$$

where

NUHR	=	Net unit heat rate
BE	=	Expected boiler efficiency
TGDEV	=	Actual - expected Gas temperature corrected to zero leakage

B. Possible Causes of Deviation

1. Bypass dampers miss-positioned
2. Air heater baskets corroded/eroded
3. Air heater baskets fouled
4. Attemperating air flows misadjusted
5. Combustion air heater in use

C. Possible Corrections

1. Operator Controllable
   a) Reduce excess air
   b) Adjust bypass dampers
   c) Adjust tempering air flows

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d) Use air heater sootblowers
e) Adjust steam flow to coils
f) Adjust air recirculation dampers
g) Remove combustion air heater from operation

2. Maintenance Correctable
   a) Repair or replace air heater baskets
   b) Repair air heater leakage

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10. Condenser Pressure

A. Effect on Heat Rate (per 1 mm Hg)
   1. US Utility Average: 8.5 kJ/kWh or 2.03 kcal/kWh
   2. US Utility Range: 1.7-11.1 kJ/kWh or 0.4-2.7 kcal/kWh
   3. Plant-Specific Value: The effects on heat rate of a deviation in condenser pressure can be obtained from the unit's thermal kit.

B. Possible Causes of Deviation
   1. Air leakage
   2. Excess condenser load
   3. Tube fouling
   4. Low circulating water flow
   5. Increases in circulating water inlet temperature caused by:
      - o Changes in ambient conditions
      - o Problems with cooling tower performance

C. Possible Corrections
   1. Operator Controllable
      a) Increase circulating water flow
      b) Add an additional vacuum pump
      c) Check cycle isolation
      d) Place additional circulating water pumps in service
      e) Place additional cooling tower cells in service
   2. Maintenance Correctable
      a) Repair condenser air leaks
      b) Repair cycle isolation valves
      c) Clean condenser
      d) Repair circulating water discharge control valve
      e) Repair cooling tower

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11. Unburned Carbon

A. Effect on Heat Rate (per percentage point):

1. US Utility Average: 12.4 kJ/kWh or 2.96 kcal/kWh
2. US Utility Range: 6.3-13.5 kJ/kWh or 1.5-3.23 kcal/kWh
3. Plant-Specific Value: The effect on heat rate due to unburned carbon in the ash refuse can be approximated by the following equations:

Heat Loss Due to (UBHL) = (UBE-UBA) * ASH * 337.3/HHV

BE (COR) = BE (EXP) + (UBHL)

HR (DEV) = NUHR * {[BE (EXP)/BE (COR)] - 1.0}

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.. Possible Causes of Deviation

1 Incorrect fuel/air ratio
2. Change in mill fineness
3. Change in mill air flow

C. Possible Corrections

1. Operator Controllable
   a) Adjust fuel/air ratio
   b) Adjust mill secondary air flow damper settings
2. Maintenance Correctable
   a) Adjust classifier vane setting
   b) Repair or replace grinding wheels, balls, or rings
   c) Check secondary air heater for blockage
   d) Calibrate fuel/air control

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12. Coal Moisture

A. Effect on Heat Rate (per percentage point):

1. US Utility Average: 8.2 kJ/kWh or 2.0 kcal/kWh
2. US Utility Range: 6.3-10.6 kJ/kWh or 1.5-2.5 kcal/kWh
3. Plant-Specific Value: The effect on heat rate of changes from design values of the coal moisture can be calculated from the following equations:

Heat Loss (MHL) = (ME-MA) * (H_1 - H_2)/HHV
BE (CDR) = BE (EXP) + (MHL)
HR (DEV) = NUHR * {[BE (EXP)/BE (COR)] - 1.0}

ME	= Expected moisture content of coal
MA	= Measured moisture content of coal
H_1	= Enthalpy of vapor at 1 psia and corrected exit gas temperature TG (NL) (see item 9)
H_2	= Enthalpy of liquid at air heater inlet temperature TA (see item 9)
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

A. Possible Causes of Deviation

1. Change in coal quality

B. Possible Corrections

1. Operator Controllable
   a) None
2. Maintenance Correctable
   a) None

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13. Auxiliary Power

A. Effect on Heat Rate (per percentage point):
   1. US Utility Average: 90.7 kJ/kWh or 21.7 kcal/kWh
   2. US Utility Range: 68-102 kJ/kWh or 16-24 kcal/kWh
   3. Plant-Specific Value: The effect on heat rate of a deviation from the expected auxiliary power usage level can be obtained by using the following equations:

$$HR\ (DEV) = NUHR * [100 - AP\ (EXP)] / [100 - AP\ (ACT) - 1]$$

where

AP (EXP) = Expected power usage
AP (ACT) = Measured power usage

B. Possible Causes of Deviation
   1. Continuous running of noncontinuous loads
   2. Decline in efficiency of operating equipment
   3. Operation of redundant equipment during low-load operation

C. Possible Corrections
   1. Operator Controllable
      a) Stop non-continuous loads
      b) Reduce equipment operation at low loads
   2. Maintenance Correctable
      a) Repair or replace inefficient equipment
      b) Maintain equipment whose power usage increases with deteriorating performance, e.g., electrostatic precipitators, pulverizers, etc.

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14. Makeup

A. Effect on Heat Rate (per percentage point):

1. US Utility Average: 25 kJ/kWh or 6.1 kcal/kWh
2. US Utility Range: 4-93 kJ/kWh or 1-22 kcal/kWh
3. Plant-Specific Value: The effect on heat rate of changes from the expected amount of makeup can be estimated from the following equations:

$$MU(DEV) = MU(ACT) - MU(EXP)$$
$$H(DEV) = X(H_1 - H_2) + (1-X) * (H_3 - H_2)$$
$$TOP\ TERM = [MU(DEV) * THRFLO * H(DEV)] / (GG * 100)$$
$$BOT\ TERM = [BE(EXP)]/100 * [1 - AP(ACT)]$$
$$HR(DEV) = TOP\ TERM / BOT\ TERM$$

where

MU(ACT) = Actual makeup flow, %
MU(EXP) = Expected makeup flow, %
H_1 = Throttle enthalpy
H_2 = Makeup water enthalpy
H_2 = Feedwater enthalpy
THRFLO = Throttle flow, l bm/hr
GG = 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. Possible Cause of Deviation

1. Boiler tube leaks
2. Excess deaerator venting to atmosphere
3. Excess continuous blowdown
4. Excess steam lost through condenser venting
5. Valve packing leaks
6. Pump seal leaks
7. Steam leaks to atmosphere

C. Possible Corrections

1. Operator Controllable
   a) None
2. Maintenance Correctable
   a) Check dearator vent orifices or valve settings
   b) Repair valve and pump packings and seals
   c) Repair boielr tube leaks

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d) Optimize continuous blowdown
e) Isolate cycle losses

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15. Feedwater Heater Performance

A. Effect on Heat Rate:

1. Top High Pressure Heater Performance:
   Utility Average: 2.2 kJ/kWh or 0.53 kcal/kWh
   Utility Range: 1.4-2.4 kJ/kWh or 0.33-0.58 kcal/kWh

2. Next to Top High Pressure Heater Performance:
   Utility Average: 0.57 kJ/kWh or 0.14 kcal/kWh
   Utility Range: 0.35-0.66 kJ/kWh or 0.083-0.16 kcal/kWh

3. Third from Top High Pressure Heater Performance:
   Utility Average: 0.69 kJ/kWh or 0.16 kcal/kWh
   Utility Range: 0.60-0.84 kJ/kWh or 0.14-0.20 kcal/kWh

4. High Pressure Heater Out of Service:
   Top Heater: 99 kJ/kWh or 23.7 kcal/kWh
   Second Heater 74 kJ/kWh or 17.7 kcal/kWh
   Third Heater: 74 kJ/kWh or 17.7 kcal/kWh

5. Plant-Specific Value: The effect on heat rate of changes in feedwater heater TTDs and DCSs 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. Possible Causes of Deviation:

1. Changes in heater level
2. Changes in extraction line pressure drop
3. Reduced condensate flow through the heater
4. Heater baffle leaks
5. Failure to vent noncondensible gases
6. Tube fouling

C. Possible Corrections

1. Operator Controllable
   a) Set feedwater heater levels
2. Maintenance Correctable
   a) Optimize feedwater heater level
   b) Maintain heater vent valves and line orifices
   c) Repair baffle leaks
   d) Clean tube bundles

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16. Startup

A. Effect on Heat Rate:
   1. US Utility Average: 7.73 kJ/kWh or 1.85 kcal/kWh
   2. US Utility Range: 0.5-20 kJ/kWh or 0.13-4.8 kcal/kWh
   3. Plant-Specific Value: The effect on heat rate of unit start-up can be determined by using the method of Chapter x.

B. Possible Causes of Deviation
   1. Forced outage
   2. Unscheduled outage

C. Possible Corrections
   1. Operator Controllable
      a) None
   2. Maintenance Correctable
      a) Eliminate unscheduled outage through effective predictive and preventive maintenance

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APPENDIX E HEAT RATE LOGIC TREE

