Thermal Performance Engineers Handbook, Volume II

Advanced Concepts in Thermal Performance

WARNING:
Please read the Export Control and License Agreement on the back cover before removing the Wrapping Material.

Technical Report

Thermal Performance Engineering Handbook, Volume II

Advanced Concepts in Thermal Performance

TR-107422-V2

Final Report
October 1998

Prepared for
EPRI
3412 Hillview Avenue
Palo Alto, California 94304

EPRI Project Manager
T. Eckert

DISCLAIMER OF WARRANTIES AND LIMITATION OF LIABILITIES

ORGANIZATION(S) THAT PREPARED THIS REPORT

Mollerus Engineering Corporation

ORDERING INFORMATION

Requests for copies of this report should be directed to the EPRI Distribution Center, 207 Coggins Drive, P.O. Box 23205, Pleasant Hill, CA 94523, (925) 934-4212.

CITATIONS

This report was prepared by

Mollerus Engineering Corporation
455 Los Gatos Blvd., Suite 208
Los Gatos, CA 95032

Principal Investigators
Fred J. Mollerus
William B. Reuland

This report describes research sponsored by EPRI. The report is a corporate document that should be cited in the literature in the following manner:

Thermal Performance Engineering Handbook, Volume II, EPRI, Palo Alto, CA, 1998. TR-107422-V2.

REPORT SUMMARY

The two-volume Thermal Performance Engineer Handbook will assist thermal performance engineers in identifying and investigating the cause of megawatt (MWe) losses as well as in proposing new ways to increase MWe output.

Volume I contains a thermal performance primer to provide a brief review of thermodynamic principles involved in the steam power plant thermal cycle. The primer also contains brief descriptions of the equipment and systems in the cycle that can be sources of thermal losses. Also in Volume I is a section on the elements of a thermal performance program and a guideline for developing a desktop instruction for the thermal performance engineer.

Volume II contains detailed discussions of the components of the nuclear steam turbine cycle as well as corrective actions and modifications that can be taken to maintain and enhance plant electrical equipment.

Background

Thermal performance is the term used in this handbook to describe how well a nuclear plant is converting thermal energy to electrical energy. The elements of a thermal performance program and the duties of a thermal performance engineer (TPE) include:

- Monitoring and trending plant thermal performance
- Identifying and investigating the causes of loss of electrical production in megawatts
- Developing potential remedial actions to correct conditions that cause MWe losses
- Developing potential changes and modifications to recover and enhance MWe production
- Developing potential actions to ensure efficient MWe production during the life of the plant

Utilities frequently rotate personnel through the plant performance engineer position, and the transition of personnel into this position can be costly (training, lessons learned, etc.). A group of utility thermal performance engineers (Plant Performance Enhancement Program (P^2EP) coordinators) recommended that EPRI capture both the science and the "art" of thermal performance engineering and present them in handbook form.

Objectives

To provide guidance to assist utility thermal performance engineers in identifying and investigating the cause of MWe losses as well as in proposing new ways to increase MWe output.

Approach

The EPRI Plant Support Engineering Program established the TPE Handbook Task Group, which met five times in 1997. The Task Group reviewed various utility thermal performance programs and identified the major elements in the most successful of those programs. The Task Group used the actual utility best practices to develop the guidance in this document.

Key Points

- A thermal performance primer provides a brief review of thermodynamic principles involved in the steam power plant thermal cycle.
- The basic elements of a good thermal performance program are identified.
- The balance of plant components that can adversely affect electrical output are discussed.
- Specific ways to improve heat rate and megawatt electrical output are covered.
- Industry consensus was achieved.

TR-107422, V2

Interest Categories

Plant support engineering
Thermal performance

ACKNOWLEDGMENTS

The following individuals were ongoing members of Plant Support Engineering's Thermal Performance Engineer's Handbook Task Group. As such, they have made significant contributions to the development of this guide by attending the majority of the task group meetings, reviewing/commenting on various drafts, and writing portions of the document.

Mark Metzger, Utility Chairman	Nebraska Public Power District
Barry Blair	Centerior Energy
Eugene Matranga	Centerior Energy
David Eder	ComEd
Joe Wooten	ComEd
Chad Main	Consumers Energy
Mark Smith	Duke Energy Corp.
Tom Bilger	Duke Engineering & Services
Chaz McFeaters	Duquesne Light Co.
Deborah Cioffi	Encotech Engineering
Duke Dow	Energy Management Services
Lawrence Bergeron	Entergy Operations, Inc.
Bill Reuland	Mollerus Engineering Corp.
Fred Mollerus	Mollerus Engineering Corp.
John Lyons	New York Power Authority
Robin Frederickson	Northern States Power Co.
V. Ray Foster	Pacific Gas & Electric
Amy ElChaar	PECO Nuclear Company
Jim Klucar	PECO Nuclear Company
Patricia A. Hinchey	PECO Nuclear Company
John Walden	Rochester Gas and Electric Co.
Ray Morris	Southern Nuclear Operating Co.
Richard Mooney	Tennessee Valley Authority
William Eberly	Tennessee Valley Authority
Al Saunders	TU Electric

CONTENTS

LIST OF FIGURES

1

INTRODUCTION

Volume II of the *Thermal Performance Engineer's Handbook* provides guidance for the application of the elements of a thermal performance program that are described in Volume I. Monitoring and trending of key parameters are aided by a knowledge of the components monitored. Volume II contains a component focus section with the following format:

- General description
- Performance calculations
- Impact on thermal performance
- Typical problems, degradation mechanisms and corrective actions
- References

The scope of this handbook is limited to the detail necessary to understand the basic approach to monitoring components with a significant effect on thermal performance. An extensive list of references is included where specific details of measurements and their function are required.

Cycle steam losses account for the most lost megawatts. Therefore, Volume II contains a section on cycle isolation and covers some specific ways to improve heat rate.

Specific measurements are included that are prone to significant uncertainties. Finally, Volume II presents some fundamentals on data validation and the effects on performance measurement caused by measurement uncertainty.

2

COMPONENT FOCUS AREAS

2.1 Condenser and Auxiliaries

2.1.1 General Description

The purpose of the condenser is to condense large amounts of turbine exhaust steam at low (subatmospheric) pressure using cooling water, usually referred to as circulating water. The condenser is located below the LP turbines and usually fits within an envelope set by the turbine foundation.

Figure 2-1 shows a schematic of a condenser. In this case, a two tube pass arrangement with circulating water inlet and outlet on the same side of the condenser shell. Condensers come in several configurations and can be classified in several ways: (1) orientation of the condenser tubes, transverse or parallel to the axis of the turbine; (2) the number of condenser shells, usually one condenser shell for each set of turbine two-flow exhausts; (3) the number of tube passes, one or two; and (4) whether the circulating water flows in parallel through each condenser shell or whether it flows in series through each shell. If the latter, each condenser shell has a correspondingly higher inlet and outlet circulating water temperature and condensing pressure.

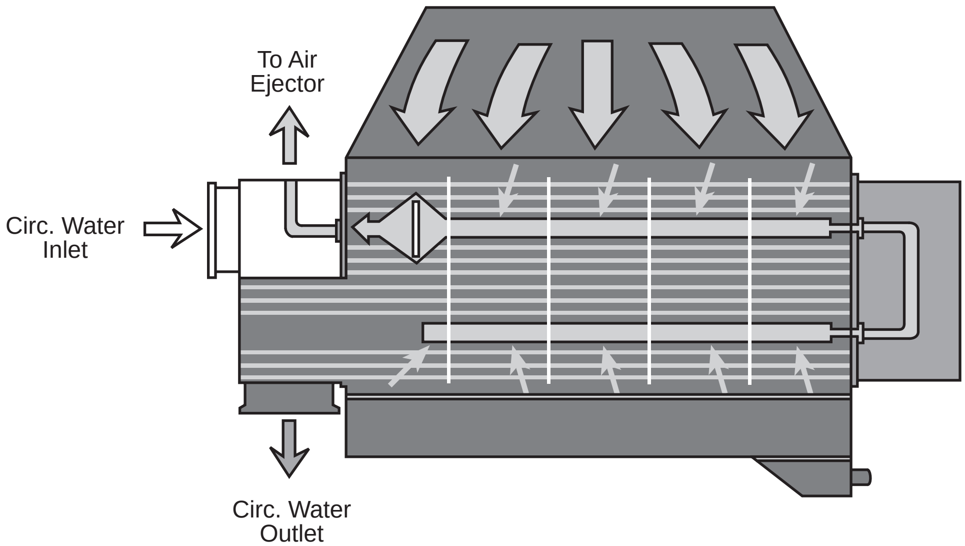


Figure 2-1
Basic Arrangement of a Condenser (Schematic)

Because condensers handle large quantities of steam at low, subatmospheric pressure, the volumetric flow is high. As a result, the condenser tube arrangement must be opened rather than compact in order to allow steam flow into the inner region of the tube bundle. A typical tube arrangement is shown in Figure 2-2. The tube pattern and shell volume are designed to minimize steam-side pressure drop. A cruciform region in the middle of the bundle is void of tubes and forms a passageway to the circulating water end of the condenser to vent non-condensable gases that can accumulate during condensation of the steam.

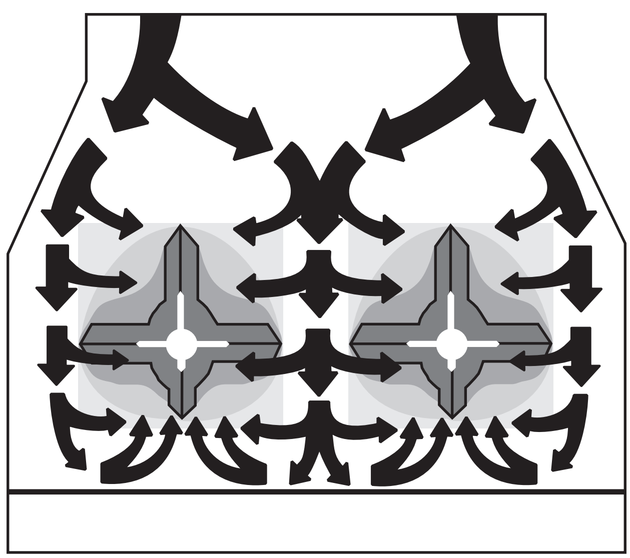


Figure 2-2
Condenser Tube Arrangement and Installation

Condensers must continually vent non-condensable gases to prevent becoming air-bound and losing heat transfer capability. Sources of non-condensable gases are air inleakage and ammonia from oxygen scavenging chemical additions in PWRs and, in the case of BWRs, oxygen and hydrogen that is generated in the reactor vessel and mixes with main steam.

Figure 2-3 shows how non-condensable gas flows within and out of a condenser. The key to removing non-condensable gas is the tendency of the non-condensable gas to flow to the coldest or circulating water inlet region of the condenser by virtue of the partial pressure of the condensing steam being lowest in the cold region. Steam jet air ejectors (or vacuum pumps) are used to draw a vacuum and pull non-condensable gas (with some steam) from the condenser.

Table 2-1 lists condenser instrumentation. A key condenser operating parameter is condenser pressure, which is usually measured in the condenser neck (just above the condenser tubes and below the turbine exhaust) using pressure taps enclosed in baskets. These baskets prevent high steam velocities from affecting measured pressure.

Other parameters shown in Table 2-1 are used to calculate log mean temperature difference (LMTD) and condenser heat load. Most plants do not directly measure circulating water flow using devices such as orifices or venturis. Rather, it can be determined by calculating circulating water flow using pump performance curves with measured pump discharge pressure while accounting for suction elevation and the number of operating pumps.

Table 2-1
Instrumentation for Condensers and Auxiliaries

Parameter	Purpose
Condenser pressure	Performance
Circulating water inlet temperature	Calculated LMTD, heat load
Circulating water outlet temperature	Calculated LMTD, heat load
Circulating water flow	Calculate heat load
Circulating water pump discharge pressure	Flow indicator
Circulating water box delta p	Trending fouling
Circulating water box levels	Flow

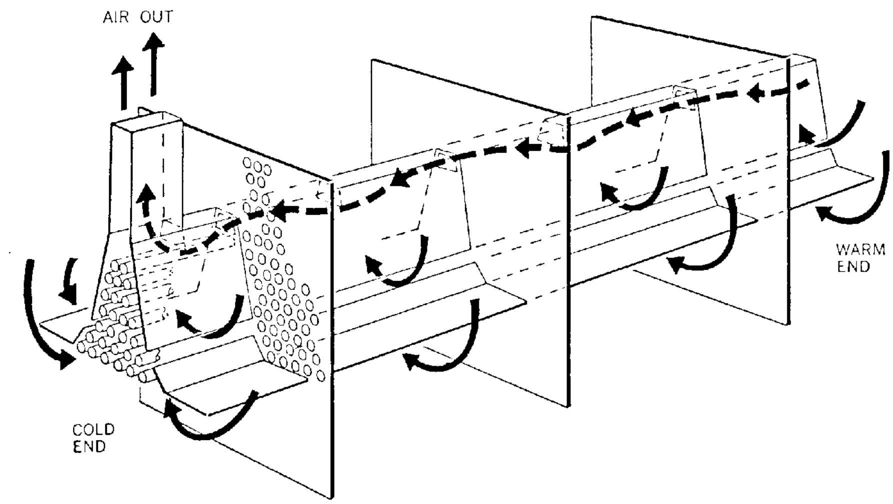


Figure 2-3
Removal of Non-condensable Gas

2.1.2 Performance Calculations

The following calculation methodology is provided as a means to measure the overall performance of condensers. The methodology is applicable to hand calculations or PC spreadsheets.

Condenser Duty (Q)

The condensing duty is the total heat transferred to the condenser cooling water (circulating water). The condenser duty can be calculated by subtracting steam cycle side non-condenser heat flows from the energy delivered to the steam cycle, Q_T (See Volume I, Section 2.3). Typically, the non-condenser heat flows are generator electrical output, generator losses, auxiliary steam, and steam driven feedwater pumps. PWR steam generator blowdown should also be considered. The accuracy of this calculation depends very much on the accuracy of the instrumentation used to measure the parameters involved.

$$Q = Q_T - Q_{elect} - Q_{gen} - Q_{aux} - Q_{fwp} - Q_{bd}$$

Where

Q_T	=	heat load to steam cycle
Q_{elect}	=	generator gross output
Q_{gen}	=	generator losses
Q_{aux}	=	auxiliary steam loads
Q_{fwp}	=	steam turbine driven pump load
Q_{bd}	=	steam generator blowdown load

All heat loads need to be converted to a common set of dimensions, such as Btu / hr.

Condensing duty can be calculated based on circulating water flow and temperature flow:

$$Q = WC_p(t_o - t_i)$$

Where

Q	=	condensing duty, Btu / hr
W	=	circulating water flow rate, lbm / hr
C_p	=	circulation water specific heat = 1.0 Btu / lbm- °F
t_o	=	circulating water outlet temperature, °F
t_i	=	circulating water inlet temperature, °F

An example calculation using circulating water flow and temperature change is shown below:

Input Data	W = 600,000 gpm = 3×10^8 lbm / hr t_i = 65°F t_o = 85°F
Calculation	Q = $3 \times 10^8 \times (85-65)$ = 6×10^9 Btu / hr

Condensing duty can also be calculated using turbine exhaust flow and enthalpy into the condenser

$$Q = W_{ex}(h_{ex} - h_c)$$

Where

W_{ex}	=	turbine exhaust steam flow, lb / hr
h_{ex}	=	turbine exhaust steam enthalpy, Btu / lb

h_c = condensate enthalpy (enthalpy of water at the condensing pressure), Btu / lb

Turbine exhaust flow and enthalpy are not measured. Instead, this method requires that these parameters be estimated using turbine electrical output and the turbine heat balance or thermal kit. This method is inherently inaccurate.

An example calculation using turbine exhaust flow and enthalpy is shown below:

Input Data	W_{ex} = 6.6 x 10^6 lb / hr h_{ex} = 975 Btu / lb h_c = 80 Btu / lb @ P_c = 2 inch Hga
Calculation	Q = 6.6 x 10^6 x (975-80) = 5.9 x 10^9 Btu / hr

Cleanliness Factor (CF)

Cleanliness factor is the ratio of actual or operating overall heat transfer coefficient to clean overall heat transfer coefficient.

$$CF = \frac{U_o}{U_c}$$

Where

U_o = the operating or as-found heat transfer coefficient, Btu / hr-°F-ft^2

U_c = the clean overall heat transfer coefficient, Btu / hr-°F-ft^2

Operating overall heat transfer coefficient can be calculated from the equation

$$U_o = \frac{Q_o}{A_c \Delta T_m}$$

Where

A_c = the surface area of the condenser, ft^2

$$\Delta T_m = \frac{(T_s - t_i) - (T_s - t_o)}{\ln\left(\frac{T_s - t_i}{T_s - t_o}\right)}$$

The condensing temperature, T_s, is the saturation temperature at condenser pressure, P_s, and can be obtained from steam tables. The operating temperatures used to calculate ΔT_m are actual measured data, while Q is obtained using the previous calculation. The condenser heat transfer area is not easily calculated unless the total number of unplugged tubes and their lengths are known. The clean overall heat transfer coefficient, U_c, can usually be obtained from the manufacturer's data specifications for the condenser. The calculation shown below assumes the values are known. The clean overall heat transfer coefficient (U_c) will increase with circulating water temperature. The temperature affects the fluid properties, which affect the temperature profile in the boundary layer at the tube wall. The clean heat transfer coefficient can increase about 1% per °F increase in circulating water temperature.

Input Data	U_c = 500 Btu / hr-°F-ft^2 P_s = 2 inch Hg T_s = 101.1°F t_i = 65°F t_o = 85°F N_t = 70,000 tubes L_t = 45 ft d_{ot} = 0.75 inches
Calculation	$A_c = \dfrac{\pi d_{ot} L_t N_t}{12}$ $= \dfrac{\pi x 0.75 x 45 x 70{,}000}{12} = 618{,}500 ft^2$ $\Delta T_m = \dfrac{(101.1-65)-(101.1-85)}{\ln\left(\dfrac{101.1-65}{101.1-85}\right)} = 24.8°F$ $U_o = 6x10^9 / 618{,}500 / 24.8°F$ $= 391.7 / Btu / hr - °F - ft^2$ $C_f = 347 / 500$ $= 0.783$

2.1.3 Impact on Thermal Performance

Condenser performance has a large impact on plant thermal performance. As condenser pressure increases, there is less available energy and more heat rejection, resulting in increased heat rate and less electric power production. Figure 2-4 shows a typical change in heat rate due to change in condenser pressure. This curve is often provided in the manufacturer's thermal kit. The reason that the curve is relatively flat below the design backpressure shown, 1.5 inches Hga, is that volumetric steam flow increases as condenser pressure decreases. The last stage of the LP turbine experiences increased windage or mechanical losses as volumetric flow increases. Plant cycle-specific calculations using a heat balance program are needed if more exact, plant-specific values are desired. Expected backpressure, based on inlet circulating water temperature, can be compared to the measured backpressure to separate poor condenser performance from the effect of change in ambient conditions. An electrical megawatt deviation can then be assigned to each for accurate accounting.

Figure 2-4 also shows that last stage bucket (LSB) loading has a significant effect on change in heat rate due to changes in condenser pressure. If LSB loading, defined as design generator output in kW / sq.ft of last stage bucket annulus, is high, then last stage bucket losses are high. High last stage bucket losses decrease with increasing turbine backpressure and offset the loss in thermodynamic efficiency that occurs with increasing turbine backpressure. Most nuclear turbines have fairly high LSB loadings since turbine configurations are limited to six or fewer LP turbine ends and it has not been economical to use more that three LP turbine casings at nuclear power plants.

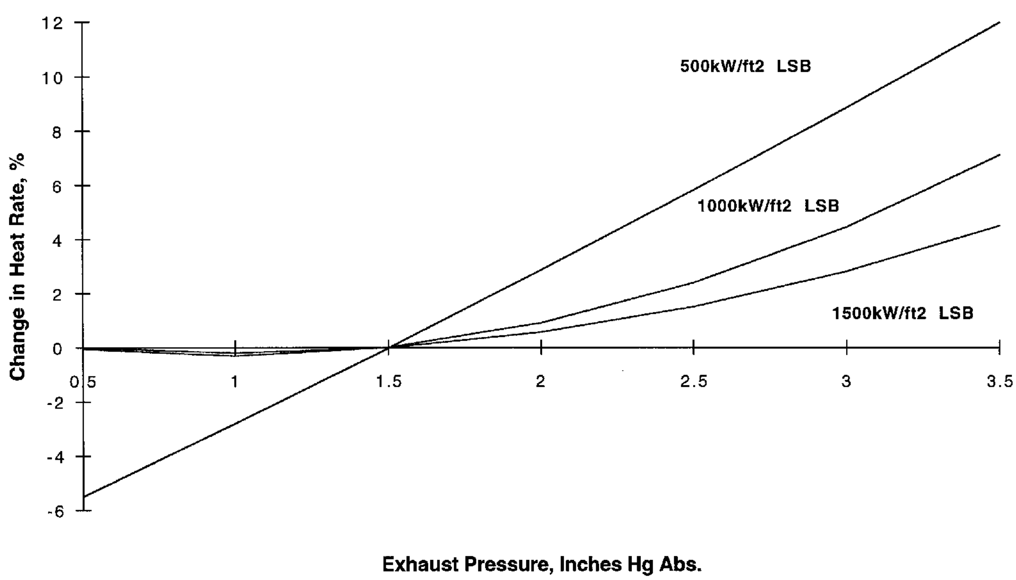


Figure 2-4
Typical Condenser Performance

2.1.4 Typical Problems, Degradation Mechanisms, and Corrective Actions

There are several problems associated with condensers that can cause a heat rate increase. Most also result in degraded heat transfer and increased condensing pressure.

Microfouling

Microfouling is the formation of deposits on the inside of the condenser tubes and can be of chemical or biological, or both.

Chemical fouling is the formation of a chemical deposit with poor heat transfer properties on the inside of the condenser tubes. The most common such deposit is calcium carbonate. The solubility limit for this compound decreases with increasing water temperature causing deposition as inlet circulating water, with calcium carbonate at its solubility limit, heats up in the condenser. One method of removal is by mechanically cleaning the tubes. See EPRI ***Design and Operating Guidelines for Nuclear Power Plant Condensers,*** NP-7382, for more details on condenser cleaning.

Biological fouling is the formation of a biological layer or slime on the inside of the tubes that impedes heat transfer and, if of sufficient quantity, can impede circulating water flow. Mitigation is usually by mechanical cleaning; on-line cleaning systems are available. Prevention is usually by treating the water with chemicals.

Macrofouling

Macrofouling results in the blockage of tubes, usually at the tubesheet, with debris such as seaweed or organisms such as Asiatic clams or zebra mussels. Severe macrofouling in the condenser may be noticeable by increased condenser pressure and / or reduced circulating water flow. An increase in ΔP across the waterbox is also an indication of fouling. (Circulating water flow is not usually measured. However, there may be an increase in circulating water pump discharge pressure indicating a flow reduction.) Macrofouling is almost always removed by mechanical means.

Prevention of macrofouling depends on site-specific conditions causing macrofouling. If the source is debris such as seaweed or fresh water vegetation, then traveling screens at the circulating water intake to prevent the debris from flowing into the condenser become an important means of prevention. If the source is an organism such as clams, then a biocide is needed. Many plants have the capabiltiy to backwash the condenser to remove macrofouling. Strategies for backwashing depend on circulating water conditions and trends in condenser performance.

Non-condensable Gases

Condensers are designed to remove air and additional oxygen and hydrogen in BWR plants using vacuum equipment such as pumps or steam jet air ejectors (SJAEs). Non-condensable gases retard efficient heat transfer. Accumulation of non-condensable gas, referred to as “air-binding,” results in degraded heat transfer, high condenser pressure, and increased heat rate. The result is similar to microfouling. In fact, it can be difficult to distinguish between the two causes of high condenser pressure. If a rather sudden increase in condenser pressure occurs, the likely cause can be either an increase in non-condensable gas inleakage or a malfunction of the non-condensable gas removal system. Inspections of equipment such as subatmospheric valve packing and connections to the condenser and the vacuum equipment, as well as appropriate corrective action, may be needed to reduce inleakage. If condenser pressure increases gradually, then microfouling or macrofouling is a likely cause.

Although it may be difficult to differentiate air binding from fouling, there are several methods that may be used to determine if the condenser is air-bound:

- **System Operational Changes** Baseline data can be taken with normal vacuum pumps in operation. Then an additional vacuum or SJAE can be placed in service while continuing to monitor condenser pressure, load, and circulating water temperature. If the additional vacuum pump results in a decrease in condenser pressure while the other factors remain constant, then the condenser is likely air-bound.

- **Load Versus Pressure** Condenser pressure, circulating water inlet temperature, and load can be monitored over a period of time when load reductions are planned or can be scheduled. Condenser pressure should drop with decreasing load. It drops less rapidly if the condenser is air-bound. A curve demonstrating how an air-bound condenser behaves at low load is shown in Figure 2-5.

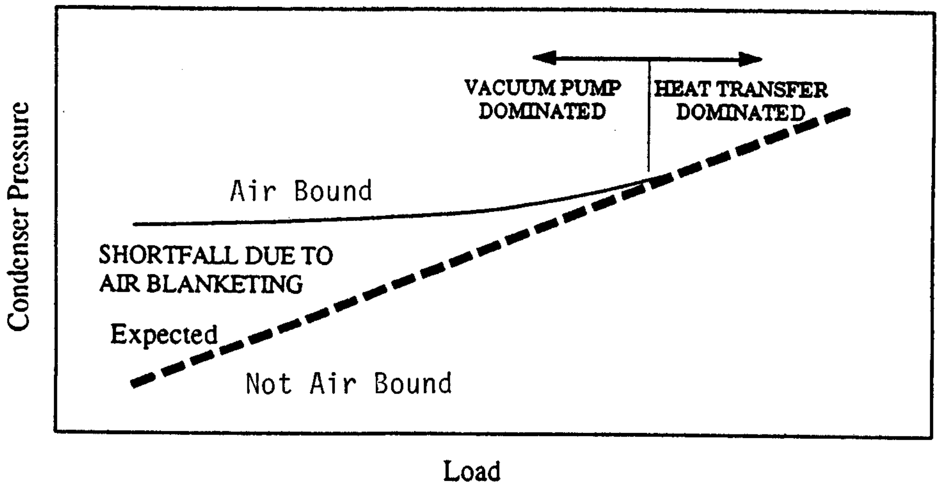


Figure 2-5
Typical Curve of Air-bound Condenser

- **Air In-Leakage Rate Versus Pressure** While holding constant load with air-removal equipment in service, a measured rate of air, increased in steps, can be added to the condenser. If the initial step of introducing air does not cause condenser pressure to rise, then the condenser is not air-bound. Figure 2-6 shows typical condenser pressure responses to controlled air-inleakage when the condenser is air-bound and when it is not. Caution should be taken when using this method since recovery can be difficult and there is the potential to lose control of inleakage.

- **Outlet Temperature Stratification** A grid of temperature detectors can be installed at the outlet of the condenser to identify regions of low heat transfer. Because air-binding prevents the entrance of steam into the regions of the tube bundle where air-binding is occurring, the circulating water temperature rise in the air bound regions is reduced. This method can be costly to install and maintain.

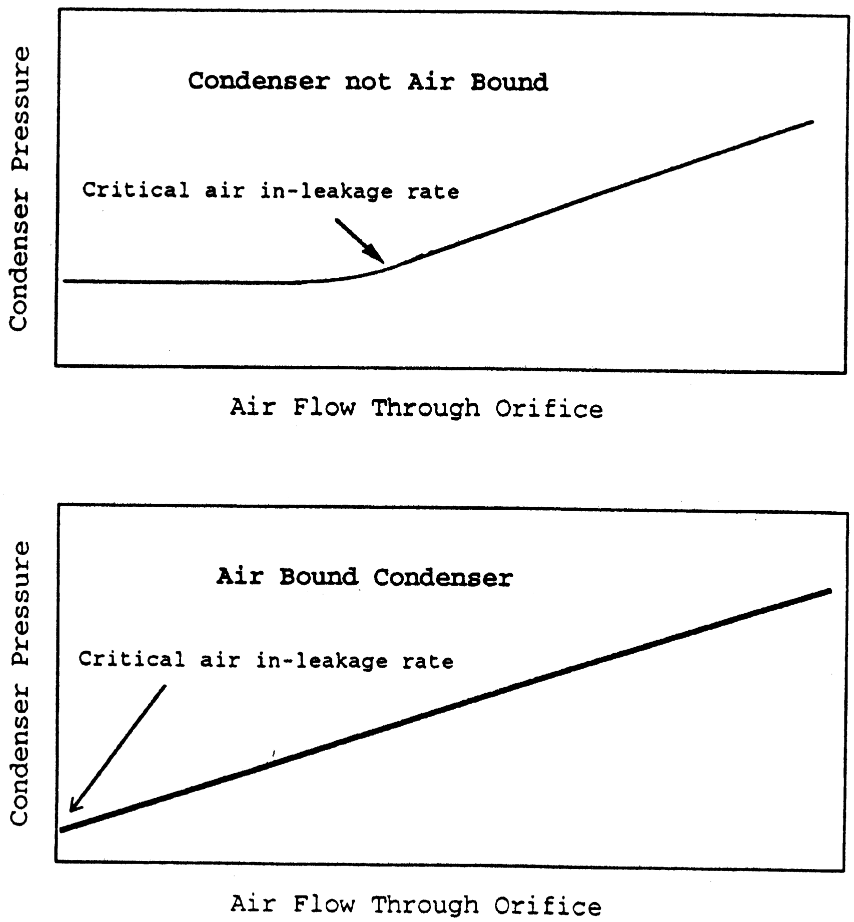


Figure 2-6
Condenser Pressure Response to Air-inleakage Test

Condensate Depression

Condensate accumulates in the hotwell below the condenser tubes. Normally, the water level in the hotwell is maintained below the condenser tubes. However, either by intention or by malfunction of level control, the hotwell level can rise and flood the lower tubes. This causes the condensate to subcool below saturation temperature corresponding to condenser pressure. The subcooling is referred to as both *condensate depression* and *condensate subcooling*. It can cause increased levels of dissolved oxygen and corrosion of the bottom condenser tubes. It results in increased heat rate because the condensate requires additional heating. As a rule of thumb, each 5°F of condensate depression results in a 0.05% increase in heat rate.

Some plants prefer to operate with condensate depression because subcooling reduces cavitation in the condensate pumps. The need for this practice is questionable because many condensate pumps are located well below the hotwell to minimize cavitation or are designed to operate in a cavitating mode.

2.1.5 References

The following is a compilation of documents on the subject of “Condenser and Auxiliaries.” They are not necessarily referenced by the preceding text.

P^{2}EP Technical Library

The Plant Performance Enhancement Program (P^{2}EP) has a technical library that provides an effective way to share information on plant thermal performance. Listed below are the documents from the P^{2}EP Technical Library that relate to the subjects “Condenser, Air Ejectors, Air Inleakage, Chlorination, Circulating Water.“ These documents are available to eligible EPRI members by calling the P^{2}EP Administrator at 704/547-6024, or via the World Wide Web at http://www.epriweb.com/npg/pse/ptp/index.html.

File #	*Document Title*
0001.0-489	WPPSS Plant Efficiency and Reliability Monitoring Programs: Steam Jet Air Ejector (SJAE) Performance Monitoring. Procedure No. 8.4.45. Rev 0. April 1988.
0002.0-101	Development And Implementation Of A Feedwater Heater Testing Program
0002.0-447	Air Binding and Condenser Optimization
0002.0-58	Condenser Leak-Detection Guidelines Using Sulfur Hexafluoride as a Tracer Gas
0002.0-91	An Air Ingress Monitor for Turbine Condensers - Its Development and Validation
0010.0-883	Surface Condensers
0012.5-80	Current Cathodic Protection Practice in Steam Surface Condensers
0016.0-440	On-Line Circulating Water Flow Measurement
0016.0-491	WPPSS Plant Efficiency and Reliability Programs: Circulating Water System Performance Monitoring, Procedure No. 8.4.24. Rev. 1, 1988
0024.0-226	Niagara Mohawk Power Corp. Nine Mile Point 2. Plant Performance Monitoring Instruction. Steam Surface Condenser Performance, N2-PPMI-3.1, Rev. 00
0024.0-227	Nine Mile Point 2 Condenser Performance Curves
0024.0-254	Nuclear Plant Component Basic Series-Condensers Workbook
0024.0-292	Pilgrim Nuclear Power Station Instruction PG.1090: Condenser Performance Monitoring, Rev. 0, 1988
0024.0-461	Waterford III Standing Test Procedure: Condenser Performance, Proc. No. N8-008
0024.0-482	WPPSS Plant Efficiency and Reliability Monitoring Programs: Condenser Performance Monitoring. Procedure No. 8.4.40. Rev 0. April 1988.

0024.0-52	Comprehensive Condenser Performance Testing at Indian Point 3 Nuclear Power Plant
0024.0-55	Con Edison Technical Analysis Procedures: Performance Testing of Condensers, TAP 7-4-1. 1986
0024.0-60	Condenser Performance
0024.0-622	An Improved Method of Estimating the Performance of Condensers with Both Single and Multiple Compartments
0024.0-628	Investigation of Condenser Deficiencies Utilizing State of the Art Test Instrumentation and Modeling Techniques
0024.0-66	Condenser Test (Watts Bar Nuclear Plant)
0024.0-854	Cost Effective Corrosion Control in Electric Power Plants
0024.0-894	Performance Monitoring of Three Compartment Condensers in Nuclear Plants
0034.6-519	Oyster Creek Thermal Performance and Availability Report
0053.0-619	Improved Condenser Performance can Recover up to 25 MW Capacity in a Nuclear Plant
0053.0-752	Biofouling Control in Power Station Circuits: Overview of Electricite de France's Experience
0054.0-645	On-Line Condenser Fouling Monitor Development
0060.0-263	Nuclear-Plant Performance, Reliability, Heat Rate Improvement & Life Extension Basic Series-Heat Balances Workbook
0060.5-275	Performance Monitoring and Replacement of Heat Exchanger Components and Materials
0060.5-309	Practical Aspects and Performance of Heat Exchanger Components and Materials
0067.0-757	INPO Significant Event Notification: Salt Water Intrusion Caused by Main Condenser Tube Rupture
0073.0-460	Water Leakage to the Condenser
0073.0-719	Demonstrations of Condenser On-Line Leak Detection Using SF6 Tracer Gas
0088.0-867	Number of High Performance Stainless Steel Condenser Installations
0088.0-869	Application Experience with UNS N08366, UNS N08367 and UNS S44735 Alloy Condenser Tubing
0088.0-870	Experience with High Performance UNS S44660 Ferritic Stainless Steel Tubing in Power Plant Condensers
0088.0-871	Survey of High Performance Stainless Condenser Tubing Experience in Electric Power Plants
0088.0-872	Application of UNS S31254 (Avesta Sheffield 254 SMO) Austenitic Stainless Steel in Power Plants
0099.0-529	P2EP Survey 92-006, Condenser Performance
0099.0-531	P2EP Survey 92-008, Condenser Hot Well Condensate
0099.0-578	P2EP Survey 94-018, Condensers, Cooling Towers, and Circulating Water Data
0099.0-589	P2EP Survey 95-010, Main Condenser Tube Cleaning
0099.0-592	P2EP Survey 95-013, Limits on Turbine Cycle on Air In-Leakage
0099.0-597	P2EP Survey 95-018, Variances in Condenser Pressure
0100.0-521	Nine Mile Point 2 Lost Electrical Generation Summary

0101.0-205	Condenser Performance Monitoring
0101.0-280	Finding Lost MegaWatts
0101.0-301	Plant Thermal Performance and Availability Monitoring Report (GPU Nuclear - Oyster Creek)
0101.0-421	The Performance Software Users Group Meeting Proceedings
0101.0-778	Clinton Power Station Condenser Performance Evaluation and Improvement Program
0105.0-466	Waterford Unit 3 Standing Test Procedure: Condenser Basket Tip Pressure Measurements, Procedure N8-009, Rev. 0
0106.0-735	Point Beach Thermal Performance Procedure TP3.0, Rev. 0, Condenser Fouling - Data Collection and Analysis
0106.0-786	Callaway Plant Engineering Test Procedure: ETP-AD-01100, Condenser Performance Monitoring
0124.0-388	Standards for Steam Jet Vacuum Systems (Fourth Edition)
0142.3-111	Effect of Tube Material on Steam Condensation
0142.3-162	Selecting Condenser Replacement Tubes
0142.3-407	Surface Condenser Tube Sample Heat Transfer Testing
0142.3-827	FitzPatrick 1995 Condenser retube and a Recent Operational Finding
0145.0-19	Aero-Thermodynamics of Low Pressure Steam Turbines and Condensers
0145.0-199	Latest Advances in Steam Turbine Design, Blading, Repairs, Condition Assessment, and Condenser Interaction

Formal EPRI Reports/Software

EPRI has created formal reports and / or software on the subjects of “Condenser, Chlorination, or Circulating Water.“ These are available to eligible EPRI members by calling:

- EPRI Document Distribution Center 925 / 934-4212
- EPRI Software Distribution Center 800 / 763-3772

Report #	***Report Title***
AP-3651	Thermo-Economic Analysis of Power Plants
CS-1554	Biofilm Development and Destruction
CS-2251	Recommended Guidelines for the Admission of High-Energy Fluids to Steam Surface Condensers
CS-2276	Design and Operating Guidelines Manual for Cooling-Water Treatment
CS-2961	Current Cathodic Protection in Steam Surface Condensers
CS-3200	High-Reliability Condenser Design Study
CS-3527	Augmented Heat Transfer Rates in Utility Condensers
CS-3550	Condenser Macrofouling Control Technologies
CS-3844	Condenser Procurement Guidelines

CS-4014	Proceedings: 1983 Fossil Plant Heat Rate Improvement Workshop
CS-4076	Laboratory Studies Supporting Cooling-Water Treatment Tests at a Power Plant with Calcium-limited Water
CS-4279	Condenser Targeted Chlorination Design
CS-4329-SR	Seminar Proceedings: Prevention of Condenser Failures--The State of the Art
CS-4339	Proceedings: Condenser Biofouling Control--State of-the Art Symposium
CS-4562	Effects of Sulfide, Sand, and Cathodic Protection on Corrosion of Condensers
CS-4736	Proceedings: 1985 Heat-Rate Improvement Workshop
CS-5032	Performance of Mechanical Systems for Condenser Cleaning
CS-5180	Commercial Assessment of Condenser-Targeted Chlorination
CS-5235	Recommended Practices for Operating and Maintaining Steam Surface Condensers
CS-5271	Guidelines on Macrofouling Control Technology
CS-5486	Non-radioactive Tracer for Steam Turbine Thermal Performance Tests
CS-5589	Effects of Selected Water Treatments and Cathodic Protection on Corrosion and Embrittlement of Condenser Tubes
CS-5601	Engineering Assessment of Condenser Deaeration Retrofits for Cycling Fossil Plants
CS-5662	Condenser Performance Test and Replacement Tubing Material Evaluation
CS-5729	Condenser Performance Test and Back-Pressure Improvement
CS-5942-SR	Proceedings: Condenser Technology Symposium (1988)
CS-6024	Condenser Condition Assessment and Retrofit Analysis: Ravenswood Unit 10
CS-6077	Effects of Targeted Chlorination on AL-6X Condenser Tube Corrosion in Seawater
CS-6080	Proceedings: 1987 Conference on Expert-System Applications In Power Plants
GS/EL-5648	MARK I Performance Monitoring Products
GS-6253	The TVA Moving-Manifold Targeted-Chlorination System: Second Field Demonstration,
GS-6566	Non-toxic Foul-Release Coatings
GS-6576	Targeted Chlorination Schedules
GS-6635	1988 EPRI Heat-Rate Improvement Conference
GS-7003	Improvement of the Exhaust Flow of a Low-Pressure Turbine
GS-7009	Retrofits for Improved Heat Rate and Availability: Circulating Water Heat Recovery Retrofits
GS-7181	Condenser Targeted Chlorination Demonstration at Brayton Point Station-Unit 2
GS-7349	Proceedings: Condenser Technology Conference
GS-7350	Condenser On-Line Leak Detection System Development
NP-1467	Assessment of Condenser Leakage Problems
NP-2294	Guide to the Design of Secondary Systems and their Components to Minimize Oxygen-Induced Corrosion
NP-2597	Condenser Inleakage Monitoring System Development
NP-3395	Calculation of Leak Rates Through Cracks in Pipes and Tubes
NP-4564	High-Sensitivity Dissolved-Gas Monitoring System with Applications for PWR Secondary-Side Chemistry

NP-7382	Design and Operating Guidelines for Nuclear Power Plant Condensers
TR-101096	Design Guidelines for Targeted Chlorination with Fixed Nozzles
TR-101772 R1	Electromagnetic NDE Guide for Balance-of-Plant Heat Exchangers, Rev. 1
TR-101846	Second EPRI BOP Heat Exchanger NDE Workshop
TR-101942	Condensate Polishing Guidelines for PWR and BWR Plants
TR-102922	High-Reliability Condenser Application Study
TR-103474	Review of Flow Problems at Water Intake Pump Sumps
TR-103475	Proceedings: Condenser Technology Conference, October 1993
TR-106741	Heat Exchangers: An Overview of Maintenance and Operations (NMAC Tech Notes)

Non-EPRI References

Organizations and publishers other than EPRI have material pertaining to this subject. Some of this other reference information is listed below to enable the reader to obtain this material if desired.

American Society of Mechanical Engineers. "Code on Steam Condensing Apparatus." ASME Performance Test Code 12.2-1983. August 31, 1983.

American Society of Mechanical Engineers. "Part 11 - Water and Steam in the Power Cycle (Purity and Quality, Leak Detection and Measurement), Instruments and Apparatus." ASME Performance Test Code 19.11-1970. August 8, 1972.

Heat Exchange Institute, Incorporated, "Standards for Steam Surface Condensers, Eighth Edition." Cleveland, Ohio: January, 1984.

Heat Exchange Institute, Incorporated, "ADDENDUM 1, Standards for Steam Surface Condensers, Eighth Edition." Cleveland, Ohio: January, 1989.

Heat Exchange Institute, Incorporated, "Standards for Direct Contact Barometric and Low Level Condensers, Fifth Edition." New York, New York: 1970.

TPE Handbook References

The following documents contain additional information on condensers:

Prevention of Condenser Failures- The State of the Art. Electric Power Research Institute, Palo Alto, California: March 1982. RD-2282-SR (Special Report).

John V. Walden, "Condenser Performance Monitoring and Cleaning," *Thirteenth Annual Plant Performance Enhancement Program (P^2EP) Annual Meeting Proceedings*, San Antonio, Texas (August 1997).

Corrosion Related Failures in Power Plant Condensers. Electric Power Research Institute, Palo Alto, California: August 1980. Report NP-1468.

B. K. Long, "Air Binding and Condenser Optimization." Alabama Power Company

Design and Operating Guidelines for Nuclear Power Plant Condensers. Electric Power Research Institute, Palo Alto, California: September 1991. Report NP-7382.

2.2 Cooling Towers

2.2.1 General Description

Cooling towers are used to cool circulating water, which provides the condenser with water at a low enough temperature to maintain the design vacuum. Cooling tower designs are natural draft or forced draft, depending on climatic conditions and economics. Natural draft cooling towers may be designed for crossflow or counterflow of the air with the water in the cooling region. Counterflow towers are usually more efficient than crossflow towers and are not as susceptible to ice damage, although ice damage can still occur. Natural draft towers use the buoyancy of the air being warmed and wetted to move the air through the tower.

Cooling towers are not always necessary where circulating water can be taken directly from a river or body of water. However, cooling towers are sometimes used only to satisfy ecological requirements or to isolate the condenser from some types of biofouling. Forced draft cooling towers that use fans to move the air are used typically in dry climates because fan power costs are low and the smaller towers have a lower capital cost. Also, natural draft towers, which are the choice in most climates, will sometimes stratify in low humidity.

The cooling tower cools water by evaporation caused by the air moving through the tower. The coolest theoretical temperature that can be attained is the adiabatic saturation temperature. This property of general gas mixtures can be measured for air-water vapor mixtures by a wet-bulb thermometer because the wet-bulb temperature is within one degree of the adiabatic saturation temperature for air-water vapor mixtures. It may also be calculated from the dew point and dry-bulb temperatures. Because wet-bulb temperatures are high in the summer, cooling tower design performance is not always attainable.

The cooling region of the tower consists of the distribution system and is filled with packing (or fill) that increases the water surface for evaporation and helps in attaining the optimum water to airflow ratio. Cooling tower packing can be a splash or a film type. The splash type fill packing disperses the water as droplets through horizontal or vertical air flow. The film type is more efficient than the splash type because it forms sheets of water which increases the evaporative surface. However, this increased surface is more apt to become biofouled. The film type fill may be sheet or cellular in design.

Table 2-2
Instrumentation for Cooling Towers

Parameter	Purpose
Circulating water flow	Duty and capability
Condenser hot well temperature	Range and approach
Circulating water return temperature	Range and approach
Make up flow	Capability
Ambient wet-bulb temperature	Approach
Ambient dry-bulb temperature	Capability (redundant)
Ambient dew point	Adiabatic saturation temperature

2.2.2 Performance Calculations

Cooling tower performance is defined by the following parameters illustrated in Figure 2-7.

- **Range, °F:** The temperature change in the water effected by the cooling tower. There is a design range and an actual range measured during operation.
- **Approach, °F:** The temperature difference between the wet-bulb temperature, T_{WB}, and the circulating water temperature, T_{CW}, typically 5°F or greater.
- **Capability, %:** The ratio of the heat removed from the water to the design amount.

Range:

Input	T_{HW} = 109.6°F T_{CW} = 89.6°F
Calculation	Range = 109.6°F - 89.6°F = 20°F

Approach:

Input	T_{WB} = 70°F
Calculation	Approach = 89.6°F -70°F= 19.7°F

Capability:

Input	T_{HW} = 109.6°F T_{CW} = 91°F T_{WB} = 70°F $T_{HW\ (design)}$ = 109.6°F $T_{CW\ (design)}$ = 89.6°F $T_{WB\ (design)}$ = 70°F
Calculation	Capability (%) = $\frac{109.6 - 91}{109.6 - 89.6}$ = 93% (assumes no adjustments for flow rate, which may change due to macroscopic condenser fouling)

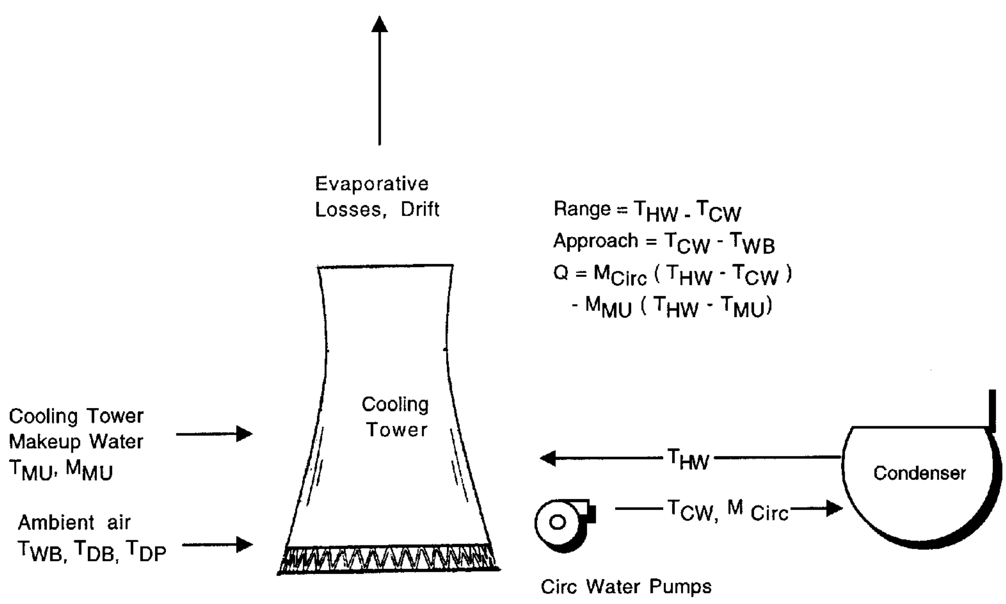


Figure 2-7
Cooling Tower Performance

Some plants do not monitor circulating water flow rate. For purposes of trending cooling tower performance, circulating water flow can be determined from the condenser heat load as follows:

Input	Q_{cp} = Core Power = 6.45 x 10^9 Btu / hr Q_{qe} = Gross Generator Output = 630 x 103 kW T_{HW} = 109.6 °F T_{CW} = 89.6 °F C_p = 1 Btu / lbm °F
Calculation	$Q_{cond} = Q_{cp} - Q_{qe}$ $= (6.45x10^9) - (630x10^3 x3414)$ $= 4.30x10^9 \frac{Btu}{hr}$ $M_{flow} = \frac{4.30x10^9 \frac{Btu}{hr}}{1\frac{Btu}{hr°F} x20°F}$ $= 215x10^6 \frac{lbm}{hr}$

Cooling tower performance can be measured using the temperature change of the water multiplied by the circulating water flow rate. Cooling tower design performance is typically based on a 5°F approach for the expected wet-bulb temperature that is not expected to be exceeded for a given number of days per year. The capability is the percent design capacity that the cooling tower can achieve. Another measure of performance is the actual approach temperature compared to the design. This comparison must account for ambient conditions. As a rule of thumb, a 2° degree reduction in temperature of the cooling water leaving the tower can reduce the turbine backpressure by 1 inch Hg.

Cooling tower models are complex, iterative, multidimensional models requiring a computer to solve. Heat and mass transfer, momentum, and energy must be coupled in a complex flow geometry. A major cause of poor cooling tower performance is maldistribution of the water. Computational fluid dynamic (CFD) modeling has been demonstrated to show the effects of maldistribution of the water on cooling tower performance. Diagnostic testing of cooling towers by mapping air temperature and velocities can aid in identifying distribution deficiencies.

2.2.3 Impact on Thermal Performance

Cooling tower performance impacts circulating water temperature which, in turn, impacts the condenser vacuum. The cooling tower capital and operating costs are a trade-off with cycle efficiency obtained by maintaining condenser pressure at an optimum level for given ambient wet- and dry-bulb temperatures.

An EPRI survey indicates that the mean value estimated for cooling tower performance is 85%. This is equivalent to a 2–2.5°F increase in circulating water temperature. Cooling tower performance is the percent of rated flow that can be cooled to design temperature.

2.2.4 Typical Problems, Degradation Mechanisms, and Corrective Actions

Typical cooling tower problems include the following:

- Water maldistribution
- Fan design
- Fill type and fouling

As mentioned above, water maldistribution is often the cause of a cooling tower not achieving full capability. The causes of maldistribution may be damage to the spray system or a spray system design that does not provide a flow distribution over the fill correct for the airflow pattern. There is a compound effect from a region not receiving proper flow. First, the water goes to another fill region. Because the hot region is overloaded, the water is not cooled to the design temperature. Second, the air flow tends to follow the drier path through the fill.

The solutions to the maldistribution problems are changing (or fixing) the spray nozzles, redesigning the distribution system, and adding to or redistributing the fill. The nozzle size can be varied to compensate for flow maldistribution.

Table 2-3 shows the impact on power output of high wet-bulb temperatures at Watts Bar Nuclear Plant (1160 MW PWR). BP-3 in the table is the backpressure in the third zone of the condenser. The table has two calculations. One is based on maximum MWe output, and the other is based on power reduction required to maintain a BP-3 less than 5.5 in. Hga for the turbine. The effect of degraded cooling tower performance in the latter case is 22.3 MWe for each degree of wet-bulb temperature.

Fouling of the fill degrades performance. Changing the fill may be a solution to fill fouling, and it may also be cost beneficial to upgrade the fill performance from the original design. Chemical treatment can also help control fouling.

One utility reports that their biggest cooling tower problems have been macroscopic condenser fouling caused by loose fill parts produced when freezing damages the fill. These loose fill parts can also clog cooling tower spray nozzles.

Table 2-3
Impact of 5.5 In. Hga BP Limit on MWe (from EPRI survey)

	No BP Limit			BP Limited to 5.5 In. Hga.		
WBT	**CWT**	**BP3**	**MWe**	**CWT**	**BP3**	**MWe**
40	70.3	3.08	1212.3	70.3	3.08	1212.3
50	75.5	3.55	1204.2	75.5	3.55	1204.2
60	80.9	4.14	1193.6	80.9	4.14	1193.6
70	86.8	4.88	1180.2	86.8	4.88	1180.2
75	89.8	5.32	1172.3	89.8	5.32	1172.3
80	92.9	5.81	1163.6	92.9	5.50	1111.6
85	96.1	6.35	1154.1	96.1	5.50	1010.1
90	99.3	6.97	1143.7	99.3	5.50	898.8

Prior to startup, TVA evaluated the expected performance of the Watts Bar cooling towers. It had been some years since the towers were constructed, and a review of cooling tower performance of peer plants indicated a potential problem. As a result of computer modeling and thermal mapping of the exit air, it was estimated that 15% of the air was passing through poorly wetted areas. A subsequent ASME PTC-23[)] test confirmed that the tower would only perform at 88% capacity. This shortfall was calculated to cause a 2 to 3° higher temperature, which would be equivalent to 34,000 lost MWe hours annually.

An extensive project was undertaken to evaluate alternatives and correct this problem. The approach included modifications to the fill and water distribution system. The cost was $1.5M, and the results provided an increase in performance from 88 to 106% capacity for this natural draft tower.

A common problem with forced draft cooling towers is fan design and performance. Proper fan performance can improve cooling, reduce fan power requirements, reduce noise, and extend fan life. Fan performance improvements include optimizing fan speed, pitch, number of blades, and plenum design. These fan performance improvements are not additive. For example, if optimizing fan speed increased flow by 2% and optimizing the number of blades increased flow by 2%, then doing both may not increase flow by 4%.

Identification of fan problems is complex. EPRI Report TR-108483, *Proceedings: Cooling Tower Technology Conference*, contains a discussion of the causes of fan problems and the parameters to adjust for improving performance. A frequent cause of reduced air flow is dirty fan blades. Replacing a fan is expensive. Justification of a fan improvement would probably include the need to replace a fan that was malfunctioning. A performance improvement potential of 20% in airflow exists for some towers.

The type, placement, and condition of fill has a significant potential to affect performance for all types of cooling towers. Depending on the suspended solids and chemistry of the circulating water, the fill can become degraded.

Fill configurations vary for counterflow vs. crossflow and natural vs. forced draft cooling towers. Traditional designs were of a splash type fill that breaks the water into small droplets. Recent fill improvements include film type fill that increases the evaporative surface. Film fill thicknesses on the order of one tenth the thickness of equivalent performing splash fill is efficient, but splash fill is less susceptible to fouling.

The condition of fill can be tracked by trending circulating water approach temperature. Visual inspections from the edge of the tower during operation have also been shown to have benefit. Also, the weight of film fill increases significantly with fouling as it picks up deposits. Typical film fill deposits are shown in Table 2-4. The samples are from six plants with different circulating water conditions.

Table 2-4
Typical Film Fill Deposits - Analysis Characterization (%)

	Samples					
Ingredients	**#1**	**#2**	**#3**	**#4**	**#5**	**#6**
Biological	30	10	80	60	45	20
Mud/silt	45	80	10	10	55	15
Calcium scales	10	5	0	0	0	60
Corrosion products	15	5	10	5	0	5
Oil material	0	0	0	25	0	0
TOTAL	100	100	100	100	100	100

The reference for Table 2-4 recommends the following requirements on circulating water for the application of film type fill.

Table 2-5
Operating Guidelines for High Efficiency Film Fill

Parameter	Circulating Water Limits
Total suspended solids, ppm	<100
Bacteria, CFU/ml	<10,000
Oxidant residual, ppm FAO	0.1 - 0.2
Polymer residual, ppm active	4-6

Table 2-6
Nuclear Plants with Mechanical Draft Cooling Towers

Unit Name	Cooling Water Type	Cooling System Type
Browns Ferry 2, 3	Fresh	Mixed mode, once-thru/cool tower
Catawba 1, 2	Fresh	Closed cycle
Crystal River 3	Saline	Combination, once-thru/helper tower
Duane Arnold 1	Fresh	Closed cycle
Farley 1, 2	Fresh	Closed cycle
Hatch 1, 2	Fresh	Closed cycle
Monticello 1	Fresh	Mixed mode, once-thru/cool tower
Palisades 1	Fresh	Combination, once-thru/helper tower
Palo Verde 1, 2, 3	Sewage effluent	Closed cycle
Peach Bottom 2, 3	Fresh	Combination, once-thru/helper tower
Prairie Island 1, 2	Fresh	Mixed mode, once-thru/cool tower
River Bend	Fresh	Closed cycle
Vermont Yankee 1	Fresh	Mixed mode, once-thru/cool tower
WNP 2	Fresh	Closed cycle

Table 2-7
Nuclear Plants with Natural Draft Cooling Towers

Unit Name	Cooling Water Type	Cooling System Type
Arkansas One 2	Fresh	Closed cycle
Beaver Valley 1, 2	Fresh	Closed cycle
Byron 1, 2	Fresh	Closed cycle
Callaway 1	Fresh	Closed cycle
Davis Besse 1	Fresh	Closed cycle
Fermi 2	Fresh	Closed cycle
Grand Gulf 1	Fresh	Closed cycle
Harris 1	Fresh	Closed cycle

Unit Name	Cooling Water Type	Cooling System Type
Hope Creek 1	Brackish	Closed cycle
Limerick 1, 2	Fresh	Closed cycle
Nine Mile Point 2	Fresh	Closed cycle
Perry 1	Fresh	Closed cycle
Sequoyah 1, 2	Fresh	Combination, once-thru/helper tower
Susquehanna 1, 2	Fresh	Closed cycle
Three Mile Island 1	Fresh	Closed cycle
Vogtle 1, 2	Fresh	Closed cycle
Watts Bar 1	Fresh	Closed cycle

2.2.5 References

The following is a compilation of documents on the subject of "Cooling Towers." They are not necessarily referenced by the preceding text.

P^{2}EP Technical Library

The Plant Performance Enhancement Program (P^{2}EP) has a technical library that provides an effective way to share information on plant thermal performance. Listed below are the documents from the P^{2}EP Technical Library that relate to the subject "Cooling Towers." These documents are available to eligible EPRI members by calling the P^{2}EP Administrator at 704/547-6024, or via the World Wide Web at http://www.epriweb.com/npg/pse/ptp/index.html.

File #	***Document Title***
0003.5-298	River Bend Station Thermal Performance Assessment
0027.0-110	Cooling Tower Performance
0027.0-473	WPPSS Plant Efficiency and Reliability Monitoring Program: Cooling Tower Performance Monitoring. Procedure No. 8.4.22. Rev 0. June 1988.
0027.0-613	Cooling Tower Performance
0027.0-75	Cooling Towers - The Neglected Bonanza
0099.0-578	P2EP Survey 94-018, Condensers, Cooling Towers, and Circulating Water Data
0099.0-585	P2EP Survey 95-006, Forced Draft Cooling Towers
0100.0-521	Nine Mile Point 2 Lost Electrical Generation Summary
0101.0-481	WPPSS Plant Efficiency and Reliability Monitoring Program: Tower Makeup Water System Performance Monitoring, Procedure 8.4.28., Rev. 0, 1988.
0136.0-509	WPPSS Operating and Engineering Test Procedure 8.4.53, Plant Thermal Performance
0142.3-111	Effect of Tube Material on Steam Condensation
0147.0-826	Power Uprate Cooling Tower Projects

Formal EPRI Reports/Software

The Electric Power Research Institute (EPRI) has created formal reports and / or software on the subject of “Cooling Towers.” These are available to eligible EPRI members by calling:

- EPRI Document Distribution Center 925 / 934-4212
- EPRI Software Distribution Center 800 / 763-3772

Report #	***Report Title***
CS-2276	Design and Operating Guidelines Manual for Cooling-Water Treatment
CS-3144-SR	On the Hypotheses of Calculation of the Water Flow Rate Evaporated in a Wet Cooling Tower
GS-6317	Proceedings: International Cooling Tower Conference
GS-6370, V-1	Cooling Tower Performance Prediction and Improvement, Vol. 1, Applications Guide
GS-6370, V-2	Cooling Tower Performance Prediction and Improvement, Vol. 2, Knowledge Base
GS-6574	Model Validation: Cooling-Tower Performance
GS-6976	Proceedings: International Cooling-Tower and Spray Pond Symposium

TPE Handbook References

The following documents contain additional information on cooling towers.

Proceedings: Cooling Tower Technology Conference. Electric Power Research Institute, Palo Alto, California: July 1997. ReportTR-108483.

C. F. Bowman, Chuck Bowman Associates, and D. J. Benton, Environmental Consulting Engineers, “Cooling Tower Performance.”

American Society of Mechanical Engineers. “Atmospheric Water Cooling Equipment Systems.” ASME Performance Test Code 23-1983 (R1988).

Cooling Tower Fundamentals. The Marley Cooling Company, 1985.

Robert Burger. *Cooling Tower Technology*. Third Edition, 1995.

Proceedings: Cooling Tower and Advanced Cooling Systems Conference. Electric Power Research Institute, Palo Alto, California: February 1995. Report TR-104867.

2.3 Feedwater Heaters

2.3.1 General Description

Figure 2-8 shows the basic arrangement of a train of feedwater heaters used to heat feedwater from its temperature leaving the condenser to final feedwater temperature (FFT) using steam extracted from various stages of the turbines. Some plants extract steam from both the high- and low-pressure turbines; others extract steam only from the low-pressure turbine. Most plants have more than one train of heaters, usually one for each low-pressure (LP) tandem turbine, for example, three trains of feedwater heaters if there are three LP turbines. Most BWRs are arranged to cascade all heater drains back to the condenser. This allows all the condensate to be demineralized before being returned to the reactor vessel as condensate. On the other hand, most PWRs pump part of the higher stage heater drains forward to the next higher stage heater.

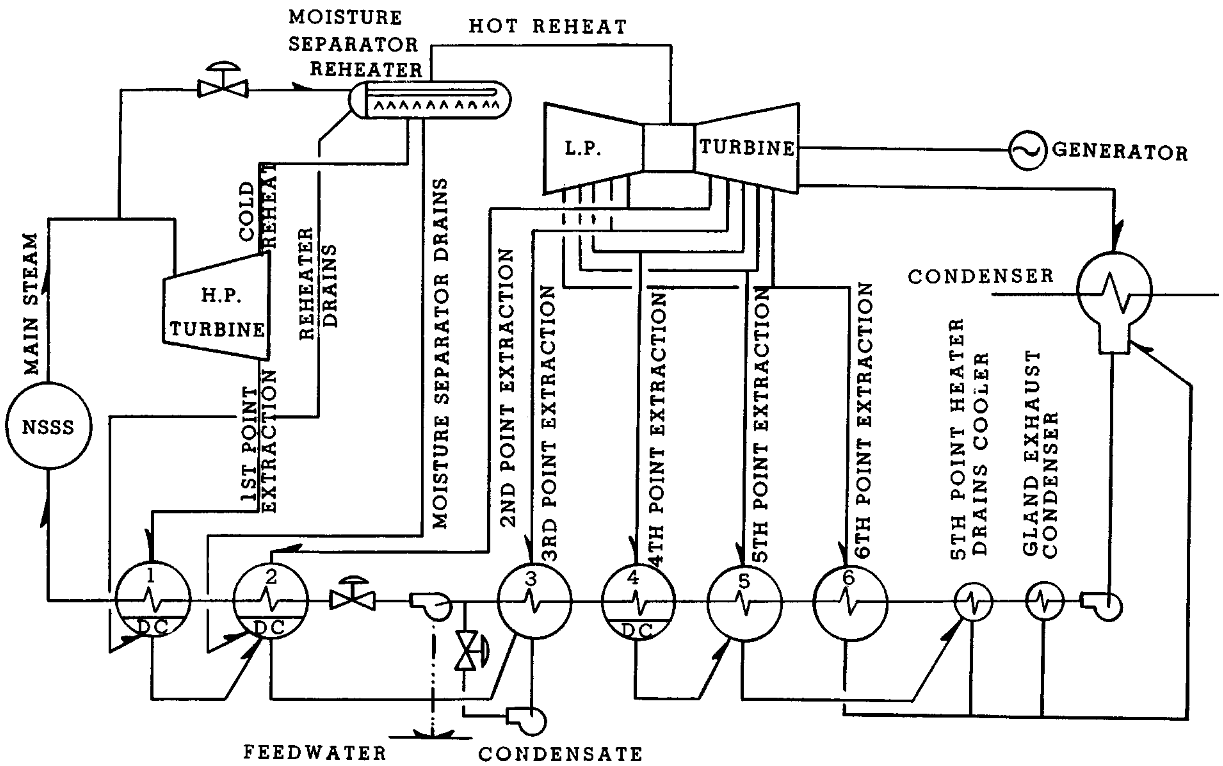


Figure 2-8
Basic Feedwater Heater Arrangement

Nearly all Light Water Reactor feedwater heaters are of the shell and U-tube, horizontal, two zone configuration. A typical configuration is shown in Figure 2-9. Feedwater flows on the inside of the U-tubes, which traverse both the condensing and drain cooler zones. The condensing zone consists of a part of the inlet legs of the U-tubes and is located in the bottom of the heater. Feedwater flows into the drain cooler zone of the heater where it is heated by subcooling the condensate formed in the condensing section of the heater. The feedwater then flows into the condensing zone, comprising the remainder of the U-tubes. Feedwater is heated in the condensing section by condensing turbine extraction steam, and steam is formed by the flashing of condensate flowing in from the drain coolers of upstream heaters. Most of the feedwater heating and heat transfer in a heater occurs in the condensing zone.

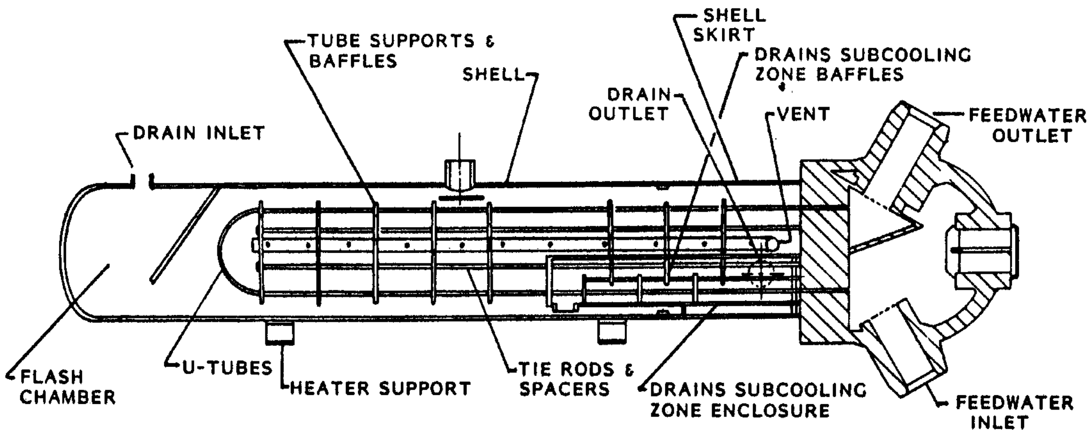


Figure 2-9
Typical Feedwater Heater Configuration

As previously noted, most feedwater heaters use U-tubes and are arranged to both condense extraction steam and to subcool the condensate. Some plants use separate drain coolers in the first stage of feedwater heating. In either case, it is necessary to maintain water level to submerge all the tubing in the drain cooler section. This is done by providing instrumentation to measure water level in the drain coolers section and control valves to control the condensate flow rate out of the heater.

Table 2-8 lists instrumentation that can be used to monitor performance of feedwater heaters. Temperature, pressure, and level instrumentation is commonly used with each heater. However, there are very few plants that have instrumentation to routinely measure feedwater and drain flows of the individual heaters. Instead, the flow is estimated from measured feedwater flow and the design split between parallel feedwater heater trains.

Table 2-8
Instrumentation for Feedwater Heaters

Parameter	Purpose
Shell-side pressure	Obtain extraction steam temperature, TTD
Feedwater inlet temperature	DCA, LMTD
Feedwater outlet temperature	TTD, LMTD
Extraction steam temperature	DCA, LMTD
Feedwater flow	Heat balance, tube-side film coefficient
Drain cooler flow	Heat balance, drain cooler shell-side film coefficient

2.3.2 Basic Calculations

Terminal Temperature Difference (TTD)

The TTD of a feedwater heater is the difference between the temperature of the extraction steam being condensed on the shell-side of the heater and the temperature of the feedwater leaving the heater. The design TTD for most plants varies from 5 to 8°F. A high TTD indicates that the feedwater heat transfer performance is deficient. It is a direct measure of the heat transfer capability of the heater and can be trended as a means to predict future performance.

$$TTD = T_s - T_{fwo}$$

Where

T_s = shell-side extraction steam temperature, °F
T_{fwo} = feedwater outlet temperature, °F
P_s = condensing section steam pressure, psia
T_s = condensing section steam temperature (from steam tables)

Input Data	P_s = 100 psia T_s = 400.1°F @ P_s = 100 psia T_{fwo} = 396°F
Calculation	$TTD = T_s - T_{fwo}$ TTD = 400.1 -396 = 14.1 °F

Drain Cooler Approach (DCA)

The DCA of a feedwater heater is the difference between the subcooled condensate leaving the heater and the feedwater entering the heater. It can be a direct measure of the heat transfer capability of the drain cooler section of the heater.

$$DCA = T_d - T_{fwi}$$

Where

T_d = drain cooler outlet temperature, °F
T_{fwi} = feedwater inlet temperature, °F

Input Data	T_d = 360.3°F T_{fwi} = 340.2°F
Calculation	$DCA = T_d - T_{fwi}$ DCA = 360.3 - 340.2 = 20.1 °F

2.3.3 Impact on Thermal Performance

Because the purpose of feedwater heaters is to increase cycle efficiency by heating the feedwater, any reduction in heat transfer capability of a heater reduces cycle efficiency and increases heat rate. In the case of a nuclear power plant that usually operates at or near licensed thermal power, an increase in heat rate results in a reduction in electrical output.

Heat transfer deficiencies are usually manifested by high TTDs and DCAs. Most plants have been designed with TTDs of 5°F and DCAs of 15°F. Increasing TTDs and DCAs cause increased heat rate and reduced electrical output. Usually, the TTD of the top heater stage has the largest impact on thermal performance with the lower pressure heaters having correspondingly less impact. The impact of a 3°F increase in final feedwater temperature or TTD of the top heater is approximately a 0.05% increase in heat rate. For the next highest heater, a 3°F increase in TTD increases heat rate approximately 0.04%. Changes in the lower stages of feedwater heating have correspondingly less impact. Changes in heater DCAs have significantly less impact than changes in TTDs (approximately a 0.05% increase in heat rate if all DCAs increase 10°F). Impacts can also be less at part power.

Table 2-9 lists typical impacts of increases in TTDs and DCAs on heat rate. Plant cycle-specific calculations using a heat balance program are needed if more exact, plant-specific values are desired.

Table 2-9
Feedwater Heater Impact on Thermal Performance

1°F increase in top heater TTD (FFT)	0.016% increase in heat rate
1°F increase in other stage heater TTD	0.013% increase in heat rate
1°F increase in DCA	0.005% increase in heat rate

2.3.4 Typical Problems, Degradation Mechanisms, and Corrective Actions

Feedwater heaters are subject to numerous problems. Of primary interest to this handbook are those that result in reduction in feedwater heating that usually manifest themselves in high TTD, high DCA, increased heat rate, and reduced power generation. The following are the more significant of these problems and some of the applicable diagnostics and corrective actions.

Plugged Tubes

Plugged heater tubes remove heat transfer surface from both the condensing and drain cooling zones of the heater. The effect can be to increase both the TTD and the DCA. Also, plugged tubes can increase the tube-side ΔP. If ΔP gets too high, the heater pass partition plate can buckle. This can lead to leakage across the pass partition plate instead of through the tubes and further deterioration of heater performance. Tubes are plugged because of observed leakage or non-destructive evaluation (NDE) indicating tube deterioration with a high potential to become a leaker. There can be several reasons for tube deterioration including tube vibration and corrosion. These maintenance and design related issues are discussed in some of the references at the end of this section.

The need to plug tubes based on NDE that indicates potential leakage rather than an actual leak is questionable, especially if the plugging criterion is very conservative (for example, 50% through-wall indication). In most cases, a tube leak developed during operation does not result in a forced plant outage. Tube plugging based on conservative NDE practices can lead to premature loss of thermal performance and high ΔP.

High Heater Level

This condition can cause submergence of some condensing zone tube surface and heat transfer capability. If the condition occurs, there should be a corresponding increase in drain cooler heat transfer surface and a decrease in DCA. One cause can be a malfunctioning drain cooler level control system. If the level is high, depending on the design of the heater, a portion of the condensing zone heat transfer surface may be submerged in condensate and not capable of condensing extraction steam and completely heating the feedwater. At the same time, there could be additional drain cooling capacity due to the high condensate level resulting in a small DCA.

Low Heater Level

A low heater level can have the opposite effect than a high level: low TTD and high DCA. In some cases, this can improve thermal performance. However, low drain cooler level should be avoided because it can subject the baffles and tubes supports in the drain cooler zone to two-phase erosion.

Shell-side Erosion

Most heaters use carbon steel tube support plates and flow baffles. Carbon steel is susceptible to flow-induced corrosion (FIC) especially if the flow is a two-phase mixture of steam and condensate as in a feedwater heater. Some long term erosion in the condensing zone is unavoidable. However, the heat transfer mode in the condensing zone is by condensation and the plates in the condensing zone are primarily for tube support, not to create high flow across the tubes and enhance heat transfer. As a result, FIC does not necessarily have a direct impact on condensing zone thermal performance and TTD. It can cause erosion of the tube supports, which can increase susceptibility to tube vibration damage. FIC in the condensing zone is primarily a life cycle problem.

On the other hand, the drain cooler is normally subject to single-phase condensate flow where FIC is not of major concern. However, if the drain cooler is continually operated with a low water level, the transverse flow baffle / tube supports can be subject to steam / two-phase FIC. The result can be increased tube vibration and a reduction in flow across the tubes, a reduction in drain cooler thermal performance, and an increase in DCA.

If FIC is suspected in either the condensing or drain cooler zones, then a shell-side visual inspection may be necessary to evaluate the extent of the problem.

Section 6 of EPRI NP-4057 contains a Problem Solving Guide. This guide is reproduced as Table 2-10 in this handbook. EPRI NP-4057, a three-volume report, provides a primer and design, procurement, operations, and maintenance guidelines for nuclear plant feedwater heaters.

Table 2-10
Problem Solving Guide

Problem Experienced	Areas to Investigate	Possible Solutions
Temperature rise of feedwater gradually decreases. Pressure drop across heater gradually increases.	Fouled heat transfer surfaces. Deposits of scale or corrosion on interior surfaces of tubes.	Clean tubes. Prevent oil or other contamination from getting into system.
Temperature rise of feedwater is less than specified. Temperature in shell side condensate is lower than specified. Vent lines are cold.	Air binding non-condensable gases accumulating in steam space of heater. Gland seals at L.P. end of turbine may be defective.	Check for proper venting of heater. On heaters operating at vacuum, check for air leaks at heater and pipe joints. Check turbine gland seals.
Temperature rise of feedwater is less than specified. Pressure drop across heater decreases.	By-passing. Leak at partition plate in water channel allows water to by-pass from one pass to the next without passing through the tubes.	Check gaskets and gasket contact surfaces for “wire-drawn” or eroded areas. Repair surfaces and replace gasket.
Shell flooding with condensate when outlet control valve is fully open.	Ruptured tube or leak between feedwater side an steam space.	Shut down heater as quickly as possible to prevent damage to other tubes or finished surfaces. Repair leak.
Unstable condensate level erratic surging of level.	Control devices not working properly or backpressure in drain line too high.	Repair control devices or reduce backpressure in drain line.
Water hammer and vibration.	Overloading, above normal temperatures or flow rates causing pulsations in connecting pipes.	Check flow rates, temperatures and pressure conditions of entering steam and drains against design operating specifications. Check tube side flow rate.

2.3.5 References

The following is a compilation of documents on the subject of “Feedwater Heaters.” They are not necessarily referenced by the preceding text.

P^{2}EP Technical Library

The Plant Performance Enhancement Program (P^{2}EP) has a technical library that provides an effective way to share information on plant thermal performance. Listed below are the documents from the P^{2}EP Technical Library that relate to the subject “Feedwater Heaters.“ These documents are available to eligible EPRI members by calling the P^{2}EP Administrator at 704/547-6024, or via the World Wide Web at http://www.epriweb.com/npg/pse/ptp/index.html.