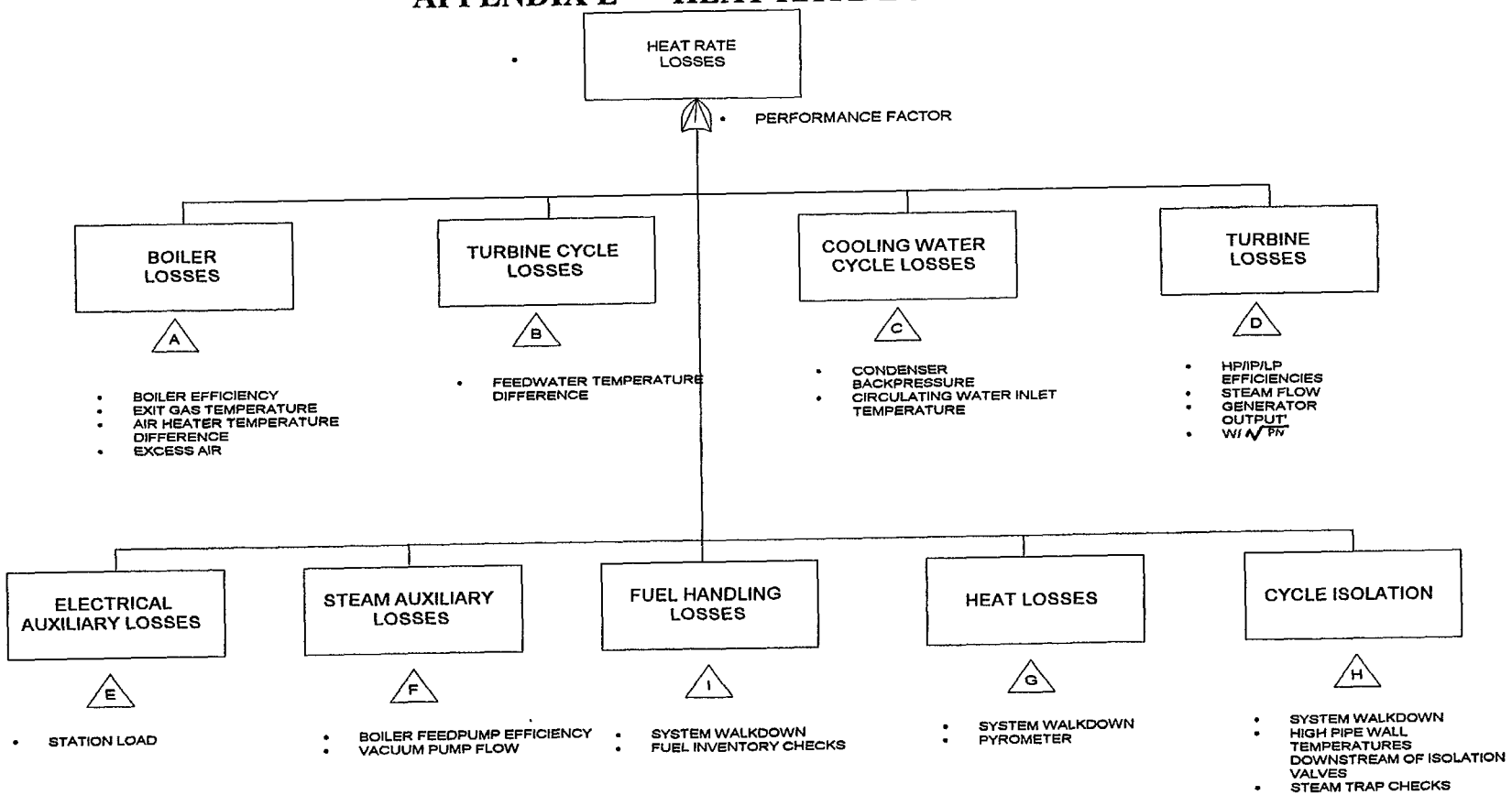

FIGURE E-1. HEAT RATE LOGIC TREE - MAIN DIAGRAM

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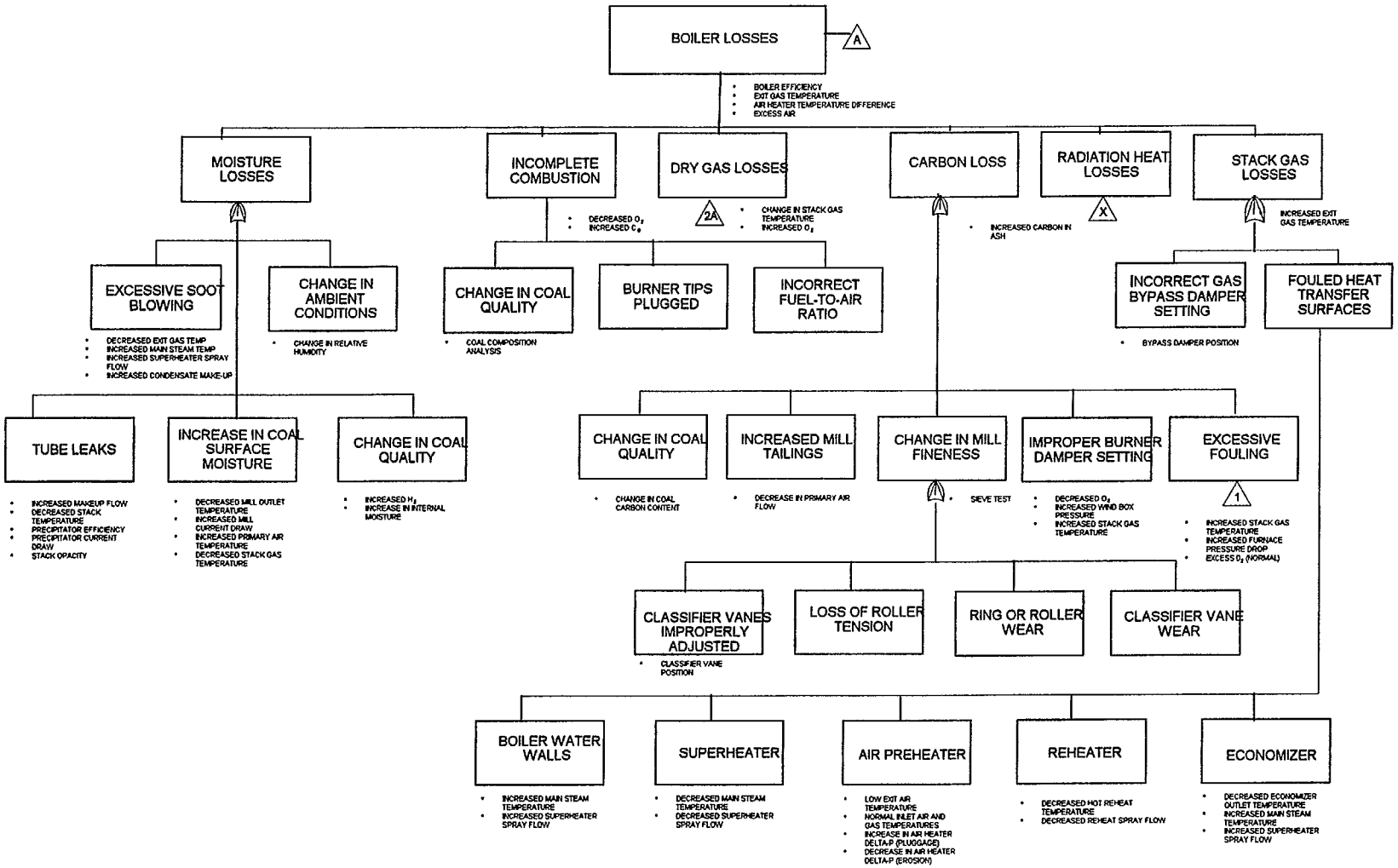

FIGURE E-2. HEAT RATE LOGIC TREE - BOILER LOSSES

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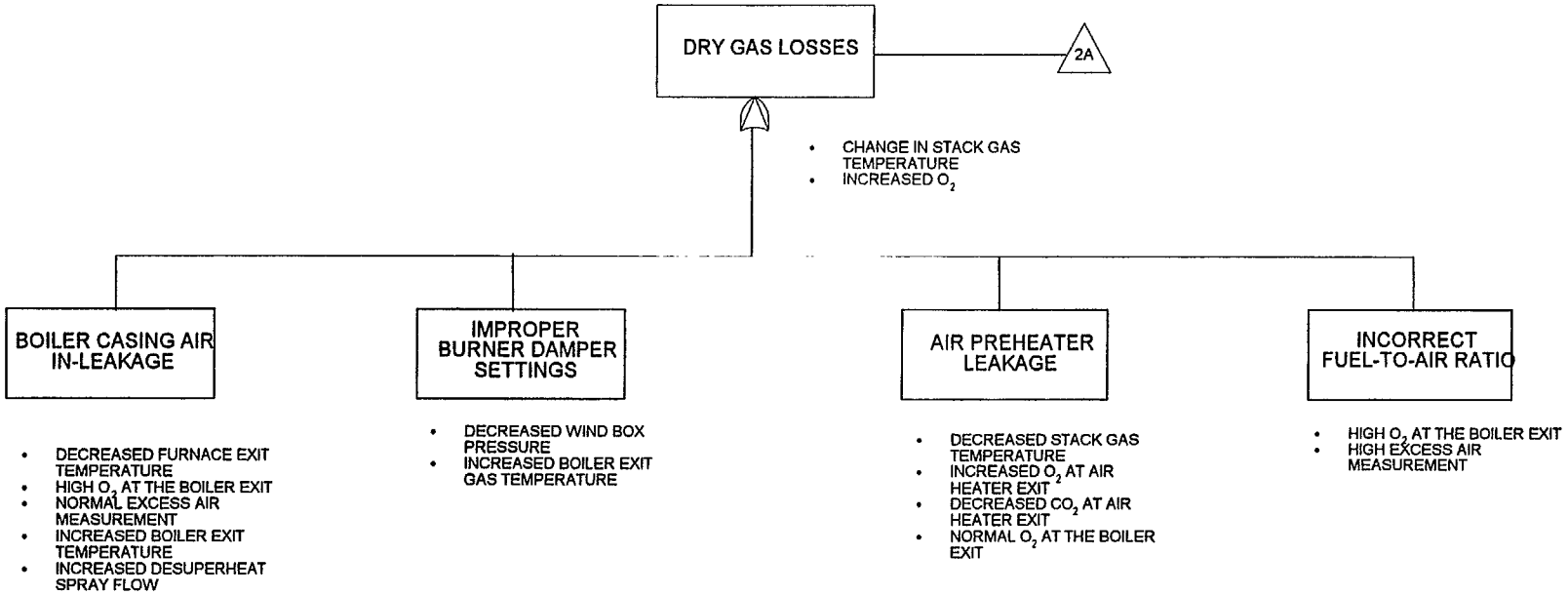

FIGURE E-3. HEAT RATE LOGIC TREE - DRY GAS

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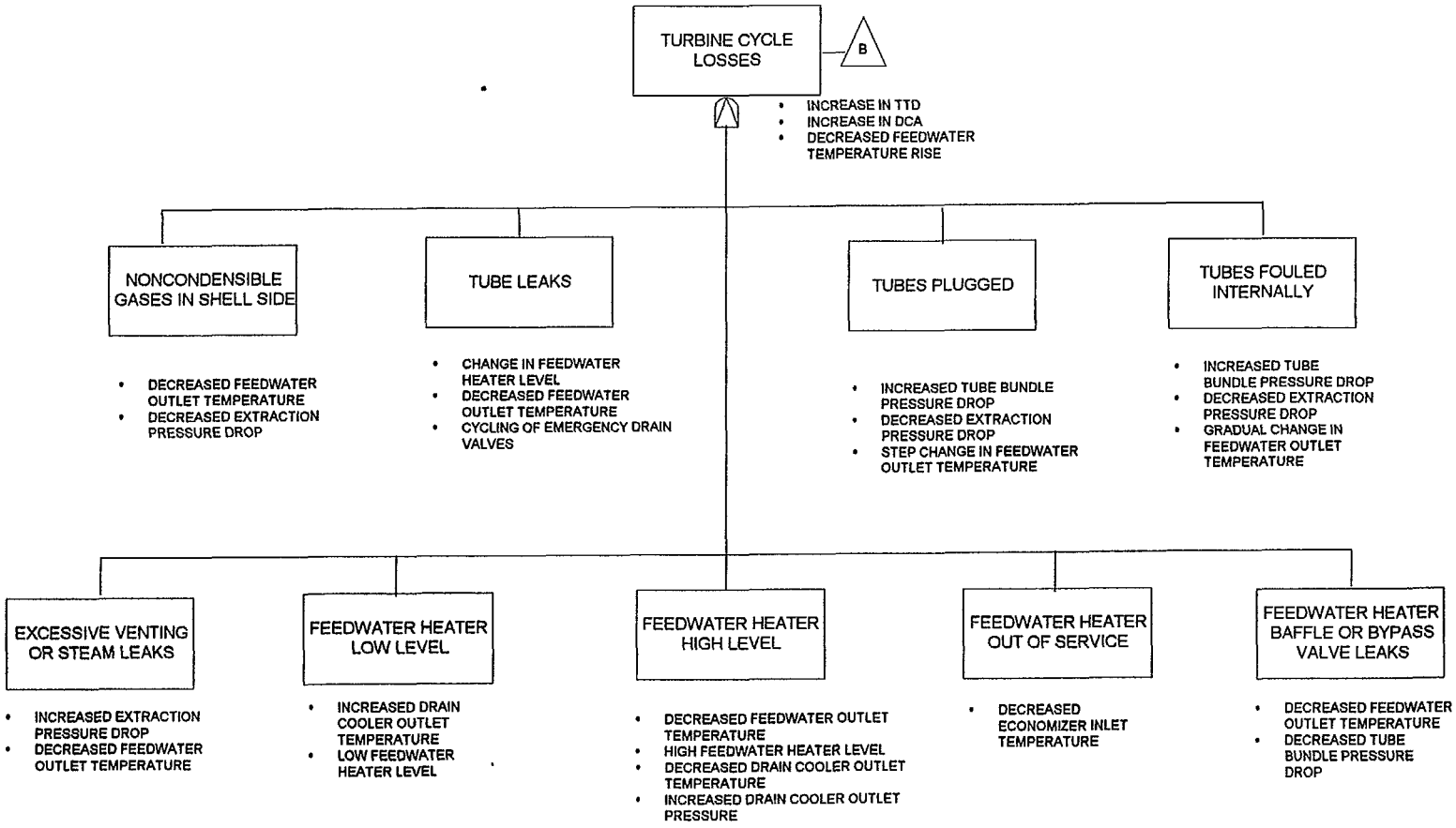

FIGURE E-4. HEAT RATE LOGIC TREE - TURBINE CYCLE LOSSES

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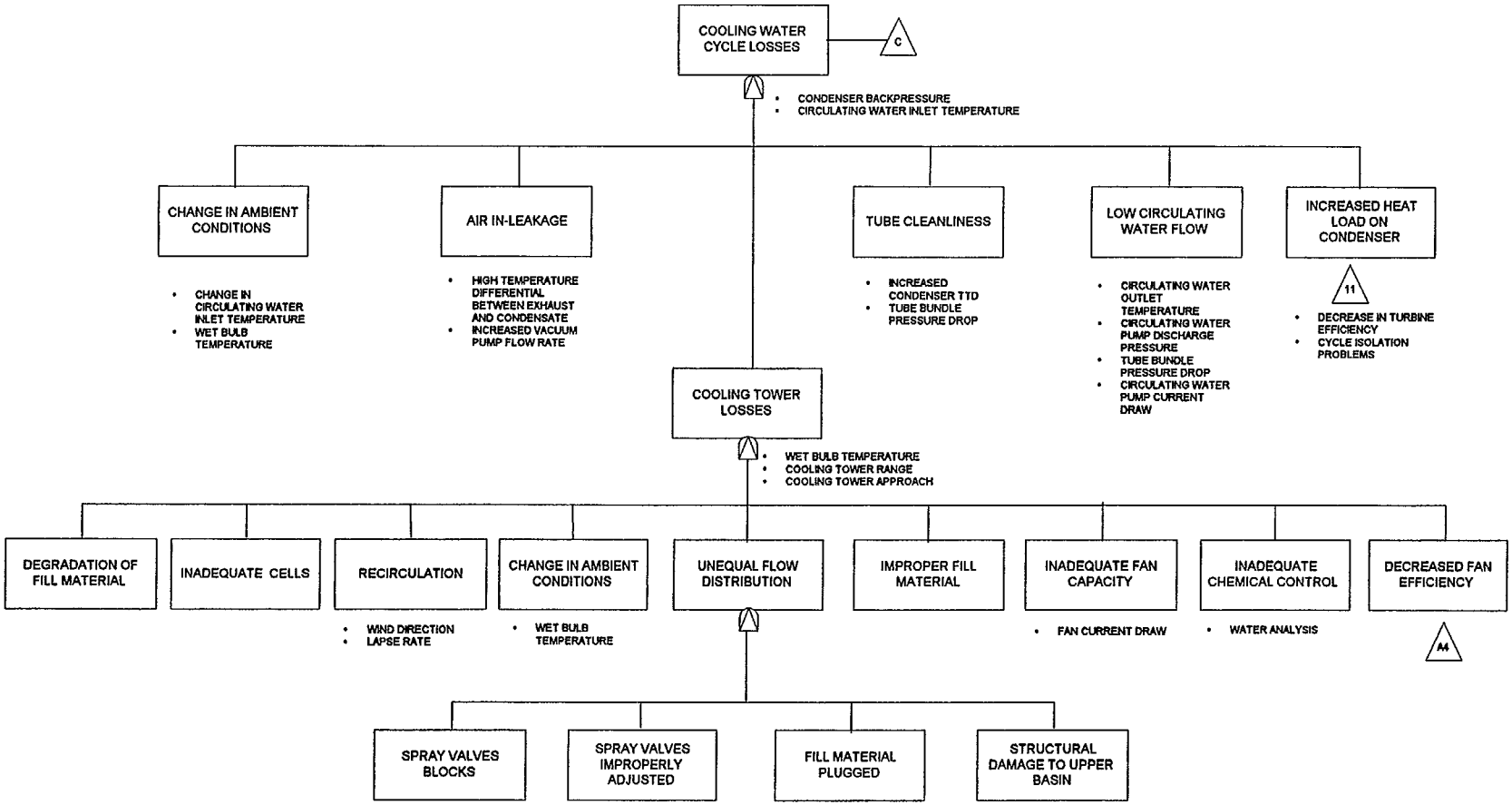

FIGURE E-5. HEAT RATE LOGIC TREE - COOLING WATER CYCLE LOSSES

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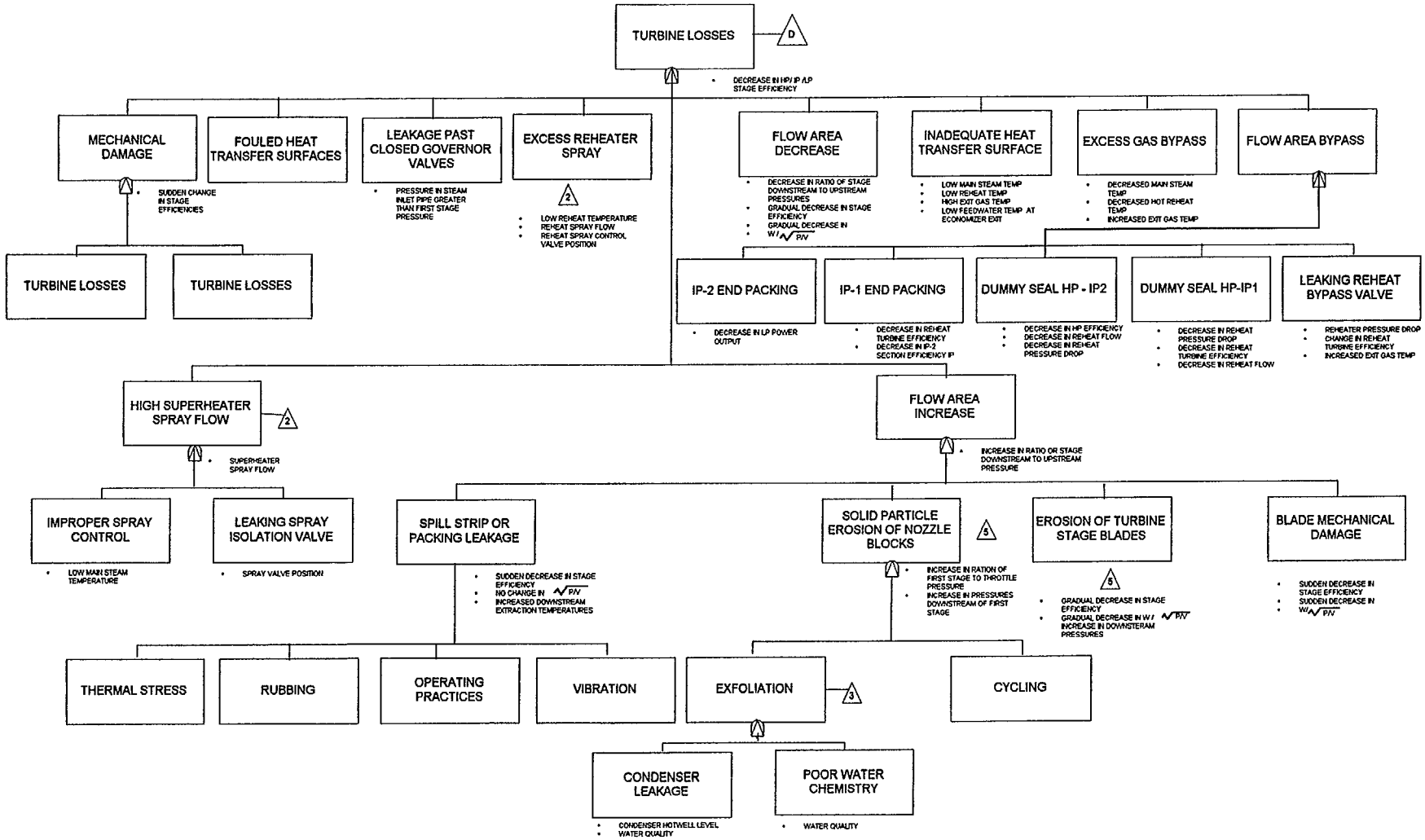

FIGURE E-6. HEAT RATE LOGIC TREE - TURBINE LOSSES

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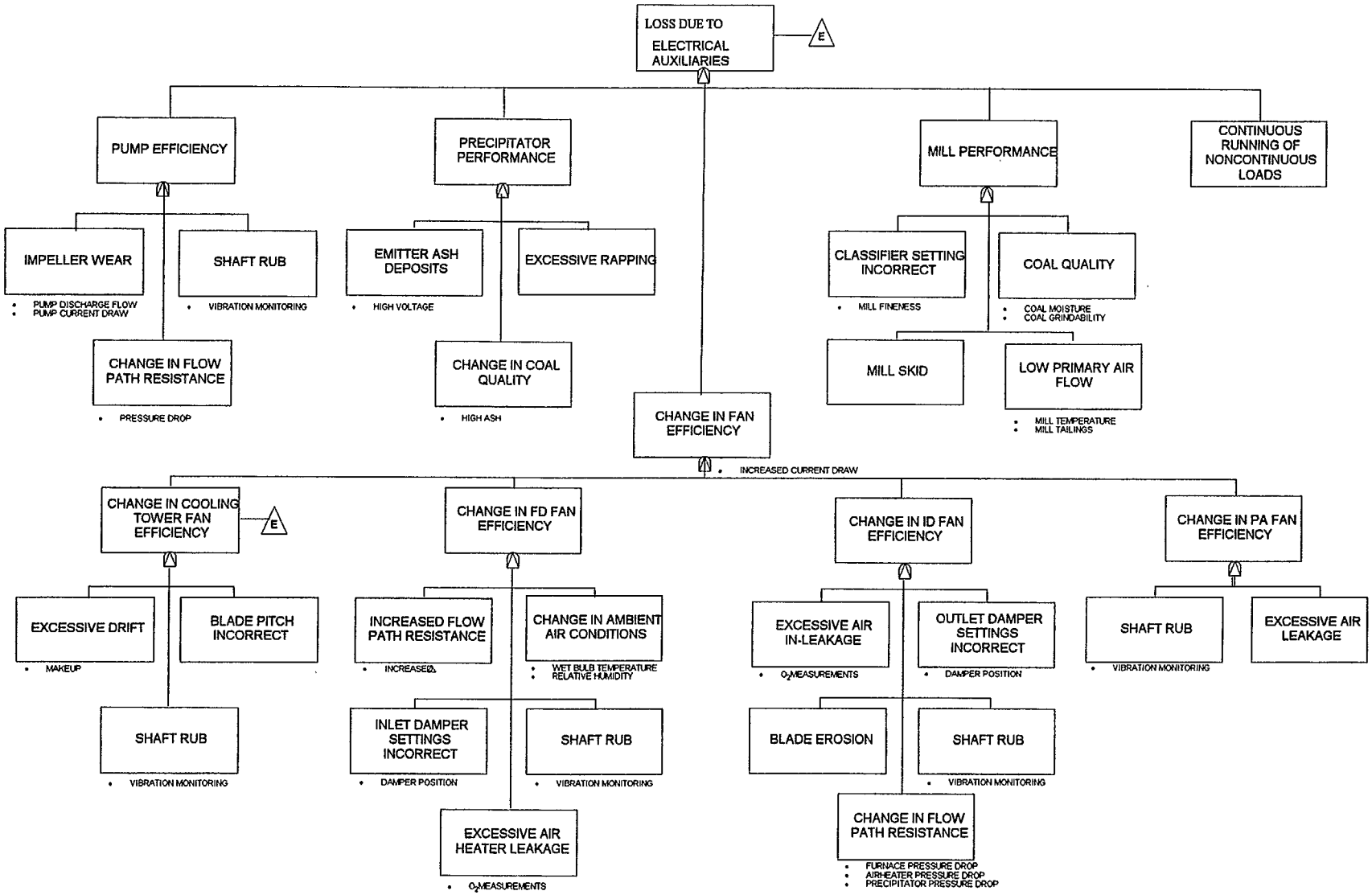