File #	Document Title
0006.0-212	Minimizing Turbine Blade Erosion With An Innovative Low-Pressure Heater Design Modification
0010.0-141	Feedwater Heater Performance Prediction Calculation Procedure
0024.0-854	Cost Effective Corrosion Control in Electric Power Plants
0031.0-138	Feedwater Heater Cycle Configuration
0033.0-677	Miller Generating Station Drain Cooler Problems
0034.4-674	The Life Cycle Economics of Feedwater Heater Replacement
0034.6-519	Oyster Creek Thermal Performance and Availability Report
0040.0-364	Reliable Specification Criteria Confirmed Through Forensic Analysis Of A High Pressure Feedwater Heater
0040.0-679	A Case Study of Tube Failure in Subcooling Zone of Low Pressure Feedwater Heaters
0040.0-697	Manual for Investigation and Correction of Feedwater Heater Failures
0042.0-378	Sequoyah Nuclear Plant--Feedwater Venturi Fouling Investigation
0043.0-114	Effects of Geometry and Temperature Ramps on Feedwater Heaters
0043.0-123	Evaluation Of A Feedwater Heater Acoustic Leak Detection System: Phase II
0043.0-129	Feedwater Heater Testing Program at Alabama Power
0043.0-137	Feedwater Heater Replacement At Indian Point 3
0043.0-139	Feedwater Heater Life Extension Via Programs For Shell Liquid Level, Venting and Allied Systems
0043.0-143	Feedwater Heater Replacements in Ontario Hydro's Thermal Generating Stations
0043.0-145	Feedwater Heater Tubesheet System Design Features For Cyclic Operation
0043.0-178	High Pressure Feedwater Heater Channel Head Cracking Update
0043.0-201	Level Control of Feedwater Heaters Using Resistivity Probe Sensors
0043.0-230	Nine Mile Point 2, Feedwater Heater Performance, N2-PPMI-6.2, Rev. 00, 1991
0043.0-251	Nuclear Plant Component Basic Series Feedwater Heaters Workbook
0043.0-295	Pilgrim Nuclear Station Instruction: Feedwater Heater Performance. PNPS SI-PG.1021, Rev. 1
0043.0-366	Replacement Feedwater Heater Project Overview: Belews Creek Steam Station, Unit 2
0043.0-368	Restoring Heat Transfer Surface To Feedwater Heaters Using Explosively Welded Sleeves At LA Cygne Generating Station
0043.0-377	Sequoyah Feedwater Heater Performance Test, Technical Instruction TI-106., Rev. 0.
0043.0-379	Single String Operation Of High Pressure Feedwater Heaters: A Systematic Design Approach
0043.0-404	Study of Feedwater Heaters For Improved Coal-Fired Power Plants
0043.0-467	Waterford Unit 3 Standing Test Procedure: Feedwater Heater Performance, Procedure No. N8-006, Rev. 0
0043.0-471	La Cygne Station Unit 2 Partial Retubing of High Pressure Feedwater Heater No. 21
0043.0-483	WPPSS Plant Efficiency and Reliability Monitoring Programs: Feedwater Heater Performance Monitoring. Procedure No. 8.4.25. Rev 1, September 1988.

0043.0-53	Con Ed Feedwater Heaters, TAP 7-2-2, 3/86.
0043.0-621	Measurement of Sludge Deposits on Pickering NGS and Bruce NGS Retired Copper Alloy Feedwater Heaters
0043.0-670	Application of a New High Strength FNi-Cr-MN Austenitic Tube Alloy for High Pressure Feedwater Heaters
0043.0-671	Cracking in High Pressure Heater Heads
0043.0-672	High-Reliability Feedwater Heater Design and Construction
0043.0-673	An Evaluation of Tube Failure Rates of Original and Replacement Feedwater Heaters in the Southern Electric System
0043.0-675	Gibson Feedwater Heater Shell Crack Repairs and Modifications
0043.0-676	Results Derived from Levels I to III Inspections of Two Stainless Steel Tubed High Pressure Feedwater Heaters
0043.0-678	H.P. Feedwater Heater Level Control Modification
0043.0-681	Energy Analysis for Sensitivity to Turbine Induction
0043.0-684	Remote Visual Testing (RVT) for the Diagnostic Inspection of Feedwater Heaters
0043.0-685	Recent Improvements in RFEC Inspection Technology
0043.0-686	Experience with Remote Field Electromagnetic Technique (RFET) in the NDT of Carbon Steel and High Ferritic Alloy Tubes
0043.0-688	Comparison of Electromagnetic NDE Procedures Using Realistic Feedwater Heater Mock-ups
0043.0-689	Investigation of Hemi-Heat Cracking in the High Pressure Feedwater Heaters at Homer City
0043.0-690	Feedwater Heater Tube Material Study
0043.0-691	Feedwater Heater Materials in the SCE System Historical and Current Practices
0043.0-692	Application of AL-6XN Tube Material in Feedwater Heaters
0043.0-695	Symposium Breakout Session 6A: Maintenance Technologies for Performing Damage Assessments and Repair Evaluations
0043.0-696	Symposium Breakout Session 6B: Operation Technologies for Achieving Design Enhancements and Performance
0043.0-698	Heat Exchanger Workstation: A Comprehensive Software Tool for Feedwater Heater Operation and Maintenance
0043.0-699	Feedwater Heater Tube to Tubesheet Connections
0043.0-7	A Case History Of Feedwater Heater Condition Assessment and Life Extension
0043.0-700	First Header-Type Feedwater Heater Technology in the United States
0043.0-701	Reliability of Header Type Feedwater Heater Tube Joints
0043.0-717	Turbine Capability with Feedwater Heaters Out of Service
0043.0-723	Feedwater Heater Program Development
0043.0-824	Feedwater Heater Testing & Level Optimization Experience Within the Southern Company
0057.0-360	Recommended Practices for the Prevention of Water Damage to Steam Turbines Used for Electric Power Generation
0060.0-263	Nuclear-Plant Performance, Reliability, Heat Rate Improvement & Life Extension Basic Series-Heat Balances Workbook

0060.5-275	Performance Monitoring and Replacement of Heat Exchanger Components and Materials
0060.5-309	Practical Aspects and Performance of Heat Exchanger Components and Materials
0060.5-328	Proceedings: Heat Exchanger Workshop
0070.0-773	Seabrook Station Feedwater Heater Digital Level Controllers
0072.0-781	Flash Drying Protects Standby Plants
0073.0-204	Magnetic Flux Leakage Examination of Feedwater Heater Tubing
0073.0-460	Water Leakage to the Condenser
0073.0-680	Acoustic Feedwater Heater Leak Detection Industry Application of Low & High Frequency Detection Increases Response and Reliability
0073.0-683	Early Detection of Feedwater-Heater Leaks
0074.0-296	Pilgrim Nuclear Station Instruction: Feedwater Heater Controls. SI-PG.1022, Rev 1.
0074.0-353	Quantifying The Costs Associated With Feedwater Heater Drain Flow Bypass To Condenser
0074.0-759	Ft. Calhoun Station Repetitive Special Services Procedure: Feedwater Heater Optimum Level Test
0074.0-88	Performance Driven Level Control of Feedwater Heaters Utilizing Independent Microprocessors and Host Distributed Digital Control Systems
0084.0-693	The Application of an Advanced FNI-CR-MN Austenitic Alloy for Feedwater Heaters
0084.0-694	Alloy 400 (UNSNO4400) for Use in High Pressure Feedwater Heaters
0092.0-643	Feedwater Heating Performance Gains Developed Through AOV Maintenance
0092.0-831	FW Bypass of High Pressure Heaters
0094.0-248	Non-Destructive Examination Method For Carbon Steel High Pressure Feedwater Heaters
0094.0-35	Available Non-Destructive Examination Techniques For Closed Feedwater Heaters
0099.0-524	P2EP Survey 92-001, Feedwater Heater Replacement
0099.0-576	P2EP Survey 94-016, Inlet Feedwater Temperature
0099.0-586	P2EP Survey 95-007, High Pressure Feedwater Heater Bypass
0099.0-669	P2EP Survey 96-009, Detection of Feedwater Heater Tube Leaks
0100.0-521	Nine Mile Point 2 Lost Electrical Generation Summary
0101.0-21	Feedwater Heater Performance Monitoring
0101.0-421	The Performance Software Users Group Meeting Proceedings
0102.0-682	Interim Status Report on the Revision of ASME PTC 12.1 - Closed Feedwater Heaters
0106.0-787	Callaway Plant Engineering Test Procedure: ETP-A01102, HP Feedwater Heaters Performance Monitoring
0142.3-185	Improved Ferritic Stainless Feedwater Heater Tubing - Crevice Corrosion Resistance In Chloride Faulted Feedwaters
0146.2-862	Testing, Instrumentation and Analysis Proposed for Revision of ASME PTC 12.1--Closed Feedwater Heaters
0152.0-130	Explosively Welded Tube Joints in Critical Service Applications

Formal EPRI Reports/Software

The Electric Power Research Institute (EPRI) has created formal reports and / or software on the subject of “Feedwater Heaters.“ These are available to eligible EPRI members by calling:

- EPRI Document Distribution Center 925 / 934-4212
- EPRI Software Distribution Center 800 / 763-3772

Report #	***Report Title***
AP-3651	Thermo-Economic Analysis of Power Plants
CS-3239	Recommended Guidelines for the Operation and Maintenance of Feedwater Heaters
CS / NP-3743	Symposium on State of the Art Feedwater Heater Technology
CS-4014	Proceedings: 1983 Fossil Plant Heat Rate Improvement Workshop
CS-4155	Feedwater Heater Procurement Guidelines
CS-4285	Detection of Water Induction in Steam Turbines, Phase 3: Field Demonstration
CS-5486	Non-radioactive Tracer for Steam Turbine Thermal Performance Tests
CS-5856	High-Reliability Feedwater Heater Study
CS-6080	Proceedings: 1987 Conference on Expert-System Applications In Power Plants
GS / EL-5648	MARK I Performance Monitoring Products
GS-6635	Proceedings: 1988 EPRI Heat-Rate Improvement Conference
GS-6913	Feedwater Heaters: Replacement Specification Guidelines
GS-6935	Feedwater Heaters Maintenance and Repair Technology: Reducing Outage Cost
GS-7290	Proceedings: Feedwater Heater Technology Conference
GS-7417	Feedwater Heater Survey
NP-3395	Calculation of Leak Rates Through Cracks in Pipes and Tubes
NP-4057, V-1	Nuclear Plant Feed Water Heater Handbook, Vol. 1, Primer
NP-4057, V-2	Nuclear Plant Feed Water Heater Handbook, Vol. 2,
NP-4057, V-3	Nuclear Plant Feed Water Heater Handbook, Vol. 3, O&M Guidelines
TR-101772 R1	Electromagnetic NDE Guide for Balance-of-Plant Heat Exchangers, Rev. 1
TR-101846	Second EPRI BOP Heat Exchanger NDE Workshop
TR-102923	Proceedings: 1992 Feedwater Heater Technology Symposium
TR-106741	Heat Exchangers: An Overview of Maintenance and Operations (NMAC Tech Notes)

Non-EPRI References

Organizations and publishers other than EPRI have material pertaining to this subject. Some of this other reference information is listed below to enable the reader to obtain this material if desired.

American Society of Mechanical Engineers. “Code on Steam Condensing Apparatus, (Reaffirmed 1988).” ASME Performance Test Code 12.2-1983. August 31, 1983.

American Society of Mechanical Engineers. “Closed Feedwater Heaters, (Reaffirmed 1987).” ASME Performance Test Code 12.1-1978. 1978.

“Standards for Steam Surface Condensers, Eighth Edition.” Heat Exchange Institute, Incorporated, Cleveland, Ohio: January, 1984.

“Standards for Closed Feedwater Heaters, Fifth Edition”. Heat Exchange Institute, Incorporated, Cleveland, Ohio: January, 1992.

TPE Handbook References

The following document contains additional information on feedwater heaters.

Nuclear Plant Feedwater Heater Handbook, Volumes 1 - 3. Electric Power Research Institute, Palo Alto, California: June 1985. Report NP-4057.

2.4 Steam Generators

2.4.1 General Description

Steam generators (SG) in PWR plants are heat exchangers with reactor primary cooling water on the tube side entering at T_{hot} and exiting at T_{cold}.

In a feedring steam generator, feedwater enters a downcomer region and flows up past the tubes. Over most of the tubes, the secondary flow is saturated. Nearly dry saturated steam exits from the top of the steam generator after passing through a moisture separator section in the steam dome. The moisture that is separated from the saturated steam recirculates with the feedwater. The moisture return path is via the downcomer region between the tube wrapper and the outer steam generator shell. Figure 2-10 shows a typical feedring steam generator.

Preheating SGs have feedwater entering in a lower baffled section of the T_{cold} region of the tubes, which uses the primary coolant at T_{cold} to preheat the feedwater.

Table 2-11
Instrumentation for Steam Generators

Parameter	Purpose
Main steam flow	Check on FW flow
Feedwater flow	Calculate heat transfer and Q
Mainsteam P, T	Calculate t_{sat}, Hs
T_{hot}	Calculate Δ Tm
T_{cold}	Calculate Tm
Feedwater temperature	Calculate heat transferred Q

In a once-through steam generator (OTSG), feedwater enters a downcomer region and then flows up past the tubes. Saturated steam is produced in the bottom third of the tube region and is superheated 50°F in the top two-thirds of the region. Fouling in OTSGs causes the water level to rise and decrease superheat.

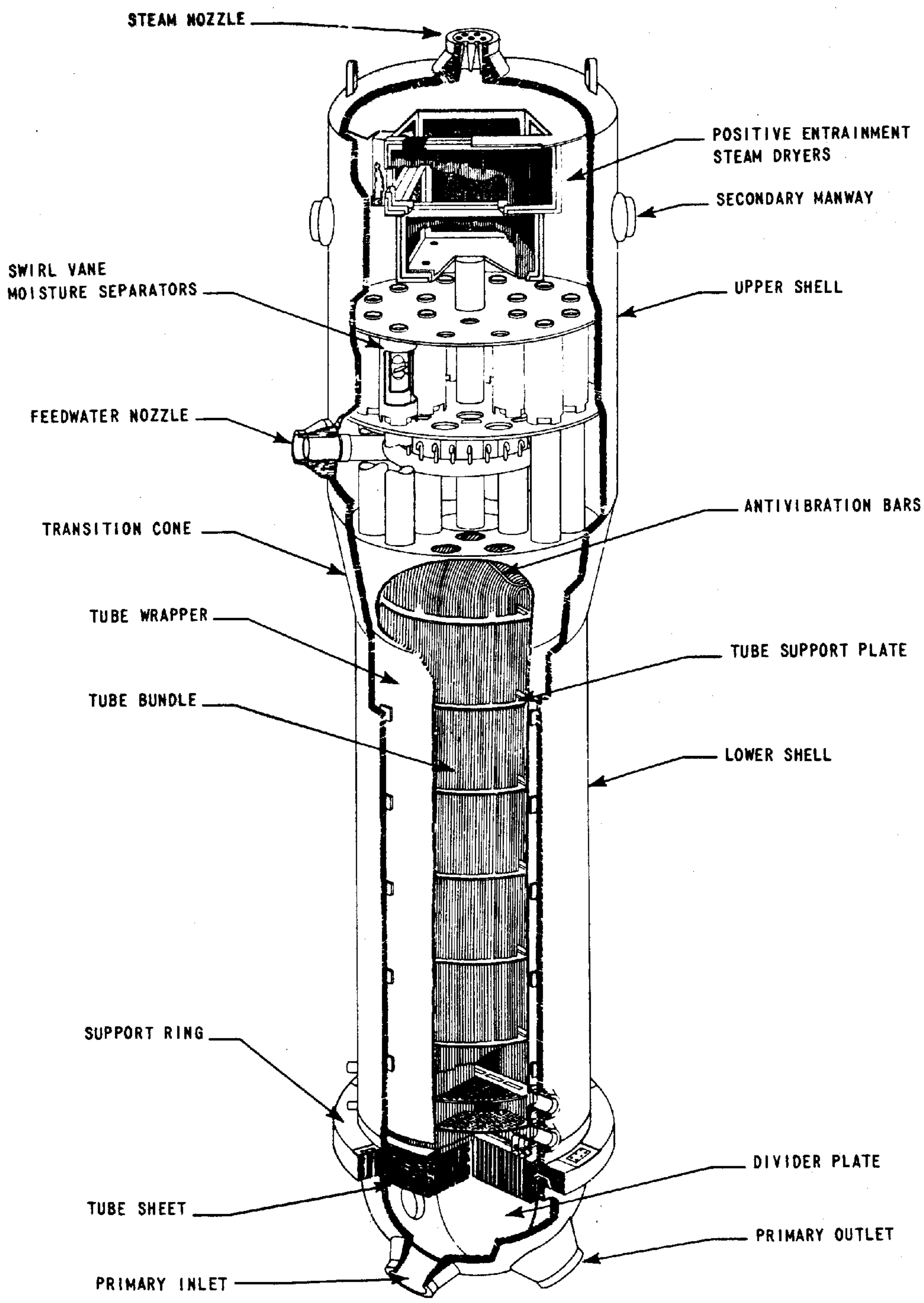


Figure 2-10
Westinghouse Model F Feedring Type Steam Generator

2.4.2 Performance Calculations

Plant thermal performance can be affected by the capability of PWR steam generators to transfer heat from the reactor coolant to the secondary system. This thermal capability can be reduced by several degradation mechanisms, the three most important being the quantity of plugged tubes, accumulation of deposits on the inside and outside tube surfaces, and fouling of the moisture separators. Fouling of the moisture separators increases pressure drop and allows more moisture carryover. The number of plugged tubes should be accounted for and trended. The effect of deposit accumulation is determined by calculation.

The resistance to heat transfer of a SG tube is the sum of the conductive resistance of the tube wall, the boundary layer resistance of the primary and secondary fluids, and the resistances resulting from the accumulation of any deposit layers on the inside and outside tube surfaces. The effect of increased resistance is to require an increase in reactor coolant temperature or a decrease in secondary side temperature, that is, steam pressure to continue transferring the same amount of heat from the reactor.

The thermal resistance of SG tubes varies locally throughout the tube bundle, depending on local coolant conditions and local deposit accumulation. It can be shown that a global or mean overall thermal resistance can adequately account for local variation and be used to evaluate and trend the performance of steam generators. The global thermal conductance, U, is the inverse of the global thermal resistance and can be calculated by:

$$U = Q / (A\Delta T_m)$$

where

$$Q = M_{fw}\left(h_s - h_{fw}\right) + M_{bd}\left(h_s - h_a\right)$$

$$\Delta T_m = \frac{T_{hot} - T_{cold}}{\ln\left[\dfrac{T_{hot} - T_{sat}}{T_{cold} - T_{sat}}\right]}$$

and

A = the active heat transfer area of the steam generator, ft^2
M_{fw} = feedwater into steam generator, lb / hr
h_s = steam enthalpy at SG pressure (Ps), Btu / lb
h_{fw} = saturated water enthalpy at feedwater temperature, Btu / lb
M_{bd} = SG blowdown rate at the time data is taken, lb / hr
h_a = makeup water enthalpy, Btu / lb
ΔT_m = log mean temperature difference, °F

T_{hot} = primary coolant hot leg (SG inlet) temperature, °F
T_{cold} = primary coolant cold leg (SG outlet) temperature, °F
T_{sat} = saturation temperature at SG pressure, Btu / lb

After the global thermal conductance has been calculated, the global film resistance, R_f, can be calculated from:

$R_f = 1 / U - 1 / U_o$

where U_o is the design global thermal conductance of the SG and can be calculated using the above equations and design data.

The values of P_s, R_f, and U are all candidates for trending because they are measures of steam generator heat transfer degradation. Another important indicator is the quantity UA because it includes the impact of plugged SG tubes.

Sample calculations:

Input Data	$M_{fw} = 3.79 \times 10^6$ lbm / hr $h_s = 1191.3$ Btu / lbm $h_{fw} = 419$ Btu / lbm $m_{bd} = 0$ lbm / hr $T_{sat} = 541°F$ $T_{hot} = 618°F$ $T_{cold} = 557°F$ $A = 48{,}300\ ft^2$
Calculation	$Q = 3.79x10^6\, x(1191 - 419)$ $= 2.93x10^9\ Btu/hr$ $\Delta T_m = \frac{618 - 557}{\ln\left[\frac{618-541}{557-541}\right]}$ $= 38.8^\circ F$ $U = \frac{2.93x10^9}{48{,}300x38.8}$ $= 1{,}563 \frac{Btu}{hr\ ft^2\ ^\circ F}$
Input Data	$U_o = 1563$ Btu / hr ft^2 °F U = 1500 Btu / hr ft^2 °F

Calculation	$R_o = \frac{1}{1563}$ $= 6.40x10^{-4} \frac{hr\ ft^2\ {}^\circ F}{Btu}$ $R = \frac{1}{1500}$ $= 6.67x10^{-4} \frac{hr\ ft^2\ {}^\circ F}{Btu}$ $R_f = 2.69x10^{-5} \frac{hr\ ft^2\ {}^\circ F}{Btu}$

Uncertainties in the measurement of moisture in the steam and feedwater flow reduce the accuracy of these calculations. An approach for trending steam generator heat transfer performance has been used that reduces the effect of measurement uncertainties. This approach requires adjusting the primary temperature slightly to reproduce the steam pressure from the previous data point. A comparison can then be made of the effects of steam enthalpy changes on the throttle position, power, and feedflow that would indicate SG fouling.

2.4.3 Impact on Thermal Performance

Steam generator performance directly impacts thermal performance. The steam generator removes heat from the primary coolant. Therefore, it must return coolant at a value of T_{cold} that provides adequate heat transfer from the primary system. At its maximum performance, it must remove heat equal to the reactor power while providing steam flow and pressure to obtain 100% of plant MWe.

Figure 2-11 shows the reduction in steam pressure due to an increase in fouling factor for a typical steam generator. Figure 2-12 shows the effect of steam generator tube plugging on steam generator pressure, and is shown to emphasize that the heat transfer area in fouling factor calculations must be corrected for plugged tubes.

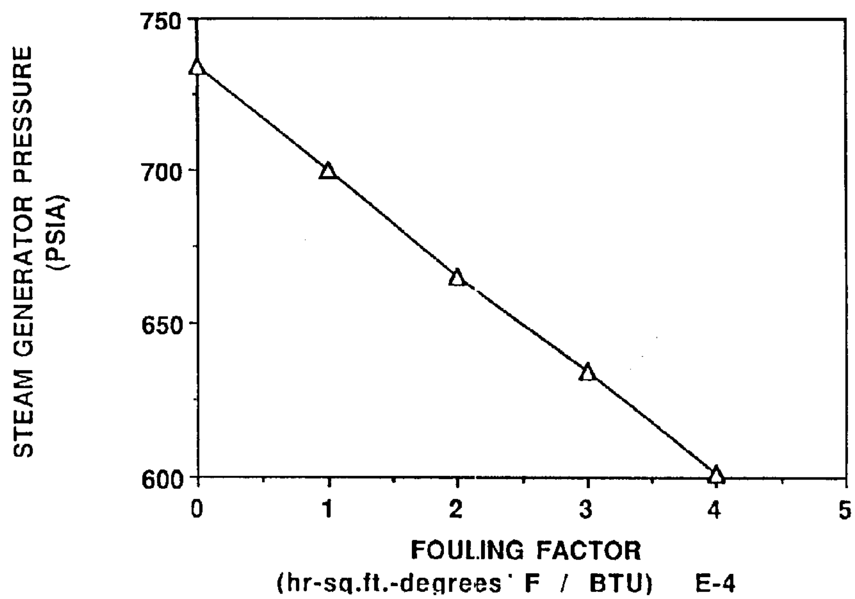


Figure 2-11
Effect of Fouling Factor on Steam Generator Pressure

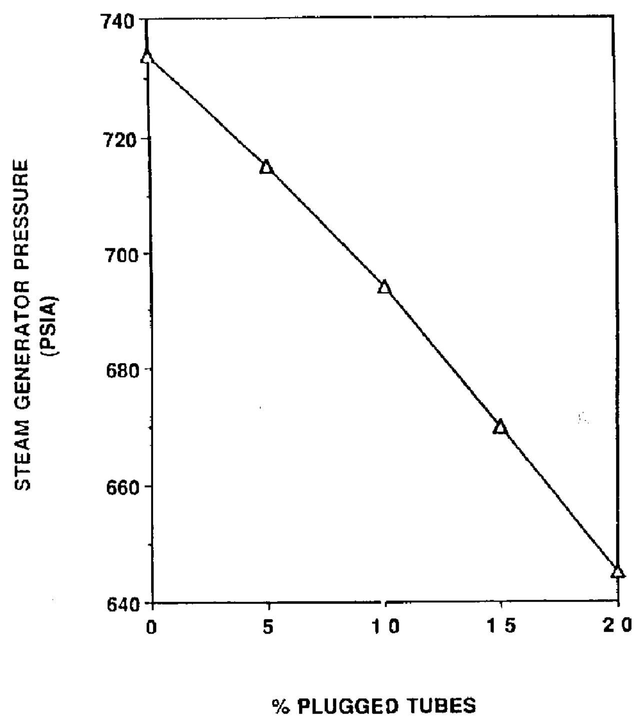


Figure 2-12
Effect of Plugged Tubes on Steam Generator Pressure

Figure 2-13 shows the effects of steam generator degradation on the ability to produce full rated electrical power.

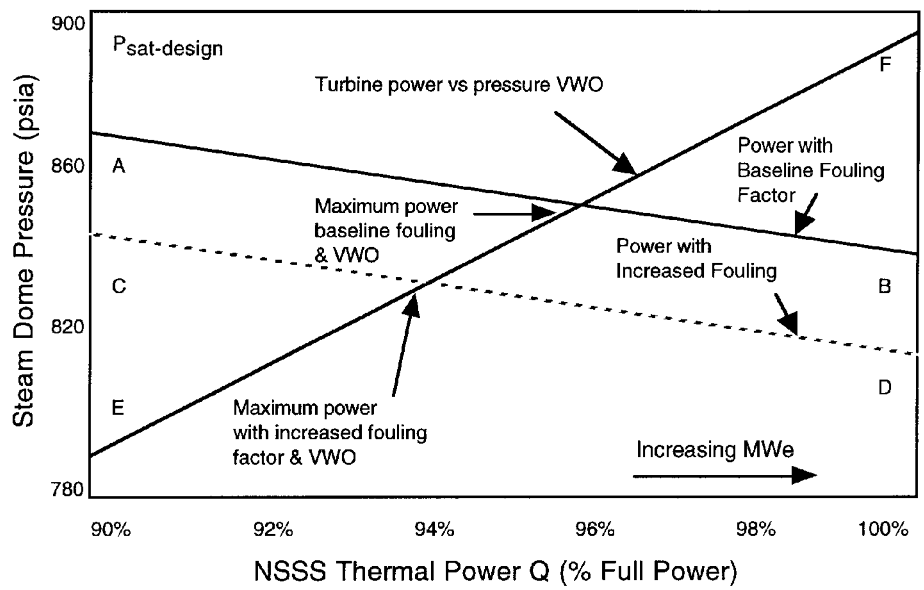


Figure 2-13
SG Pressure vs. Plant Thermal Power

Line A-B represents the power versus pressure that can be attained at the current or baseline fouling factor. Line A-B assumes that the turbine throttle valves can be adjusted to increase steam flow, which lowers the steam dome pressure and transfers more heat, thereby increasing the power back to 100%. Line C-D represents increased fouling, which requires a still higher flow and lower T_{sat}, P_{sat}.

As the throttle valves are opened to compensate for fouling (or tube plugging, etc.), the valves-wide-open (VWO) condition occurs. When it does, the power is limited by line E-F. Hence, the plant is limited in power to the intersection points shown in the figure. Also, as the throttle valves open to provide more steam flow to compensate for lower enthalpy, there is more moisture carryover, which also affects power.

Changes in secondary cycle performance can offset some of the SG heat transfer performance, which is primarily affected by fouling of the secondary surfaces of the tubes.

Steam generator parameters that should be baselined and monitored include:

- Steam generator pressure (the primary indicator of steam generator performance)
- Reactor coolant system T_{avg} (which controls the steam generator pressure to some extent and provides a subset for losses associated with steam generator pressure)
- Blowdown flow
- Moisture carryover (not normally monitored, but must be considered if determined to deviate from design)

Each of these parameters affects not only the calorimetric calculations of core power, but also has an independent effect on the turbine cycle efficiency. The baseline values and further deviations must be considered for accurate electrical megawatt accounting.

2.4.4 Typical Problems, Degradation Mechanisms, and Corrective Actions

The principle steam generator problems affecting thermal performance are:

- Secondary side deposits (including moisture separators)
- Plugged tubes - some plants have margin or excess tubes, others do not
- Primary side deposits - appear not to be significant

The first two are most significant, and primary side deposits do not appear to be significant. All have the effect of decreasing the steam generator heat transfer capacity (UA). In order to maintain the same reactor power and steam generation rate with reduced UA, the steam generator mean temperature difference between the primary and secondary coolants has to increase. This increase leads to decreasing secondary side steam pressure (P_{sat}) and temperature (T_{sat}). Figure 2-13 illustrates this relationship.

Most plants initially had excess turbine steam flow capacity (VWO) and steam generator heat transfer capacity (UA). As SG tubes are plugged and the remainder become fouled, T_{ave} has to be increased or P_{sat} has to be reduced to maintain reactor power and turbine output. There may be some reduction in turbine output depending on the change in heat rate with changes in P_{sat} and turbine control valve throttling. At some point, depending on available margins, T_{ave} runs up against Tech Spec limits (and is not usually an option) and P_{sat} reduction results in turbine VWO conditions. Further reduction in UA requires reduction in both reactor power and turbine output. The point at which this occurs is plant unique and depends on steam generator and turbine capacity margins.

The subject of corrective action for both secondary side fouling and tube degradation are complex and beyond the scope of this handbook. However, there are several caveats that can be brought to the attention of the user of this handbook:

- There is a great deal of uncertainty as to the effect of fouling deposits due to the measurement process and other factors that may be reducing SG thermal performance.
- Several plants have performed chemical cleaning to remove the secondary side fouling deposits. Results have been mixed. In many cases, chemical cleaning has not had beneficial results.
- Mechanical cleaning of moisture removal sections has increased steam pressure and the performance of SGs.
- Prevention of tube degradation that results in the need for tube plugging is dependent on many factors including the history of secondary side water treatment and primary coolant T_{hot}. Several plants have or are in the process of replacing steam generators because of current and predicted steam generator plugged tube conditions.

The references at the end of this section deal with the prevention and remediation of tube fouling and tube degradation and with SG replacement. These are subjects and responsibilities that are usually beyond the scope of just the thermal performance engineer. As a minimum, personnel involved in secondary side water chemistry and reactor power have responsibilities in these areas. The duties of the thermal performance engineer can include informing others of the thermal performance of the steam generators by determining and trending the number of plugged tubes, the U, the UA, and the r_f of the steam generators, as well as the impact of the condition of the steam generators on heat rate and plant output.

Corrective actions for SG fouling should be based on a program that includes the following steps:

- Compute the global fouling factor from available early data to develop a baseline of initial performance. The initial performance should be compared with design values.
- Track the global fouling factor and record all potential causes for changes such as tube plugging. These items facilitate determining the root cause for trends.
- Determine actual measurement uncertainties for evaluation of data. These uncertainties might suggest upgrading instrumentation to obtain usable accuracy.

- Consider adding instruments to obtain additional parameters. These include pressure drop between the tube bundle and the steam pressure measurement location to determine if there are flow changes in the internals, also, recirculation ratio and preheater flow distribution (for preheater SGs) to detect tube support or baffle clogging.
- Characterize and track tube scale properties.
- Track secondary side impurity ingress as an additional check on scale build-up.
- When scale build-up indicates potential corrective action, an economic evaluation would include chemical cleaning, mechanical cleaning, cycle modifications and T_{hot} increases.

A power uprate reduces the steam generator performance margin. Therefore, degraded SGs may prevent obtaining the full power uprate. In order to keep primary temperatures constant, more heat must be removed, which requires a lower temperature, hence a lower pressure. Primary side deposits are also a potential for degraded SG performance. Estimating from industry data on primary side scale deposits, primary deposits are probably not responsible for a significant portion of steam generator pressure losses.

2.4.5 References

The following is a compilation of documents on the subject of "Steam Generators." They are not necessarily referenced by the preceding text.

P^2EP Technical Library

The Plant Performance Enhancement Program (P^2EP) has a technical library that provides an effective way to share information on plant thermal performance. Listed below are the documents from the P^2EP Technical Library that relate to the subject "Steam Generators." These documents are available to eligible EPRI members by calling the P^2EP Administrator at 704/547-6024, or via the World Wide Web at http://www.epriweb.com/npg/pse/ptp/index.html.

File #	***Document Title***
0017.0-833	Dimethylamine Technology to Remove Deposits and Reduce Fouling of Nuclear Steam Generators
0031.5-648	Steam Generator Mass Calculation: The Cure for the "Shrink-and-Swell" Phenomenon
0051.0-626	Steam Generator Performance Monitoring Using Non-Intrusive Downcomer Flow Measurements
0053.0-608	Fouling of Heat Transfer Equipment

0053.0-642	Alternate Amine Improves Plant Performance at Comanche Peak Steam Electric Station
0053.0-834	Steam Generator Pressure Reduction: Historical Perspective, Current EPRI Work, Monitoring Tools, and Role of Deposits
0060.0-716	Heat Rate Performance of Nuclear Steam Turbine-Generators
0060.5-328	Proceedings: Heat Exchanger Workshop
0061.0-884	1996 Activity Report Heat Transfer and Aerodynamics Branch--Electricite de France
0069.0-707	The Role of Visual Inspection in Managing Steam Generator Sludge
0080.0-253	Nuclear Plant Component Basic Series--Turbine-Generators Workbook
0099.0-552	P2EP Survey 93-021, Turbine Performance Related to S/G Plugging
0099.0-556	P2EP Survey 93-025, Thermal Performance Models
0099.0-562	P2EP Survey 94-001, Steam Generator Degradation Modeling
0099.0-575	P2EP Survey 94-015, Main Steam Header Pressure Loss
0099.0-576	P2EP Survey 94-016, Inlet Feedwater Temperature
0099.0-595	P2EP Survey 95-016, Steam Generator Pressure Loss
0100.0-714	Predicting the Performance of 1800-RPM Large Steam Turbine-Generators Operating with Light Water-Cooled Reactors
0101.0-355	Recent Innovations and Experience with Plant Monitoring and Utility Operations
0106.0-788	Callaway Plant Engineering Test Procedure: ETP-BB-01100, Steam Generator Performance Monitoring
0106.0-789	Callaway Plant Engineering Test Procedure: ETP-BM-01100, Steam Generator Blowdown Performance Monitoring
0120.0-880	A Boiling Perspective of Tube Fouling, Water Chemistry and Steam Pressure
0126.0-109	Effect of Corrosion Product Fouling on Ginna Steam Generator Tube Heat Transfer
0126.0-354	R.E. Ginna Nuclear Station Steam Generator Performance
0126.0-394	Steam Generator Performance Degradation
0126.0-395	Steam Generator Reference Book
0126.0-769	Steam Generator Degradation and Improvement
0126.0-808	Causes of PWR Steam Generator Thermal Performance Degradation
0126.0-832	Steam Generator Pressure Improvement at Comanche Peak Unit 2
0126.0-860	Measurement Uncertainty and the New Performance Test Code for Steam Generators (PTC 4)
0126.0-895	Predicting the Steam Pressure Recovery Following Chemical Cleaning at San Onofre Unit 2
0126.0-898	Steam Generator Thermal Performance Degradation Case Studies
0146.0-803	Application of Ultrasonic Techniques for Measuring Steam Generator Downcomer Flow
0146.2-861	Measurement Uncertainty and the New Performance Test Code for Steam Generators (PTC 4) - Presentation

Formal EPRI Reports/Software

The Electric Power Research Institute (EPRI) has created formal reports and / or software on the subject of "Steam Generators." These are available to eligible EPRI members by calling:

- EPRI Document Distribution Center 925 / 934-4212
- EPRI Software Distribution Center 800 / 763-3772

Report #	***Report Title***
GS / EL-5648	MARK I Performance Monitoring Products
NP-2987	Chemical Cleaning Process Evaluation - Westinghouse Steam Generators
NP-2990	Steam Generator Chemical Cleaning: Demonstration Test in a Model Boiler
NP-2997	Ultrasonic Enhancement of Chemical Cleaning of Steam Generators
NP-3009	Steam Generator Chemical Cleaning Process Development
NP-3302	Data Acquisition Reduction System for Chemical Cleaning Processes for Nuclear Steam Generators
NP-3477, V-2	PWR Steam Generator Chemical-Cleaning Data Base
NP-4597	Chemical Cleaning of Millstone Unit 2,
NP-4600	Effect of Venting on Crevice Cleaning for PWR Steam Generators
NP-4604, V-3	ATHOS3: A Computer Program for Thermal-Hydraulic Analysis of Steam Generators
NP-4708	Chemical Cleaning of PWR Steam Generator Sludge Piles
NP-4990, V-1	Thermal Performance Diagnostic Manual for Nuclear Power Plants, Vol. 1, Development of Thermal Performance Diagnostic Manual
NP-4990, V-2	Thermal Performance Diagnostic Manual for Nuclear Power Plants, Vol. 2, Plant-Specific Conversion Guidelines and User's Manual
NP-7524	Steam Generator Performance Degradation
TR-102952 R1	Advanced Amine Application Guidelines
TR-105003	Experience With Inhibitor Injection to Combat IGSCC in PWR Steam Generators
TR-106048	Characterization of PWR Steam Generator Deposits
TR-106212-V1	Inhibition of IGA/SCC on Alloy 600 Surfaces Exposed to PWR Secondary Water: Vol 1
TR-107262	Effect of Inhibitors on the Electric Resistance of Alloy 600 Surface Films

Non-EPRI References

Organizations and publishers other than EPRI have material pertaining to this subject. Some of this other reference information is listed below to enable the reader to obtain this material if desired.

American Society of Mechanical Engineers. “Steam Generator Apparatus.” ASME Performance Test Code 4.1, 1964.

TPE Handbook References

The following document contains additional information on steam generators.

White, et al., “Causes of PWR Steam Generator Thermal Performance Degradation,” Presented at EPRI Nuclear Plant Performance Improvement Seminar, Asheville, North Carolina (September 1996).

2.5 Steam Turbine

2.5.1 General Description

Nuclear plant turbines are composed of a high-pressure turbine and one or more low-pressure turbines connected to the generator by a common shaft. Each turbine stage consists of a stationary set of blades that act as nozzles that develop high velocity steam and a set of rotating blades that convert the kinetic energy of the steam to torque on the turbine shaft. A portion of the steam flowing through a stage may be extracted for feedwater heating.

Turbine stages typically are categorized as impulse or reaction designs. An impulse stage produces all of the stage pressure drop in the stationary blades (sometimes called nozzles), and there is little relative velocity change across the rotating blades (sometimes called buckets). In a reaction stage, approximately half of the stage pressure drop is in the rotating blades. From the point of view of the thermal performance engineer, there is negligible difference between the two types.

Turbines consist of a number of stages. A typical nuclear plant HP turbine has six stages and the LP has eight stages. The pressure ratios are approximately the same for all stages of a turbine over the entire load range, except for the first stage which has a variable inlet area due to the turbine throttle valves, and the last stage, which has a relatively constant exhaust pressure.

As load changes, the upstream pressure and flow on each stage changes, which causes all of the stage pressure drops to change proportionally except the last stage of the LP turbine, which always sees the condenser pressure on the downstream side. The other pressure ratios in the LP turbine are constant across the load range. Therefore, the last LP stage has a pressure ratio that varies with load and condenser pressure. As load decreases or condenser pressure increases, the last stage pressure ratio decreases. Most of the opportunity for turbine performance improvements is in the first HP stage design and the last LP stage design. An important component of the flow path is the exhaust hood, which can affect the last stage efficiency.

2.5.2 Performance Calculations

A key indicator of turbine performance is overall gross turbine heat rate. A sample calculation of this parameter is contained in Table 2-2 Volume 1 of this handbook. However, gross turbine heat rate is also affected by the cycle leaks and performance of other cycle components such as condensers and MSRs. Nevertheless, gross turbine heat rate is the most important parameter that can be monitored and trended.

Calculation of performance parameters that directly indicate turbine performance is handicapped by lack of cost effective technology to routinely measure moisture and enthalpy of wet steam. There are, however, several performance parameters that can be calculated that do not require moisture measurement. Two that are discussed below are turbine first stage pressure and HP turbine enthalpy drop efficiency.

The following describes how turbine first stage pressure can serve as a turbine performance parameter. The flow-passing ability of a steam turbine is given by the relationship:

$$M = k\sqrt{\frac{p}{v}} \tag{2.1}$$

Where

M	=	steam flow lb / hr
k	=	the flow coefficient
p	=	the stage inlet pressure, psia
v	=	the stage inlet specific volume, ft^3 / lb

This relationship holds at all turbine stages. Also, it can be shown that the relationship between p and v is approximately:

$$v \approx \frac{1}{p} \tag{2.2}$$

Substituting Equation 2.1 into Equation 2.2 results in:

$$\frac{p}{M} \approx \frac{1}{k}$$

The above relationships are good approximations except when turbine stage pressure ratios are substantially off-design, and the turbine is at low load.

Figure 2-14 shows the general relationship between stage inlet pressure and steam flow or feedwater flow, and lists the causes of high and low first stage pressure. A single relationship is shown because feedwater flow and steam flow are nearly identical. (Nozzles refers to nozzles downstream of control or first stage.)

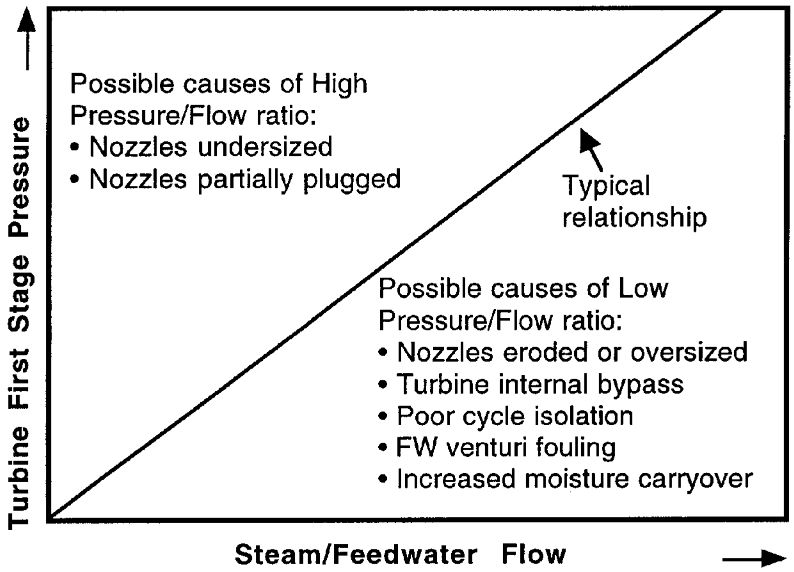


Figure 2-14
Turbine First Stage Pressure vs. Steam or Feedwater Flow

Depending on availability of pressure measurement points, this method of measuring turbine performance can be used at all steam path points downstream of the HP inlet steam, including feedwater heating steam extraction points. Note from Figure 2-14 that the method can be used to indicate feedwater flow nozzle fouling.

Turbine enthalpy drop efficiency is defined as the actual enthalpy drop from turbine inlet to turbine outlet divided by the ideal or isentropic enthalpy drop, or:

$$\eta = \frac{h_{hpi} - h_{hpo}}{h_{hpi} - h'_{hpo}}$$

Where

h_{hpi}	=	specific enthalpy of the HP turbine inlet (throttle) steam, Btu/lb
h_{hpo}	=	specific enthalpy of the HP turbine exhaust steam, Btu/lb
h'_{hpo}	=	ideal specific enthalpy of the HP turbine exhaust steam, Btu/lb

Calculation of enthalpy drop efficiency across the HP turbine is possible in (and only in) plants that have MSRs and the outlet temperature of the MSRs is measured and used to calculated HP turbine outlet enthalpy (routine direct measurement of HP turbine moisture to determine turbine outlet enthalpy is not practical). Also, cycle steam flow and certain flows and enthalpy must be measured or estimated from cycle heat balances and thermal kits. For this reason, absolute accuracy of an enthalpy drop efficiency calculation is not high. However, the repeatability of the calculation, if trended at nearly the same turbine load, can indicate turbine degradation due to erosion, foreign deposits, or feedwater nozzle fouling.

The first step in this methodology is to calculate actual HP turbine exhaust enthalpy using a heat balance and MSR data.

$$h_{hpo} = \frac{h_{msro} \times \left(W_t - W_{ext} - W_{ms} - \left(W_{r1} + W_{r2}\right)\right) + h_{ms} \times W_{ms} - \left(h_{ri1} - h_{ro1}\right) \times W_{r1} - \left(h_{ri2} - h_{ro2}\right) \times W_{r2}}{W_t - W_{ext} - \left(W_{r1} + W_{r2}\right)}$$

Where

h_{msro} = calculated MSR outlet enthalpy, Btu / lb
W_t = derived HP turbine throttle steam flow, lb / hr
W_{ext} = derived total extraction flow from HP turbine to FW heaters, lb / hr
W_{ms} = derived moisture separator drain flow, lb / hr
W_{r1} = derived first- or single-stage reheat steam flow from HP turbine, lb / hr
W_{r2} = derived second-stage reheat steam flow from HP turbine, lb / hr
h_{ms} = derived enthalpy of moisture separator drain flow, Btu / lb
h_{ri1} = derived enthalpy of first- or single-stage inlet reheat steam, Btu / lb
h_{ro1} = derived enthalpy of first- or single- stage reheat drain, Btu / lb
h_{ri2} = derived enthalpy of second-stage inlet reheat steam, Btu / lb
h_{ro2} = derived enthalpy of second stage reheat drain, Btu / lb

Except for MSR outlet reheat temperature and pressure, it is not practical to routinely measure the MSR data and the turbine throttle extraction and reheat steam flows needed in the above equation. The above data has to be derived from design heat balances and thermal kits for the turbine load existing at the time the MSR outlet temperature and pressure are measured.

MSR outlet enthalpy is determined from steam tables or the Mollier diagram using the measured MSR outlet superheat temperature and pressure, T_{msro} and P_{msro}, respectively.

Figure 2-15 shows an example heat balance for a nuclear plant steam cycle with a two-stage reheater. The calculations for the HP turbine exit enthalpy and the enthalpy drop efficiency are shown in the sample calculation below.

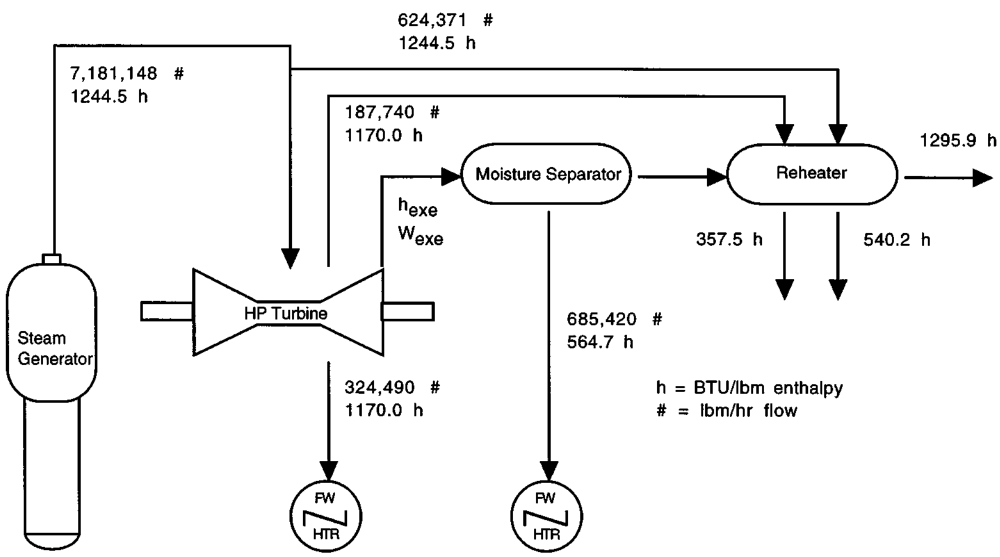


Figure 2-15
Heat Balance Diagram for HP Turbine Exhaust Enthalpy

Extraction flows to the feedwater heaters can be taken from the guarantee heat balance if the measured temperature rise in the heaters is about the same as the values given in the heat balance. Reheat steam enthalpy can be obtained from the measured pressure and the design expansion line for the turbine on a Mollier diagram. Determining the enthalpy of the throttle steam requires the moisture content value. Using the Mollier diagram and assuming that the steam leaving the steam generator is dry and saturated, a constant enthalpy process from the steam generator dry saturated conditions to the pressure just ahead of the throttle valve can be used to obtain the throttle steam moisture.

In the example, the throttle steam is slightly superheated, which would be the case for a once-through SG.

Input Data	P_{msro} = 88.0 psia T_{msro} = 530°F W_t = 7,181,148 lb / hr W_{ext} = 324,490 lb / hr W_{ms} = 685,420 lb / hr W_{r1} = 187,740 lb / hr W_{r2} = 624,371 lb / hr h_{ext} = 1170 Btu / lb h_{ms} = 564.7 Btu / lb h_{ri1} = 1170 Btu / lb h_{ro1} = 357.5 Btu / lb h_{ri2} = 1244.5 Btu / lb h_{ro2} = 540.2 Btu / lb h'_{hpo} = 1102.0 Btu / lb
Calculations	Determine MSR outlet enthalpy at a measured MSR outlet temperature and pressure of 530°F and 88 psia, respectively, using Mollier diagram. Note that these should be the average values for all MSRs receiving steam flow from the HP turbine. h_{msro} = 1295.9 Btu / lb Calculate HP turbine outlet enthalpy $$h_{hpo} = \frac{\begin{bmatrix} 1295.9 \times (7{,}181{,}148 - 324{,}490 - 685{,}420 - 187{,}740 - 624{,}371) \\ +(564.7 \times 685{,}420) - ((1170 - 357.5) \times 187{,}740) \\ -((1244.5 - 540.2) \times 624{,}371) \end{bmatrix}}{[7{,}181{,}148 - 324{,}490 - 187{,}740 - 624{,}371]}$$ = 1115 *Btu/lb* Calculate HP turbine enthalpy drop efficiency $$\eta_{hp} = \frac{1244.5 - 1115}{1244.5 - 1102}$$ = 0.909 = 90.9%

K.C. Cotton's ***Evaluating and Improving Steam Turbine Performance*** contains a systematic approach to diagnosing potential turbine problems. Nuclear plant turbines have controlled water conditions and also run at 1800 rpm to reduce moisture-caused erosion. Therefore, they seldom have increased or decreased flow area caused by solid particle erosion or buildup of deposits. However, there are instances of turbine problems, and part of diagnosing trends in the monitored parameters should include checking for HP and LP turbine degradation.

A complete tutorial for interpreting the data from turbine monitoring and testing is beyond the scope of this handbook and can be found in Cotton's book. However, the following approach is a good starting point for the TPE:

1. Trend the parameters listed in Table 2-12. Also trend thermal efficiency, the stage pressure ratio P_{1st} / P_{EXH} for the HP turbine, the first stage pressure ratio for the LP turbine, the reheater TTD, and the pressure drop across the MSR.

2. Verify the data by looking for

   - Changes in MSR ΔP

   - Suspicious changes in thermal efficiency

   - Unexplainable changes in P_{1st} / P_{EXH} for the HP turbine

   - Variations in duplicate measurements

   - Inconsistencies in measurements

3. Reconcile data. This is done through redundant calculations of the heat and mass balances for available flows.

4. Check for cycle isolation. This is usually indicated by a decrease in thermal efficiency at licensed maximum MW thermal. Methods for detection include:

   - Isolate the cycle one step at a time.

   - Measure leakage flows to the condenser using pressures downstream from leakage points.

   - Check the temperature of lines entering the condenser.

5. Check power output versus valve positions to check for whether the problem is in the valve or turbine.

6. Check the HP turbine efficiency versus percent load to check for HP turbine solid particle erosion.

Table 2-12
Instrumentation for Steam Turbines

Parameter	Purpose
Feedwater flow	Throttle flow
Final feedwater temperature	Thermal efficiency
Throttle pressure	Throttle enthalpy and steam flow
Electric load	Thermal efficiency
First stage pressure (P_{1st})	Flow characteristics
Reheater exit pressure (P_{x-0})	RH exit enthalpy
HP turbine exit pressure	HP pressure drop and flow
HP turbine extraction temperature	Check on heat balance
MSR drain flows	Enthalpy drop
MSR drain temperatures	Enthalpy drop
Reheater inlet pressure	Extraction flow
LP turbine extraction temperatures	Heat balance
Condenser pressure	Enthalpy and turbine backpressure
LP turbine extraction pressure	LP stage ΔPs

2.5.3 Impact on Thermal Performance

The turbine efficiency has a one-to-one effect on thermal efficiency or heat rate. Although usually there is little deterioration in turbine performance, the turbine performance should be trended. The turbine can degrade to a point where it is cost effective to repair or upgrade it. HP turbine enthalpy drop efficiency can be an indicator of SG moisture separator effectiveness. The LP turbine produces approximately 65% of the total kilowatts in a non-reheat cycle and 69% in a reheat cycle.

The opportunities for heat rate improvement in the steam path include 0.25% for last stage LP turbine blade improvements and 0.5% for exhaust hood improvements.

Improved heat rate alone has not justified major turbine design modifications. However, blade replacement due to stress corrosion cracking of the hubs has presented the opportunity to justify upgrading blade design in the process of repairing the turbine.

2.5.4 Typical Problems, Degradation Mechanisms, and Corrective Actions

Since the development of the turbines used in the nuclear plants, two advancements in turbine design and test capability have helped identify inefficiencies in the turbine design and how to improve the designs. These advancements have been in three-dimensional computational fluid dynamics (CFD) and interstage sample traverses to measure pressure, velocity, flow angles, and moisture.

Turbines are designed to optimize as well as possible the mechanical forces that are imparted to the blades by the steam. The maximum available work that the steam can do is equal to the reversible isentropic process change in enthalpy from the turbine inlet pressure to the turbine back pressure. Any less work than the reversible isentropic work represents lost efficiency and shows up as increased entropy and enthalpy at the condenser. Inefficiencies are caused by the following:

- Moisture droplets impinging on blades
- Leakage of steam through seals on the stationary blading, the blade tips, and at the ends of the casings
- Aerodynamic drag on tie wires
- Profile losses from flow separation on portions of the blades
- Secondary losses or endwall losses in the low aspect ratio (length ÷ chord) HP stages
- Secondary losses due to high tip speeds and radial flow on LP blades (especially the last stage)
- Exhaust losses due to high kinetic energy in the steam which is not converted to work

Moisture is a problem in nuclear turbines because first stage inlet steam is saturated or only slightly superheated in plants with once-through SGs. As the initial micron-sized droplets impinge on surfaces, they coalesce into droplets that impact blades at various angles. The net effect is a loss of aerodynamic efficiency.

Profile losses are the result of aerodynamic problems with blade designs. The relative velocities of the blades and steam flow vary over the length of the blade. This variation changes the angle of attack, which can be in a range causing stall or flow separation. The energy is lost to friction that would otherwise produce “lift” on the blades.

Endwall losses are caused by radial crossflow. Vortices are formed at the blade tips. Energy that would otherwise provide rotating forces on the blades is dissipated in these vortices.

The exhaust hood forms the region of the flow path following the last stage of the turbine. Optimum flow in the exhaust hood directs flow evenly to the condenser tubing. Also, it should be designed so that steam leaving the last stage does not leave with an excess of kinetic energy that could have been converted to work by the turbine. Most of the turbine pressure drop should be in the turbine rather than the path to the condenser.

A tool for measuring the turbine performance during overhaul is a steam path audit, which quantifies performance improvements from both routine maintenance and parts replacement. The purpose of the steam path audit is to quantify steam path deterioration on a stage-by-stage and casing-by-casing basis. One engineer typically takes from three to six days to collect the data, generate the loss reports, and present the preliminary results before making any repair decisions. The timing of the steam path audit is concurrent with turbine disassembly, and does not interfere with the outage schedule. A steam path audit is part of a five-step heat rate improvement program shown in Figure 2-16.

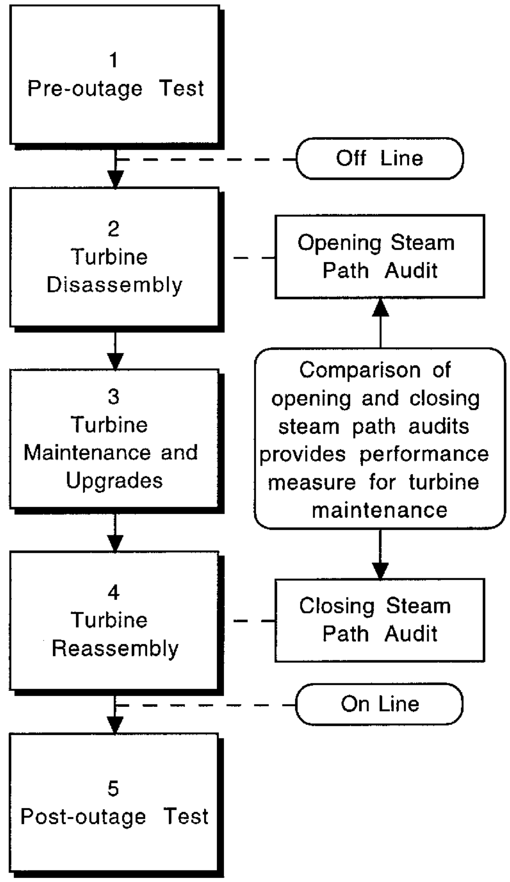


Figure 2-16
Steam Path Audits as Part of Heat Rate Improvement Program

1. **Pre-Outage Test** - A heat rate test before an overhaul establishes the condition of the unit before routine maintenance and installation of any new parts. The heat rate test results may indicate that, for example, the heat rate is up 4%, but it does not indicate the cause of the poor performance.

2. **Opening Steam Path Audit** - An inspection of the steam path immediately upon opening the steam turbine helps to quantify the damage and degradation found in each stage in terms of efficiency, power, and heat rate. For example, a steam path audit quantifies water erosion, rubbed packing seals, and foreign object damage in the HP and LP casings that account for the measured 4% increase in heat rate. The audit engineer measures material losses in the following categories:

   - Interstage packings (diaphragm packings or shaft packings)
   - Tip spill strips (shroud seals)

- End packings (shaft packings, dummy packings, n-1, n-2, and other packings)
- Miscellaneous leakages (snout ring, bell seal, manhole cover gasket, horizontal joint, etc.)
- Solid particle erosion
- Deposits
- Mechanical damage (broken blading, foreign object damage, etc.)
- Surface roughness (deposits, machining, etc.)
- Deposits under rotating blade covers (deposits under shroud bands)
- Trailing edge thickness (from previous weld repairs)
- Hand calculations (anything that does not fit into the above categories)

3. **Upgrade and Maintenance** - The results of the steam path audit are used to determine the most cost-effective routine repairs to the steam path. The cost-effectiveness, or benefit-to-cost ratio, of individual maintenance actions is calculated using the heat rate penalty assigned to the damage identified in the steam path audit results and the corresponding repair costs. The benefit-to-cost ratio for each possible maintenance action item is then ranked. A minimum benefit-to-cost ratio is then established and the maintenance is completed.

4. **Closing Steam Path Audit** - After any new parts are installed and the remaining maintenance has been performed, the closing steam path audit predicts the return-to-service condition of the unit. Like the opening steam path audit, the closing audit identifies stage-by-stage losses and quantifies the performance penalty associated with each. For example, after repairing foreign object damage, suppose the nozzle trailing edges were left at 40 mils rather than at their design thickness of 30 mils. Additionally, some grinding marks were left on the suction side of the nozzle exhaust. The closing steam path audit would quantify the trailing edge thickness and grinding losses as, say, 10 BTU/kWh. The difference between the opening results and the closing results is the performance gain due to routine maintenance. The purpose of a closing steam path audit is twofold: 1) to estimate the return-to-service condition of the unit and 2) to serve as a quality control check on maintenance conducted during the outage.

5. **Post-Outage Test** - Shortly after the unit is back on line, a post-outage heat rate test measures the actual return-to-service condition. The difference between the pre- and post-outage tests is the performance improvement due to upgrades and routine maintenance.

The steam path audits provide a calculated change in heat rate based on conditions of packings, glands, blade surfaces, spill strips, etc., found when the turbine casings are open. The five steps in Figure 2-16 yield the following heat rate values:

- Pre-test heat rate
- Opening steam path audit heat rate
- Closing steam path audit heat rate
- Post-test heat rate

The total improvement in heat rate due to maintenance and any turbine upgrades is equal to the pre-test heat rate minus the post-test heat rate. The heat rate improvement due to outage maintenance is the opening steam path audit heat rate minus the closing steam path audit heat rate. Heat rate degradation due to startup is the post-test heat rate minus the closing steam path audit heat rate.

2.5.5 References

The following is a compilation of documents on the subject of "Steam Turbine." They are not necessarily referenced by the preceding text.

P^2EP Technical Library

The Plant Performance Enhancement Program (P^2EP) has a technical library that provides an effective way to share information on plant thermal performance. Listed below are the documents from the P^2EP Technical Library that relate to the subject "Steam Turbine." These documents are available to eligible EPRI members by calling the P^2EP Administrator at 704/547-6024, or via the World Wide Web at http://www.epriweb.com/npg/pse/ptp/index.html.

File #	*Document Title*
0006.0-212	Minimizing Turbine Blade Erosion With An Innovative Low-Pressure Heater Design Modification
0006.0-334	Proceedings: Steam Turbine Blade Reliability Seminar and Workshop
0020.0-79	Corrosion-Resistant Coatings for Low-Pressure Steam Turbines
0020.3-103	Development Of More Accurate Correction Factors Through Heat Balance Modeling
0031.5-223	Thermal Performance Data (Thermal Kit) for ComEd Braidwood Station
0034.6-160	Ginna Station Stationary Blading Design Optimization
0034.6-453	Upgrades To Turbine - MSR Hardware For Performance Improvement
0034.6-519	Oyster Creek Thermal Performance and Availability Report
0043.0-681	Energy Analysis for Sensitivity to Turbine Induction

0055.0-876	Heat Kit for Peach Bottom
0057.0-360	Recommended Practices for the Prevention of Water Damage to Steam Turbines Used for Electric Power Generation
0060.0-263	Nuclear-Plant Performance, Reliability, Heat Rate Improvement & Life Extension Basic Series-Heat Balances Workbook
0060.0-523	1992 Heat rate Improvement Conference: Sessions 5A, 5B, 6A, 6B, 7A, and 7B
0080.0-104	Westinghouse Turbine Performance
0080.0-11	A Manufacturer's Experiences Of Improving The Heat Rate Of Large Steam Turbines
0080.0-13	A Practical Guide To N2 Packing Testing On GE Combined HP-LP Turbines
0080.0-253	Nuclear Plant Component Basic Series--Turbine-Generators Workbook
0080.0-308	PP&L Heat Rate Improvement of Three Large Super Critical Turbines
0080.0-382	Some Thoughts On Plant Performance Evaluation
0080.0-398	Steam Turbine Performance Survey As Compiled From Steam Path Audits
0080.0-418	Comparison of Steam Turbine Path Audit Results to Performance Test Results
0080.0-485	WPPSS Plant Efficiency and Reliability Monitoring Programs: Main Turbine Performance Monitoring. Procedure No. 8.4.41. April 1988.
0080.0-498	Design and Efficiency Increase of LMZ Turbines
0080.0-641	Turbine Steam Path Audits at San Onofre Nuclear Generating Station
0080.0-760	Fort Calhoun Station Repetitive Special Services Procedure: Steam Cycle Performance Test
0085.0-516	Duct Retrofits Cut Leakage and Boost Heating-System Efficiency
0090.0-124	Evaluation of a Moisture Removal Device for Turbine Steam Piping
0090.0-18	MSR Performance Monitoring
0090.0-224	Optical Probe Measurements of Low Pressure Steam Turbine Wetness and Efficiency at Morgantown Unit 2
0090.0-25	Interim Report #2 on Moisture Separator-Reheater Drain Systems
0090.0-269	General Description--Two-Stage Moisture Separator Reheater (Internal Header, Vertical U-Bend Design)
0090.0-390	The MSR Evolution
0090.0-732	MSR Optimization Test Report
0099.0-528	P2EP Survey 92-005, Erosion / Corrosion
0099.0-533	P2EP Survey 93-002, ASME Steam Turbine Testing
0099.0-534	P2EP Survey 93-003, HP Turbine Efficiency
0099.0-535	P2EP Survey 93-004, Unaccounted-For MWe Loss and Turbine Degradation
0099.0-552	P2EP Survey 93-021, Turbine Performance Related to S / G Plugging
0099.0-566	P2EP Survey 94-005, Precision Shaft Torque Meter for Large Utility Turbines
0099.0-602	P2EP Survey 96-005, Nuclear Turbine Upgrade Activities
0100.0-257	Nuclear Plant Performance, Reliability, Heat Rate Improvement and Life Extension Basic Series-Thermal Kits Workbook
0100.0-521	Nine Mile Point 2 Lost Electrical Generation Summary

0101.0-301	Plant Thermal Performance and Availability Monitoring Report (GPU Nuclear - Oyster Creek)
0101.0-355	Recent Innovations and Experience with Plant Monitoring and Utility Operations
0101.0-356	Recent Innovations and Experience with Plant Monitoring and Utility Operations
0101.0-802	Use of Reconciled Process Data in Performance Tests of Low-Pressure Turbines at a 1000 MW Power Plant
0102.0-273	Performance Improvements of Fossil Unit Retrofit Designs
0102.0-439	Adaptation of the Alternative Test Code in Evaluating a Low Pressure Steam Path Replacement
0106.0-784	Callaway Plant Engineering Test Procedure: ETP-AC-01100, Main Turbine Generator Performance Monitoring
0113.0-252	Nuclear Plant Component Basic Series Moisture-Separators / Reheaters Workbook
0116.0-457	Variable Pressure Operation: An Assessment
0116.0-725	Rotor Retrofit at WNP-2
0134.5-897	Optical Probe for Measurement of Steam Wetness Fraction in LP Turbines
0139.0-518	P2EP Pilot Training: Thermal Kits Course Module
0144.0-12	A Method To Monitor High-Pressure Turbine Efficiency and Moisture-Separator Effectiveness In LWR Power Plants Without Tracer Techniques
0145.0-102	Development of Low-Pressure Turbine Coatings Resistant to Steam-Borne Corrodents, Vol. 2: Detailed Studies
0145.0-131	Steam Turbine Efficiency Improvement
0145.0-19	Aero-Thermodynamics of Low Pressure Steam Turbines and Condensers
0145.0-199	Latest Advances in Steam Turbine Design, Blading, Repairs, Condition Assessment, and Condenser Interaction
0145.0-202	Low Pressure Turbine Exhaust Flow Improvement Study
0145.0-217	Model Testing Of Low Pressure Turbine Exhaust Ends
0145.0-288	Pilgrim Nuclear Power Station Instruction PG-1070: Main Turbine Performance, Rev. 0, 1987.
0145.0-436	Direct Measurement of LP Cylinder and Final Stage Efficiencies Using Probe Traverses
0145.0-448	TVA Technical Instruction TI-119: Turbine Cycle Performance Test, Rev. 0, 1989
0145.0-620	Low Pressure Steam Turbine Thermal Performance Improvements
0146.5-797	Survey of Performance Upgrade Experiences on U.S. Nuclear Turbines
0146.5-891	Turbine Upgrades at Grand Gulf

Formal EPRI Reports/Software

The Electric Power Research Institute (EPRI) has created formal reports and / or software on the subject of “Steam Turbine.” These are available to eligible EPRI members by calling:

- EPRI Document Distribution Center 925/934-4212
- EPRI Software Distribution Center 800/763-3772

Report #	***Report Title***
AF-903	Turbine Cycle Performance Improvement Through Titanium LP Blades
CS-1604	Detection of Water Induction in Steam Turbines
CS-2124	Review of Corrosion-Resistant Coatings for Steam Turbine Components
CS-2932	Corrosion Fatigue of Steam Turbine-Blading Alloys in Operational Environments
CS-3135	Detection of Water Induction in Steam Turbines, Phase II: Field Evaluation
CS-3251	Steam Turbine Condensation--Shock Wave Interaction
CS-3891	Survey of Steam Turbine Blade Failures
CS-4285	Detection of Water Induction in Steam Turbines Phase 3: Field Demonstration
CS-4410	Demonstration of an Alternative ASME Steam Turbine Generator Acceptance Test
CS-4515	Titanium L-1 Steam Turbine Blade Retrofit
CS-4545-SR	Proceedings: 1984 Power Plant Performance Monitoring Workshop
CS-4683	Solid-Particle Erosion of Utility Steam Turbines: 1985 Workshop
CS-5085	Steam Turbine Blade Reliability Seminar and Workshop, 1986
CS-5415	Erosion Resistant Coatings for Steam Turbines
CS-5486	Non-radioactive Tracer for Steam Turbine Thermal Performance Tests
GS-6535	Solid Particle Erosion of Steam Turbine Components: 1989 Workshop
GS-6725	Proceedings: Fossil Plant Retrofits for Improved Heat Rate Availability
GS-7003	Improvement of the Exhaust Flow of a Low-Pressure Turbine
GS-7295	Demonstration of EPRI Heat-Rate Improvement Guidelines
GS/EL-5648	MARK I Performance Monitoring Products
TR-100215	Field Measurement of Solid Particle Erosion in Utility Steam Turbines
TR-100603	Procedure for Economic Evaluation. of Steam Turbine Drives vs. Electric Drives
TR-104885	Introduction to Nuclear Plant Steam Turbine Control Systems
TR-106230	Main Turbine Performance Upgrade Guideline (final draft)
TR-106345	MSR Source Book (final draft)

Non-EPRI References

Organizations and publishers other than EPRI have material pertaining to this subject. Some of this other reference information is listed below to enable the reader to obtain this material if appropriate.

American Society of Mechanical Engineers. "PTC 6 on Steam Turbines Interpretations 1977-1983." ASME Performance Test Code 6. June 30, 1984.

American Society of Mechanical Engineers. “Guidance for Evaluation of Measurement Uncertainty in Performance Test of Steam Turbines.” ASME Performance Test Code 6 Report-1985. August 31, 1988.

American Society of Mechanical Engineers. “Steam Turbines (Reaffirmed 1991).” ASME Performance Test Code 6-1976. 1976.

American Society of Mechanical Engineers. “Interim Test Code for an Alternative Procedure for Testing Steam Turbines.” ASME Performance Test Codes 6.1-1984. June 30, 1984.

American Society of Mechanical Engineers. “Appendix A to Test Code for Steam Turbines (Numerical Examples of Various Turbine Calculations) - Reaffirmed 1988.” ASME Performance Test Code 6A-1982. June 30, 1983.

American Society of Mechanical Engineers. “Procedures for Routine Performance Tests of Steam Turbines.” ASME Performance Test Code 6S Report-1988. December 15, 1989.

Edwin R. Church. *Steam Turbines*. McGraw-Hill, New York 1950.

Kenneth J. Salisbury. *Steam Turbines and Their Cycle*. John Wiley & Sons, New York 1950.

TPE Handbook References

The following documents contain additional information on steam turbines.

F. G. Bailey, K. C. Cotton, and R. C. Spenser, “Predicting the Performance of Large Steam Turbine-Generators Operating with Saturated and Low Superheat Steam Conditions,” GER 2454A, presented at the 29th Annual Meeting of the American Power Conference, Chicago, Illinois (April 1967).

K. C. Cotton. *Evaluating and Improving Steam Turbine Performance*. Cotton Fact, Inc., Rexford, New York 1993.

Deborah H. Cioffi, P.E., “Using Steam Path Audits in a Performance Guarantee Program,“ presented at the 1997 Performance Software User’s Group Meeting sponsored by Scientech, Inc. and American Electric Power Service Corporation, Columbus, Ohio (June, 1997).

2.6 Moisture Separator Reheaters

2.6.1 General Description

During expansion through the HP section, the moisture content in the steam increases to approximately 12% at the HP turbine exhaust. Moisture in the cycle steam reduces the mechanical efficiency in the LP turbines and causes erosion of LP turbine components in the steam path. The function of MS/Rs*, located between the HP and LP sections, is to remove moisture from the steam in the HP turbine exhaust and, in the case of MSRs, to reheat the dried steam before it flows into the LP turbine sections. Moisture separation and reheat, however, do not improve the thermodynamic efficiency of the LP turbines because moisture removal and reheat, using main and extraction steam, do not increase the overall availability of energy in the steam cycle. In fact, they cause small reductions in thermodynamic efficiency. If it were not for the fact that reduced moisture improves mechanical efficiency, moisture separation and reheat would not improve turbine heat rate. This benefit was not well recognized during the period when many of the operating plants were designed. Since startup, several plants have had to replace LP rotors and stationary internals. A major reason was erosion and corrosion damage caused by moisture.

Although all MSR designs are based on the concepts of moisture separation and reheat, the designs may vary in the details. The description below is also applicable to MSs to the extent that they use the same principles of moisture separation and use large vessels.

Figure 2-17 shows a simplified cutaway schematic of a two-stage MSR. The shell is approximately 6' in diameter and 50' long, and usually is designed to be located in a horizontal position. The vessel is designed to contain moisture separation equipment and tube bundles used to reheat the cycle steam. The vessel is larger than the tube bundles in order to produce low cycle steam velocities needed for efficient moisture separation. Figure 2-17 shows two tube bundles. In the case of single reheat, there is one tube bundle per shell, and no tube bundles in the case of MS only. However, even if the vessel is MS only, it is large due to the need to have low cycle steam velocity in order to achieve high moisture separation effectiveness.

---

* When this report is referring to moisture separators with reheat capability, the abbreviation MSR is used. When referring to moisture separators without reheat capability or the moisture separator section of an MSR, the abbreviation MS is used. When referring to both types, MS/R is used. MS/Rs are used only in light water reactor nuclear power plants; they are not used in fossil power plants.

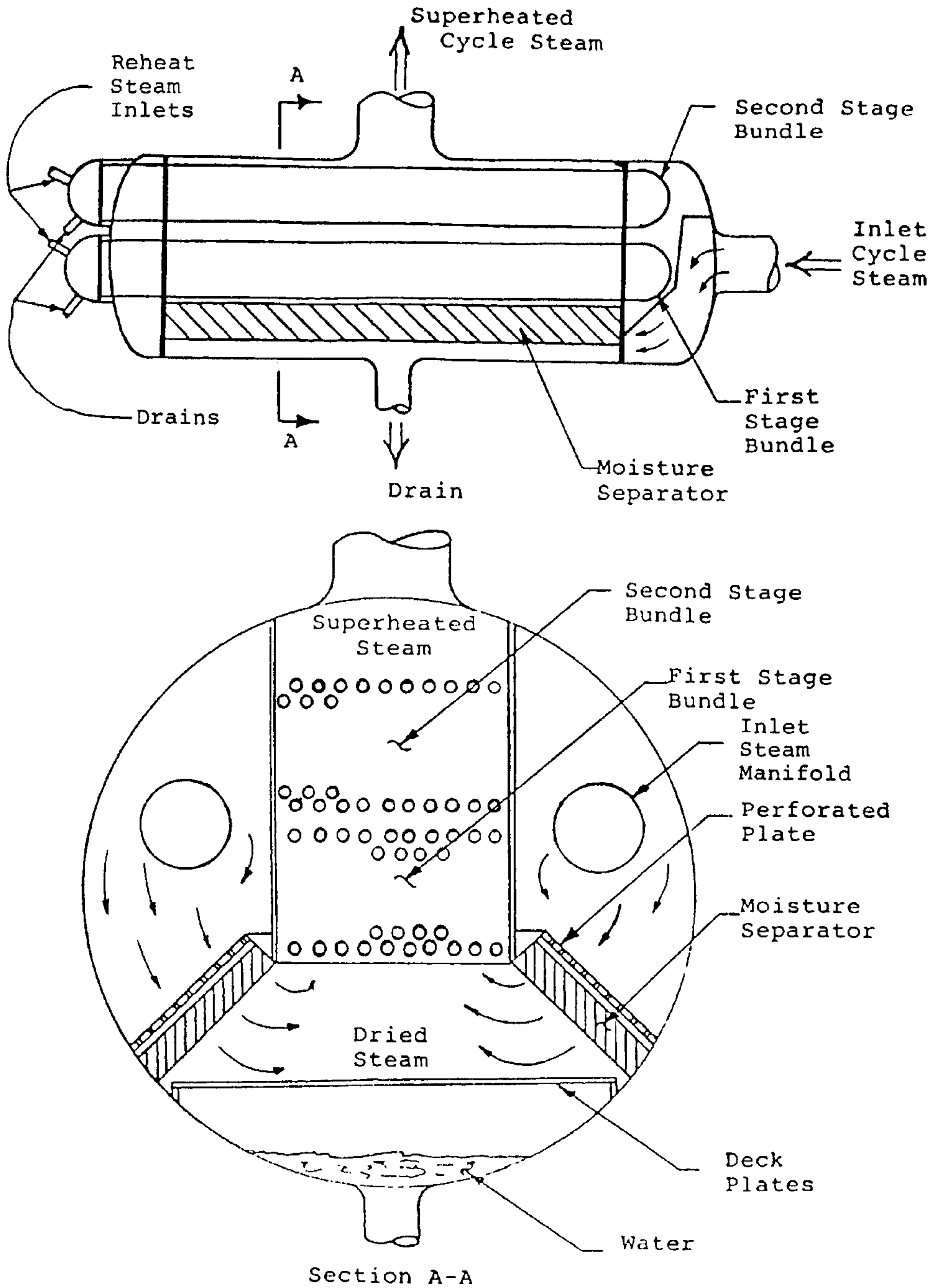


Figure 2-17
Typical Two-stage MSR

The vessel has several large piping connections, usually one inlet connection at the end or one or more inlet connections at the bottom of the vessel for inlet cycle steam, and several cycle outlet steam connections on the top of the cylindrical part of the vessel.