FIGURE E-7. HEAT RATE LOGIC TREE - ELECTRICAL AUXILIARY LOSSES

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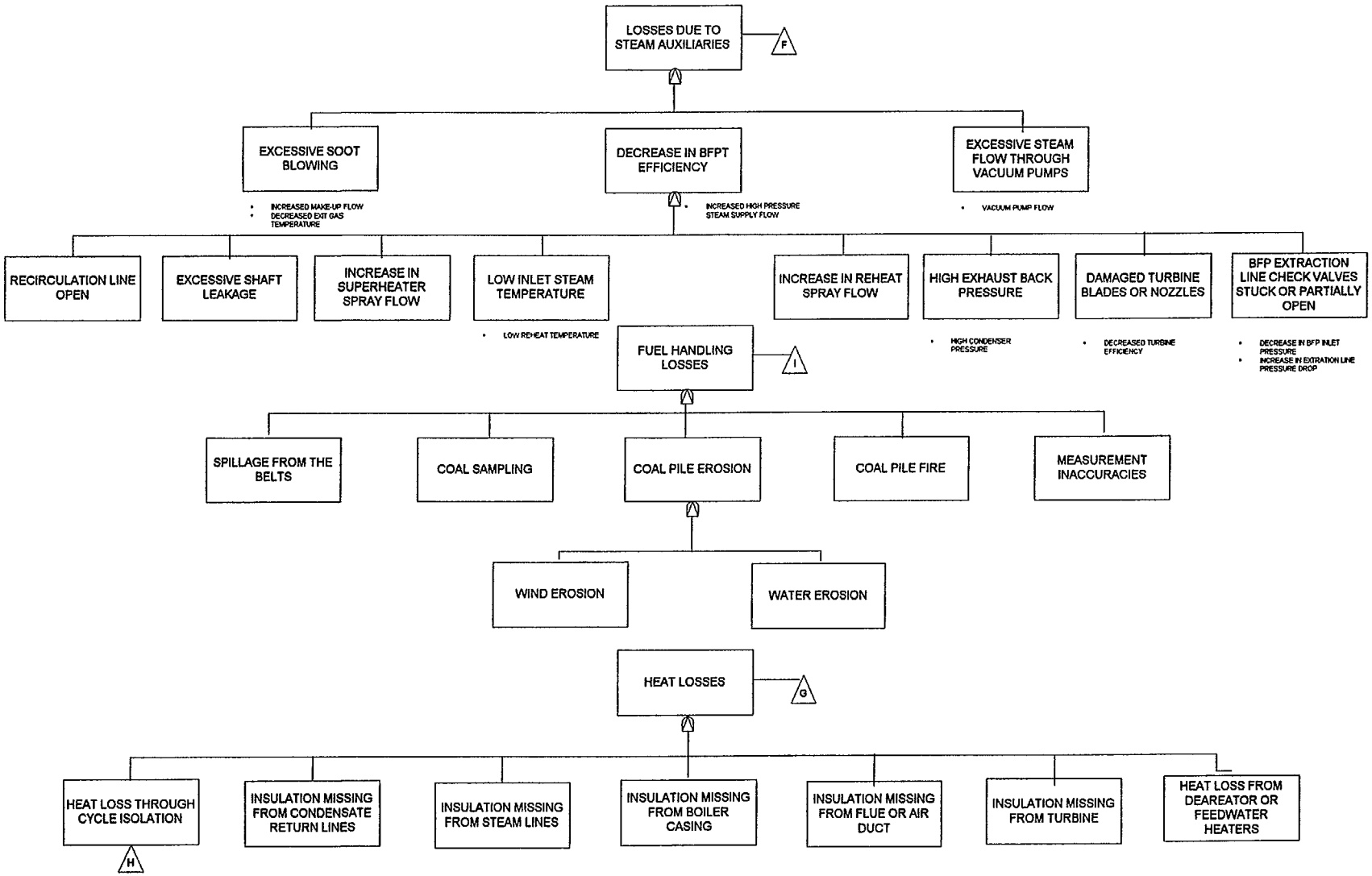

FIGURE E-8. HEAT RATE LOGIC TREE - STEAM AUXILIARY, FUEL HANDLING, AND HEAT LOSSES

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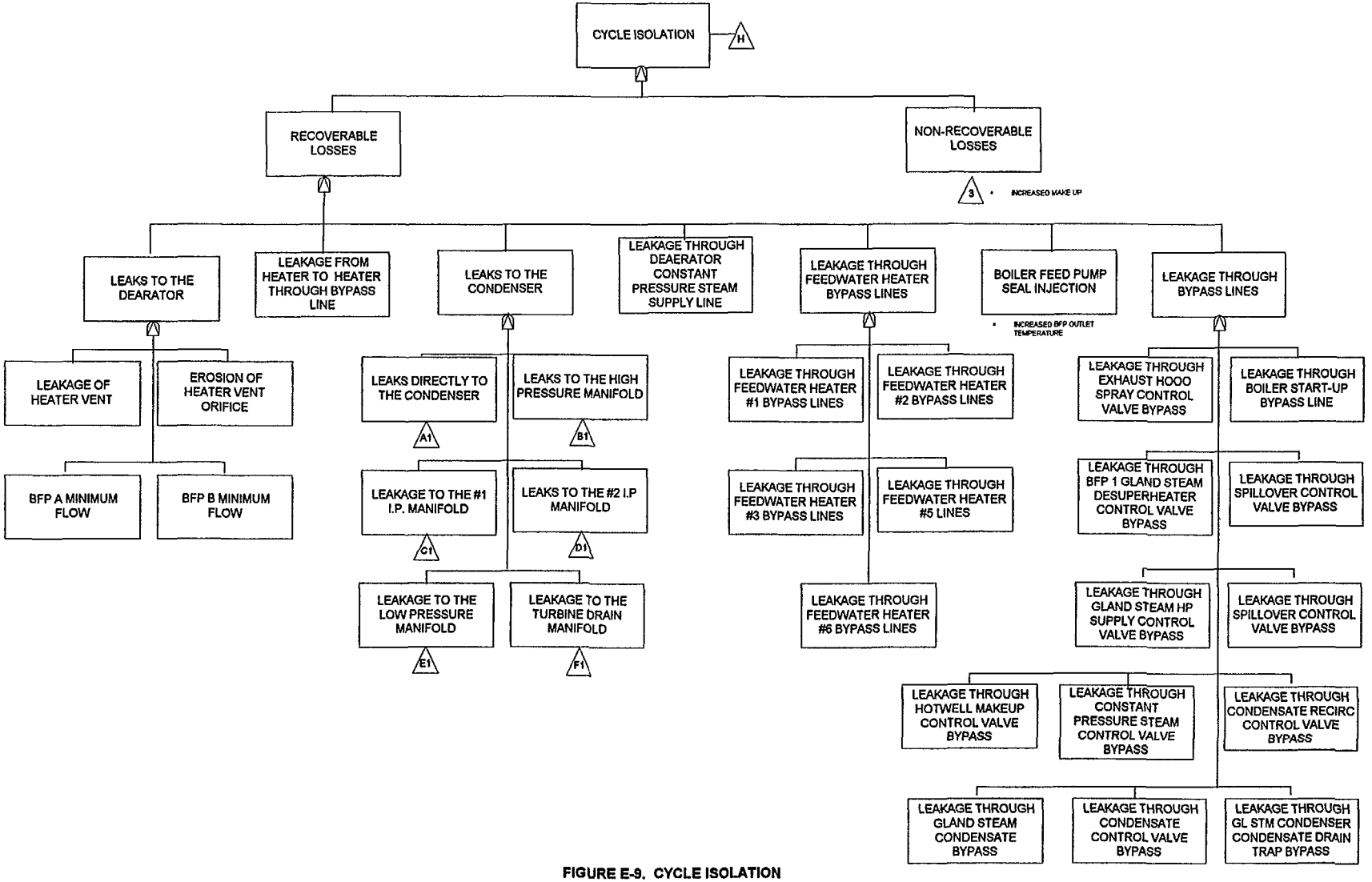

FIGURE E-9. CYCLE ISOLATION

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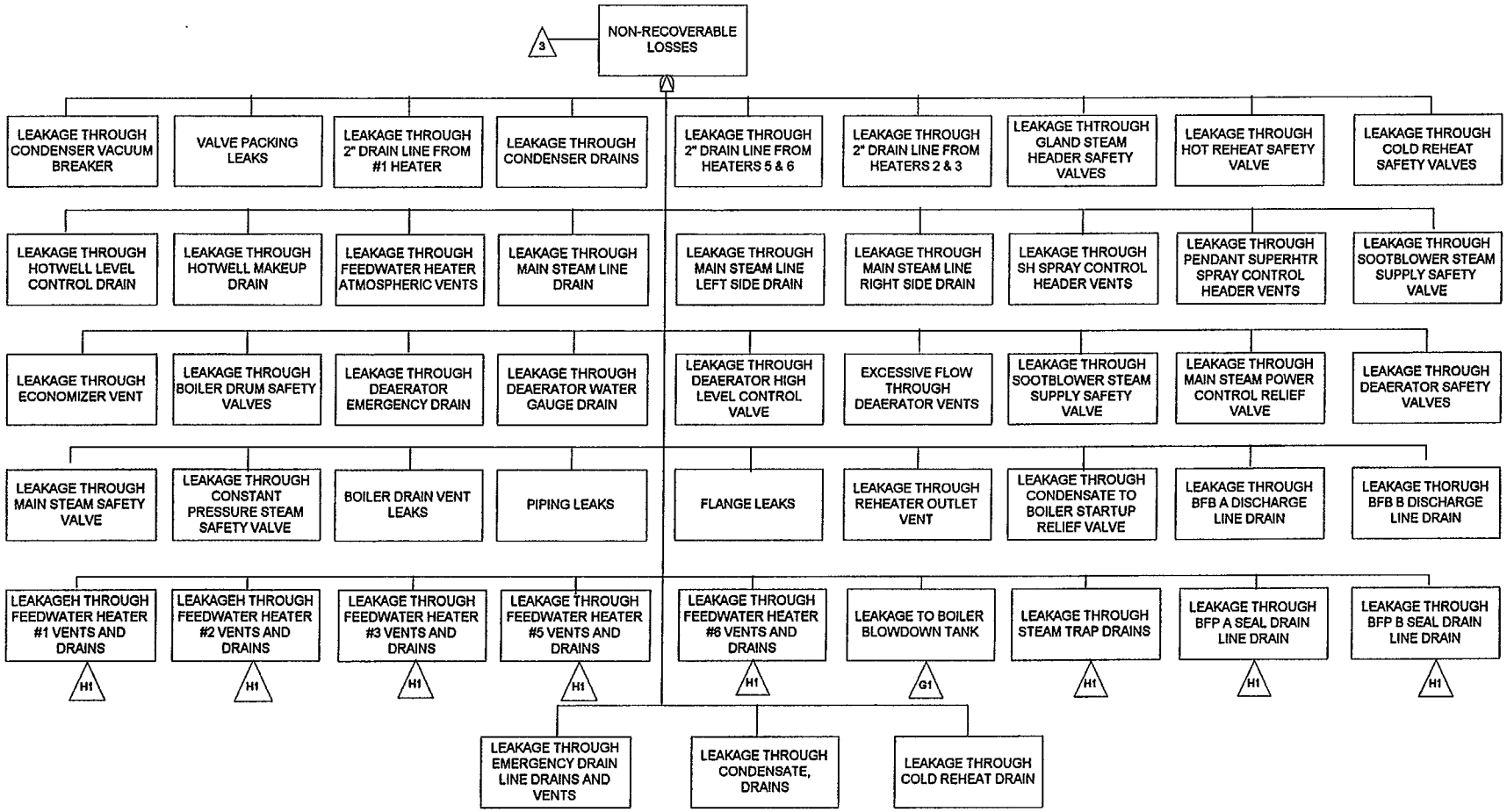

FIGURE E-9 (continued). CYCLE ISOLATION

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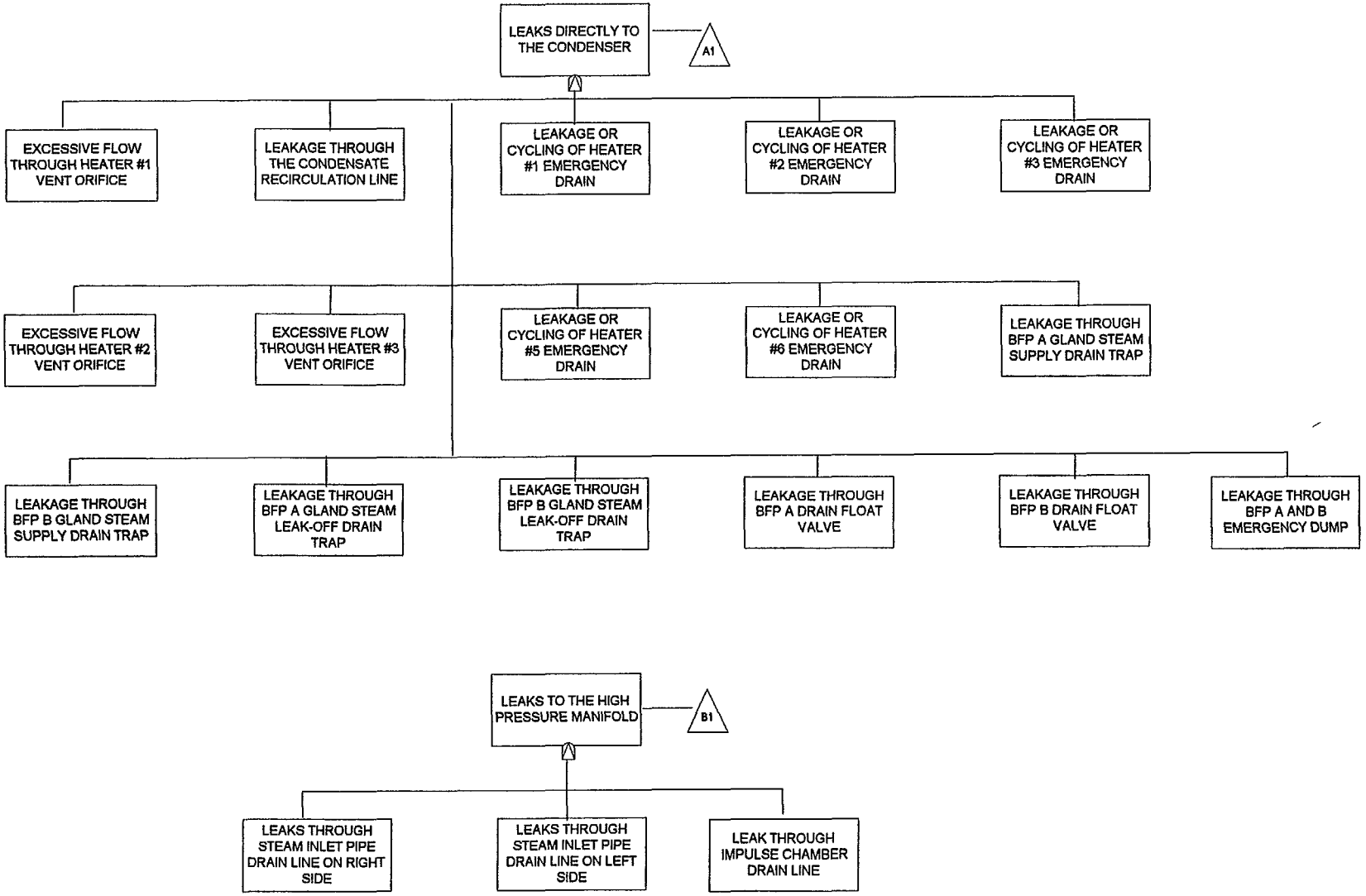

FIGURE E-9 (continued). CYCLE ISOLATION

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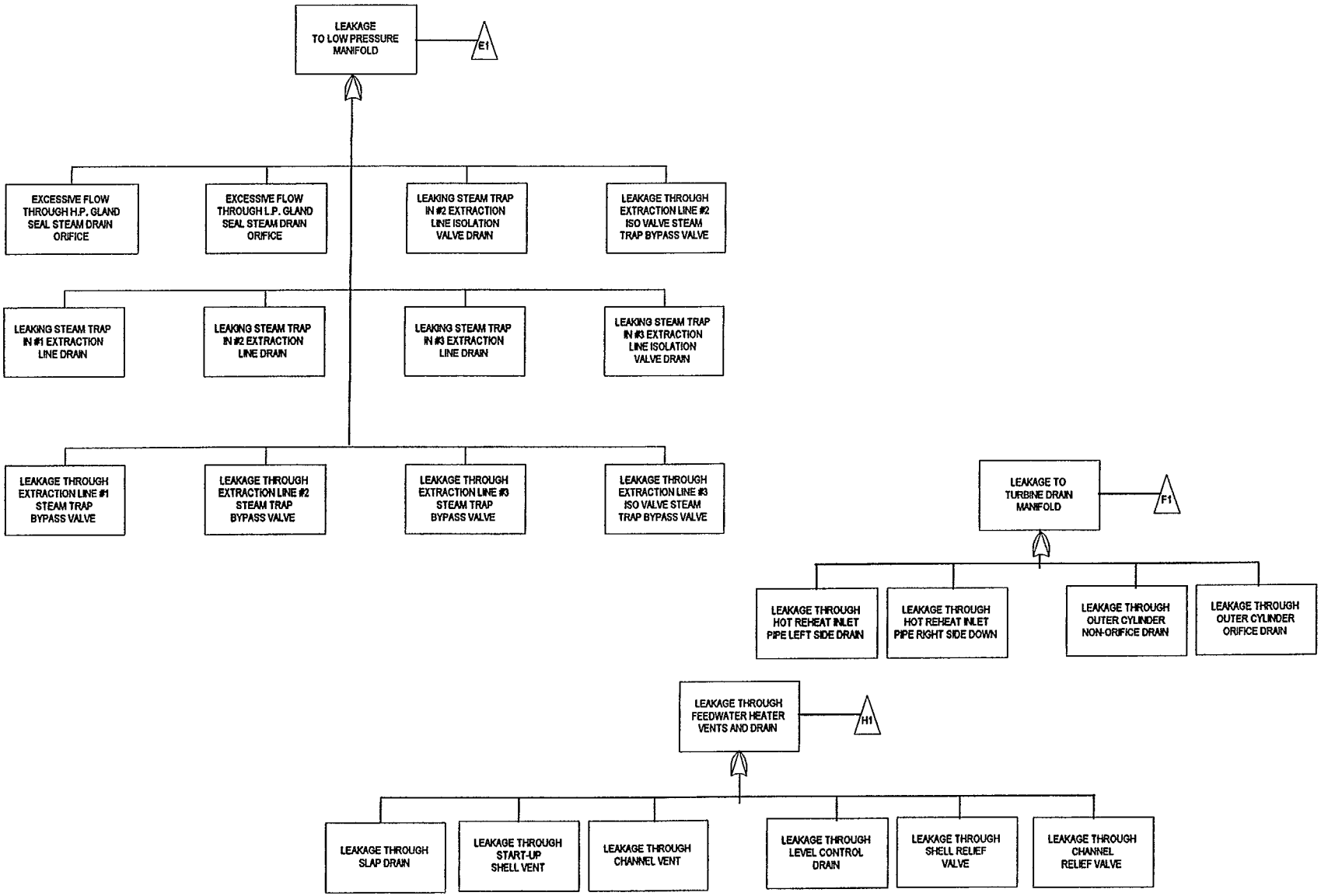

FIGURE E-9 (CONTINUED). CYCLE ISOLATION

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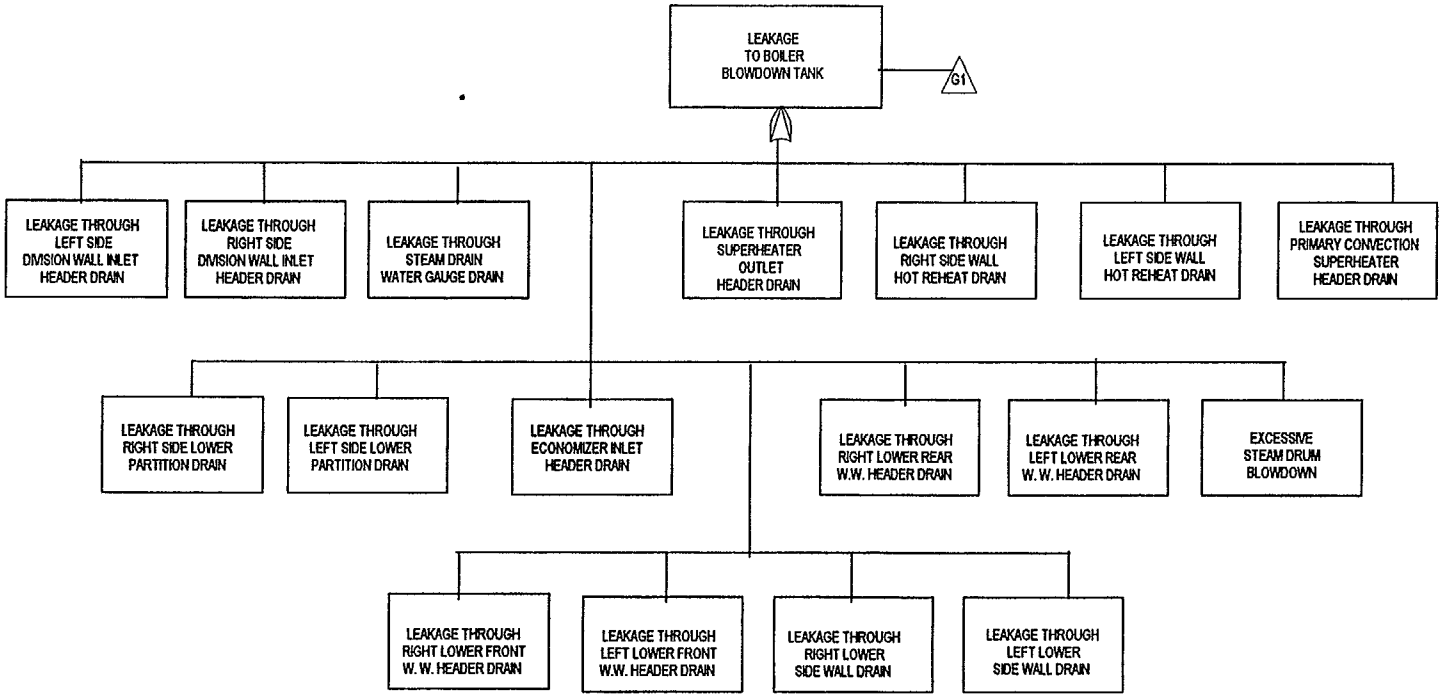

FIGURE E-9 (CONTINUED). CYCLE ISOLATION

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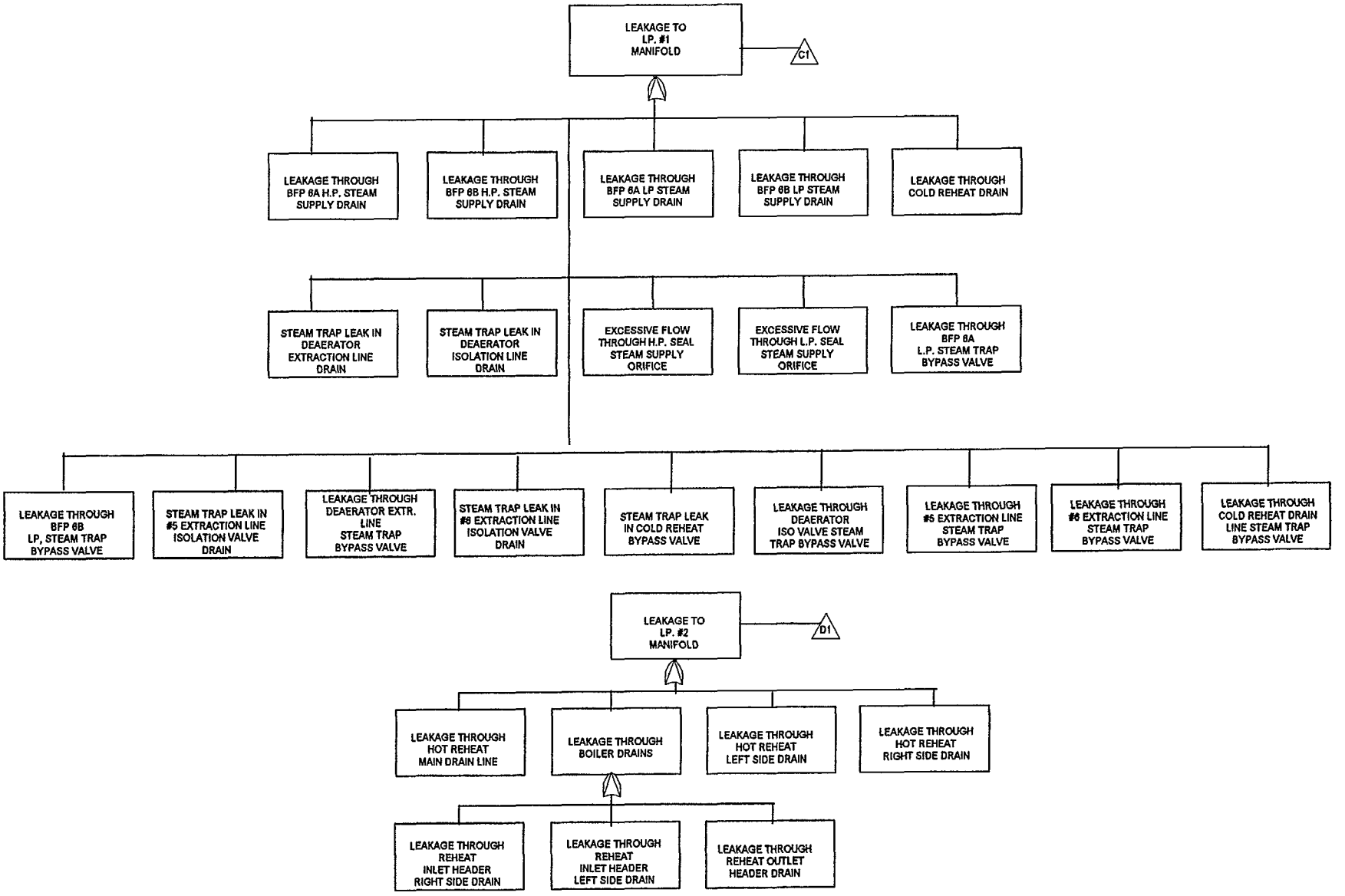