The reason for multiple outlet nozzles is to aid in distributing cycle steam flow uniformly across the moisture separators and reheat tube bundles and to provide multiple feeds to the LP turbines.

The vessel usually has one or more drain connections at the bottom of the shell to drain water removed from the inlet cycle steam by the separators. The water, at saturation temperature corresponding to inlet cycle steam pressure, is directed to a drain tank and then to a feedwater heater stage, operating at less than cycle steam pressure in the MS/R.

MSR vessels also have nozzles and channels that supply reheat steam to the tube side of the reheat tube bundles, along with channels and nozzles that drain condensed reheat steam from the outlet of the tube bundles. The condensate, usually at saturation temperature, normally drains to a feedwater heater.

The most common form of moisture separator is the chevron vane type shown in Figure 2-18. The chevrons are arranged to provide large steam passageways that achieve low steam velocity and make changes in flow direction, which helps to separate the moisture from the steam by the change in momentum. They also provide a large amount of contact surface to collect moisture in the low velocity steam.

Cycle steam is reheated using heating steam and tube bundles located in the MSR vessel. There are two sources of heating steam depending on the number of reheat stages:

- **Single-Stage Reheat** This form of reheat usually uses one tube bundle per MSR vessel with the main steam header as the source of higher temperature heating steam. Some large MSRs have two tube bundles.
- **Two-Stage Reheat** This form of reheat usually uses two tube bundles per MSR vessel with an HP turbine extraction point as the source of heating steam for the first or “LP” stage and the main steam header as the source of the second or “HP” stage heating steam. Some large MSRs use two sets of tube bundles for each stage.

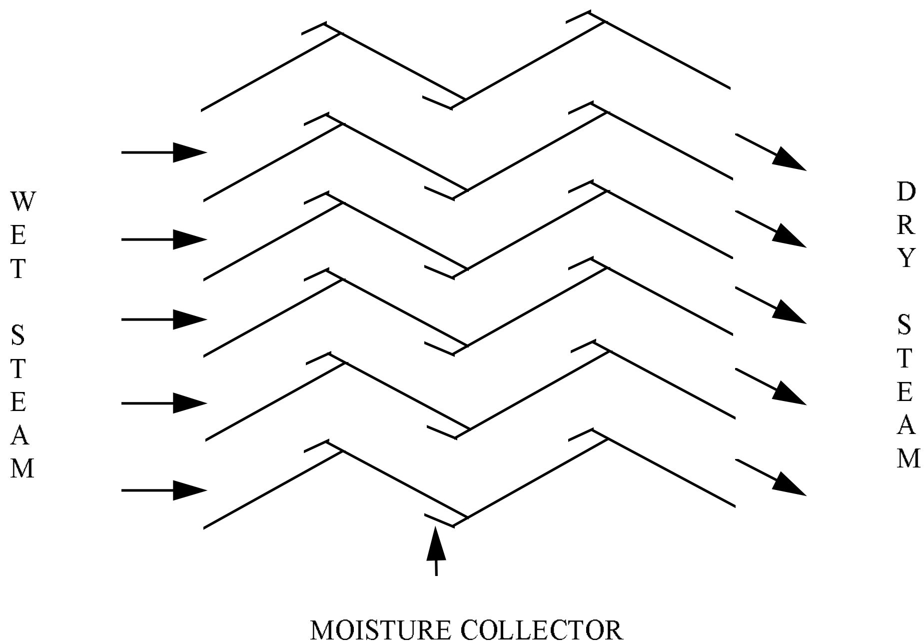


Figure 2-18
Chevron Moisture Separator

Both single-stage and two-stage MSRs use U-tubes with most having the plane of the U-tube oriented vertically. The heating steam is on the inside of the tubes and cycle steam is on the outside. Heating steam flows into the top leg of the U-tube and condenses along the length of the tube. The condensate and some steam flows out of the bottom leg of the U-tube. Cycle steam is heated by flowing across the outside of the tubes. The outside heat transfer coefficient for heating steam is not high. For this reason, the outside of the tubes is usually finned with integral, low profile fins. The U-tube bundles are enclosed in shrouds that provide a structure for tube support and direct cycle steam flow up between the tubes. The tube sheets are generally located at the end of the vessel opposite the end that may contain the cycle steam inlet nozzle. The tube bundles extend the length of the MSR vessel. Usually, several cycle steam outlet nozzles are provided along the length of the vessel to enhance uniform distribution of steam flow up through the reheater.

There is usually little instrumentation provided by the original equipment manufacturer (OEM) to monitor the performance of MSRs. This makes diagnostics and isolation of problem areas difficult. Most MSRs have thermocouples or resistance temperature detectors that measure reheater exit cycle steam temperature. These temperatures are often the only direct indication of MSR performance. If temperature is low, the reheater tube bundle heat transfer capability may be low, or the moisture separation efficiency may be low and part of the reheater surface is being used to evaporate carryover moisture.

Table 2-13 lists instrumentation that can be provided on a MSR and the reasons to provide the instrumentation. The instrumentation is discussed in more detail in PTC 12.4. Moisture is not routinely measured. It is possible to set up special tests using tracer techniques, but there is no technology available that can be used to routinely measure moisture content in wet steam.

Table 2-13
Instrumentation for MSRs

Parameter	Purpose
HP heater cycle steam outlet temperature	HP heater TTD and heat balance
HP heater heating steam flow	HP heater heat balance
HP heater steam inlet pressure and temperature (temperature required only if steam is superheated)	Inlet enthalpy and HP heater TTD and heat balance
HP heater outlet flow pressure and temperature	Subcooling, enthalpy, HP heater TTD, and heat balance
HP heater excess steam flow	Adequacy of flow, HP heat balance
LP heater cycle steam outlet temperature	LP heater TTD and heat balance
LP heater heating steam flow	LP heater heat balance
LP heater steam inlet pressure and temperature (temperature required only if steam is superheated)	Inlet enthalpy, LP heater TTD, and heat balance
LP heater outlet flow pressure and temperature	Subcooling, enthalpy and LP heater TTD and heat balance
LP heater excess steam flow	Adequacy of flow, LP heat balance
Moisture separator drain flow	Moisture carryover, MSR heat balance
LP turbine inlet pressure	Cycle steam flow
Feedpump turbine steam flow	Cycle steam flow

2.6.2 Basic Calculations

The following calculation methodology is provided as a simple way to measure the overall performance of an MSR. It is applicable to hand calculations or PC spreadsheets.

Heater Terminal Temperature Difference (TTD)

The TTD is a measure of the ability of the reheater to achieve a high heat transfer rate indicative of high heat transfer coefficients, low fouling, and good heat transfer surface properties. It can be compared with design and previous operating TTDs. An increasing trend indicates degradation in heat transfer coefficient and area, or a decrease in moisture separator effectiveness, or a combination of these factors.

$$TTD = t_{hs} - t_{cso}$$

Where

t_{hs}	=	temperature of the heating steam, °F
t_{cso}	=	reheater outlet cycle steam temperature, °F

Note that it may not be practical to determine TTDs for the first or LP stage of a two-stage reheater since MSRs are normally not instrumented to measure t_{cso} for the LP heater.

An example calculation is shown below:

Input Data	t_{hs} = 535°F t_{cso} = 520°F
Calculation	TTD = 535 - 520 = 15°F

Reheater Log Mean Temperature Difference (LMTD)

The LMTD is the effective temperature difference that is heating the cycle steam in the reheater.

$$LMTD = \frac{(t_{hs} - t_{csi}) - (t_{hs} - t_{cso})}{\ln \frac{t_{hs} - t_{csi}}{t_{hs} - t_{cso}}}$$

Where

t_{csi}	=	temperature of the cycle steam flowing into the reheater

Single-stage HP reheaters and LP reheaters, t_{hs} and t_{csi} are saturation temperatures. In two-stage HP reheaters, t_{csi} needs to be directly measured. In all cases, t_{cso} is directly measured.

An increasing trend indicates degradation in heat transfer coefficient and area, or a decrease in moisture separator effectiveness, or a combination of these factors.

An example calculation of LMTD is shown below:

Input Data	In addition to the data from TTD, the following additional data is needed to calculate LMTD: t_{csi} = 350°F
Calculation	$LMTD = \dfrac{(535-350)-(535-520)}{\ln\dfrac{535-350}{535-520}} = 67.7°F$

2.6.3 Impact on Thermal Performance

Table 2-14 shows that the most gain in cycle efficiency is made by moisture separation; incremental, but important, gains are made with single- and two-stage reheat. Results may vary with individual plant design.

Table 2-14
Typical Moisture and Efficiency Gains

Condition	LP Turbine Inlet Condition Moisture, % or Superheat, °F	LP Turbine Outlet Condition Moisture, %	Gain in Cycle Efficiency %
No MS or reheat	12-15%	24- 30%	-
MS without reheat	0-3%	12- 18%	4- 5%
MS and single stage reheat	140- 160°F	10- 12%	5- 6%
MS and two stage reheat	140- 160°F	10- 12%	6- 7%

2.6.4 Typical Problems, Degradation Mechanisms, and Corrective Actions

Plant operating experience has shown that MS/Rs are susceptible to problems that have required significant modifications. The cause of most of the problems involves moisture entrained in the cycle steam and the potential for it to cause erosion, and large temperature differences and rates of temperature change in MSRs that can cause significant thermal distortion and high thermal stresses.

The original design intent was to achieve a high percentage removal of cycle steam moisture in the moisture separators. There have been several problems related to moisture separation: (1) use of wire mesh for moisture separation; (2) use of carbon steel chevrons; (3) low moisture removal efficiency; (4) poor inlet steam flow distribution; and (5) moisture re-entrainment. Wire mesh and carbon steel chevrons have been largely eliminated. Low moisture removal efficiency, poor inlet steam flow distribution, and re-entrainment remain problems at some plants. Others have remedied these problems by modifying the inlet flow path and changing to more efficient stainless steel chevrons. See the references for more detail.

2.6.5 References

The following is a compilation of documents on the subject of "Moisture Separator Reheaters." They are not necessarily referenced by the preceding text.

P^2EP Technical Library

The Plant Performance Enhancement Program (P^2EP) has a technical library that provides an effective way to share information on plant thermal performance. Listed below are the documents from the P^2EP Technical Library that relate to the subject "MSRs." These documents are available to eligible EPRI members by calling the P^2EP Administrator at 704/547-6024, or via the World Wide Web at http://www.epriweb.com/npg/pse/ptp/index.html.

File #	*Document Title*
0034.0-306	MSCDT Level Control Issue
0034.6-453	Upgrades To Turbine - MSR Hardware For Performance Improvement
0034.6-666	Determination of MSR Moisture Separation Efficiency at the Washington Nuclear Power Station - Unit 2
0090.0-119	MSR Design
0090.0-124	Evaluation of a Moisture Removal Device for Turbine Steam Piping
0090.0-127	Moisture Separator Reheater Performance
0090.0-175	Improved Moisture Separator Reheaters for Nuclear Steam Supply Systems
0090.0-176	Tube Failures in Moisture Separator-Reheater Tube Bundles due to Restrain Thermal Expansion
0090.0-18	MSR Performance Monitoring
0090.0-187	A Few Design Considerations for Moisture Separator Reheaters
0090.0-190	MSR Redesign and Reconstruction at Indiana Michigan Power Company's Donald C. Cook Nuclear Power Plant, Unit 1
0090.0-191	MSR Performance Enhancements and Modifications at St. Lucie Power Plant
0090.0-20	In-Situ Reconstruction of the MSRs at the Beaver Valley Nuclear Power Station
0090.0-203	Moisture Separator-Reheaters: Entering the Second Decade
0090.0-219	Moisture Separator Reheaters
0090.0-224	Optical Probe Measurements of Low Pressure Steam Turbine Wetness and Efficiency at Morgantown Unit 2
0090.0-23	Maintenance and Fuel with Ellison's Steam Calorimeter
0090.0-237	Testing to Detect Changes in Performance of a Steam Turbine Cycle
0090.0-24	Moisture Separator Reheater Design Technology
0090.0-247	Drainage and Venting in a Swirl Vane Moisture Separator Application

0090.0-25	Interim Report #2 on Moisture Separator-Reheater Drain Systems
0090.0-255	Design of Short Manifolds for Lateral Flow Distribution
0090.0-258	Moisture Separator Reheater Replacement Tube Bundle
0090.0-259	MSR Reheater Technology: Improved 4-Pass Vent Chamber and Excess Steam Flow Regulation
0090.0-26	Redesign and Replacement of Connecticut Yankee Moisture Separator / Reheater (MSR) Tube Bundles
0090.0-260	Report on Operational Problems and Modifications of Trojan Moisture Separator Reheaters
0090.0-264	Improvement in Performance and Availability of Currently Operating MSRs
0090.0-267	Progress in Moisture Separator-Reheaters for Nuclear Power Plants
0090.0-269	General Description--Two-Stage Moisture Separator Reheater (Internal Header, Vertical U-Bend Design)
0090.0-27	Moisture Separator Reheater (MSR) Performance Improvement Project
0090.0-350	MSR Testing and Evaluation; PTC-12.4--The ASME Performance Test Code for MSRs
0090.0-373	Equipment Upgrades Increase MW Output at Prairie Island
0090.0-376	Sensitivity of Moisture Separator Reheater Performance To the Nuclear Steam Turbine Generator Cycle
0090.0-38	Beaver Valley MSR Rebuild
0090.0-390	The MSR Evolution
0090.0-4	Reconstruction of the MSRs In-Situ at Beaver Valley
0090.0-402	Moisture Separator Source Book Compendium
0090.0-41	Moisture Separator-Reheaters (MSRs)
0090.0-463	Waterford III Standing Test Procedure: Moisture Separator Reheater (MSR) Performance, Procedure No. NE-8-007
0090.0-470	WNP-2 MSR Program
0090.0-486	WPPSS Plant Efficiency and Reliability Monitoring Programs: Moisture Separator Reheater Performance Monitoring, Procedure No. 8.4.27, Rev. 0
0090.0-610	MSR Literature
0090.0-611	MSR Performance Monitoring Presentation
0090.0-617	Thermo-Hydraulic Simulation Used to Rectify MSR Performance Deficiency
0090.0-618	MSR Flow-Induced Vibration and Fretting-Wear: A Case History
0090.0-647	Turbine Moisture Removal and Reheater Alternatives - Experiences in Performance Improvement
0090.0-649	Excess steam Flow Optimization in Operating MSRs
0090.0-667	MSR Performance Improvement at Prairie Island Nuclear Station
0090.0-732	MSR Optimization Test Report
0090.0-810	PA Preview of the EPRI MSR Source Book
0090.0-843	MSR Installation List
0090.0-844	Experience with the Moisture Preseparator (MOPS) and with the Special Cross-Under Pipe Separator (SCRUPS) in the Leibstadt Nuclear Plant

0090.0-845	Thermo-Flow Induced Tube Damage in Moisture Separator Reheaters
0090.0-846	Startup of Second Stage Reheat for the MSRs
0090.0-847	Moisture Separator Reheaters - Entering the Third Decade
0090.0-848	Feedwater Heater and MSR Performance Monitoring
0090.0-850	Performance Simulation Pinpoints MSR Problem
0090.0-893	PMSR Reconstruction Challenges at KEPCO's 900-MW KORI Units
0098.5-821	Outage Maintenance & Inspections
0099.0-532	P2EP Survey 93-001, MSR Drains Pumped Forward (B&W Plants Only)
0099.0-537	P2EP Survey 93-006, Single Stage MSRs
0099.0-542	P2EP Survey 93-011, Reheater Tube Leaks
0099.0-551	P2EP Survey 93-020, MSR Impingement Baffles
0099.0-558	P2EP Survey 93-027, MSR Shell Drains
0099.0-563	P2EP Survey 94-002, MSR Reheat Valves
0099.0-567	P2EP Survey 94-006, MRES TAG Subgroup MSR Survey
0099.0-568	P2EP Survey 94-007, Moisture Separator Reheater Drain Tank Level Oscillations
0099.0-572	P2EP Survey 94-011, Moisture Separator Effectiveness/Testing
0099.0-590	P2EP Survey 95-011, Flow Loss in Moisture Separator Drain Tank Line
0099.0-591	P2EP Survey 95-012, Moisture Separator Reheater Drain Tank Orientation
0099.0-594	P2EP Survey 95-015, MSR Sourcebook Input
0099.0-596	P2EP Survey 95-017, Moisture Separator Reheater Steam Blanketing System
0099.0-600	P2EP Survey 96-003, MSR Performance
0100.0-521	Nine Mile Point 2 Lost Electrical Generation Summary
0101.0-209	Moisture Separator/Reheaters Performance Monitoring
0102.0-31	ASME Performance Test Code 12.4 on Moisture Separator Reheaters: An Update
0105.0-849	Some Data for Crossflow Pressure Drop in Tube Banks that Contain 3/4 inch (0.0191 m) Integral Low-Finned Tubes with 27 Fins per inch (1063 FINS/m)
0106.0-361	Moisture Separator/Reheater Corrective Maintenance Procedure for Duke Power
0106.0-756	Reheat and Moisture Separator Procedure, No. OP 3317, Rev. 11
0106.0-785	Callaway Plant Engineering Technical Procedure: ETP-AC-01101, Moisture Separator/Reheater Performance Monitoring
0144.0-12	A Method To Monitor High-Pressure Turbine Efficiency and Moisture-Separator Effectiveness In LWR Power Plants Without Tracer Techniques

Formal EPRI Reports/Software

The Electric Power Research Institute (EPRI) has created formal reports and/or software on the subject of “MSRs.” These are available to eligible EPRI members by calling:

- EPRI Document Distribution Center 925/934-4212
- EPRI Software Distribution Center 800/763-3772

Report #	Report Title
NP-3692	Procurement and Operations Considerations for MSRs
NP-5911M	Acceptance Criteria for Structural Evaluation of Erosion-Corrosion Thinning in Carbon Steel Piping.
TR-101772 R1	Electromagnetic NDE Guide for Balance-of-Plant Heat Exchangers, Rev. 1
TR-101846	Second EPRI BOP Heat Exchanger NDE Workshop
TR-103833	Testing and Evaluation of a Moisture Separator Drain Demineralizer at Davis-Besse Nuclear Station
TR-106345	MSR Source Book
TR-106741	Heat Exchangers: An Overview of Maintenance and Operations (NMAC Tech Notes)

Non-EPRI References

Organizations and publishers other than EPRI have material pertaining to the subject of "MSRs." Some of this other reference information has been provided below to enable the reader to obtain this material if appropriate.

American Society of Mechanical Engineers. "Moisture Separator Reheaters." ASME Performance Test Code 12.4-1992. May 24, 1993.

"Predicting Corrosion Damage with the CHECWorks Software Package," *Nuclear Engineering International* (December 1992).

A. L. Yarden and C. W. Tam, "Excess Steam Flow Optimization in Operating MSRs," presented at the EPRI Nuclear Plant Performance Improvement Seminar (August 1994).

A. L. Yarden and C. W. Tam, "The MSR Evolution," *Nuclear Engineering* (December 1994).

A. L. Yarden, and M. W. Thomas, "MSR Performance Improvement at Prairie Island Nuclear Station," presented at the EPRI Nuclear Plant Performance Improvement Seminar (May 1993).

B. Chexal, R. Jones, J. Lang, D. Munson, C. Wood, R. Wang, J. Horowitz, "CHECWorks Integrated Software for Corrosion Control," presented at the ANS/ASME Nuclear Energy Conference (August 1992).

Tubular Exchanger Manufacturers Association, Inc. *Standards of the Tubular Exchanger Manufacturers Association*, Seventh Edition. 1988.

Two Stage Moisture Separator Reheater, General Electric General Description, (Internal Header, Vertical U-Bend Design), January 1982, Revision A. Report GEK-72221A.

TPE Handbook References

The following documents contain additional information on moisture separator reheaters.

Procurement and Operation Considerations for Moisture Separator Reheaters. Electric Power Research Institute, Palo Alto, California: January 1982. Report NP-3682.

Moisture Separator Reheater Source Book. Electric Power Research Institute, Plant Support Engineering, Charlotte, North Carolina: November 1997. Report TR-106345.

2.7 Reactor Power (Calorimetrics)

2.7.1 General Description

The calorimetrics refer to the cycle heat balance calculations that are used to control reactor power. Typical thermal power calculations for a PWR with loops A and B are as follows:

$$\text{Primary Power} = W_{RCS\text{-}A}\,(H_{OUT\text{-}A} - H_{IN\text{-}A}) + W_{RCS\text{-}B}\,(H_{OUT\text{-}B} - H_{IN\text{-}B}) + Q_{PLVS} + Q_{PAMB}$$

$$\text{Secondary Power} = W_{FW\text{-}A}\,(H_{STM\text{-}A} - H_{FW\text{-}A}) + W_{FW\text{-}B}\,(H_{STM\text{-}B} - H_{FW\text{-}B}) + Q_{SAMB} + Q_{BLOWDOWN}$$

Where

Q_{PLVS} = Net miscellaneous losses from letdown, makeup, seal injection and coolant pumps

Q_{PAMB} & Q_{SAMB} = Primary and Secondary Ambient losses

$W_{RCS\text{-}A}$ = reactor coolant flow in loop A
$H_{OUT\text{-}A}$ = enthalpy of reactor coolant leaving reactor, loop A
$H_{IN\text{-}A}$ = enthalpy of reactor coolant entering the reactor, loop A
$F_{FW\text{-}A}$ = feedwater flow, loop A
$H_{STM\text{-}A}$ = steam enthalpy, loop A
$H_{FW\text{-}A}$ = feedwater enthalpy, loop A
$W_{RCS\text{-}B}$ = reactor coolant flow in loop B
$H_{OUT\text{-}B}$ = enthalpy of reactor coolant leaving reactor, loop B
$H_{IN\text{-}B}$ = enthalpy of reactor coolant entering the reactor, loop B
$F_{FW\text{-}B}$ = feedwater flow, loop B

H_{STM-B} = steam enthalpy, loop B
H_{FW-B} = feedwater enthalpy, loop B

The above calculation of primary power is the total main coolant or reactor system power. To calculate the core thermal power, the reactor coolant pump and pressurizer heat inputs would be subtracted.

The reactor power is controlled on secondary power. Uncertainties in secondary power propagate to the reactor power control with a one-to-one ratio. Therefore, if the secondary power is less than feed flow and temperature measurements say it is, the reactor is operating at a lower power than indicated. Therefore, if feedwater flow reads 1% high, the plant thermal power is 1% low. This decrease in reactor power produces a proportional decrease in MWe output. Uncertainty of the core power is limited by technical specifications.

Table 2-15
Instrumentation for PWR Reactor Power

Parameter	Purpose
Feedwater flow	Secondary power
SG pressure	Main steam enthalpy
SG exit temperature	Main steam enthalpy
Feedwater flow	Secondary power
Feedwater temperature	Feedwater enthalpy
SG blowdown flow	Power correction
SG blowdown temperature	Blowdown enthalpy

A BWR is controlled by controlling reactor pressure with the turbine electro-hydraulic control system (EHC) which positions the turbine throttle valves. The core thermal power is calculated by a heat balance performed on the reactor. An alternate computation of core thermal power is provided by the averaging power range monitors (APRM). Calibration of the APRMs is kept updated by the core thermal power heat balance.

The energy balance for the reactor (in consistent units) is:

$$\text{Core Power} = W_{STM} H_{STM} - W_{FWA}H_{FWA} - W_{FWB}H_{FWB}) + Q_{RAD} + Q_{RWCU} - Q_{CP}$$

$$W_{STM} = W_{FWA} + W_{FWB} + W_{CRD}$$

Where

W_{STM}	=	steam (STM) flow rate
H_{STM}	=	steam enthalpy, steam quality assumed for calculation
W_{FWA}, W_{FWB}	=	mass flow of trains A and B feedwater (FW) pumps
H_{FWA}, H_{FWB}	=	enthalpy of trains A and B feedwater
Q_{RAD}	=	core power loss to radiation (RAD)
Q_{RWCU}	=	power loss due to reactor water cleanup (RWCU)
Q_{CP}	=	power added by circulating pumps (CP)

Table 2-16
Instrumentation for BWR Reactor Power

Parameter	Purpose
Feedwater flow	Reactor flow
Feedwater temperature	Inlet enthalpy
Steam pressure	Steam enthalpy
Recirc pump current	Recirc pump power
CRD feedwater temperature	CRD feedwater enthalpy
CRD flow	CRD energy
RWCU temperature	RWCU enthalpy
RWCU flow	RWCU energy

A sample heat rate calculation may also be found in Volume 1, Table 2-2.

2.7.2 Performance Calculations

PWR secondary power calculation using the equations in Section 2.7.1 is shown below as an example.

Input Data	W_{FW} = 7,212,360 lbm / hr H_{STM} = 1244.5 Btu / lbm H_{FW} = 350.1 Btu / lbm Q_{SABM} = 0 W_{BD} = 35,000 lbm / hr H_{BD} = 540 Btu / lbm
Calculation	Secondary Power = W_{FW} (H_{STM} - H_{FW}) + Q_{SABM} + W_{BD} (H_{BD} - H_{FW}) = 7,212,360 x (1244.5 - 350.1) + 0 + 35,000 x (540-350.1) = 6.457 x 10^9 Btu / hr = 1,891 MW

2.7.3 Impact on Thermal Performance

The impact on thermal performance is due to errors introduced into the measurement of feedwater flow. The venturis used to measure feedwater flow become fouled, which produces an increased pressure drop across the venturi for a given flow velocity. The indication is that the feedwater flow is constant when it is actually less. Reactor power reduces to match the actual feedwater flow (calculated secondary power). Fouling of feedwater flow venturis has been widely documented in the industry.

2.7.4 Typical Problems, Degradation Mechanisms, and Corrective Actions

The cause of the fouling is not fully understood. The foulant is a buildup of magnetite and copper. Causes for the buildup have been hypothesized and investigated. Certain changes in plant operation may cause defouling. Defouling causes the feedwater flow measurement to decrease and the reactor power to increase. Therefore, care in compensating for fouling is necessary to preclude exceeding maximum core power including uncertainties. Defouling has been observed following scrams, power reductions, and possibly water chemistry changes (zinc addition).

Fouling is noticeable soon after startup if the venturis have been cleaned, and it continues at a fairly constant rate. Fouling can affect the feed flow measurement by 2% or more during a fuel cycle.

Ideally, there would be a way to prevent the buildup of foulant on the feedwater flow venturis. However, the only way to correct for fouling at present is to compensate for the scale buildup throughout the cycle. Several redundant measurements of feedwater flow are available, and ratios such as condensate flow divided by feedwater flow can be calibrated and trended to indicate degradation of the feed flow measurements.

Some plants have ASME PTC 6 flow sections with ASME nozzles installed. These can be used to calibrate the final feedwater flow venturis at the beginning of a cycle. The ASME nozzles may also foul during the cycle. Therefore, a method of providing continuous compensation throughout the cycle is desired. Ultrasonic flow meters (UFMs) are not subject to fouling and may be used throughout the cycle. However, UFMs have other problems that must be accounted for. UFMs are sensitive to high temperatures and to the method used to attach their transducers to the pipe. They can be used to improve the accuracy of feedwater flow, but further improvements could be obtained with a more accurate and reliable method of measuring flow.

The process used to account for venturi fouling varies in detail depending on the existing equipment and the investment that a plant can justify for additional feedwater flow measurement. The generic approach begins with a baseline calibration of the beginning of a fuel cycle. The calibration basis can be a measurement with clean venturis and / or calibration with clean ASME PTC 6 flow sections. As the cycle progresses, various indicators of fouling can be tracked, which can provide a correction factor for the feedwater flow.

Different approaches are applied to determining a correction factor. If a UFM is installed and calibrated, it can be used to calculate the correction factor. However, the UFM should be validated with calorimetrics and other measurable parameters.

It is possible to use a set of trended parameters as a diagnostic of feedwater flow. One BWR utility baselines and trends the following data:

- Main steam line (MSL) flow (measured)
- Feedwater (FW) flow (measured)
- Condensate flow (measured)
- First stage main turbine pressure
- R_{MSL} = MSL flow ÷ FW flow
- R_{1st} = First stage main turbine flow ÷ FW flow
- R_{cond} = Condensate flow ÷ FW flow

The three ratios will decrease as actual feedwater flow decreases, while indicated flow remains constant. They should increase after one of the conditions that cause defouling.

If economically feasible, flanged ASME PTC 6 flow sections can be cleaned and used to calibrate the flow meters during a cycle.

2.7.5 References

The following is a compilation of documents on the subject of “Reactor Power” or “Calorimetric.” They are not necessarily referenced by the preceding text.

P^2EP Technical Library

The Plant Performance Enhancement Program (P^2EP) has a technical library that provides an effective way to share information on plant thermal performance. Listed below are the documents from the P^2EP Technical Library that relate to the subjects “Reactor Thermal Power” or “Calorimetric.” These documents are available to eligible

EPRI members by calling the P^2EP Administrator at 704/547-6024, or via the World Wide Web at http://www.epriweb.com/npg/pse/ptp/index.html.

File #	***Document Title***
0003.5-172	Final Report, Peer Assessment of NSP's Prairie Island Thermal Performance Program
0003.5-337	Final Report for the Peer Assessment of NSP's Monticello Thermal Performance Program
0003.5-338	Final Report for the Peer Assessment of Union Electric's Callaway Plant Thermal Performance Program
0009.0-609	Meeting Minutes for the 1995 BOPPMG Meeting, Albuquerque, NM
0011.0-454	Use of ASME Nozzles to Provide Feedwater Venturi Adjustment Factor
0012.0-612	Plant Perspective of Steam Flow Calorimetric Implementation
0012.0-730	Prairie Island Modification 95L510 - Plant Calorimetric Improvement Program
0020.3-863	A Mathematical Model for Assessing the Uncertainties of Instrumentation Measurements for Power and Flow of PWR Reactors (NUREG/CR-3659)
0042.0-459	Venturi/Flow Calorimetric Panel Discussion Transcript
0051.0-125	Evaluation of Available Technology in Flow Measurements
0053.0-772	Correcting for Feedwater Nozzle Fouling at Diablo Canyon Nuclear Power Plant
0060.0-408	Surveillance of Calorimetric and Heat Rate Results (Waterford Unit 3). Procedure No. NE-8-004. 10/12/87.
0070.0-773	Seabrook Station Feedwater Heater Digital Level Controllers
0090.0-23	Maintenance and Fuel with Ellison's Steam Calorimeter
0092.0-704	Recovered Lost MegaWatts from Feedwater Nozzle Calibration Project (One Percent Core Thermal Power)
0099.0-540	P2EP Survey 93-009, Reactor T-hot Reduction
0099.0-550	P2EP Survey 93-019, Thermal Performance Models
0099.0-553	P2EP Survey 93-022, Feedwater Flow
0099.0-561	P2EP Survey 93-030, RCS Temperature Streaming
0099.0-574	P2EP Survey 94-013, Proposed P2EP Guideline on Operation at Licensed Power
0099.0-577	P2EP Survey 94-017, Inputs to Secondary Calorimetric
0099.0-579	P2EP Survey 94-019, Diagnosing Feedwater Flow Nozzle Degradation
0099.0-582	P2EP Survey 95-003, Maximizing Reactor Thermal Power
0099.0-582	P2EP Survey 95-003, Maximizing Reactor Thermal Power
0099.0-599	P2EP Survey 96-002, Accounting for Blowdown on Calorimetric Calculation
0099.0-794	P2EP Survey 96-010, Secondary Calorimetric Divergence
0099.0-858	P2EP Survey 97-002, Heat Rate Improvements vs. Power Uprates
0100.0-771	Attaining 99.5% Thermal Performance at Sequoyah Nuclear Plant
0110.0-281	Discussion of "Licensed Power Level"
0110.0-474	WPPSS Plant Efficiency and Reliability Monitoring Program: CRD Hydraulic System Performance Monitoring, Procedure No. 8.4.48. Rev. 0, 1988

0110.0-615	Increasing MW Output in Units with Reduced Main Steam Pressure can be Performed Without Efficiency Gains
0126.0-808	Causes of PWR Steam Generator Thermal Performance Degradation

Formal EPRI Reports/Software

The Electric Power Research Institute (EPRI) has created formal reports and / or software on the subject of "Reactor Thermal Power" or "Calorimetric." These are available to eligible EPRI members by calling:

- EPRI Document Distribution Center 925 / 934-4212
- EPRI Software Distribution Center 800 / 763-3772

Report #	***Report Title***
CS-5486	Non-Radioactive Tracer for Steam Turbine Thermal Performance Tests
GS-7295	Demonstration of EPRI Heat Rate Improvement Guidelines
NP-3915, V-1	Guidelines for Nuclear Plant Performance Data Acquisition, Vol. 1, Data Item Matrix Identification
NP-4990, V-1	Thermal Performance Diagnostic Manual for Nuclear Power Plants, Vol. 1, Development of Thermal Performance Diagnostic Manual
NP-4990, V-2	Thermal Performance Diagnostic Manual for Nuclear Power Plants, Vol. 2, Plant-Specific Conversion Guidelines and User's Manual
TR-100514, V-1	Survey and Characterization of Feedwater Venturi Fouling at Nuclear Power Plants: Vol. 1, Feedwater Venturi Fouling
TR-101256	Identifying Prospective Anti-Fouling Coatings for Venturis (Zeta Potential Measurements of Oxides at Elevated Temperatures)
TR-101388	Feedwater Flow Measurement in US Nuclear Power Generation Stations
TR-101867	MegaWatt Improvement Casebook and Guidelines (Vol. 1)

Non-EPRI References

Organizations and publishers other than EPRI have material pertaining to this subject. Some of this other reference information has been provided below to enable the reader to obtain this material if appropriate.

Memorandum SSINS #0200 from E.L. Jordan, Assistant Director for Technical Programs, Division of Reactor Operations Inspection, IE. Discussion of "Licensed Power Level," August 22, 1980.

Memorandum to Richard W. Cooper, Division of Reactor Projects - Region I, from Phillip F. McKee, Director of Reactor Projects - I / II. Subject: "Response to Region I Request for Technical Assistance Regarding the Adequacy of the Oyster Creek

Methodology for Determining the Reportability of Plant Operation Above the Licensed Core Thermal Power Limit (TAC No. M87769),” July 5, 1995.

TPE Handbook References

The following documents contain additional information on Reactor Power (calorimetrics.)

John Helton, Southern California Edison Company, “Effects of Electrical Isolation of the Feedwater Venturi.”

C. S. Sullivan, Duane Arnold Energy Center, “Feedwater Flow Venturi Fouling Study.”

T. A. Nelson and G. L. Starnes, PG&E Company, “Correcting for Feedwater Nozzle Fouling at Diablo Canyon Nuclear Power Plant.”

3

CYCLE ISOLATION

Cycle isolation is minimization of steam and heated water flow paths that bypass all or part of the turbine cycle. Cycle losses of high enthalpy steam to the condenser can be large. In addition to reducing cycle losses, there are other benefits to a cycle isolation monitoring program. In many cases of leaking valves or poor level control, a maintenance problem can be identified and planning done for corrective action. There are also indirect benefits from having the plant working as designed. Plant personnel are motivated by success in having the plant perform at its best. INPO reports thermal performance index (TPI) as an industry performance parameter and encourages plants to meet industry goals.

Cycle isolation improvement programs typically involve the following:

- Steam trap testing
- Isolation valve testing
- Heat rate monitoring
- Control valve monitoring and maintenance

The major focus is on high enthalpy leaks to the condenser, for example, any valves isolating main steam from the condenser. One of the first places cycle isolation problems may show up is in unaccounted losses in heat rate. Heat rate measurements should be made before and after valve leakage correction to observe if there is improvement to validate the corrective action.

There are several actual and potential steam and heated water flow paths that bypass all or part of the turbine cycle and result in increased heat rate. The flow paths can be divided into two categories: (1) unavoidable flow paths necessary for operation of components in the turbine cycle, but which should be minimized, and (2) avoidable flow paths which should be eliminated.

3.1 Unavoidable Flow Paths

The unavoidable flow paths include leakage through the inner and outer turbine control valve stem packing, leakage across the turbine shaft seals, MSR scavenging steam, and feedwater heater vent valve losses:

- **Turbine Control Valve Stem Leakage**. Turbine control valves usually have an inner and an outer stem packing. Normal valve stem leakage is approximately 0.1% of rated turbine inlet steam flow. Most of this is vented to a heater where it contributes to feedwater heating. The valve outer packing leakage is usually directed through the steam seal regulator to the condenser or the first stage feedwater heater. Deterioration of the outer packing increases this leakage and reduces the inner packing leakoff rate. In the limiting case, the effect on heat rate is approximately the valve stem leakage flow divided by turbine throttle flow discounted by the fact that most of the leakage contributes to feedwater heating. The effect is estimated to be approximately a +0.1% increase in heat rate if valve stem leakage doubles.