FIGURE E-9 (CONTINUED). CYCLE ISOLATION

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APPENDIX F SPLINE FIT EXAMPLE

Given 4 pairs of x,y data (x_1,y_1; x_2,y_2; x_3,y_3; x_4,y_4) calculate the value of y at a given x:

$$
\begin{aligned}
y = \quad & (x-x_2) * (x-x_3) * (x-x_4) * y_1 / \{(x_1-x_2) * (x_1-x_3) * (x_1-x_4)\} + \\
& (x-x_1) * (x-x_3) * (x-x_4) * y_2 / \{(x_2-x_1) * (x_2-x_3) * (x_2-x_4)\} + \\
& (x-x_1) * (x-x_2) * (x-x_4) * y_3 / \{(x_3-x_1) * (x_3-x_2) * (x_3-x_4)\} + \\
& (x-x_1) * (x-x_2) * (x-x_3) * y_4 / \{(x_4-x_1) * (x_4-x_2) * (x_4-x_3)\}
\end{aligned}
$$

For example, given the following data:

CCW Inlet Temperature °C	Net Unit Heat Rate (kcal/kWh)
8	2375
13	2383
18	2405
22	2419

What would be the net unit heat rate at 20 °C ?

$$
\begin{aligned}
y = \quad & (20 - 13) * (20 - 18) * (20 - 22) * 2375 / \{(8-13) * (8-18) * (8-22)\} + \\
& (20 - 8) * (20 - 18) * (20 - 22) * 2383 / \{(13-8) * (13-18) * (13-22)\} + \\
& (20 - 8) * (20 - 13) * (20 - 22) * 2405 / \{(18-8) * (18-13) * (18-22)\} + \\
& (20 - 8) * (20 - 13) * (20 - 18) * 2419 / \{(22-8) * (22-13) * (22-18)\}
\end{aligned}
$$

$$
\begin{aligned}
y = \quad & 7 * 2 * -2 * 2375 / \{-5 * -10 * -14\} + \\
& 12 * 2 * -2 * 2383 / \{5 * -5 * -9\} + \\
& 12 * 7 * -2 * 2405 / \{10 * 5 * -4\} + \\
& 12 * 7 * 2 * 2419 / \{14 * 9 * 4\}
\end{aligned}
$$

$$y = 95.00 + (-508.37) + 2020.2 + 806.33 = 2413.2$$

Note that if a straight line interpolation had been done,

$$
\begin{aligned}
y = \quad & 2405 + \{(2419 - 2405) * (20 - 18) / (22-18)\} \\
& 2412
\end{aligned}
$$

This technique is especially useful, if instead of having x,y pairs, you have four equations of y as a function of z, each equation at a given x value. Then you can use the four given equations to determine the four y values, then perform a spline fit using the calculated y values. For example, instead of being given the four heat rates listed above, you probably would have four equations for net unit heat rate as a function of gross load.

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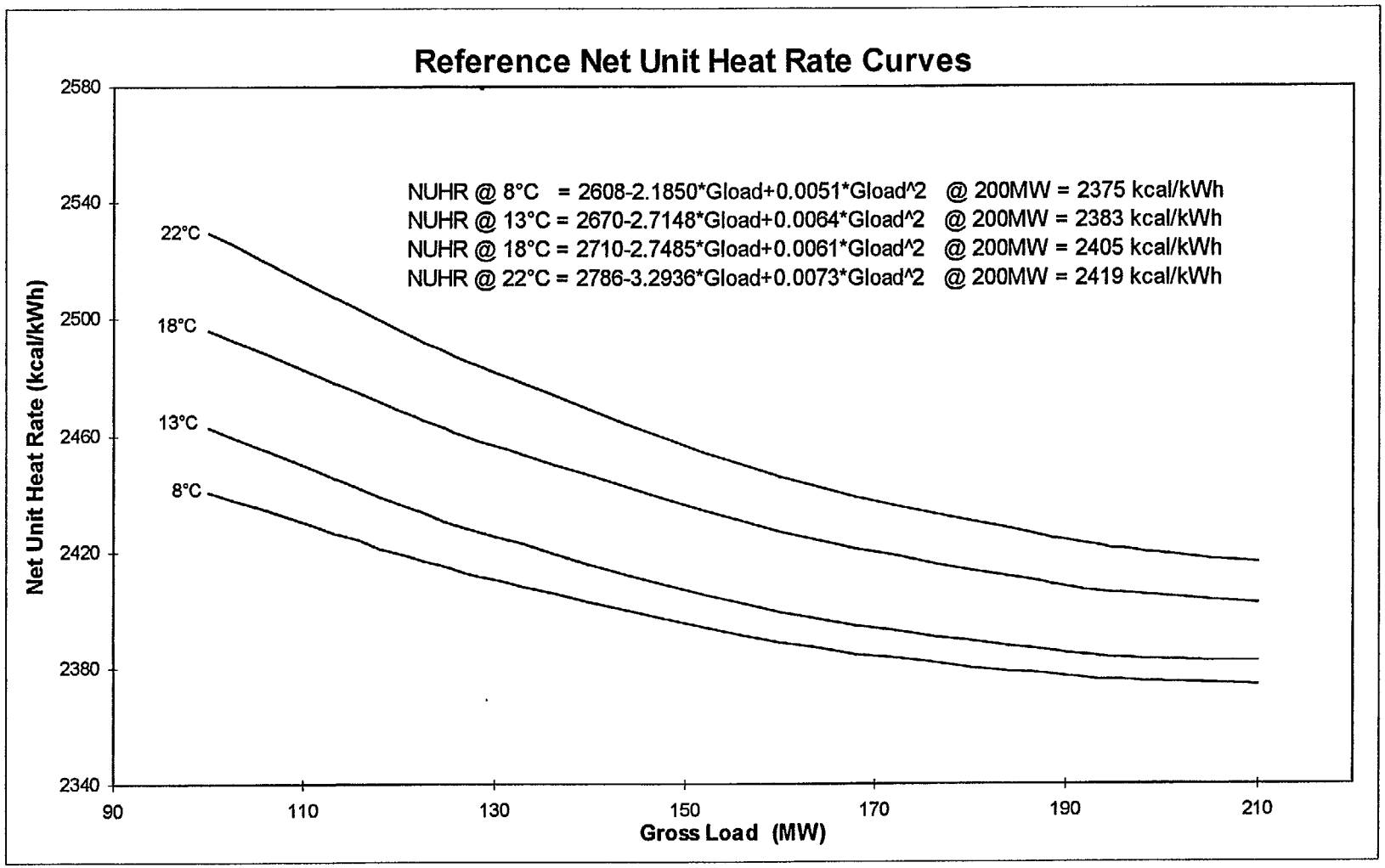

Figure F-1 Example of a Family of Curves Where Spline Fitting Would Be Used to Interpolate Between the Curves

APPENDIX G QUALIFICATION CARDS

This Appendix lists some Qualification Cards used by TVA to ensure that the technicians running test work are capable of performing their required duties. It also includes two examples, one of an "Equipment" card, dealing with how to use a certain type of equipment, and a "Test" card, which deals with the running of a particular test. Note that there are "Equipment" cards listed as prerequisites for the "Test" card.

Equipment Cards	*Test Cards*
Gauge & Absolute Pressure Transmitter	Turbine Efficiency - Enthalpy Drop
Differential Pressure Transmitters	Condenser Performance
RTD	High Pressure Feedwater Heaters Performance
Fecheimer Probe	Low Pressure Feedwater Heaters Performance
Dirty-Air & Fuel Sampling Probes	Boiler Feedwater Pump Performance
Pitot Tubes - "L" and "S" Type	Auxiliary Power Use Survey
HVT	Unit Capability
Coal Sampling	Air Heater Performance
Fineness Determination of Pulverized Coal	Clean Air Flow/Primary Air Flow Calibration
Coal Riffle	Dirty Air Flow/Pulverizer Fineness
Coal Grindability Machine	FD Fan Test
Boiler Test Probe Installation	ID Fan Test
Bubble Jar Setup	Condenser Air In-leakage Survey
Flue Gas Analyzer - Servomex 4900	Cooling Tower Test
Flue Gas Analyzer - Servomex 1440	Boiler Pressure Drop Test
Flue Gas Analyzer - Teledyne 320	Boiler Infrared Scanning
Helium Leak Detector - Varian	Turbine Cycle Heat Rate
Helium Leak Detector - Leybold	Boiler Efficiency/Optimization
Ultrasonic Flowmeter - Controlotron	
Ultrasonic Flowmeter - Panametrics	
Ultrasonic Flowmeter - Polysonics	
Fly Ash Sampling - High Volume Sampler	
Fly Ash Sampling - Isokinetic	
Fly Ash Sampling - Precipitator	
Station Service Data Readings	
Tank Level Data Readings	
Coal Scale Data Readings	
Basic Programming of Data Acquisition Sys.	
Data Acquisition System Module Setup	
Computer Link to DAS	
Calibration Data Regression Analysis	

PERFORMANCE TESTING TECHNICIAN
&
POWER PLANT TECHNICAL ANALYST

TASK QUALIFICATION CARD

Pitot Tubes - "L" and "S" Types

	SIGNATURE	DATE
PREPARED BY: R. J. Tramel		
MANAGER, TECHNICAL SERVICES DEPT. Edmund A. Kopetz		
TECHNICAL TRAINING PROGRAM MANAGER Douglas J. Peters		

1.0 Performance Standards

Utilizing required safety measures and practices, the Technician / Analyst shall demonstrate the proper method of measuring fluid speed using an "L" type (or Pitot-Static) and "S" type (or Forward-Reverse or Double Reverse or Stauschiebe) pitot tube, and converting the speed to mass flow.

2.0 General References

ASME Performance Test Code (PTC) 11-1984 Fans

ASME Performance Test Code (PTC) 19.10-1981 Flue and Exhaust Gas Analyses

AMCA Publication 203 Fan Application Manual Part 3 - Field Performance Measurements

TVA's "Performance Testing and Test Equipment Basics Course" Notebook

3.0 Prerequisites

3.1 Observe a qualified person perform this task.

3.2 Perform the task with a qualified person.

4.0 Knowledge Items (Technician / Analyst should answer)

4.1 What is the "standard" "K" factor for an "L" type pitot and a "S" type pitot that has not been calibrated in a laboratory ?

4.2 Define the terms "static pressure", "velocity pressure", and "total or impact or stagnation pressure".

4.3 Name two conditions when it is preferable to use an "S" type pitot over an "L" type.

4.4 What advantage does a "Fechheimer" probe have over "L" and "S" type pitot tubes ?

4.5 What measurements are necessary, and what is the general equation for calculating velocity when using a pitot tube ?

4.6 Describe in general terms how to determine the number of reading that are required to be taken in a duct ?

4.7 If multiple readings are taken in a duct, how are the differential pressures "averaged" ?

5.0 Task Performance Items

5.1 Locate the test port to be used. Check to see if it meets the required criteria (distance from upstream and downstream obstructions). Assemble all material on site (pitot, tubing, manometers, temperature measurement device, etc.)

5.2 If multiple readings are to be made across the duct, mark the probe as per reference 1.

5.3 For the differential pressure reading, set-up, level and zero the incline manometer (or if a u-tube manometer is used, check that it is installed vertically). Take two equal lengths of Tygon tubing. On both, mark one end "high" (as with red tape) and mark the other end "low" (as with green tape). Connect one pair of ends to the manometer and the other pair of ends to the pitot tube (on the "S" type, the tube on the side of the probe facing the flow is the high side; on the "L" type, the tap that is in line with the handle is the high side). Be sure the connections are tight and do not leak.

5.4 For the static pressure reading when using an "L" type pitot, connect a second manometer to the low side tap (the one that is parallel to the direction of flow) using a "tee". The static pressure can be read at the same time the differential pressure is read. For the "S" type pitot, a "tee" can be placed at the low side tap and connected to a second manometer. After reading the differential pressure, the probe can be rotated 90 degrees and the differential pressure should go to zero. At that position, the static pressure can be read from the second manometer.

5.5 Leak check the system.

5.6 Pinch both hoses closed and insert the probe to the first measurement point and simultaneously release both hoses. (This is to prevent blowing the fluid out of the manometer if an unusually high differential pressure is sensed as the pitot is inserted.)

5.7 Record the position, differential pressure and static pressure, move to the next point and repeat the process until all points have been measured. (If using a u-tube manometer, be sure to record the readings from both legs.)

5.8 Pinch both hoses and remove the pitot. Check to insure that the manometer is still zeroed.

5.9 Convert the differential pressure, static pressure and temperature readings to velocity. Velocity = K * $\sqrt{(2 * \text{differential pressure/density})}$. Note that depending on the units of the differential pressure (psi, inches of water, inches of mercury, etc.) and the desired units for velocity (feet / second, etc.) there will need to be conversion factors.

5.10 Be sure the data sheet includes, in addition to the differential pressures for each position for each port, the plant, unit number, exact location of test port, date and time, the temperature and static pressure, names of the personnel involved, serial numbers of the instruments used, and any other relevant comments.

6.0 Remedial Action

If the technician / analyst is unable to adequately perform all of the tasks outlined in this qualification card to the satisfaction of the evaluator, then the evaluator will specify actions or additional training that is needed before the technician / analyst attempts the tasks at a later time.

If the technician / analyst is unable to meet the satisfaction of the evaluator, after three consecutive attempts, then the evaluator shall notify the Test Section Manager / Production Manager (Engineering) for proper remedial action.

7.0 Signatures

When the evaluator is satisfied that the technician / analyst has fully demonstrated adequate or better than adequate performance of the criteria specified by this document, then the evaluator will sign the qualification sheet.

Training Status: **Initial** _____ **Continuing** _____ **Other** _____

TECHNICIAN / ANALYST:

_____ _____ _____
Print Name **S. S. #** **Signature**

(1) Date : _____ **Tech :** _____ (Initials) __**Pass** __**Fail** **Evaluator** _____ (Signature)

Comments : _____

(2) Date : _____ **Tech :** _____ (Initials) __**Pass** __**Fail** **Evaluator** _____ (Signature)

Comments : _____

(3) Date : _____ **Tech :** _____ (Initials) __**Pass** __**Fail** **Evaluator** _____ (Signature)

Comments : _____

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PERFORMANCE TESTING TECHNICIANS
&
POWER PLANT TECHNICAL ANALYST

TASK QUALIFICATION CARD

Condenser Performance Test

	SIGNATURE	DATE
PREPARED BY: Robert J. Tramel		
MANAGER, TECHNICAL SERVICES DEPT. Edmund A. Kopetz		
TECHNICAL TRAINING PROGRAM MANAGER Douglas J. Peters		

1.0 Performance Standards

Utilizing test instrumentation, the Technician / Analyst shall conduct a condenser performance test, following the proper procedure for the unit being tested. This includes installing the test equipment, setting unit conditions, collecting raw data, and calculating and interpreting the results.

2.0 General References

ASME Performance Test Code (PTC) 12.2-1978 Code on Steam Condensing Apparatus

Heat Exchange Institute's Standards for Steam Surface Condensers, Ninth Edition, 1995

TVA's "Performance Testing and Test Equipment Basics Course" Notebook

Fundamentals of Power Plant Performance; General Physics Corporation, Columbia, Maryland

Unit Group Specific Condenser Performance Test Procedure
i.e. TS/PERF/RTST/FOS/9.0-KIF 1-4 or TS/PERF/RTST/FOS/9.0-PAF 3

3.0 Prerequisites

3.1 Complete the relevant equipment qualification cards:

3.11 Pressure Transmitter Installation & Removal	FH-TCH 006.002
3.12 Cable Installation & Removal	FH-TCH 006.004
3.13 Data Acquisition System Module Setup	FH-TCH 006.005x
3.14 Computer Link to Data Acquisition System	FH-TCH 006.006x
3.15 Calibration Data Regression	FH-TCH 006.012x
3.16 Commissioning Sequence of DP Transmitters	FH-TCH 006.013
3.17 Basic Programming of D/A System	FH-TCH 006.014x
3.18 RTD Installation	FH-TCH 006.032

3.2 Observe a qualified person perform this task.

3.3 Perform the task with a qualified person.

4.0 Knowledge Items (Technician / Analyst should answer)

4.1 When should a condenser performance test be conducted ?

4.2 What unit conditions are required for a proper and repeatable test ?

4.3 Define "Cleanliness Factor", how is it determined, and how does it affect unit performance ?

4.4 What are the possible causes of temperature stratification in a CCW outlet pipe ?

4.5 What are the possible causes of a low cleanliness factor ?

4.6 What effect does circulating water flow have on cleanliness factor and back pressure ?

4.7 What does the term "subcooling" mean and what is the effect of subcooling the :
A) condensate in the hotwell
B) steam/air mixture going to the vacuum pumps or steam jet air pumps ?

5.0 Task Performance Items

5.1 Lead a test crew in conducting a condenser performance test, following a unit-group specific test procedure, including :
A. Insuring that the test equipment is installed properly
B. Setting the proper unit conditions
C. Collecting the data
D. Calculating the results
E. Analyzing the results
F. Recommending corrective actions as necessary.

6.0 Remedial Action

If the Technician / Analyst is unable to adequately perform all of the tasks outlined in this qualification card to the satisfaction of the evaluator, then the evaluator will specify actions or additional training that is needed before the Technician / Analyst attempts the tasks at a later time.

If the technician / analyst is unable to meet the satisfaction of the evaluator, after three consecutive attempts, then the evaluator shall notify the Test Section Manager / Production Manager (Engineering) for proper remedial action.

7.0 Signatures

When the evaluator is satisfied that the Technician / Analyst has fully demonstrated adequate or better than adequate performance of the criteria specified by this document, then the evaluator will sign the qualification sheet.

Training Status: **Initial** _____ **Continuing** _____ **Other** _____

TECHNICIAN / ANALYST:

_____ _____ _____

Print Name **S. S. #** **Signature**

(1) Date : _____ **Tech :** _____ ___**Pass** ___**Fail** **Evaluator** _____
Initials Signature

Comments : _____

_____

(2) Date : _____ **Tech :** _____ ___**Pass** ___**Fail** **Evaluator** _____
Initials Signature

Comments : _____

_____

(3) Date : _____ **Tech :** _____ ___**Pass** ___**Fail** **Evaluator** _____
Initials Signature

Comments : _____

_____

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APPENDIX H CONDENSER TUBE HEAT TRANSFER CHARACTERISTICS

Table H.1 Uncorrected Heat Transfer Coefficients - U_1 (HEI Standards Table 1)

Tube Diameter	Tube Velocity (ft/s)								
	3.0	*3.5*	*4.0*	*4.5*	*5.0*	*5.5*	*6.0*	*6.5*	*7.0*
0.625 & 0.75"	462.5	499.5	534.0	566.4	597.0	626.2	654.0	680.7	706.4
0.875 & 1.00"	455.0	492.0	526.0	557.9	588.1	616.8	644.2	670.5	695.8
1.125 & 1.25"	448.6	484.5	518.0	549.4	579.1	607.4	634.4	660.3	685.2
1.375 & 1.50"	441.7	477.1	510.0	540.9	570.2	598.0	624.6	650.1	674.7
1.625 & 1.75"	434.7	469.6	502.0	532.5	561.3	588.6	614.8	639.9	664.1
1.875 & 2.00"	427.8	462.1	494.0	524.0	552.3	579.8	605.0	629.7	653.5

Tube Diameter	Tube Velocity (ft/s)									
	7.5	*8.0*	*8.5*	*9.0*	*9.5*	*10.0*	*10.5*	*11.0*	*11.5*	*12.0*
0.625 & 0.75"	731.2	755.2	775.5	795.3	814.1	831.9	848.9	865.2	880.7	895.6
0.875 & 1.00"	720.3	743.9	763.9	783.2	801.6	819.0	835.6	851.5	866.6	881.1
1.125 & 1.25"	709.3	732.6	752.0	770.7	788.4	805.3	821.4	836.7	851.3	865.3
1.375 & 1.50"	698.3	721.2	740.4	758.7	776.1	792.6	808.3	823.2	837.5	851.2
1.625 & 1.75"	687.4	709.9	727.8	745.7	762.7	778.8	794.1	808.8	822.7	836.0
1.875 & 2.00"	676 4	698.6	716.8	734.4	751.0	766.8	781.8	796.2	809.8	822.9

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Table H.2 Inlet Water Temperature Correction Factor - F_W (HEI Standards Table 2)

Inlet Water		F_W	Inlet Water		F_W	Inlet Water		F_W
°F	°C		°F	°C		°F	°C	
30	-1.1	0.650	60	15.6	0.923	90	32.2	1.075
31	-0.6	0.659	61	16.1	0.932	91	32.8	1.078
32	0.0	0.669	62	16.7	0.941	92	33.3	1.080
33	0.6	0.678	63	17.2	0.950	93	33.9	1.083
34	1.1	0.687	64	17.8	0.959	94	34.4	1.085
35	1.7	0.696	65	18.3	0.968	95	35.0	1.088
36	2.2	0.706	66	18.9	0.975	96	35.6	1.090
37	2.8	0.715	67	19.4	0.982	97	36.1	1.092
38	3.3	0.724	68	20.0	0.989	98	36.7	1.095
39	3.9	0.733	69	20.6	0.994	99	37.2	1.097
40	4.4	0.743	70	21.1	1.000	100	37.8	1.100
41	5.0	0.752	71	21.7	1.005	101	38.3	1.103
42	5.6	0.761	72	22.2	1.010	102	38.9	1.105
43	6.1	0.770	73	22.8	1.015	103	39.4	1.108
44	6.7	0.780	74	23.3	1.020	104	40.0	1.110
45	7.2	0.789	75	23.9	1.025	105	40.6	1.113
46	7.8	0.798	76	24.4	1.029	106	41.1	1.115
47	8.3	0.807	77	25.0	1.033	107	41.7	1.117
48	8.9	0.816	78	25.6	1.037	108	42.2	1.119
49	9.4	0.825	79	26.1	1.041	109	42.8	1.121
50	10.0	0.834	80	26.7	1.045	110	43.3	1.123
51	10.6	0.843	81	27.2	1.048	111	43.9	1.125
52	11.1	0.852	82	27.8	1.051	112	44.4	1.127
53	11.7	0.861	83	28.3	1.054	113	45.0	1.129
54	12.2	0.870	84	28.9	1.057	114	45.6	1.131
55	12.8	0.879	85	29.4	1.060	115	46.1	1.133
56	13.3	0.888	86	30.0	1.063	116	46.7	1.135
57	13.9	0.897	87	30.6	1.066	117	47.2	1.137
58	14.4	0.905	88	31.1	1.069	118	47.8	1.139
59	15.0	0.914	89	31.7	1.072	119	48.3	1.141
						120	48.9	1.143

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Table H.3 Tube Material and Gauge Correction Factor - F_M (HEI Standards Table 3)

Tube Material	**Tube Wall Gauge - BWG**								
	25	*24*	*23*	*22*	*20*	*18*	*16*	*14*	*12*
Admiralty Metal	1.03	1.03	1.02	1.02	1.01	1.00	0.98	0.96	0.93
Arsenical Copper	1.04	1.04	1.04	1.03	1.03	1.02	1.01	1.00	0.98
Copper Iron 194	10.4	1.04	1.04	1.04	1.03	1.03	1.02	1.01	1.00
Aluminum Brass	1.03	1.02	1.02	1.02	1.01	0.99	0.97	0.95	0.92
Aluminum Bronze	1.02	1.02	1.01	1.01	1.00	0.98	0.96	0.93	0.89
90-10 Cu-Ni	1.00	0.99	0.99	0.98	0.96	0.93	0.89	0.85	0.80
70-30 Cu-Ni	0.97	0.97	0.96	0.95	0.92	0.88	0.83	0.78	0.71
Cold Rolled Carbon Steel	1.00	1.00	0.99	0.98	0.97	0.93	0.89	0.85	0.80
Stainless Steel Type 304/316	0.91	0.90	0.88	0.86	0.82	0.75	0.69	0.62	0.54
Titanium	0.95	0.94	0.92	0.91	0.88	0.82	0.77	0.71	0.63
UNS N08367	0.90	0.89	0.87	0.85	0.81	0.74	0.67	0.60	0.52
UNS S43035	0.95	0.94	0.92	0.91	0.88	0.82	0.77	0.71	0.63
UNS S44735	0.93	0.91	0.90	0.88	0.85	0.78	0.72	0.65	0.57
UNS S44660	0.93	0.91	0.90	0.88	0.85	0.78	0.72	0.65	0.57