- **Turbine Shaft Seal Leakage**. Turbines are usually arranged as double flow having two half-capacity elements with the inlet at the center and the steam expanding toward both ends. As a result, only the low pressure ends require shaft seals. Shaft seals usually consist of inner and outer labyrinth glands. The design shaft seal flow is approximately 17,000 lb/hr, with approximately 70% of the inner packing leakage vented to a feedwater heater and the remainder leaking across the outer seal and vented to the condenser or a low pressure heater via the steam seal regulator (SSR). In the limiting case, the effect on heat rate is approximately the shaft seal leakage flow divided by turbine throttle flow discounted by the fact that most of the leakage contributes to feedwater heating. The effect is estimated to be approximately a +0.1% increase in heat rate if shaft seal leakage doubles.

- **MSR Scavenging Steam**. Many single- and two-stage MSRs are now provided with the capability to continuously vent heating steam from the outlet of the MSR U-tubes. In many cases, this provision was added as a field modification to prevent harmful temperature oscillation of the condensate produced inside the U-tubes. The amount of scavenging steam varies from 2-10% of heating steam flowing to the MSR tube bundle. This steam is either vented back into the cycle steam flowing into the MSRs or is vented to a stage of feedwater heating. One plant estimates that the penalty for scavenging steam is 1.6 MWe or 0.14% increase in heat rate. This loss is set by plant configuration and should not change appreciably during operation.

- **Feedwater Heater Vent Valve Losses.** Feedwater heaters have vent lines to the condenser that are used to remove non-condensable gases. These vent lines are usually small and may be continuously or intermittently opened during operation to vent non-condensable gases to the condenser. These losses are essentially unavoidable. However, restricting orifices in the lines could be oversized or worn from erosion resulting in excess vent flow.

Feedwater heaters are provided with larger vent lines and valves to remove non-condensable gases during startup. Leakage through the vent line shutoff valves or leakage if the valves are left open during operation can have an effect on heat rate similar to dump valve leakage.

3.2 Avoidable Flow Paths

There are many potential paths for steam or heated water to dump directly to the condenser, including leakage through the turbine bypass valves, MSR drain tank dump valves, and heater dump valves. If any of these paths leaks, cycle isolation is not completely effective.

- **Turbine Bypass Valves.** Turbine bypass valves are provided for plant startup and shutdown. In some cases, 80-100% bypass is provided to prevent a scram when a turbine trip occurs. Bypass valve leakage during normal operation is a total loss of steam energy because the leakage flows directly to the condenser. The change in heat rate is +1% for each 1% increase in bypass valve leakage expressed as a percentage of initial turbine steam flow.

- **Feedwater Heater Dump Valve Leakage.** The impact of valve leakage depends on the stage of feedwater heating where the leakage is occurring. The higher the feedwater heater pressure and temperature, the greater the impact.

- **Other Flow Paths.** There are other flow paths that can affect cycle isolation and heat rate. These include feedwater pump, main turbine drain valves, recirculation valves, steam traps, and pipe low point drain valves.

3.3 Steam Traps

Steam traps are normally checked for leakage by walkdowns using thermal and acoustic devices. Steam traps must open to vent non-condensable gasses. Therefore, some flow is to be expected. A normal temperature profile from upstream of the trap to downstream can be used as a baseline to detect changes. The simplest approach is to use a touch thermometer. For difficult to reach traps, it may be cost effective to install one or more thermocouples on the trap.

Acoustic testing of steam traps can be done with a mechanic's stethoscope or ultrasonic listening device. A steam trap is isolated for a few minutes for condensation to build up upstream of the trap. If no flow is detected or if the flow does not stop shortly after the trap is un-isolated, it can be scheduled for repair. Traps that cannot be isolated for repair can be rebuilt during each refueling outage to reduce the potential for failure during the fuel cycle.

Several different steam trap designs are available, and it is not uncommon to encounter misapplication of steam traps in a plant. If a steam trap is not installed properly, is the wrong size, or is not the correct type for the application, unnecessary loss of steam is likely. Some plant thermal performance engineers keep a collection of steam trap catalogs against which they review problem check valves for proper application, installation, adjustment, and maintenance.

3.4 Isolation Valves

A typical BWR might have 60 isolation valves included in the cycle isolation program. One utility has divided their BWR isolation valves into three categories:

- Valves isolating main steam from the condenser
- All valves larger than 4” isolating pressure sources 100 psi to 1000 psi
- All other cycle isolation valves

A cost-benefit analysis indicated the first two categories to be worth instrumenting. Thermocouples were installed downstream of the valves during an outage. The thermocouple wire was run outside the condenser to a connection board where temperatures can be read using a thermocouple reader.

Isolation valves to be tested can be any of the following:

- Steam dump valves
- Level control dump valves
- Relief valves
- Startup vents
- Recirculation valves
- Bypass valves

Some valves can be visually inspected if they have visible leakage paths such as weep holes or drains to the atmosphere. In general, the process is to measure upstream and downstream temperature. Figure 3-1 demonstrates the effect of leaks on the temperature profile upstream and downstream of a valve. Downstream temperature may be lower than saturation if condensate builds up behind the valve and can be subcooled by conduction through or along the pipe. The temperature profiles may be different than shown in Figure 3-1 due to flashing just downstream of the valve and condensation at some point beyond.

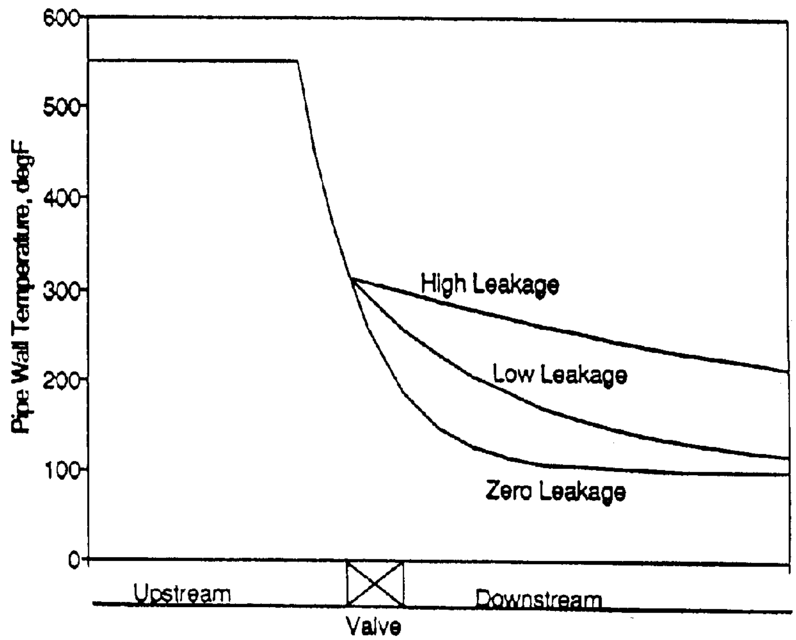


Figure 3-1
Cycle Isolation Valve Leakage Effects

3.5 Air-operated Valves (AOVs)

AOVs are used for level and flow control. Many are in potential cycle isolation leakage paths. AOVs can leak or pass more steam than necessary from any of the following causes:

- Valve seat leakage
- Bad adjustment of a control system
- Mechanical failure
- Steam erosion

- Bypassing
- Valve design misapplication

Problems with control valves can be identified by several methods. The temperature profile can be obtained with a pyrometer. Measurement and trending of downstream pressure can sometimes reveal a problem with control. Monitoring stem position to observe a flow trend can show valve degradation or isolation leakage. For valves in parallel, average stem position is an important parameter because the total flow is proportional to the average. Isolation of a valve can be used to detect problems.

When leakage is suspected in control valves, the following maintenance or replacement may be required:

- Calibration of positioners
- Spring replacement
- Travel adjustment
- Replacement or tightening of packing
- Adjustment or tightening of stem connections
- Adjustment of air supply pressure
- Replacement of eroded seats
- Replacement of bent stems
- Replacement of leaking diaphragms

Testing should be repeated before and after valve repairs.

3.6 Instrumentation

Instrumentation for monitoring and controlling cycle isolation falls into the following categories:

- RTDs and thermocouples installed for continuous monitoring of valves
- Pyrometers for thermography
- Stethoscopes for acoustic monitoring

- AOV diagnostic
- Periodic inspection (orifices)

The applications of these devices have been discussed above. However, some additional discussion is provided in the following paragraphs.

Infrared thermography produces a temperature picture of a valve, piping, or other cycle component. The best application of thermography is using its ability to make comparisons. It can be used to pinpoint a problem that shows up in a heat balance. For routine cycle isolation checks, a touch pyrometer or installed sensor is adequate. Thermography may be cost beneficial for its ability to communicate pictorially a cycle isolation problem.

Acoustic monitoring can be used to assess relative magnitudes of leakage flow through valves. Its use before and after valve overhauls can enhance cost-effective maintenance. In some cases, costly repairs can be avoided. The flow turbulence downstream of the seat is detected to frequencies in the 200 kHz range. One such system determines a measure of leakage by subtracting the measured valve background noise level from the measured leakage noise pattern. It may also be used to differentiate between acoustic signatures from two probes at different locations upstream and downstream of a valve. An increase downstream indicates leakage. The effectiveness of determining leakage versus isolation is dependent on proper location of the probe and the analysis of the acoustic signature patterns versus frequency.

Another device measures valve stem travel and actuator air pressure to identify actuator operating characteristics. The signals are downloaded to a floppy disk for analysis. From these measurements, the device can diagnose most of the control valve adjustments. Using these results allows the control valves to be adjusted to maintain design conditions for the valve. Proper adjustment increases valve life and reduces cycle losses.

3.7 Loss Quantification

Quantifying losses from faulty cycle isolation depends on the parameters that can be measured or calculated. For example, the leak rate past a valve or steam trap may be difficult to determine. Leaks of high enthalpy steam to the condenser have the largest effect on heat rate. The effect on heat rate becomes less as the enthalpy decreases or the leak is to a location in the cycle where some of the energy is recovered.

Cotton provides the following tables for leakage to the condenser.

Table 3-1
Effect of 1% Leakage to the Condenser on Heat Rate

Origin of 1% Leakage Flow	Fossil Reheat	Nuclear Non-Reheat	Nuclear Reheat
Throttle	0.83%	0.91-1.09%	0.81-0.96%
HP TB exhaust	0.53%	0.56%	0.56%
Ahead of intercept valve	0.69%	0.56%	0.60%
X-over	0.44%	0.56%	0.60%

Table 3-2
Effect of 1% Leakage to the Condenser on the Load

Origin of 1% Leakage Flow	Fossil Reheat	Nuclear Non-Reheat	Nuclear Reheat
Throttle	0.94%	0.91-1.09%	0.81-0.96%
HP TB Exhaust	0.69%	0.56%	0.56%
Ahead of intercept valve	0.69%	0.56%	0.60%
X-over	0.44%	0.56%	0.60%

3.8 References

Bailey, K. C. Cotton, and R. C. Spencer, “Predicting the Performance of Large Steam Turbine-Generators Operating with Saturated and Low Superheat Steam Conditions,” GER 2454A, presented at the 29th Annual Meeting of the American Power Conference, Chicago, Illinois (April 1967).

Trojan Thermal Performance Program. PSEP 30-1.

K. C. Cotton. *Evaluating and Improving Steam Turbine Performance.* Cotton Fact, Inc., Rexford, New York 1993.

The following is an additional compilation of documents on the subject of “Cycle Isolation.” They are not necessarily referenced by the preceding text.

P^{2}EP Technical Library

The Plant Performance Enhancement Program (P^{2}EP) has a technical library as an effective way to share information on plant thermal performance. The list below is from the P^{2}EP Technical Library, specifically those documents with the keywords of

"Cycle Isolation." These are available to eligible EPRI members by calling the P^2EP Administrator at 704/547-6024, or via the World Wide Web at http://www.epriweb.com/npg/pse/ptp/index.html.

File #	***Document Title***
0003.5-172	Final Report, Peer Assessment of NSP's Prairie Island Thermal Performance Program
0003.5-299	Final Report for the Peer Assessment of TU Electric's Comanche Peak Thermal Performance Program
0003.5-337	Final Report for the Peer Assessment of NSP's Monticello Thermal Performance Program
0003.5-338	Final Report for the Peer Assessment of Union Electric's Callaway Plant Thermal Performance Program
0030.0-721	Oyster Creek's Cycle Isolation Program
0030.0-776	Indian Point 2 Cycle Isolation Monitoring
0030.0-777	Cycle Isolation Techniques
0030.0-879	Waterford 3 Technical Procedure NE-008-012, Rev. 1, Turbine Cycle Isolation
0030.0-906	The Other Side of Turbine Cycle Isolation - Continuous Flow Orifices
0060.0-128	Heat Rate Recovery by Cycle Isolation and Control Diagnosis and Correction
0060.0-174	Heat-Rate Demonstration Project, Salem Harbor Station Unit 4, EPRI Report GS-7329
0099.0-570	P2EP Survey 94-009, Estimating Loss Through Steam Dumps
0102.0-349	PTC 6S Report-1988: Procedures for Routine Performance Tests of Steam Turbines
0106.0-737	Point Beach Thermal Performance Procedure TP 5.0, Rev. 0, Cycle Isolation
0131.4-468	Waterford Unit 3 Technical Procedure: Steam Trap Leakage, Procedure No. NE-008-010, Rev. 1
0138.0-192	Infrared Thermography As A Thermal Performance Tool: Myth Or Magic?

Formal EPRI Reports/Software

The Electric Power Research Institute (EPRI) has created formal reports and/or software on the subjects of "Cycle Isolation." These are available to eligible EPRI members by calling:

- EPRI Document Distribution Center 925/934-4212
- EPRI Software Distribution Center 800/763-3772

Report #	***Report Title***
AP-3651	Thermo-Economic Analysis of Power Plants
CS-1554	Biofilm Development and Destruction
CS-4014	Proceedings: 1983 Fossil Plant Heat Rate Improvement Workshop
CS-4410	Demonstration of an Alternative ASME Steam Turbine Generator Acceptance Test

CS-4545-SR	Proceedings: 1984 Power Plant Performance Monitoring Workshop
GS-6635	Proceedings: 1988 EPRI Heat-Rate Improvement Conference
IN-106658	Innovator, Using CHECWORKS to Enhance Valve Seat Leakage Monitoring Programs
NP-3395	Calculation of Leak Rates Through Cracks in Pipes and Tubes
NP-4990, V-1	Thermal Performance Diagnostic Manual for Nuclear Power Plants, Vol. 1, Development of Thermal Performance Diagnostic Manual
NP-4990, V-2	Thermal Performance Diagnostic Manual for Nuclear Power Plants, Vol. 2, Plant-Specific Conversion Guidelines and User's Manual
TR-101867	MegaWatt Improvement Casebook and Guidelines (Vol. 1)
TR-105416	Qualified Life of Main Steam Isolation Valves

4

DATA VALIDATION

4.1 Introduction

The foundation of thermal performance monitoring is data. Each performance parameter for each system in the plant thermal cycle has data requirements that must be satisfied if valid system performance is to be measured. Data requirements include the following:

- Proper parameters for the performance measurement
- Required accuracy
- Sampling frequency

Measuring the proper parameter includes the measurement location as well as the requirement for the measurement in the performance calculation. Developing accuracy requirements includes uncertainty based on the sensitivity of the performance calculations to errors in the parameter being measured. Sampling frequency requirements are based on the rate at which thermal performance is expected to undergo a significant change.

4.2 Data Requirements

The cost of obtaining measurements and tracking a measurement must be compared to the potential benefit for performance improvement. The data needed to calculate a parameter that has a significant effect on performance have a high monetary value. However, data are not very useful if the uncertainty of measurement is greater than accuracy required to indicate a performance change.

The uncertainty of a data point is usually stated as a range and a probability. One component of the range of uncertainty is the precision that includes repeatability and resolution, and the second part of the accuracy range is measurement bias. Bias is a constant error that can be accounted for by calibration. Both precision and bias must be accounted for in accuracy. For example, main steam pressure might be given as 1000 psia ±10 psia (P=.95). This example means that considering both precision and bias, 95% of the pressure measurements would be expected to fall with ±10 psia of the actual pressure. The percent accuracy of a measurement is dependent on the input value. In this case, at 1000 psia the accuracy is ±1%. In general, one must be careful that the measurement accuracy is suitable for the value of the input parameter.

Calibration of a measurement device is done to reduce measurement bias. Measurement calibration is done to a known value or standard. The accuracy of the standard must be greater than the accuracy of the measurement device being calibrated.

A general figure of merit for plant performance is a 1% deviation in the measured heat rate. Table 4-1 gives the approximate change in heat rate for a given deviation of a measured parameter value in the cycle. This table does not account for the effect of deviations of more than one parameter.

Table 4-1
Effect of Instrument Accuracy

Parameter	Estimated Change in Heat Rate
Exhaust pressure	1.25-2% per inch Hg
Main condensate flow	1% per % flow
Power	1% per % power
Throttle flow	1% per % flow
Extraction line ΔP	0.66% per % ΔP
Miscellaneous	0.15%
Final feed temperature	0.12% per °F
Hot reheat pressure	0.08% per % psi
Throttle temperature	0.07-0.12% per °F
Cold reheat pressure	0.08-0.05% per % psi
Turbine extraction pressure	0.05% per psi
Cold reheat temperature	0.04% per °F
Throttle pressure	0.084-0.02% per % psi
Hot reheat temperature	0.05-0.13% per °F
Feedwater pump enthalpy rise	0.006% per % enthalpy rise
Increase in feedwater heater temperature	0.006% per °F
Increase in heater approach temperature	0.0008% per °F
Increase in feedwater pump seal flow	0.0007% per % seal flow

Table 4-2 demonstrates the combined uncertainties in an ASME PTC 6 alternative test. Group 1 correction calculations are defined in ASME PTC 6 for six parameters.

Table 4-2
Alternative Test Uncertainty
Nuclear Feedwater Primary Flow Measurement

Parameter	Effect of Errors on Heat Rate	Measurement Uncertainty	Uncertainty HR	HR^2
Temperatures				
Feedwater flow	0.12% per °F	1°F	0.12	0.0144
Throttle quality	1% per 1%	0.1%	0.1	0.0100
Flows				
Primary	1% per 1%	0.15%	0.15	0.0225
Feed pump turbine	0.02%	2%	0.04	0.0016
LP exhaust pressure	*0.2%/in. Hg	0.1 in. Hg	0.02	0.0004
Power	1% per 1%	0.1%	0.1	0.0100
Isolation	1% per 1%	0.05	0.05	0.0025
Group 1 corrections	1% per 1%	0.1%	0.1	0.0100
		Σ H.R.^{2}		0.0714
OVERALL TEST UNCERTAINTY		$\sqrt{\Sigma HR^2}$		**0.27%**

**NOTE: Based on the ASME PTC 6 permissible LP exhaust pressure deviation from design of "0.05 psi or 2.5% of absolute pressure, whichever is larger."*

In Table 4-2, the total test uncertainty has been propagated using the square root of the sum of the squares of the individual uncertainties. For the objectives of a typical thermal performance program, a PTC 6 test with an overall test uncertainty of ±0.27% (P=.95) is probably not cost effective. However, an overall heat rate uncertainty goal in the range of 1% is achievable with the instrumentation installed in most plants. ASME PTC 19-1 contains a detailed discussion of measurement uncertainty.

The final major data requirement of concern to the thermal performance engineer is sampling frequency. Sampling frequency depends on the trend being measured as part of the thermal performance program. Other requirements for the data at the plant may dictate a higher sampling frequency. Volume I, Section 3.4.4 shows some trending examples that demonstrate the value of, and an approach for, selecting an effective sampling frequency.

4.3 Validation

The first step in data validation is to detect gross instrument or sensor failures. Criteria for identifying these failures are out-of-range indications, large inconsistencies with other system parameters, and excessive rates of change in measured values for the processes involved.

Analytical redundancy calculations can be performed using data that have survived the initial check of the above criteria. This technique is often referred to as *reconciliation*. These calculations can be done using basic mass and energy balance equations. A computer-generated heat balance model or a dynamic simulation can also be used. Data filtering techniques are also available that can smooth data and fill in data missing from a faulty source. The choice of the data validation method used should be based on the characteristics of the parameter, the number of data samples, and the characteristics of the instrumentation.

Analytical redundancy does not provide new information, but takes advantage of existing data. For example, feedwater mass flow rate equals condensate pump mass flow rate plus heater drain mass flow rates not returned to the condenser hot well. Therefore, indication of a change in one of these flows would be expected to show up as a change in one of the others.

Limit checking can detect gross failures in a single instrument channel. If the signal is either outside the measurement span of the instrument or outside the expected limits of the signal, a gross failure is indicated.

A like-sensor comparison can also be used to detect gross failure of a sensor. If three sensors are measuring the same parameter and more than half of the measurements fall within the expected band, a sensor with a large deviation from the other two can be assumed to be failed.

Instrument loop integrity checking tests the entire instrument circuit. The specific items checked depend on the loop design. Loop resistance and insulation resistance are commonly checked. Digital I&C systems are able to check for failures continuously and reboot, shift to an alternate processor, or reload software automatically.

Calibration is part of signal validation. Calibration can be performed manually or automatically if the instrumentation loop is digital. Calibration is the comparison of a measurement to a standard and removes measurement bias from the data. The bias is usually due to instrument loop drift. Digital systems are typically drift free except for sensor drift. However, safety-related loops are calibrated periodically per the technical specifications.

Statistical methods can be applied to test a few data points that lie outside the expected range. These data may be due to a few faulty readings which, for example, are not uncommon for thermocouples.

One statistical method, Chauvenet's Criterion, specifies that a reading may be rejected if the probability of obtaining the deviation of that sample from the mean is 1/2n, where n is the number of samples. Table 4-3 can be used to test a sample for rejection.

Table 4-3
Chauvenet's Criterion for Rejecting a Reading

Number of Readings n	Ratio of Maximum Acceptable Deviation to Sample Standard Deviation S
10	1.96
15	2.13
25	2.33
50	2.57
100	2.81
300	3.14
500	3.29
1000	3.48

The standard deviation is calculated by:

$$S = \sqrt{\frac{1}{1-n}\sum_{i=1}^{n}\left(x_i - \bar{x}\right)^2}$$

Where $\bar{x}$ is the mean of the samples,

$$\bar{x} = \frac{\sum_{i=1}^{n} x_i}{n}$$

An example application of Chauvenet's Criteria is provided by Figure 4-1.

Sample Number	Sample Reading	Deviation from the Mean	Ratio of Standard Deviation to Deviation	Rejected Samples Based on Maximum of ±1.96 n=10
1.00	988	-1.70	-0.22	
2.00	987	-2.70	-0.34	
3.00	987	-2.70	-0.34	
4.00	990	0.30	0.04	
5.00	995	5.30	0.67	
6.00	997	7.30	0.93	
7.00	993	3.30	0.42	
8.00	996	6.30	0.80	
9.00	970	-19.70	-2.51	Rejected
10.00	994	4.30	0.55	
Mean	**989.70**			
Standard Deviation	**7.86**			

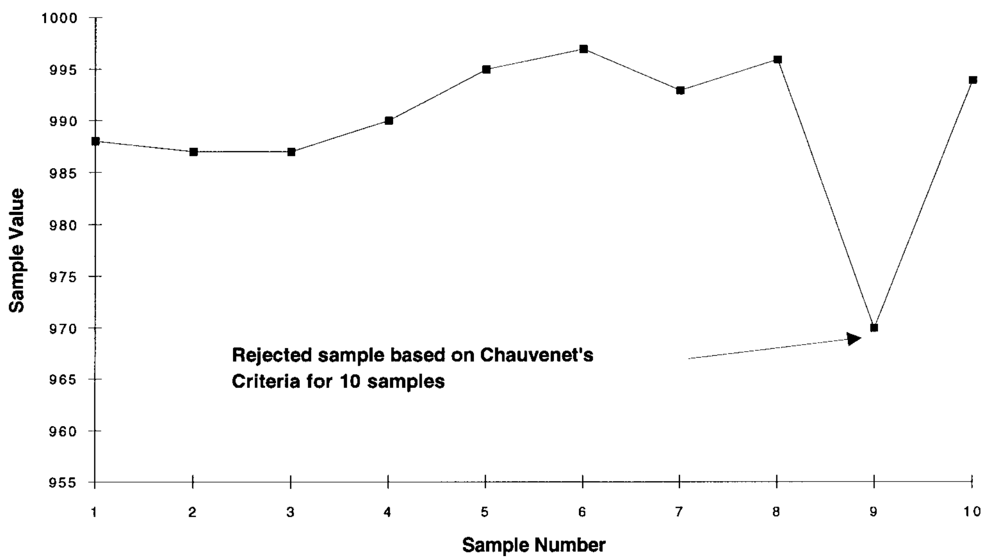


Figure 4-1
Example Application of Chauvenet's Criteria

Chauvenet's Criteria may be used for manually recorded data or programmed into digitally obtained data. Combustion Engineering (CE) Qualified Safety Parameter

Display Systems use this technique [1]. Other data validation schemes are being applied to digital instrumentation, control, and monitoring systems. The thermal performance engineer should be aware that data from these systems may have been filtered to remove potentially false data.

4.4 Practical Applications

Perhaps the most important use of data validation is to determine if observed problems are real or are the effect of drifting or inaccurate instrumentation. As discussed above, this type of validation is performed by comparing redundant instrumentation (either installed or temporary), trending of calculated parameters, comparing the current indication with expected effects on other parameters, or using a heat balance program to validate the instrument readings observed in the plant. Some practical data validation examples are provided in the following subsections.

4.4.1 Main Steam Pressure

Main steam pressure indication is very important to all power plants. Due to the importance of this parameter, there are usually several instruments measuring this parameter. For our example, let’s assume there are 3 pressure indications on each of 4 main steam lines for a total of 12 pressure indications. Let us also assume the accuracy of these instruments is 2% of full span (instruments range 0-1500 psig) with normal operating pressures of 1000 psig. One could use the example in Figure 4-1 to provide a statistical review of the data. An alternative trending method is to track the difference between the average of the 12 pressure indications and each individual pressure indication. For a 1000 psig normal reading it has been observed by one plant engineer that this difference should track within 3 psi, which is much better than the 2% accuracy of the instrument which would be 30 psi. For example, if one pressure transmitter is indicating 1003 psig and the average of the 12 instruments is 998 psig, the trended parameter should be -5 (998 psig - 1003 psig) and should remain between -2 and -8 (-5 -3 = -8, -5 + 3 = -2). This trending is easier than computing the standard deviation of a group of instruments and remains consistent over a wide range of pressure readings.

4.4.2 Feedwater Flow Rate

Most plants have precision flow measurement devices installed to accurately determine cycle heat input. In addition to these precision instruments there are usually other flow measurement devices used to determine the performance of individual pumps, for example condensate heater drain and main feedwater pumps. Although these instruments do not have the accuracy of the precision instruments, they generally trend consistently. A good group to trend would be the sum of condensate flows, the sum of heater drain flows, the sum of feedwater pump flows, the sum of condensate flows plus heater drain flows, and the total feedwater flow as measured with the precision instruments. An additional parameter to monitor would be the difference between total feedwater flow and the sum of condensate and heater drain flows. By trending this difference, flow changes will be easier to spot. With a total flow indication of about 15,000,000 lbs/hr, it has been observed that this difference should track within 40,000 lbs/hr. By trending total condensate flows and heater drain flows, one can detect a shift in mass flow from the high pressure portion of the plant to the low pressure portion of the plant (heater drain flow will decrease and condensate flow will increase by the same amount if a high pressure heater emergency drain opens). If there is a change in either condensate or heater drain flow without a change in total feedwater flow, then look for an invalid indication of flow rate or the possibility of a feedwater heater tube leak or other ways of short circuiting flow. A drifting transmitter could be identified by swapping pumps. If the apparent bad pump is not shut down, the total flow rate will not change. If the pump that is shut down is bad, the total flow rate will change. If there is an actual change in total feedwater flow rate, either condensate flow, heater drain flow, or both will need to increase.

4.4.3 Calculated or Measured Heater Drain Flow

When instrumentation allows, a calculated heater drain flow should be compared with heater drain pump flows (what comes in must go out). This calculated flow could be determined from a heat balance on a drain cooler just up stream of the heater drain tank, from heat balances from upstream feedwater heaters, or estimated from heater drain control valve position and differential pressure.

4.4.4 Final Feedwater Temperature

A consistency check between final feedwater temperature and last stage feedwater heater outlet temperature can be useful. This check could be performed locally if necessary. Because this is the same water, less than 1°F deviation should be expected if the measurements are valid.

4.4.5 Condenser Performance

Because the main condenser is saturated, there should be a good correlation between its pressure and temperature. Compare calculated temperatures from pressure with measured temperatures or hotwell temperature where applicable. This may be difficult in some instances due to inaccurate condenser pressure readings. The presence of non-condensable gasses will also affect the saturation temperature comparison.

4.4.6 Feedwater Heater Performance

Redundant instrumentation checks, whether temporary or installed, can be used to validate data. If a feedwater heater outlet temperature decreases by 3°F, the heater drain temperature from the next higher pressure heater should also decrease (this is assuming the higher pressure heater has an integral drain cooling section). If there is no drain cooling section, the colder water entering the higher pressure heater affects the performance of the drain cooling section, thus reducing the drain outlet temperature. One TPE looks for the heater drain temperature of the higher pressure heater to decrease about 2°F if the heater outlet temperature decreases by 3°F. If this does not happen, the decrease in heater temperature could be a false indication.

4.4.7 Comparison with Heat Balance Program

Heat balance computer programs can help validate data when redundant instrumentation is not available. One example of this was when a plant observed a sudden decrease in extraction pressure from a low pressure feedwater heater along with a decrease in generator output. The heat balance program was used to simulate a failed extraction bellows inside the main condenser. The analysis indicated that the loss of mass from the extraction line required to decrease feedwater heater pressure would also decrease generator output by the amount observed. In another instance, there was a brief plant transient (total of 5 minutes) that resulted in changing feedwater heater inlet temperatures. In this case, there was no chance of validating in-plant data with local readings. The heat balance program was used to simulate a failing open of one of two heater drain pump flow control valves. This analysis mimicked the actual data from the plant instruments and validated the observed event.

4.5 References

1. *Guidelines for Nuclear Plant Performance Data Acquisition, Volume 2: Data Item Requirements.* Electric Power Research Institute, Palo Alto, California: October 1985. Report NP-3915.

2. American Society of Mechanical Engineers. Steam Turbines. ASME Performance Test Code 6-1996.

3. K. C. Cotton. *Evaluating and Improving Steam Turbine Performance.* Cotton Fact, Inc., Rexford, New York, 1993.

4. American Society of Mechanical Engineers. Measurement Uncertainty. ASME Performance Test Code 19.1-1985.

5

MOISTURE MEASUREMENTS

A large part of the steam path in typical nuclear steam cycles is in the moisture region. Moisture makes it impossible to directly measure enthalpy in the HP and LP exhaust, turbine extraction lines, and the moisture separators. Also, moisture affects turbine efficiency and the effectiveness of MSR reheat.

Determination of steam quality and enthalpy is possible using several techniques:

- Radioactive-tracer technique
- Heater-drain-flow measurement and heat balance
- Throttling calorimeter methods

The first two methods and some of their limitations are discussed in ASME PTC 6. The third method is discussed in PTC 19.11. None of the methods are considered on-line and suitable for routine monitoring. They are in the nature of special measurements suitable for a PTC 6 acceptance test. Moisture measurement is not required for a PTC 6.1 test and should not be used for diagnostic component testing unless justified.

Tracer and throttling calorimeter methods are applicable to measurement of carryover from both steam generators and moisture separators. The samples should be taken from locations near the point of interest. Locations such as elbows should be avoided because elbows tend to introduce swirl and separation of the moisture to the outside of the pipe. On the other hand, moisture tends to flow closer to pipe walls and the bottom of long pipe runs. Steam samples should be taken in accordance with ASTM D 1066 or as described in ASME PTC 19.11-1970:

Throttling calorimeters operate on the principle that the initial and final enthalpies are equal when steam passes through an orifice from a higher to a lower pressure, providing there is no heat loss and the initial and final kinetic energies are negligible. Throttling calorimeters have a limited range of use which varies with pressure (see ASME PTC 19.11).

5.1 Main Steam Line Moisture Carryover

Table 2-1 of Volume I of this handbook shows that a 0.1% increase in main steam line or HP turbine inlet moisture content increases heat rate by about 0.03%. On the other hand, the design moisture content for most nuclear power plants is approximately 0.2%. A 50% increase in main steam line moisture or +0.1% moisture increases heat rate by approximately 0.03% and can decrease plant output 0.3 MWe on a 1000 MWe plant. The changes described are the actual effects. A 0.1% error or uncertainty in the **measurement** of main steam line moisture can have a two to three times higher effect on **calculated** HR and plant output.

A change in the magnitude described above is not normally expected unless there have been significant modifications to the PWR steam generator or BWR reactor vessel dryers or visual inspections reveal substantial degradation of the dryer equipment. As a result, main steam line moisture measurement is not done frequently. In the event that main steam line moisture needs to be measured, then the tracer method is the only practical approach to the problem. This method is described in Paragraphs 4.5.3 and 4.6.5 of ASME PTC 12.4-1992.

5.2 MS/MSR Outlet

Moisture carryover in the cycle steam leaving an MS can reduce mechanical efficiency in the LP turbine; in an MSR, it can reduce reheat temperature and increase heating steam consumption since the moisture is evaporated using some of the heat transfer surface of the reheater tube bundle provided to superheat dry steam. Measurement of moisture separation effectiveness* can be used to determine if the moisture separator is not meeting design expectations or if the major cause of off-performance is associated with the reheater. At this time, there is no simple, inexpensive method to measure steam quality as cycle steam exits the moisture separators. Several methods discussed in PTC 12.4 are outlined below along with their limitations:

- **Reheater Energy Balance Method** This method uses an energy balance conducted around each reheater stage to determine the specific enthalpy (moisture content) of the cycle steam entering each reheater stage. This method requires that all flows entering or exiting each reheater stage, as well as interstage temperature (enthalpy) in a two-stage MSR, be accounted for and used in the determination of specific enthalpy (moisture) leaving the moisture separator and entering the reheaters. This method is not applicable to units that have only MSs. Furthermore, most MSRs are

* Moisture separation effectiveness is defined in PTC 12.4 as the ratio of the mass flow rate of moisture removed from the entering cycle steam to the mass flow rate of moisture entering the separator. A moisture separation effectiveness of 100% means total removal of moisture from the incoming steam.

not provided with the extensive amount of instrumentation required by this method. Most of the instrumentation required for the energy balance method is needed for a full MSR test described in PTC 12.4.

- **Throttling Calorimeter Method** This method uses throttling calorimeters and steam sample probes installed at the outlet of the moisture separators to obtain representative samples. The probes need to be designed so they do not preferentially sample either steam or moisture, but sample in proportion to the steam and water content of the cycle steam exiting the moisture separator. Also, there needs to be sufficient sample points to account for variations in exit quality over the large exit area of the moisture separator. In addition to PTC 12.4, see PTC 19.1 for guidance on this method.
- **Differential Tracer Method** This method uses a two-step process. The first step uses the tracer method to measure the moisture separator shell drain flow by injecting a tracer into the shell drain flow downstream of the drain tank and sampling the drain flow further downstream after thorough mixing takes place. The second step moves the tracer injection point to a point in the cycle steam upstream of the moisture separator. The moisture carryover is then calculated by accounting for differences between drain flow tracer concentrations and tracer injection rates observed during the two steps.
- **Shell Drain Flow Method** This method is applicable if there is flow measurement of moisture separator shell drain flow. This method does not rely on measured cycle steam flow and moisture content. Rather, it uses the turbine vendor’s thermal kit data for information describing moisture in the cycle steam leaving the HP turbine and entering the moisture separator. As a result, the calculated moisture carryover is only as accurate as measured drain flow and how well the thermal kit represents actual cycle steam conditions when the method is being used.

All four methods are discussed in Sections 4.5 and 5.4 of PTC 12.4. The differential tracer method is the most accurate method. This method and the calorimeter method, however, should be considered special tests that are not practical for continuous use. On the other hand, energy balance and drain flow methods can be used for continuous monitoring but are inherently inaccurate due to the amount of data required and assumptions concerning the applicability of the thermal kit data. They can be used for trending where accuracy is less important.

5.3 LP Turbine

Steam enters the LP turbine in nearly a dry state or slightly superheated if the steam cycle includes an MSR. The moisture content of the steam increases as it expands in the LP turbine, leaving the last stage with approximately 12-16% moisture. Because moisture in the LP turbine reduces mechanical efficiency and causes erosion-corrosion type damage, it would be desirable to measure the amount of moisture in the various stages of the LP turbine.

Routine measurement of LP turbine moisture content is not currently practical or economic. In general, the moisture content in the steam can be reasonably predicted at each stage of expansion through the turbine by assuming an isentropic expansion process and using the vendor-predicted stage mechanical efficiencies (usually on the order of 80%). However, the local distribution of the moisture in each stage remains uncertain and somewhat academic unless modifications to the turbine to improve mechanical efficiency and repair or replace moisture damaged components are at issue. This is discussed further in Section 2.5, Steam Turbine.

5.4 References

1. American Society for Mechanical Engineers. Steam Turbines. ASME Performance Test Code 6.

2. American Society for Mechanical Engineers. Water and Steam in the Power Cycle (Purity and Quality, Leak Detection and Measurement. ASME Performance Test Code 19.11-1970.

3. American Society for Mechanical Engineers. Moisture Separator Reheaters. ASME Performance Test Code 12.4-1992.

6

MEGAWATT ENHANCEMENTS

The objective of a thermal performance program is to maintain excellent thermal performance and improve thermal performance when it is cost effective to do so.

To be effective in meeting this objective, a structured heat rate performance improvement program is needed. The basic process, depicted in Figure 6-1, includes:

- An evaluation process to determine current performance, estimate achievable performance, and establish performance goals based on cost benefit analyses.
- A process to implement the cost effective, corrective, and preventive options. Operations, Maintenance and Engineering are all involved in good thermal performance.
- A tracking and reporting system to inform management and to obtain support and resources. Keeping everyone informed about thermal performance is important to the bottom line.
- An effective method to obtain and maximize use of existing technology and to promote R&D projects that fill technology needs.
- Methods to receive from and feed back to industry thermal performance information and technology research and development needs. EPRI has established P^2EP to interface directly with utility performance engineers (designated as P^2EP coordinators) expressly for this purpose.

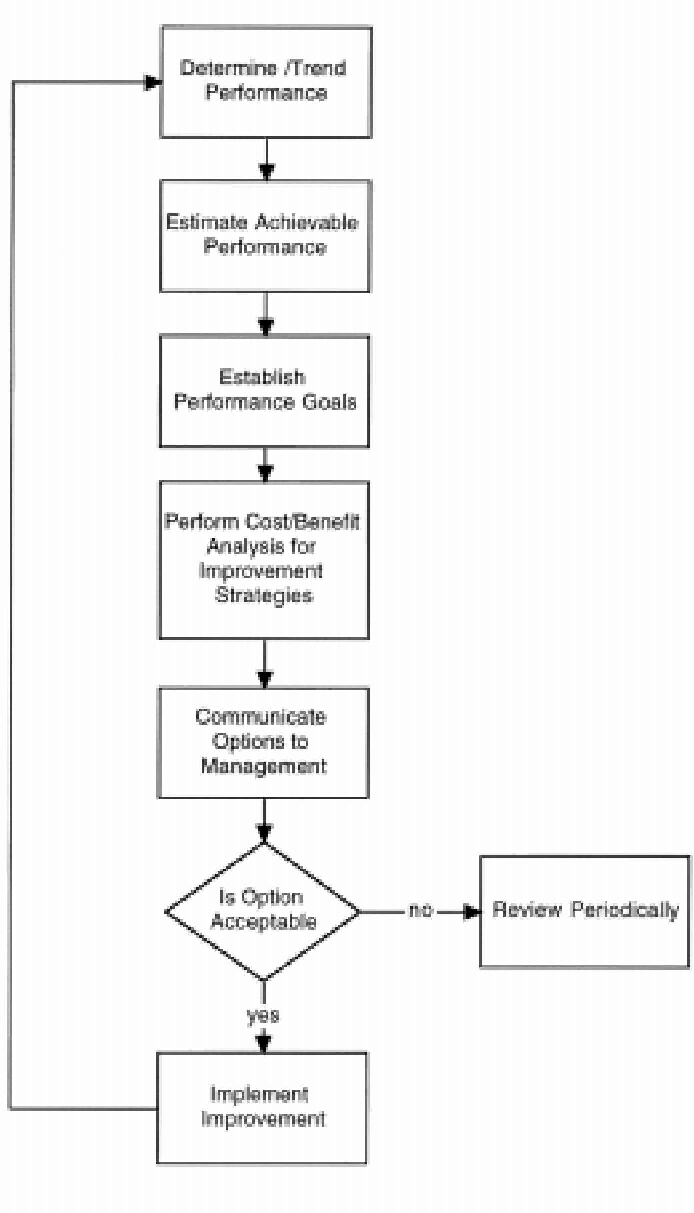


Figure 6-1
MWe and Heat Rate Improvement Process

6.1 Improved Feedwater Flow Instrumentation

In operation of nuclear power plants, it is essential that the total power in the reactor core be determined accurately in order to control plant output. Inaccurate core thermal power calculations may result in sacrificing valuable power generation capability [1].

In all nuclear power plants, a calculation is performed by the plant process computer (PPC) to determine the mass and energy balance across the reactor core. For a boiling water reactor (BWR), the mass and energy balance is

$$_{core} = Q_{st} + Q_{cu} + Q_{rd} - Q_{fw} - Q_{p} - Q_{cd}$$

Where

Q_{st} = heat loss, primary steam flow
Primary steam flow = (mass flowrate, feedwater) + (mass flowrate, reactor water cleanup)
Q_{cu} = heat loss, reactor water cleanup
Q_{rd} = radiative heat loss
Q_{fw} = heat gain, feedwater
Q_{p} = heat gain, recirculating pumps
Q_{cd} = heat gain, control rod drive

For a Pressurized Water Reactor (PWR), which generally uses a steam generator mass and energy balance (secondary calorimetric) to calculate the mass and energy balance across the reactor core, the mass and energy balance is

$$_{core} = Q_{st} + Q_{bd} + Q_{lo} - Q_{fw}$$

Where

Q_{st} = heat loss, steam generator steam flow
Steam generator steam flow = (mass flowrate, feedwater) - (mass flowrate, blowdown)
Q_{bd} = heat loss, steam generator blowdown
Q_{lo} = other system heat losses
Q_{fw} = heat gain, feedwater

As noted in both equations, accurately measuring feedwater flow is extremely important to accurately measuring the reactor core power. Feedwater flow rates are generally measured using differential pressure devices with a venturi or flow nozzle installed in the feedwater pipe. Plant experience has shown that these flow devices have fouled at some plants over time increasing the measured pressure drop across the device. This produces erroneously high flow rate indications, which result in erroneously high reactor power calculations and unwarranted plant power derating. nwarranted plant deratings of 1-3% have been experienced by many nuclear plants due to feedwater venturi or nozzle fouling.

It should be noted that a reduction in power output in a typical 800 MWe nuclear plant reduces the revenues from that plant considerably. As an example, Palisades Nuclear

Plant (811 MWe gross) in western Michigan has determined that fouling of their feedwater venturis has caused the plant to derate 2.2% as determined through measurements using an ultrasonic flow meter. With a cost of producing power at $0.025 / kWh and the residential utility rate of $0.09 / kWh, Consumer Energy can realize an increase in revenue of $27,800 / day or approximately $9 million / year (assuming 90% capacity factor). Since the incremental cost of producing the additional 2.2% power is almost negligible, the Palisades cost center should benefit by approximately $3.5 million / year from this performance improvement [Section 6.1.5, Ref. 26].

6.1.1 Causes of Fouling

Fouling of feedwater venturis and nozzles is caused by deposits of corrosion products that plate out on certain areas of the flow measurement devices. EPRI research has shown that flow measurement devices become fouled by one or more of the following mechanisms:

- **Copper Deposits** Plants with large amounts of copper-bearing alloys in their flow systems that are not directed through full flow polishers experience deposits that are 50-60% copper. Removal of the copper from the systems has reduced the amount of fouling.

- **Iron Oxide Deposits** Plants with no copper-bearing alloys experience iron oxide deposits that foul feedwater flow measurement devices throughout the feedwater system.

- **Exposure to Corrosion Products** Plants that expose the flow measurement devices to higher than normal concentrations of corrosion products during the pre-operational cleanup cycle experience fouling of these devices.

- **Surface Erosion** Surface erosion has also been observed and has a tendency to cause non-conservative feedwater flow measurement.

These fouling mechanisms appear to be accelerated by the following:

- Plants that have been in a wet or dry layup condition for a extended period of time experience a much greater thickness of fouling deposits. Normal deposits are 10-15 microns, but plants with extended layups may have fouling deposits as thick as 150-400 microns.

- Plants that have high mass flow rates through feedwater piping (5.5-6.5 million pounds per hour) tend to have more fouling. This may be attributed to increased corrosion products in the feedwater from increased flow-accelerated corrosion (FAC) or increased deposition rates on the flow devices at increased velocities.

- Plants with high pressure drops across the flow devices experience greater fouling. Plants with <15 psid across their flow devices see very little fouling, but plants with pressure drops > 24 psid experience high levels of fouling.
- Deposits tend to accumulate along machining or polishing marks in the flow measurement devices. Electropolishing, precision surface grinding, and mirror finishing significantly reduce the amount of fouling.
- Non-uniform deposits have been found in regions of flow transitions. Sharper transitions cause thicker deposits.
- Deposits have a tendency to be greatest within the pipe taps where stagnant flow conditions exist.

6.1.2 Compensation for Fouling

Fouling can be successfully removed by mechanical or chemical cleaning of the flow devices. Water jet and hand cleaning are the mechanical cleaning methods that normally have been used. Chemical cleaning has been used successfully to remove the fouling, but it must be matched to the chemical composition of the deposits and the material being cleaned [Section 6.1.5, Ref. 1, 2, 3].

Cleaning can be performed only with the unit off-line and on plants with access ports or removable flow devices. In most older plants, flow devices are welded in place with no access to the device. For these plants, cleaning would be a major undertaking. Another problem that has been noted [Section 6.1.5, Ref. 5] is that fouling, when removed, reappears during each operating cycle. Methods have been devised by utilities to correct for the fouling to allow the plant to operate at rated reactor power. These methods include:

- Use of plant on-line and historical data to determine a feedwater flow device fouling correction factor [Section 6.1.5, Ref. 6, 7].
- Use of steam flow devices to measure flow or to determine a feedwater flow device fouling correction factor [Section 6.1.5, Ref. 8, 9].
- Installation of a secondary or bypass differential pressure flow device to measure flow or determine a feedwater flow device fouling correction factor [Section 6.1.5, Ref. 10, 11, 12, 13].

- Use of an ultrasonic flow measurement device to measure feedwater flow or determine a feedwater flow device fouling correction factor [Section 6.1.5, Ref. 11, 12, 13, 14, 25, 26].
- Tracer testing to measure feedwater flow to determine a feedwater flow device fouling correction factor [Section 6.1.5, Ref. 12, 15].

6.1.3 Differential Pressure Flow Measurement Devices

Flow Nozzles

An ASME flow nozzle with throat taps is the accepted standard for testing steam turbine performance [Section 6.1.5, Ref. 1, 16, 17, 19]. The flow nozzle is a differential pressure device installed in the flow stream. The flow through the nozzle contracts gradually, creating a pressure drop across the nozzle which can be measured using a differential pressure instrument. A diffusing cone can be installed downstream to reduce the pressure drop across the nozzle by about 60%. Procedures for the design, fabrication, and installation of the flow nozzle have been discussed by ASME [Section 6.1.5, Ref. 16, 18].

Accuracies of ±0.25% of actual flow are possible if the manufacturer's specified flow range and piping configuration are met. Flow nozzles are used for feedwater, condensate, and pump-ahead drain flows [Section 6.1.5, Ref. 10, 13] but they must be calibrated before use. Proper calibration includes the flow nozzle, the entire flow section including the diffusing cone (if used), flow straightener, and upstream and downstream piping sections as a unit. Flow nozzles need to remain extremely clean to ensure accurate readings. A deposit of 0.0002 x (throat diameter) is acceptable if it is uniformly deposited. However, the nozzle must be cleaned and retested if the deposits exceed this thickness. Maintenance requirements include regular visual inspections of surfaces for deposits and pressure taps for blockage.

Venturis

A venturi is a differential pressure device installed in the flow stream. The flow enters a tapered inlet section and exits through a tapered diffuser section [Section 6.1.5, Ref. 1, 17, 18, 19, 27]. The pressure drop is measured through taps at the inlet and throat. The diffuser section allows the fluid pressure to recover smoothly and results in excellent pressure recovery when compared to other differential pressure devices. Procedures for the design, fabrication, and installation of venturis have been discussed by ASME [Section 6.1.5, Ref. 17, 18].

Accuracies of ±0.25% of actual flow are possible if the manufacturer's specified flow range and piping configuration are met. Venturis are widely used for nuclear plant feedwater flow measurement. Maintenance requirements include regular visual inspections of surfaces for deposits and pressure taps for blockage.

6.1.4 Ultrasonic Flow Measurement Devices

An ultrasonic flowmeter utilizes a pair of transmitting/receiving piezo-ceramic crystal transducers. These transducers can be mounted through the pipe walls to form a "wetted" interface with the liquid being metered, or they can be attached to the external walls of the pipe to form a dry contact [Section 6.1.5, Ref. 1, 20]. Figure 6-2 shows an ultrasonic metering section.

Transit Time

Various systems that measure the effect on sonic transit time produced by the flow velocity are used to measure the flow [Section 6.1.5, Ref. 1, 20.

Accuracies of ±0.5 to 1% are possible if the manufacturer's specified installation requirements are followed.

The Chordal ultrasonic flowmeters use transducers that penetrate the pipe wall and are referred to built-in or "wetted" systems. Wetted systems are highly accurate to ±0.5% and provide a high reliability and repeatability [Section 6.1.5, Ref. 5, 21]. However, the "wetted" systems can experience fouling and need to be cleaned at regular intervals [Section 6.1.5, Ref. 1].

Strap-on devices measure feedwater flow with an alternate instrument that is non-intrusive. The strap-on transit time ultrasonic flowmeter has been used at many plants [Section 6.1.5, Ref. 1, 11, 12, 13, 14,17, 20, 25]. This meter requires transmitters/receivers to be mounted on the outside of the pipe to transmit/receive ultrasonic signals projected diagonally across the pipe. These systems are accurate to ±1% with good repeatability. From discussions with plant engineers that have installed these systems, coupling between the pipe and the crystal and alignment of the crystals are critical to obtain accurate and reliable results.

To measure feedwater flow with a second instrument that is non-intrusive, the strap-on cross-correlation ultrasonic flowmeter is available [Section 6.1.5, Ref. 1, 19, 20, 22, 23, 24, 26]. The meter has four crystals. Two are upstream (one transmit/one receive) and two are downstream (one transmit/one receive). The upstream and downstream transmitters send continuous-wave ultrasonic beams across the feedwater flow perpendicular to the pipe centerline. The signals are modulated by the turbulence signature of the flow and are received by the crystals on the opposite side of the pipe.

With a computer, the modulated waveforms are compared to identify turbulence signatures that are identical at the upstream and downstream crystals. When identical signatures are identified, the computer measures the elapsed time required to go from the upstream to the downstream crystals, thus giving the feedwater flow velocity which is used to calculate feedwater flow. These systems are accurate to less than ±0.5% with good repeatability and reliability. From discussions with plant engineers that have installed these systems and strap-on transit time flowmeters, the cross-correlation flowmeter is much easier to install, and time-consuming crystal alignments are not required.

6.1.5 Feedwater Flow References

1. *Feedwater Flow Measurement in U.S. Nuclear Power Generation Stations.* Electric Power Research Institute, Palo Alto, CA: November 1992. Report TR-101388.

2. *Survey and Characterization of Feedwater Venturi Fouling at Nuclear Power Plants -- Vol. 1: Feedwater Venturi Fouling.* Electric Power Research Institute, Palo Alto, CA: May 1992. Report TR-100514, Volume 1.

3. *Survey and Characterization of Feedwater Venturi Fouling at Nuclear Power Plants -- Vol. 2: Photomicrograph and Chemical Analysis.* Electric Power Research Institute, Palo Alto, CA: May 1992. Report TR-100514, Volume 2.

4. EPRI P^2EP Survey #94-017. *Inputs to Secondary Calorimetric.* November 29, 1994.

5. EPRI P^2EP Survey #94-004. *Feed Flow Element Fouling.* May 17, 1994.

6. C. S. Sullivan. *Feedwater Flow Venturi Fouling Study*. EPRI P^2EP Paper

7. PO-0042.0-806.

8. Preston Gillespie and Leland Hawthorne. Development of the Feedwater Venturi Fouling Coefficient at the Oconee Nuclear Station. EPRI P^2EP Paper PE-0042.0-105.

9. EPRI P^2EP Survey #96-010. Secondary Calorimetric Divergence. October 15, 1996.

10. EPRI P^2EP Survey #96-011. Using Steam Flow to Correct for Feedwater Venturi Fouling. January 1, 1997.

11. N. R. Deming, M. A. Moradian, G. I. Arnold, K. T. Hurt. ASME Performance Test On TVA Sequoyah Unit 1. EPRI P^2EP Paper PO-0005.0-32.

12. C. C. Mason and T. F. Siegler. Sequoyah Nuclear Plant -- Feedwater Venturi Fouling Investigation. EPRI P^2EP Paper PE-0042.0-378.

13. Allen L. Williams. Installation and Use of a Full Flow Feedwater Bypass Line Venturi to Calibrate Feedwater Flow Ultrasonic Flowmeters at the Kewaunee Nuclear Plant. EPRI P^2EP Paper PO-0042.0-659.

14. T. A. Nelson and G. L. Starnes. Monitoring Feedwater Nozzle Fouling at Diablo Canyon Nuclear Power Plant. EPRI P^2EP Paper PO-0053.0-220.

15. T. Lestina and C. Contard. Using a Flow Correction Factor to Compensate for Feedwater Venturi Bias and Increase Plant Generation. EPRI P^2EP Paper PO-0049.0-805.

16. Keith A. Young. Recovered Lost Megawatts From Feedwater Nozzle Calibration Project (1% Core Thermal Power). EPRI P^2EP Paper PO-0092.0-704.

17. American Society of Mechanical Engineers."Steam Turbines." ANSI/ASME Performance Test Code 6-1976 (Interpretations 1977-1983).

18. Richard W. Miller. Flow Measurement Engineering Handbook. McGraw-Hill Co., Inc., New York 1996.

19. American Society of Mechanical Engineers. "Measurement of Fluid Flow in Pipes Using Orifice, Nozzle, and Venturi." ASME MFC-3M-1989, 1989.

20. Robert P. Benedict. Fundamentals of Pipe Flow. John Wiley & Sons, Inc., New York 1980.

21. Lawrence C. Lynnworth. Ultrasonic Measurements for Process Control. Academic Press, Inc., Boston 1989.

22. Bradley Devendorf, Donald Augenstein, and Steve Johnson. The Installation of the Leading Edge Flowmeter (LEFM) at Texas Utilities' Comanche Peak for Measurement of Feedwater Flow. EPRI P^2EP Paper PO-0052.0-660.

23. R. S. Flemons. "A New Non-Intrusive Flowmeter." National Bureau of Standards Special Publication 484, Proceedings of the Symposium on Flow in Open Channels and Close Conduits, February, 1977.

24. J. R. Sherin and D. Zorbin. Feedwater Flow Measurements Using Ultrasonic Cross-Correlation Flow Meter. EPRI P^2EP Paper PO-0146.0-136.

25. D. Zorbin and J. R. Sherin. Recent Developments in Flow Measurement Using Ultrasonic Cross-Correlation Flow Meter. EPRI P^2EP Paper PO-0051.0-624.

26. D. Augenstein, S. Corey, and S. Egli. *Correction of Dynamic Venturi Fouling at Watts Bar Nuclear Plant Using an External Leading Edge Flow Meter (LEFM).* EPRI P^2EP Paper PO-0053.0-892.

27. Chad Main. *Ultrasonic Flow Measurement at Palisades Nuclear Plant.* EPRI P^2EP Paper PE-0146.0-901.

28. K. Thomas and R. W. James. Feedwater Flow Measurement Experience at Harris Nuclear Plant. EPRI P^2EP Paper PE-0041.0-900.

6.2 Reactor Primary Coolant Flow Measurement

Primary flow measurement in a PWR is calculated from secondary calorimetrics and primary coolant temperature and density. Direct measurement of primary flow can be used to reduce the effects of feedwater flow biases caused by venturi fouling. These biases, which typically trend toward indicating increased flow, cause the turbine to be operated at reduced power. Therefore, a redundant calorimetric on the primary loop has the potential to reduce unnecessary power reductions that mean lost megawatts.

Redundant flow measurements on the secondary side combined with careful baselining of flow measurements following venturi cleaning can be used to provide some correction for subsequent fouling. However, many plants have the concern that Tech Spec limits on reactor power might be exceeded by removing the bias from venturi flow calculations.

One approach to reducing the uncertainty of primary calorimetrics is to measure primary flow directly. EdF has developed a tool that infers flow by measuring the time difference between spatially separated sensors on the reactor coolant piping. EPRI developed a tool that detects the N16 isotope signature using ultrasonic flow measurement as described in Section 6.1. Figure 6-2 shows a primary flow measurement system that has been tested at Ginna. The flow measurement accuracy in the field was unsatisfactory but it was reasoned that with improved electronics and using proper flow profile correction factor, the uncertainty could be reduced. However, the reduction was not significant enough to warrant as a substitute for using secondary calorimetrics. However, its value in reducing the uncertainty in trending venturi fouling plus providing a redundant measurement for controlling reactor power may be cost effective. The value would depend on the amount of venturi fouling experienced and other factors discussed in the references.

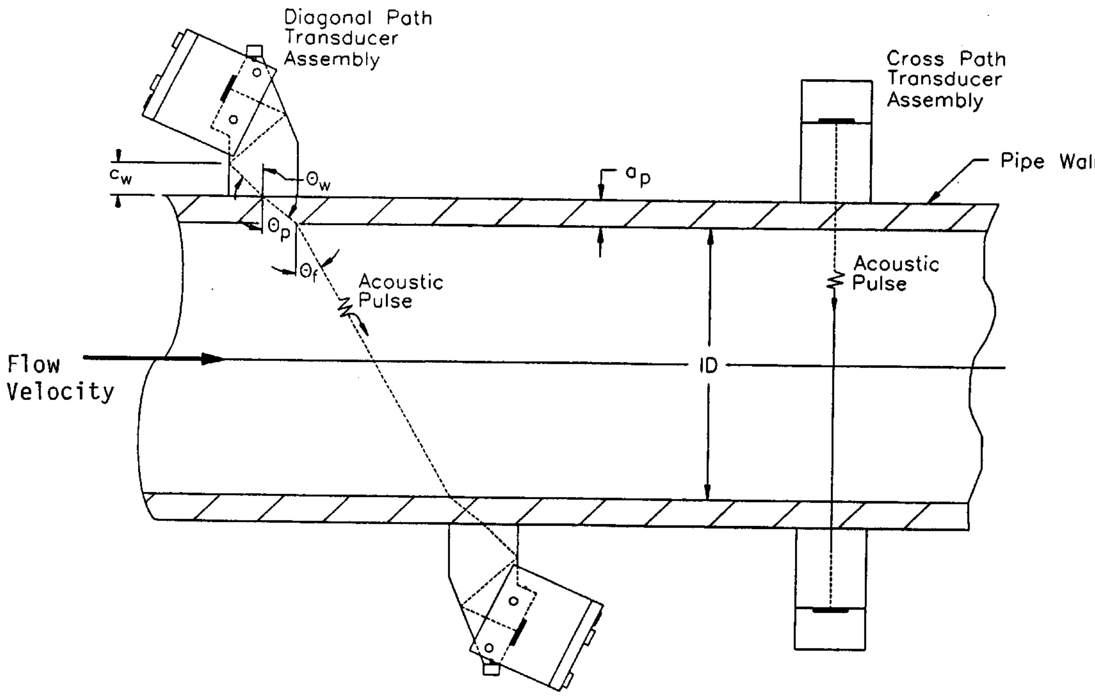


Figure 6-2
Schematic of Ultrasonic Metering Section

6.3 Exhaust Hood Modifications

Potential gains in low-pressure turbine performance are available through exhaust hood modifications. The exhaust hood forms the region of the flow path following the last stage of the turbine. Optimum flow in the exhaust hood directs steam flow evenly to the condenser tubing. Also, it should be designed so that steam leaving the last stage does not leave with an excess of kinetic energy that could have been converted to work by the turbine.

Modifications of exhaust hoods are generally a part of an overall LP section replacement. Flow measurements and computer modeling have shown that the exhaust hood flow is complex. Small changes in design can cause large effects in overall pressure loss. The flow conditions in the exhaust hood depend on inlet conditions that require that the design of the exhaust hood geometry and the last stage blade design must be integrated.

The thermal performance engineer may be involved in planning LP turbine replacement. Recognizing the relationship between the exhaust hood and last stage blade designs helps with cost-benefit analysis and design decisions. Following any turbine upgrade, the new baseline performance will need to be established. This new baseline will provide a basis for trending and will provide a check that guaranteed improvements in heat rate have been achieved.

6.4 MSR Modifications

Plant experience has shown that MS/Rs are susceptible to problems that require significant modifications. These problems are discussed in Section 2.6.2 of this handbook and in EPRI's *Moisture Separator Reheater Source Book* and *Procurement and Operation Considerations for Moisture Separator Reheaters*. This section summarizes several modification packages that have been used to improve MS/R performance and enhance megawatt output. There can be several reasons to modify MS/Rs including economic reasons, the need to prevent further degradation of the MS/Rs, and reduction of moisture content and moisture damage in the LP turbines. Modifications are described in much more detail in the references listed above.

6.4.1 Moisture Separator Modifications

Deficiencies in moisture separation have been caused by use of wire mesh separators; carbon steel separator vanes, which can rapidly erode in the wet steam environment; single vane or pocket vanes, which can have low separation efficiency especially when there is high inlet velocity and flow maldistribution into the separators; maldistribution of steam flow into the separators; and re-entrainment of separated moisture in the dry steam exiting the separators.

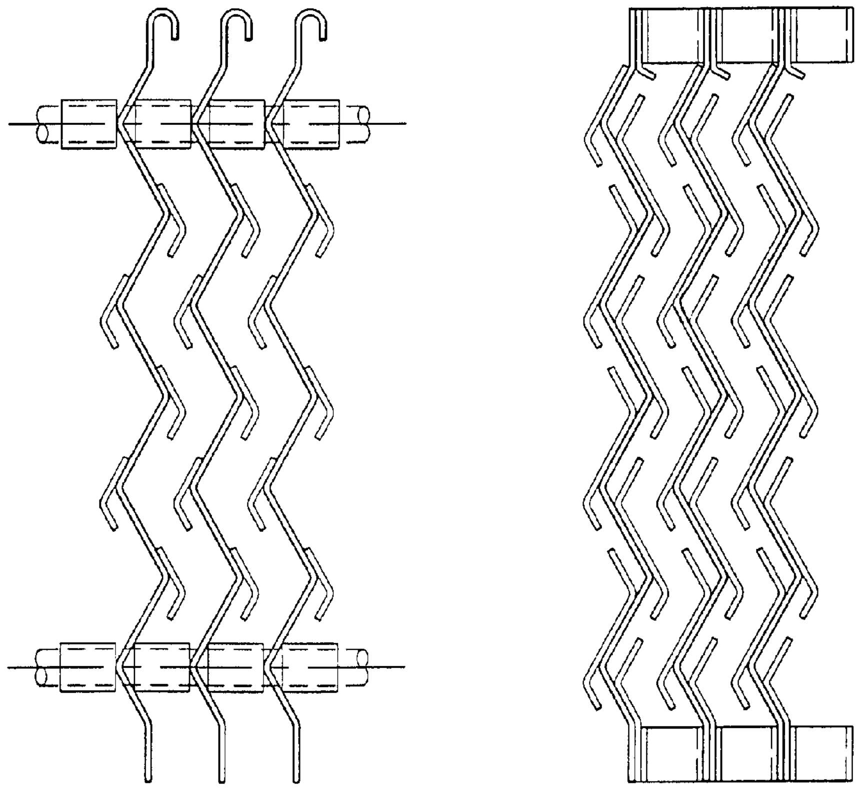
Typical modifications include:

- Double-pocket, stainless steel chevron replacements for single-pocket chevrons. Figure 6-3 shows both types. Moisture removal effectiveness depends on the moisture and dynamic head, ρV^2, of the inlet steam. Figure 6-4 illustrates the effect of these parameters on moisture removal effectiveness.

- Chevrons slanted outward as shown in Figure 6-5. This arrangement allows stray separated moisture to drain back toward the incoming cycle steam rather than into the path of the dry steam leaving the separator where it can re-entrain.

- Improved inlet steam manifolds and perforated plates at the inlet to the chevrons to improve flow distribution into the moisture separators.

These modifications are applicable to both MSRs and MS-only units.



EXISTING TYPICAL OEM
SINGLE POCKET CHEVRONS
by PEERLESS MANUFACTURING

HIGH PERFORMANCE
DOUBLE POCKET CHEVRONS
by PEERLESS MANUFACTURING

Figure 6-3
MSR Chevrons
(Courtesy of Senior Engineering)

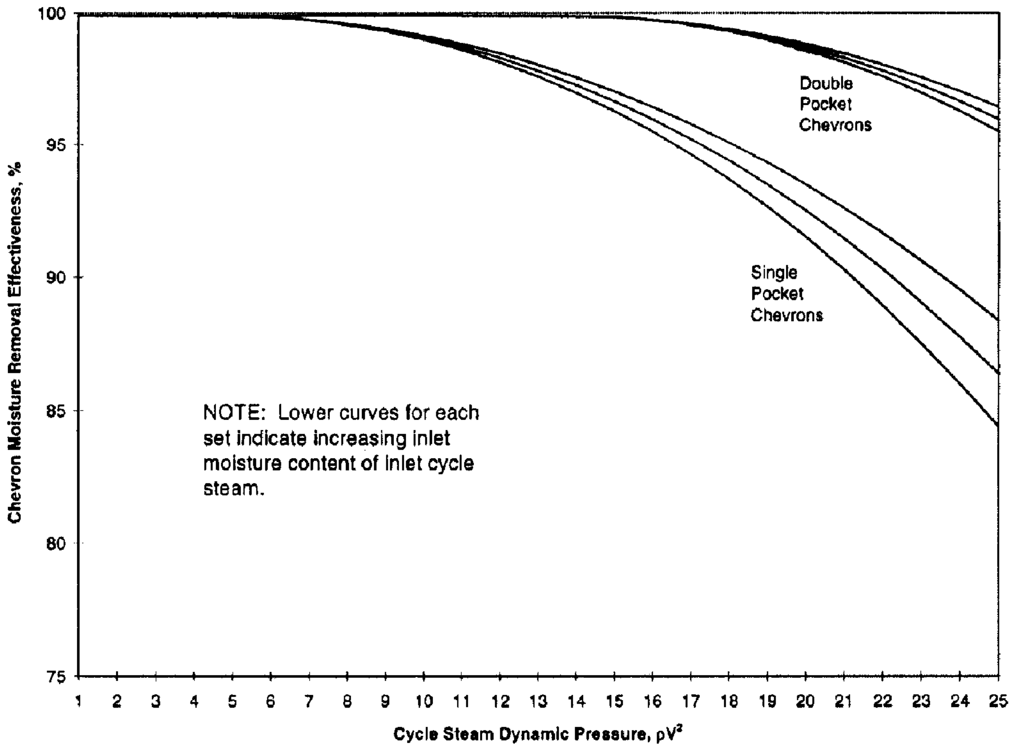


Figure 6-4
Typical Moisture Removal Effectiveness of Chevron Separators

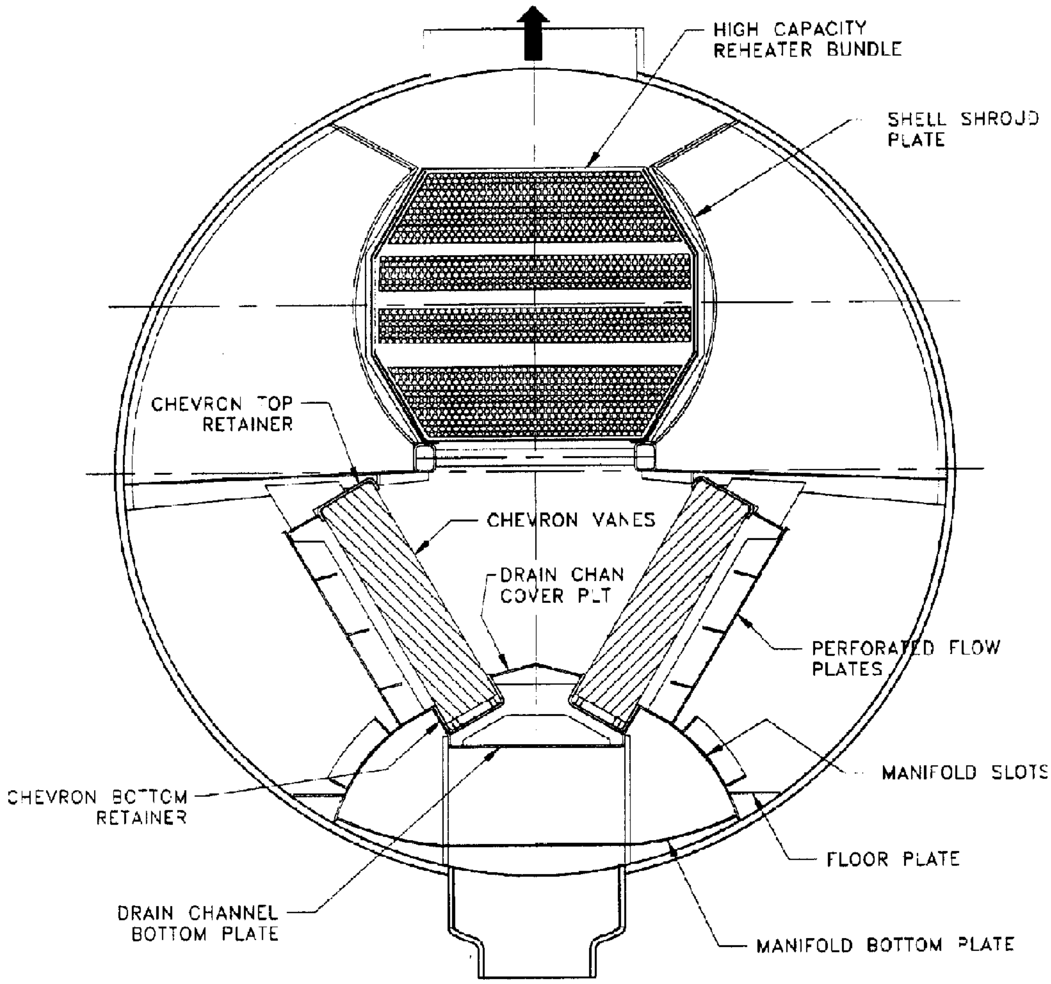


Figure 6-5
General Arrangement of G Type MSR Reconstruction
(Courtesy of Senior Engineering)

6.4.2 Excess Steam Modifications

Section 2.6.2 and References 1 and 2 discuss the problem of condensate subcooling and oscillation. The problem is mitigated by venting steam through the reheater tubes to a feedwater heater. The vented steam is referred to as excess steam. This excess steam in the outlet leg of the reheater U-tubes prevents excessive subcooling of condensate in the tubes. The amount of excess steam can be as high as 10% of total heating steam flow. The energy in the excess steam flow contributes to feedwater heating but is not available for expansion in the turbine. For this reason, an excess steam modification results in an increase in heat rate and reduction in electrical output.

There have been several solutions to the problem of providing excess steam. The simplest is to vent steam from the second pass tubes by installing a line, isolation valve, and flow restrictor from the reheater outlet channel to the top stage of FW heaters. The arrangement is shown in Figure 6-6. Since all tubes in the reheater tube bundle do not have the same heat load, some modifications use tube inlet orifices to control the amount of excess steam in the individual reheater tubes.

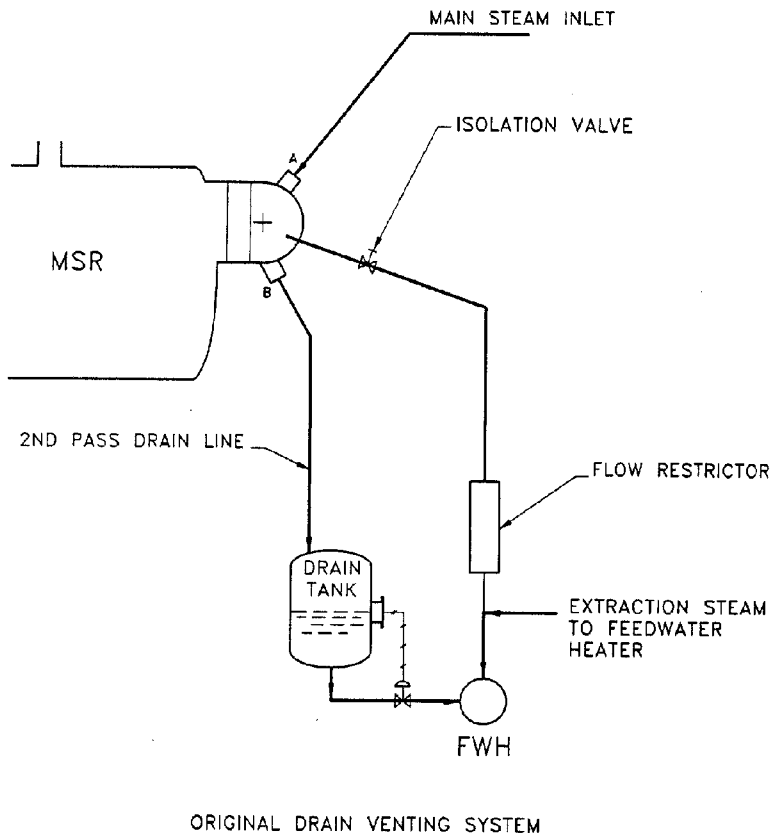


Figure 6-6
Original Drain Venting System
(Courtesy of Senior Engineering)

Despite the benefits of excess steam in a two-pass reheater bundle, there can be inadequate excess steam flow to tubes with the higher heat loads. Increasing the amount of orificing for lightly loaded tubes and increasing the total amount of excess steam flow can be inefficient and wasteful. An alternate design is the four-pass vent chamber design. This alternative requires modification to the inlet and outlet headers of the reheater to make the existing U-tubes a four-pass configuration. Figure 6-7 compares the flow paths and Figure 6-8 compares typical temperature distributions in the reheater for the two- and four-pass configurations.

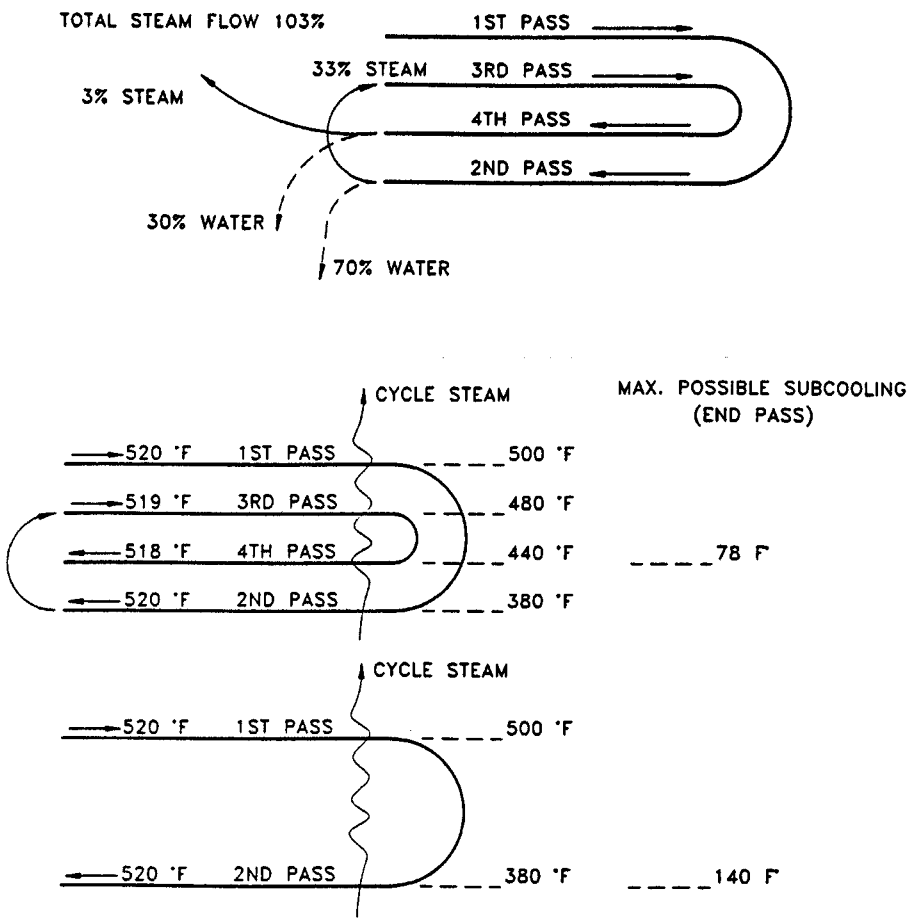


Figure 6-7
Two- and Four-pass Temperatures
(Courtesy of Senior Engineering)

This modification, using the existing tubes and tubesheet, lacks the smooth tubesheet surface needed for the first-third pass partition and fourth-pass vent and drain header gaskets and bolting. Correcting this requires that some tube locations be converted to threaded studs and requires careful, intricate design to the internal gasketing. If not, leakage between passes can readily occur. As a result, the four-pass modification is usually made in conjunction with replacement of the entire tube bundle so that the tubesheet can be redesigned for pass partitions, bolting, and gasketing.