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APPENDIX I SPECIFIC HEATS OF DRY AIR, WATER VAPOR , AND WATER VAPOR

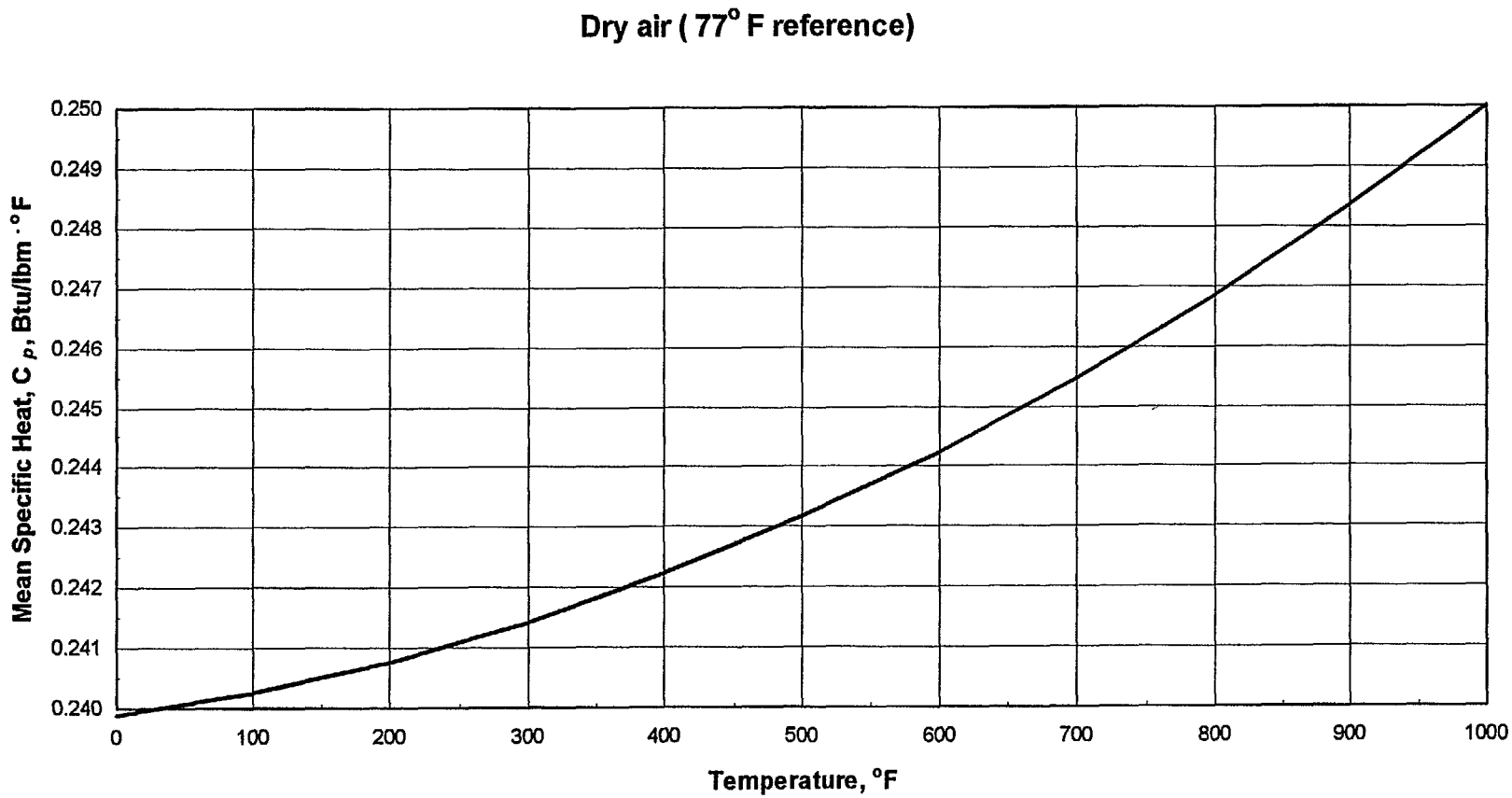

Figure I.1 Mean Specific Heat of Dry Air versus Temperature (ASME Figure 5.19-1)

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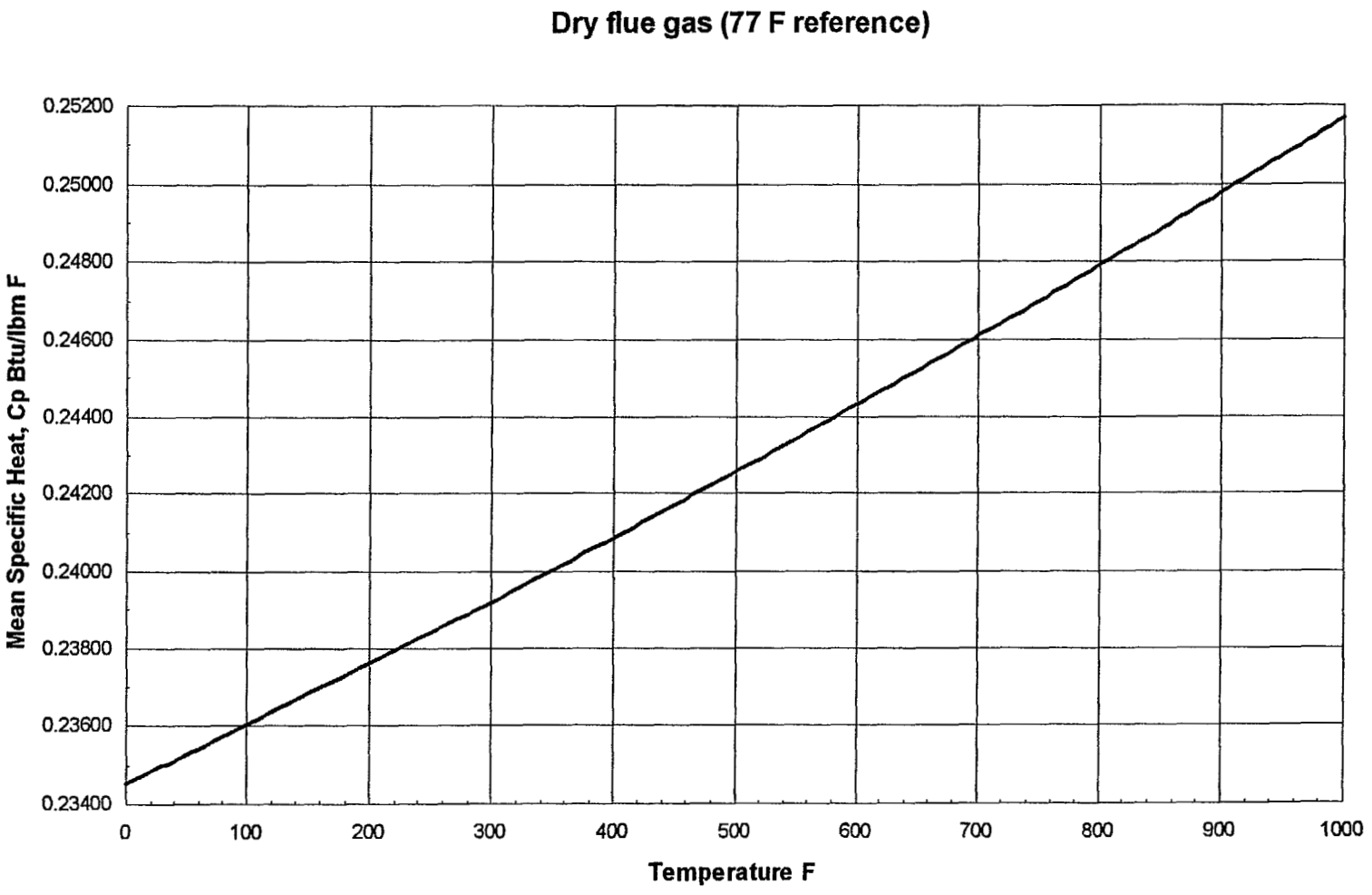

Figure I.2 Mean Specific Heat of Dry Flue Gas versus Temperature (ASME Figure 5.19-3)

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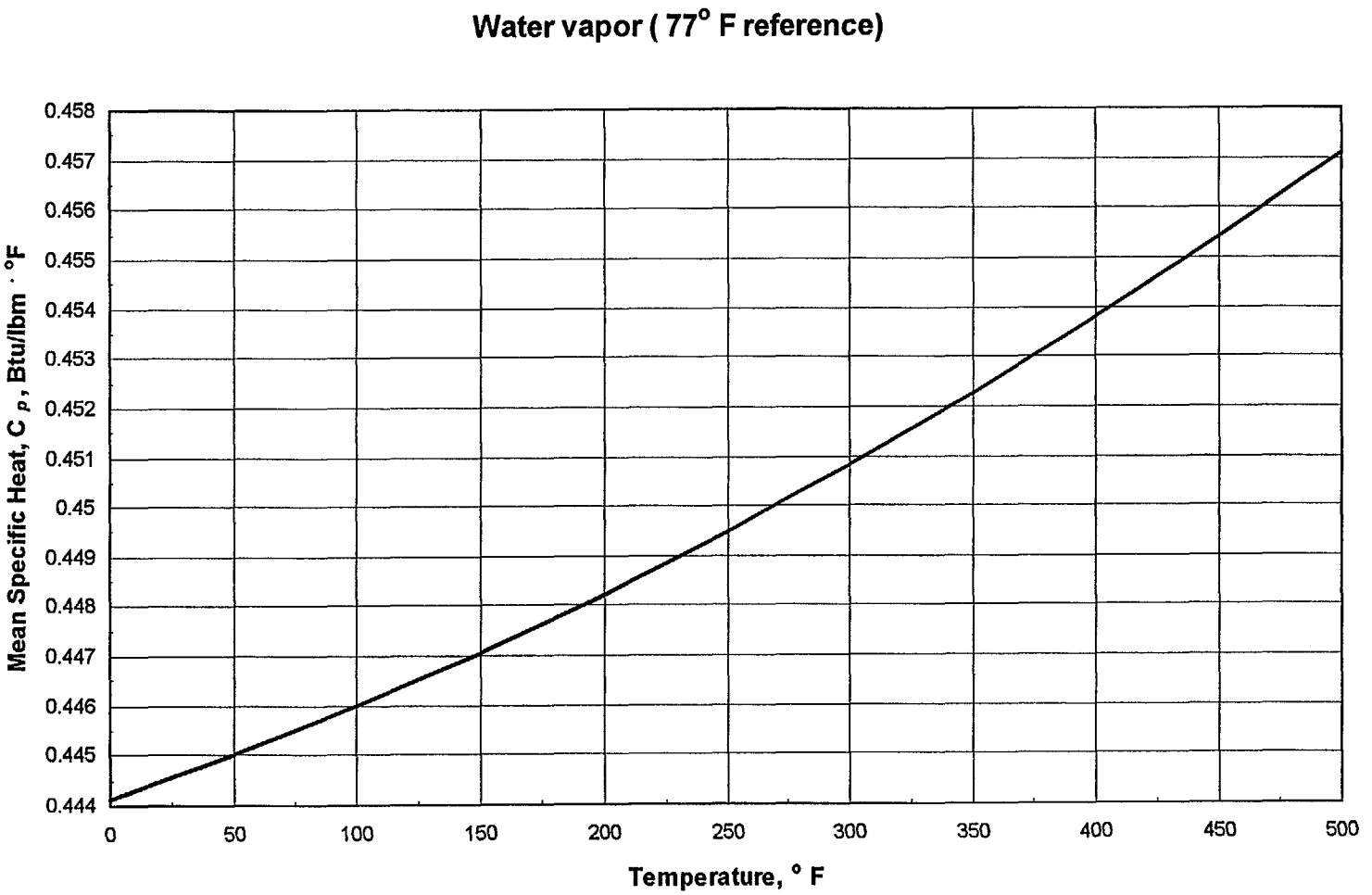

Figure I.3 Mean Specific Heat of Water Vapor versus Temperature 0-500 F (ASME Figure 5.19-2)

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