6.4.3 Tube Bundle Replacement

Reasons to replace an MSR tube bundle include the following:

- There is a need to remove sources of copper from the turbine cycle.
- The tube bundle heat transfer capacity has or is predicted to deteriorate to the point where it is economically justified to replace the tube bundle.
- The improved replacement adds MWe and is cost effective.

The problems discussed in Section 6.1 of the *MSR Source Book* suggest that it is seldom prudent to replace the tube bundle in kind. There are modification features that are now considered necessary for good performance and long tube life that were not in the design being modified:

- Operating experience has shown that tubes made of Type 439 stainless steel has provided the best performance and durability. Stainless steel is susceptible to stress corrosion cracking. Precautions need to be taken to specify material with low residual carbon content and to specify adequate process for rolling fins and for bending, welding, and heat treating the tubes.
- Section 12 of the TEMA standards provides guidance on providing adequate tube support to prevent tube damage by vibration. As a rule of thumb, a tube span of 24” or less prevents harmful vibration.
- Even if the tube bundle shroud is not replaced, it should be modified to prevent buckling and to minimize bypass of cycle steam around the tube bundle. Figure 6-8 shows a method to restrict bypass flow. The restrictor bars also strengthen the shroud against buckling. A unique buckling analysis, based on the conditions of the MSR under modification, should be made to ensure that there is adequate shroud structural rigidity.
- The U-tubes and tubesheet should be designed to provide smooth surfaces and bolt locations for pass partition and fourth pass header bolting and gaskets. The tube arrangement shown in Figure 6-5 provides these features.

The above features should be incorporated in both HP and LP reheater tube bundles if either or both are candidates for replacement. In no case should the tube bundle exposed to cycle steam flow from the moisture separators be replaced unless there is assurance that there is not significant moisture carryover. If this assurance does not exist, first conduct moisture carryover tests and / or modify the moisture separators to reduce carryover.

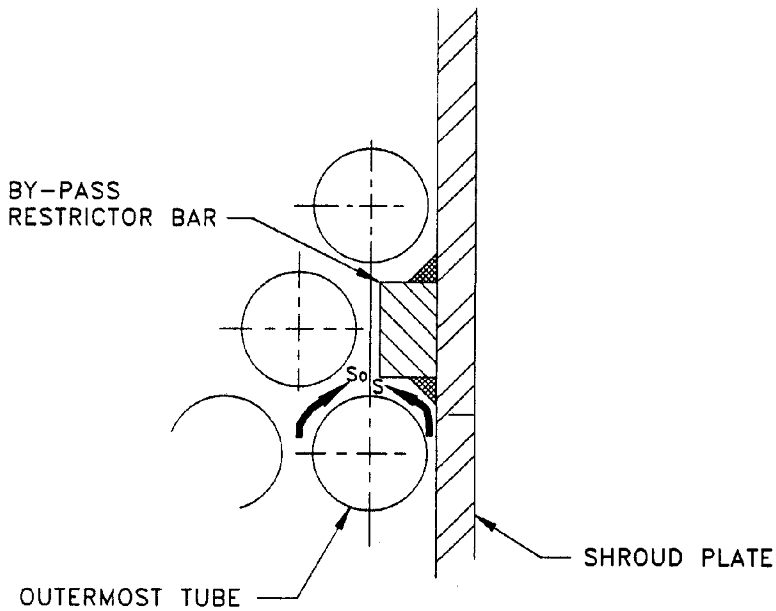


Figure 6-8
Function of By-pass Restrictor
(Courtesy of Senior Engineering)

If a decision is made to replace a tube bundle, there are other modification features that should be considered as enhancements; their relative cost may be low compared to that of the basic modification:

- Add more tubes by taking advantage of the original tubesheet diameter. Figure 6-5 shows a pattern that does this. Most OEM tube bundles had a rectangular pattern that did not use the 3 and 9 o'clock space in the tubesheet circle. The wider, pancake tube pattern also reduces cycle steam ΔP and the tendency for high ΔP to lift the tube bundle.

- Add thermocouples in the outlet of select second and fourth pass tubes to determine if condensate oscillation is occurring and to adjust excess steam to prevent condensate oscillation over the MSR operating range.

- Add features to reduce the problem of unequal thermal expansion of U-tube legs. These are discussed in EPRI's *MSR Source Book.*

- Reduce tube diameter to increase the tube-side heat transfer coefficient. Reduced tube diameter also increases cycle steam pressure drop. The tradeoff should be evaluated if this method of tube bundle enhancement is considered.

6.5 Heater Retubing

Because heater tubes are usually welded into the tube sheet and are U-tubes, replacement of individual tubes in a tube bundle is, as a rule, never undertaken. Instead, leaking and otherwise suspect tubes are plugged and the increase in heat rate is accepted. The decision to replace a heater tube bundle is generally made when there are a significant number of plugged tubes or when NDE indicates gross loss of tube wall thickness. In any case, a cost-benefit analysis should be made to determine if the cost of replacement is justified by the increase in heat rate and plant output.

When assessing the option of replacing one or more heater tube bundles, it is important that the history of the present heaters be considered:

- **Age** It is important to consider the age of the installed heater, the age of the plant, and the expected remaining life of each. If an older plant is going to remain in service for a long period of time and the heater is degrading slowly but performance has been satisfactory to date, it may be economical to replace the bundle and reap the benefit of increased plant output over a longer period of time.

- **Tube Material** Presently, Type 304 stainless steel appears to be the material of choice for heater tube bundles. However, ferretic stainless steels such as Type 439 are becoming increasingly popular due to better thermal conductivity and lower thermal expansion properties than the austenetic stainless steels. Heater bundles using other than these materials should be considered as candidates for replacement, especially if the present tubes are copper alloy. Copper deposits on steam generator tubes and supports result in a type of steam generator damage known as tube denting.

- **Material Condition and Root Cause Analysis** Detailed knowledge of the total condition of the feedwater heater is important before tube bundle replacement. First, it is important to know which tubes have failed and have been plugged, and if possible, the axial location of the failure. The root cause of the failures should be identified. The cause may be tube vibration and tube or support wastage. Although the replacement bundle will have to fit into the same envelope and have nearly the same overall dimensions as the present bundle, it is possible to change tube supports and tube materials to address such causes. Also, it is possible to change tube diameter to address a tube vibration if tube-side pressure drop remains acceptable and the replacement strategy includes tubesheet replacement.

The EPRI *Nuclear Plant Feedwater Heater Handbook* contains a more detailed discussion of heater replacement, material condition, and root cause. Generally, the most economical approach to tube replacement is one that minimizes or has no impact on reactor outage critical path time and minimizes field craft labor. A way to satisfy these conditions is to have the tube bundle with new tubesheet shop fabricated and delivered to the plant and installed in the existing heater shell. Most nuclear plant feedwater heaters have a welded tube-to-shell and shell-to-connecting piping joints. This avoids removing the shell and having to cut and reweld the attached piping, but does require that the tubesheet-shell joint be cut and rewelded in the field. An alternative is to shop fabricate an entire new heater, including the shell, and install it in the field by cutting and rewelding the attached piping and removing the old heater. It is not possible to avoid cutting and rewelding a thick-walled joint in the field when replacing a feedwater tube bundle. Also, removal of the shell requires more plant work space than replacing only the tube bundle.

The subject of tube bundle replacement is complex and extensive and is only summarized in this handbook. The reader is referred to the references at the end of this section for further information. The *Nuclear Plant Feedwater Heater Handbook* has guidelines on failure cause and prevention and tube replacement.

6.6 Throttling (Governor) Valve Modifications (Throttling Loss Modifications)

The function of a governor valve is to control the flow of steam to the HP turbine. Control of flow is accomplished by throttling. That is, as the valve is closed, the pressure drop across the valve increases. At full load, there is some pressure drop across the governor valve.

Governor valve throttling is basically a constant enthalpy process. The governor valve throttling process and subsequent expansion through the turbine is illustrated in Figure 6-9 where the dashed line A-C is the throttling process and C-D is the expansion process after throttling. This figure shows that, while enthalpy is constant and is conserved during governor valve throttling, the entropy increases. The result is that there is less available energy during subsequent steam expansion in the turbine and the thermal efficiency is reduced. This is illustrated by comparing the enthalpy drop difference in expansion lines A-B and C-D in Figure 6-9.

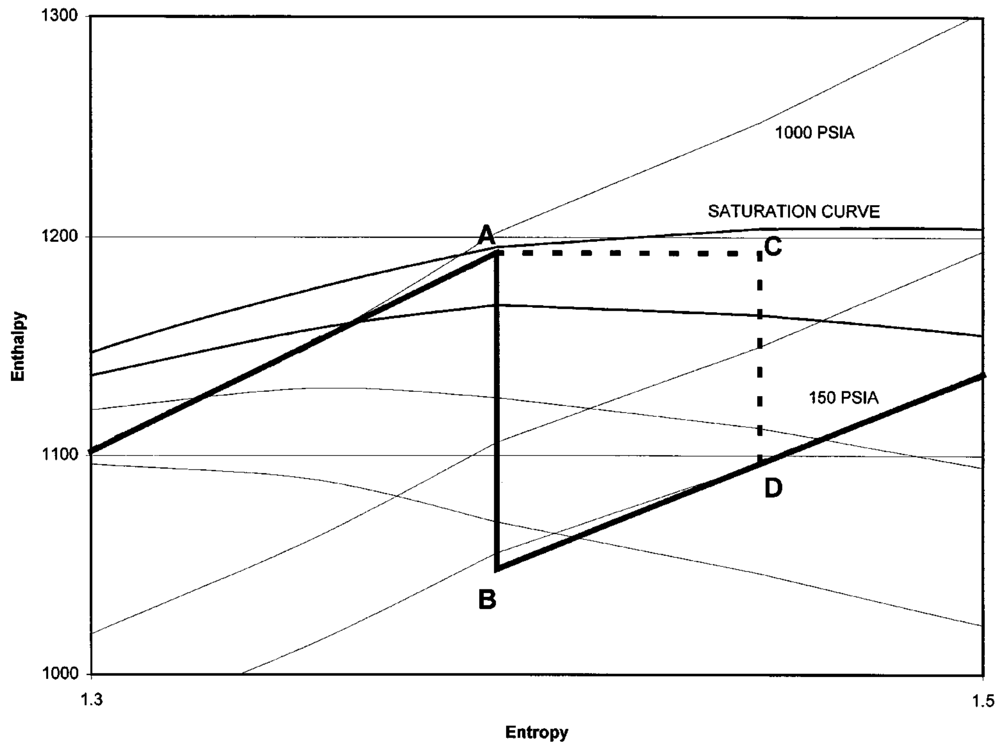


Figure 6-9
Effects of Governor Valve Throttling

6.7 Full Arc and Partial Arc Operation

Almost all nuclear turbines use more than one throttle valve. A four valve design is common. Most turbines are designed to open the valves sequentially with load and to operate with the first three valves fully open and the fourth valve partially closed and throttling at full reactor power. Some throttling at full load is needed for control.

There are two basic methods of admitting steam to the HP turbine first stage nozzles, full arc and partial arc. Full arc admits steam from all the throttle valves to a common plenum. Steam then flows from the plenum into all first stage nozzles located in a full arc in front of the first stage HP rotor blades. Partial arc admits steam into separate plenums, usually one for each throttle valve. Steam then flows from the plenums to a part of the first stage nozzle, each set of nozzles occupying a part of the full arc delivering steam to the first stage blades.

Whether at part load or full load regulation, full arc admission results in parallel steam flow through all of the first stage nozzles while partial arc admission results in steam flow through some, but not all, of the nozzles. As a result, more governor throttling is required with full arc than partial arc. This difference results in more throttling loss for full arc admission and higher heat rate. However, at valves wide open (VWO), full admission can give better HP turbine performance because partial arcs have inactive portions.

Most plants have full arc admission for several reasons:

- The intent has been to operate the plants at or near full load where thermal performance differences between full and partial arc admission can be small. Differences tend to be small whenever main steam pressure is reduced to offset problems such as steam generator fouling and tube plugging and T_{hot} reduction.

- Partial arc increases the cyclic loading of the HP turbine first stage blades since the blades see a more intermittent loading. Some turbine vendors discourage the use of partial arc admission.

In summary, there are thermal performance gains to be achieved by using partial arc admission. A nominal value is 3 MWe for a 1000 MWe plant. The amount varies depending on factors such as present governor valve type and design and future plant operating strategy for parameters such as main steam pressure and reactor power. Turbine reliability can be less with partial arc. The turbine vendor needs to be consulted when considering a change in the governor valve system. The cost of the change, parts plus field labor, including outage time chargeable to the modification costs versus MWe gain needs to be considered when making a decision about the modification.

6.8 HP Turbine Nozzle Block Upgrade

Another potential improvement to thermal performance is modification of the HP turbine nozzle block. A modification may be appropriate to complement a modification to incorporate partial or full arc admission. A modification may also improve first stage turbine mechanical efficiency by reducing nozzle profile losses, which can be approximately 15% of total stage mechanical losses.

Figure 6-10 illustrates the basic change from a conventional straight control or first stage nozzle to a contoured nozzle. Because the control stage has a small aspect ratio and the flow can vary from subsonic at full arc to transonic at partial arc, the low aspect ratio can lead to high flow losses. Modifying the original endwall of the nozzles to converging nozzles causes local flow to accelerate and reduces the boundary layer along the endwalls. Increases of 1% in control stage efficiency have been reported.

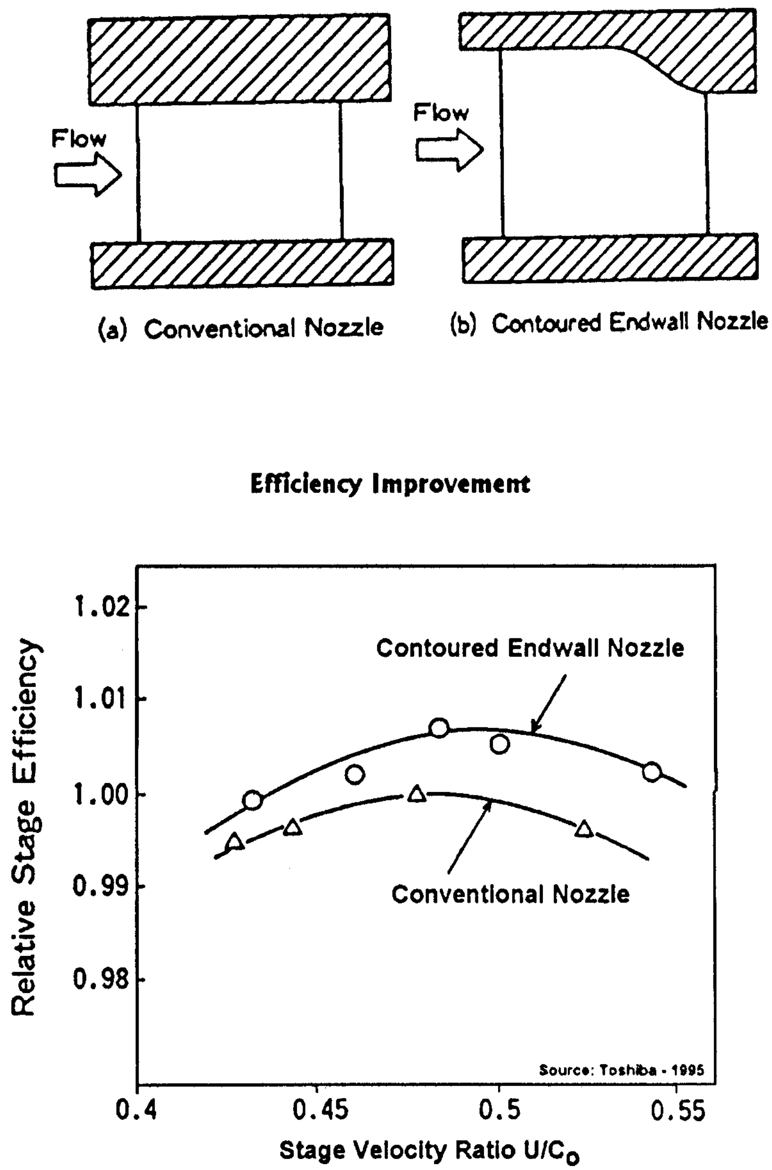


Figure 6-10
Effect of Contoured Endwall Control Nozzles

6.9 Reactor Uprate

Increasing the reactor core power is a low cost approach to gaining megawatts for some plants. Approximately 30% of the plants in the U.S. have uprated to gain electric output by using available margin in plant design.

Uprates have been done for both BWR and PWR plants. Most plants can be uprated 5-7% percent with few modifications to existing facilities. However, the added core power has an effect on nearly every system in the plant. Also, the plant license will require updated analysis leading to technical specification and FSAR changes. The NRC will require a review of the Tech Spec changes for significant hazards and review and approval of any unreviewed safety questions involved in the uprate. The NRC places uprates into four categories as follows:

1. Requiring changes to FSAR/SER documentation

2. Requiring uprate design verification within existing equipment design

3. Exceeding existing plant equipment capacities but not exceeding ECCS capability

4. Requiring upgrade of current ECCS capability

Most of the existing uprates are in the first two categories and none are in the last.

The uprate process involves a feasibility study, technical evaluation for all systems affected, preparation of licensing revisions and submittals, and the engineering and implementation of plant modifications.

The scope of uprating programs for Westinghouse and GE plants is outlined in the following documents:

- WCAP-01263, “A Review Plan for Uprating the Licensed Power of a Pressurized Water Reactor Power Plant.”

- Licensing Topical Report NEDC-31897P-1, “Generic Guidelines for General Electric Boiling Water Reactor Power Uprate.”

The following list gives an idea of some of the plant parameters that were affected by a typical BWR power uprate:

- Total core flow

- Reactor recirculation pump speed

- Neutron flux

- Condensate flow

- Condensate demineralizer flow

- Final feedwater temperature

- Reactor feed pump suction pressure
- Feedwater flow
- Reactor feed pump turbine RPM
- Reactor feed pump discharge pressure
- Reactor steam dome pressure
- Main steam line flow
- Main steam line pressure
- Turbine inlet pressure
- Condenser pressure
- Offgas flow
- Circulating water temperature ΔT
- HPCI/RCIC steam supply pressure
- HPCI/RCIC discharge pressure
- Control rod drive pump discharge pressure
- Feedwater heater extraction steam flow (all five heaters)
- Moisture separator drain tank flow
- Feedwater heater drain cooler flow (all six heaters)
- Feedwater heater pressure
- Reactor Building cooling water heat load
- Reactor Building closed cooling water heat load
- Turbine Building cooling water heat load
- Turbine Building closed cooling water heat load
- HVAC heat load

- Service water heat load
- Cooling tower evaporation rate
- Reactor water makeup flow
- Isophase bus duct heat load
- Generator stator cooling and hydrogen
- Main transformer heat load
- Liquid/solid radwaste quantity
- Plant radiation levels

There are some additional benefits of a power uprate that can be considered in the cost-benefit analysis. One benefit is the review and updating of the licensing basis and design information. Also, degraded BOP components can be identified and their replacement or repair included in the megawatts gained and the cost of the uprate, especially if more capacity is needed for the uprate.

The thermal performance program can be a major help in determining the feasibility and cost benefit of a power uprate. For example, the thermal performance program may have already determined performance margins for the steam generators, the condenser, cooling tower, MSR, HP and LP turbines, FW heaters, FW pump, and so on. Therefore, the major effects of an uprate can be anticipated.

6.10 Turbine Upgrades

EPRI has prepared a *Main Turbine Performance Upgrade Guidelines*, which addresses in some detail the potential for various turbine steam path improvements. The following conclusions concerning turbine upgrade options were developed in the EPRI guideline:

1. A majority of turbine section replacements are being implemented based on benefits related to the perceived elimination of stress corrosion cracking in the disk bore, keyways, and blade attachments. During LP turbine section replacement, utilities have implemented other modifications to improve performance in order to offset the substantial capital costs of the repairs.

2. A variety of upgrade strategies are presently available for consideration by the industry. In general, the improvements offered can be characterized as further adjustments or refinements to the original steam flow path. These refinements can achieve between 1-5% improvement in unit output by:

- Reducing profile losses
- Reducing secondary losses
- Reducing leakage through seals
- Adding stages to the LP section
- Lengthening the L-0 blades and increasing the annulus area

3. Several available retrofit packages of advanced blades, stages, and sections have utilized CFDs to improve turbine stage designs. Dramatic improvements in section efficiencies cannot be expected from the application of CFD to existing designs. Current upgrade efforts represent the process of fine-tuning.

4. Because of its disproportionate contribution to the overall turbine output, the L-0 stage constitutes a unique area of potential improvement for the turbine. In lieu of a complete section replacement, there is an opportunity to progressively increase output by 1% or more by installing an improved L-0 stage. The replacement of eroded or worn diaphragms with stationary airfoils that are more precisely oriented to their original rotating counterparts also offers an opportunity for achieving nominal improvements as part of a planned, long-term upgrade strategy.

5. To date, the U.S. nuclear industry has done little to advance and apply the optical wetness probe, which was originally introduced in the 1980s as a technique to directly determine the power developed in an LP turbine. A probe currently being developed by the Electric Power Research Institute should advance this new technology. It will, however, require the support of the industry through field applications to eventually establish the validity of this technology as a cost-effective technique for establishing the LP used energy end point (UEEP) enthalpy. This new technique will be available as an alternative to using ASME PTC 6 measurements.

6. The reason for HP section replacement is different from that reported for most LP section replacements. HP section replacements can be economically justified by the increase in efficiency. Output improvements of 20-40 MWe were reportedly associated with HP section upgrades for units of 100-1200 MWe (2-3%). Because there is a ratio of 1:2 or 1:3 in the number of HP turbines when compared with LP turbines, the addition of an improved HP turbine may become more economically palatable for those utilities that have embarked on a replacement strategy of LP sections.

7. As previously mentioned, the upper limit of a current section replacement is likely to provide an additional 5-6% output depending upon the vintage of the unit and the extent of the upgrade. Tobtain additional improvements, U.S. utilities should

monitor the success of more advanced designs that have been instituted by some of the European and Asian plants. For example, the application of L-0 blades up to 72" long are planned for the 1500 MWe Arabelle unit operated by the Electricite de France.

Based on the previous conclusions, the EPRI Task Group responsible for the *Main Turbine Performance Upgrade Guidelines* has developed the following lessons learned related to upgrading turbines:

1. Extending maintenance and inspection intervals by installing new rotor replacements remains the principal factor that justifies this costly investment.

2. Owners of new generation LP rotors have a long-term vested interest in the development of a superior SCC-management technology. This will ensure that further reliability issues do not compromise projected efficiency gains.

3. A variety of similar performance upgrade options are currently available for the original Westinghouse and General Electric turbines.

4. Advanced measurement techniques that support in-situ testing of turbines to define the inlet/outlet flow conditions of selected LP stages (including the L-0 stage) have been available since the 1980s. The CFD analytical predictions are only as reliable as the input used to characterize the boundaries of the portion of the turbine being modeled. Field testing, therefore, remains a vital part of successful design optimization with CFD, particularly where the goal is to achieve or qualify LP section replacement enthalpy gains.

5. Utility experience indicates that significant MWe increases can be achieved by upgrading the turbine flow path. Increases have been achieved by replacing all of the four major LP turbine sections originally designed by Westinghouse and General Electric. The margin for improvement, however, is limited. Potential increases can also be limited by the degradation of other components within the steam plant cycle, such as the steam generator.

6. The primary reason for performing an L-0 blade or diaphragm retrofit is to resolve blade degradation concerns. The increased number of MWe's alone does not justify the cost of replacing the large number of blades in this stage.

7. It is important for utilities to accurately measure LP power output before and after turbine upgrades. Accurate measurements are required to ensure that MWe improvement promised by vendors was actually achieved.

8. Utilities should distinguish between HP output improvements attributable to the use of an advanced steam path versus also increasing the mass flow into the HP

inlet within a vendor's offer. The latter constitutes an uprate rather than an upgrade of the original design.

9. Development of longer blades designed from lighter materials, such as titanium, may provide nuclear plant operators with their next opportunity to effectively upgrade the performance of their LP sections.

Modifications of exhaust hoods are generally part of an overall LP section replacement. Diffuser testing and CFD analysis has shown that the flow is extremely complex. Small changes can produce large effects on overall pressure loss. The flow in the exhaust hood depends upon inlet conditions. This dependence indicates the need to optimize the integrated design of hood geometry and last stage blade design.

6.11 House Load Reduction

Most of the discussion in this document addresses issues related to turbine output and heat rate. However, the bottom line in terms of revenue and O&M costs is net unit output, defined as generator output less the unit electrical loads. Many of the unit's electrical loads are required to operate the plant. Included are reactor coolant pumps, circulating water pumps, condensate pumps, and motor-driven feedwater pumps. It is difficult to reduce these loads, except possibly at reduced reactor power. Therefore, only a fraction of the total unit electrical load can be a potential for consideration in improving net unit output. Despite this admonition, the performance engineer should have a clear list of the unit's electrical loads and considerations for reduction of power consumption and power factor improvement. Methods of power reduction include using more efficient motors, securing some pumps during extended part power operation, and using variable speed motors. The effects on operating limits and component performance, such as the condenser pressure with reduced cooling flow, must be evaluated. Appendix C.4 of EPRI's *Megawatt Improvement Casebook and Guidelines* discusses a typical program including an extensive list of potential improvement candidates.

6.12 References

1. C. S. Sullivan, "Feedwater Flow Venturi Fouling Study." EPRI Nuclear Plant Performance Seminar, Asheville, North Carolina (September 1996).

2. *Development of an Ultrasonic PWR Primary Coolant Temperature/Flow Measurement System (RCSM)*. Electric Power Research Institute, Palo Alto, California: December 1996. Report TR-106821-V1.

3. T. McCloskey et al, “Low Pressure Steam Turbine Thermal Performance Improvements,” *Thirteenth Annual Plant Performance Enhancement Program (P^2EP) Annual Meeting Proceedings*, August 1997.

4. *Main Turbine Performance Upgrade Guideline*. Electric Power Research Institute, Palo Alto, California: January 1997. Report TR-106230.

5. *Moisture Separator Reheater Source Book*. Electric Power Research Institute, Palo Alto, California. Report TR-106345.

6. *Procurement and Operation Considerations for Moisture Separator Reheaters.* Electric Power Research Institute, Palo Alto, California: September 1984. Report NP-3692.

7. TEMA Standards.

8. *Nuclear Plant Feedwater Heater Handbook*, Volumes 1 - 3. Electric Power Research Institute, Palo Alto, California: June 1985. Report NP-4057.

9. Jim Nicholson, “Nuclear Power Plant Core Power Capacity Uprating,” Stone and Webster Engineering Corporation.

10. *Megawatt Improvement Casebook and Guidelines*. Electric Power Research Institute, Palo Alto, California: December 1992. Report TR-101867.

7

OPTIMIZING GENERATION FOR OFF-NORMAL PLANT CONDITIONS

Sometimes nuclear power plants need to change the strategy of operating the steam cycle in order to achieve other objectives. Usually, the change results in degraded thermal performance conditions and an increase in plant heat rate, but an overall improvement in other conditions that meet the plant operating strategy. The most common examples are end of cycle coastdown, reduced T_{ave}, and reduced feedwater temperature.

7.1 End of Cycle Coastdown

Nuclear plants usually replace part of the nuclear fuel at fixed intervals, for example, 18 or 24 month intervals. The amount of enrichment in the fuel is set to allow full power operation for the full interval between refueling. It is possible to extend fuel burnup and the interval between refuelings by gradually reducing power toward the end of the fuel cycle. Power reduction reduces the amount of core reactivity tied up in power coefficients that becomes available for additional fuel burnup and electrical production before refueling.

End of cycle coastdown has the disadvantage of decreasing plant electrical output and increasing heat rate, resulting in additional loss of electrical output. However, it may make economic sense to coastdown considering the added fuel burnup and deduction in spent fuel storage over the remaining life of the plant.

7.2 Reduced Feedwater Temperature

There can be situations when the capacity of the HP turbine and control valves exceeds the capability of the reactor system to deliver full steam flow without reducing main steam pressure. A case in point is when the heat transfer capability of the steam generators is reduced by tube plugging or fouling. One way to compensate for this condition is to reduce the feedwater temperature by reducing or stopping extraction steam to the top feedwater heaters. This has the effect of (1) increasing the steam flow through the turbine because less steam is extracted and (2) improving heat transfer conditions including steam pressure because colder feedwater results in an increased mean effective temperature difference between the reactor coolant and the steam.

7.3 Reduced T_{ave}

In recent years, it has become evident that a high PWR hot leg temperature, T_{hot}, can have detrimental effects on the life of steam generators. Some plants have or are considering a reduction in T_{hot}. If T_{hot} is reduced, then the cold leg temperature, T_{cold}, must also be reduced by the same amount in order to maintain the same reactor power. If T_{cold} is reduced, then steam side temperature and pressure in the steam generators must be reduced in order to maintain the same heat transfer rate. This will result in reduced steam cycle efficiency and generator output. Because turbine capacity depends on maintaining high steam pressure, the reactor power level may have to be reduced as a result of reduced steam pressure, further reducing plant output.

One way to compensate for a reduction in T_{ave} is to reduce the feedwater temperature as discussed previously.

In summary, the workarounds discussed above can be somewhat interactive. For example, it maybe advantageous to reduce FW temperature when T_{ave} reduction is implemented. Solutions as described above and their mix are plant-unique.

8

GLOSSARY

8.1 Definitions

adjusted actual gross heat rate. The gross heat rate attained in the normal equipment lineup during one 24-hour period each month, corrected to design condenser backpressure, expressed in BTUs per kilowatt-hour (electric).

average adjusted actual gross heat rate. The heat rates attained in the normal equipment lineup during each week of the month and adjusted to 100 % power (used in calculating the INPO thermal performance indicator).

best achievable heat rate (BAHR). The heat rate at which the unit is capable of operating, that, the design heat rate corrected for ambient conditions and other conditions not chargeable to the cycle. The INPO thermal performance Indicator uses a best achievable gross heat rate that has been attained by the plant referenced to 100 % power. It may be obtained from performance monitoring, calculations, or a thermal performance test. INPO documents specify standard adjustments to the BAHR to account for unavoidable conditions.

cold reheat. Steam from high pressure turbine exhaust to the moisture separator reheater. Also known as high pressure turbine exhaust.

design heat rate. The heat rate predicted during design.

design information. Information concerning the turbine cycle that was used during plant design.

duty. Work done by a condenser, expressed in Btu / hr. (Also referred to as heat rejected.)

gross turbine heat rate. The heat rate based on heat delivered to the turbine and the generator output (for cycles with motor-driven feedwater pumps).

gross unit heat rate. The heat rate based on the heat generated by the reactor core and the generator output.

heat rate. The heat consumption rate per unit of output, that is, Btu / kW-hr. Generically, heat rate is defined as the heat delivered to the turbine minus the heat returned to the steam generator, divided by the electrical generator output.

moisture removal effectiveness. The amount of moisture removed by the moisture separator divided by the amount of moisture entering the moisture separator times 100.

net electrical power. Electrical generation sent out from a station, expressed in MWe, net.

net turbine heat rate. The heat rate based on heat delivered to the turbine and the generator output (for cycles with steam-turbine-driven feedwater pumps).

net unit heat rate: The heat rate based on the heat generated by the reactor core and the plant net output (generator output less power used by the unit).

performance engineer. The person, persons, or organization responsible for monitoring and analyzing thermal performance and providing related information to management and other organizations.

realistic achievable heat rate. The best achievable heat rate corrected for cycle conditions and component degradations that, at the time of evaluation, are not economical or practical to correct.

temperature rise. Increase of circulating water temperature from inlet to outlet of condenser water boxes.

terminal temperature difference (FW). The feed water heater (FWH) shell-side temperature minus the temperature of the exiting feedwater.

terminal temperature difference (MSR). The outlet temperature of the superheated reheat system minus the temperature of the exiting steam / liquid in the MSR tube bundles.

thermal kit. A set of heat balances and graphs that describes the design performance of the unit.

thermal performance. The effectiveness or efficiency of the turbine cycle and its components in converting thermal into electrical energy measured by generator electrical output and heat rate. Heat rate is inversely proportional to efficiency.

thermal performance index. An INPO performance indicator. The ratio of the design gross heat rate (corrected) to the average adjusted actual gross heat rate.

8.2 Acronyms

A/D Analog to digital

ANSI American National Standards Institute

AOV Air operated valve

ASME American Society of Mechanical Engineers

BOPPMG Balance of Plant Performance Monitoring Group

BWR Boiling Water Reactor

CFD Computational Fluid Dynamics

CRD Control rod drive

DCA Drain Cooler Approach

ECCS Emergency Core Cooling System

ESFAS Engineered Safety Feature Actuation System

FFT Final Feedwater Temperature

FIC Flow induced corrosion

FW Feedwater

FWH Feedwater Heater

HP High Pressure

INPO Institute for Nuclear Power Operations

LP Low Pressure

LWR Light Water Reactor

MS Moisture Separator

MSL Main steam line

MSR Moisture Separator Reheater

MWe Electrical Megawatts

MWt Thermal Megawatts

NDE Nondestructive Examination or Evaluation

NSSS Nuclear Steam Supply System

O&M Operations and Maintenance

OEM Original Equipment Manufacturer

OTSG Once through steam generator

P^{2}EP Plant Performance Enhancement Program

PC Personal Computer

PEPSE Performance Evaluation of Power System Efficiencies

PLEX Plant Life Extension

PTC Performance Test Code

PWR Pressurized Water Reactor

R&D Research and Development

RCS Reactor Coolant System

RH Reheater

RPS Reactor Protection System

RTD Resistance Temperature Detector

RWCU BWR Reactor Water Cleanup

SCC Stress Corrosion Cracking

SG Steam Generator

SJAE Steam Jet Air Ejector

SSR Steam seal regulator

T/C Thermocouple

TPE Thermal Performance Engineer

TPI Thermal Performance Index

TR Temperature Rise, deg F

TTD Terminal Temperature Difference

UEEP Used Energy End Point

UFM Ultrasonic Flow Meter

VWO Valves wide open

8.3 Nomenclature

δ Total error of a measurement

μ Bias error

σ Random precision error

h Enthalpy, Btu / lb

ID Inside diameter, inches

k Flow coefficient

m Flow, lb / hr

P kW or hp

p Pressure, psia

Q Heat transfer or generation rate, Btu / hr

t Temperature, degF

TR Temperature rise, degF

U Overall heat transfer coefficient, Btu / hr-ft^2-degF

v Specific volume

X Quality of steam, percent of dryness

Subscripts

a Addition flow

c Clean

eg Gross electrical output

en Net electrical output

f Saturated water

fg Evaporation

fw Feedwater

fwe Feedwater extraction

h Heater

id Incoming heater drips

o Overall

od Outgoing heater drain

out Primary system heat losses

p Reactor pump energy

r Reactor or steam generator

s Steam

sat Saturated

sg Steam generator

t Heat delivered to steam cycle

WARNING: This Document contains information classified under U.S. Export Control regulations as restricted from export outside the United States. You are under an obligation to ensure that you have a legal right to obtain access to this information and to ensure that you obtain an export license prior to any re-export of this information. Special restrictions apply to access by anyone that is not a United States citizen or a Permanent United States resident. For further information regarding your obligations, please see the information contained below in the section titled "Export Control Restrictions."

Export Control Restrictions
Access to and use of EPRI Intellectual Property is granted with the specific understanding and requirement that responsibility for ensuring full compliance with all applicable U.S. and foreign export laws and regulations is being undertaken by you and your company. This includes an obligation to ensure that any individual receiving access hereunder who is not a U.S. citizen or permanent U.S. resident is permitted access under applicable U.S. and foreign export laws and regulations. In the event you are uncertain whether you or your company may lawfully obtain access to this EPRI Intellectual Property, you acknowledge that it is your obligation to consult with your company's legal counsel to determine whether this access is lawful. Although EPRI may make available on a case by case basis an informal assessment of the applicable U.S. export classification for specific EPRI Intellectual Property, you and your company acknowledge that this assessment is solely for informational purposes and not for reliance purposes. You and your company acknowledge that it is still the obligation of you and your company to make your own assessment of the applicable U.S. export classification and ensure compliance accordingly. You and your company understand and acknowledge your obligations to make a prompt report to EPRI and the appropriate authorities regarding any access to or use of EPRI Intellectual Property hereunder that may be in violation of applicable U.S. or foreign export laws or regulations.

About EPRI

EPRI creates science and technology solutions for the global energy and energy services industry. U.S. electric utilities established the Electric Power Research Institute in 1973 as a nonprofit research consortium for the benefit of utility members, their customers, and society. Now known simply as EPRI, the company provides a wide range of innovative products and services to more than 1000 energy-related organizations in 40 countries. EPRI's multidisciplinary team of scientists and engineers draws on a worldwide network of technical and business expertise to help solve today's toughest energy and environmental problems.

EPRI. Electrify the World

Programs:

Nuclear Power

TR-107422-V2

EPRI • 3412 Hillview Avenue, Palo Alto, California 94304 • PO Box 10412, Palo Alto, California 94303 • USA
800.313.3774 • 650.855.2121 • askepri@epri.com • www.epri.com