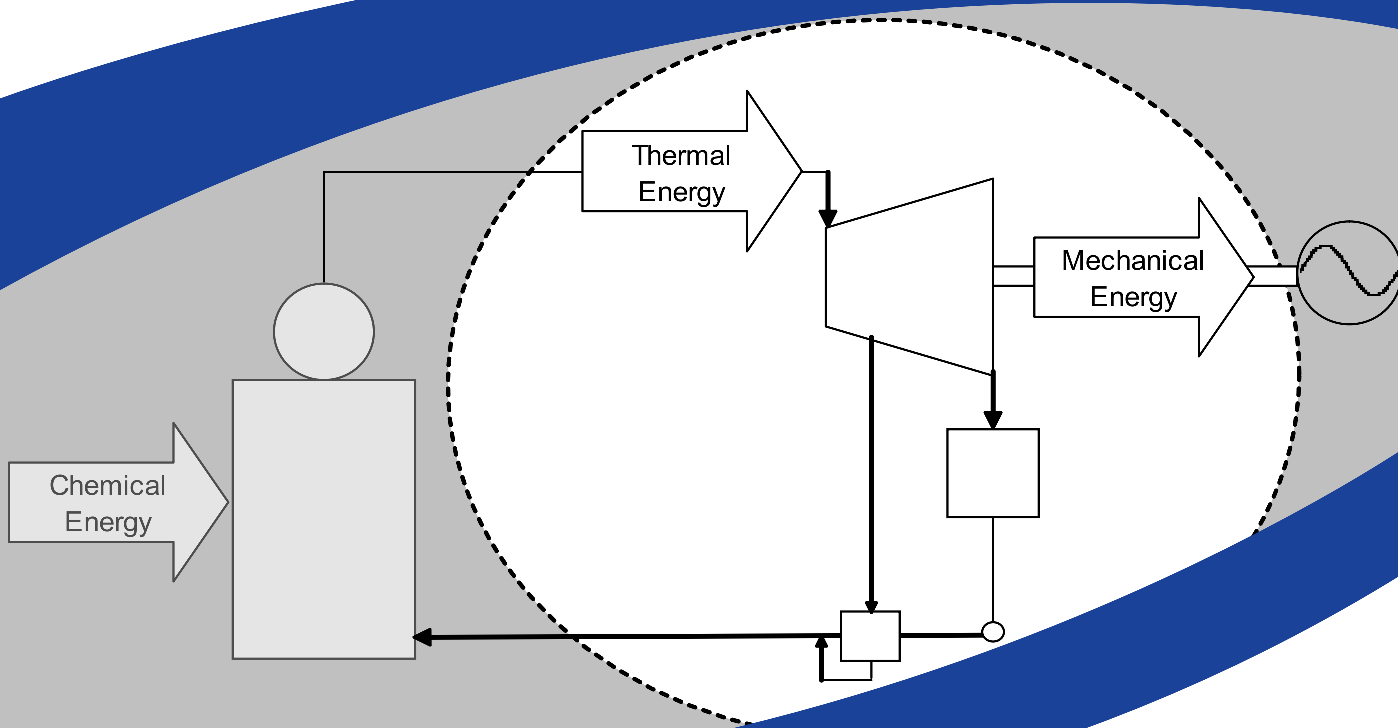
ELECTRIC POWER RESEARCH INSTITUTE

2010 TECHNICAL REPORT

Steam Turbine Performance Engineer's Guide



Steam Turbine Performance Engineer's Guide

1019657

Final Report, December 2010

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ACKNOWLEDGMENTS

The following organization, under contract to the Electric Power Research Institute (EPRI), prepared this report:

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This report describes research sponsored by EPRI.

EPRI acknowledges the valuable help from the following Mechanical Dynamics & Analysis, Ltd., personnel:

Larry Anderson, Manager, Field Engineering Services

Matt Chamberlin, Project Manager, Outage Services

David Hicks, Project Manager, Steam Turbine Retrofits

Lawrence A. Penna, Performance and Test Consultant

This publication is a corporate document that should be cited in the literature in the following manner:

Steam Turbine Performance Engineer's Guide. EPRI, Palo Alto, CA: 2010. 1019657.

REPORT SUMMARY

Background

The importance of power plant performance has been recognized recently because good performance can reduce fuel costs and reduce CO_2 emissions. The steam turbine is the workhorse of most power plants. Its performance and reliability relate directly to the performance and reliability of the power plant that it serves. The actions of the turbine performance engineer are crucial to its high level of performance. However, in many cases, that engineer is assigned many other duties and/or is an early career engineer placed into this position without previous experience.

The *Steam Turbine Performance Engineer's Guide* is meant to present the steam turbine performance engineer with the expected and important functions and responsibilities necessary to succeed in this position that are not necessarily taught in college. The instructions and recommendations in this guide, when properly executed, will improve the effectiveness of steam turbine performance engineers, positively affecting both the performance and reliability of the steam turbines under their care.

Power plants and power-generating companies with structured or active performance programs have typically been industry leaders in reliability and performance.

Objective

- To provide a description and the frequency of the performance engineer’s periodic activities
- To supply technical information to serve as a reference manual
- To make recommendations on steam turbine and performance-related training

Approach

In cooperation with interested EPRI members, the project team developed an outline for a guideline describing the activities of a successful turbine performance engineer. The expertise to develop the guide was identified in a contractor who developed draft guides based on experience and industry standards and references. Several EPRI members reviewed these drafts and provided recommendations for the fine-tuning needed to maximize their usefulness.

Results

This comprehensive guide was developed by drawing on decades of industry expertise. It contains the program structure for the coordinated actions necessary to maintain or improve steam turbine performance and reliability. The guide provides the details for engineers serving in this capacity to successfully and positively affect the performance and reliability of steam turbines.

EPRI Perspective

Stations with a performance program perform better than those without one. A performance program provides the data for decision making with respect to timely maintenance. Monitoring the performance of a steam turbine includes the trending of parameters that also describe the performance of other plant components, providing insight and information on improving their operation. A performance program creates a culture centered on improving plant performance. The sharing of performance data with the entire plant staff strengthens their understanding of how each staff member may contribute, making the improvement of plant performance a team effort.

Through the use of this guideline, EPRI members should be able to establish and maintain a turbine performance improvement program and ultimately improve component performance and unit heat rate.

Keywords

Heat rate
Performance engineer
Performance program
Steam turbine
Turbine performance
Turbine system engineer

ABSTRACT

The purpose of the *Steam Turbine Performance Engineer's Guide* is not to review the thermodynamics of the steam cycle—there are many textbooks available for that. Rather, the guide is meant to present the steam turbine performance engineer with the expected and important functions and responsibilities necessary for the job that are not necessarily taught in college. The instructions and recommendations in this guide, when properly executed, will improve the effectiveness of steam turbine performance engineers, positively affecting both the performance and reliability of the steam turbines under their care.

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1 INTRODUCTION

Steam turbine performance is important. When the steam turbine efficiency is high, emissions decrease because less fuel is used for every kilowatt of power produced. Less fuel use results in significant savings because fuel is often the largest cost of operating a power plant. Wear and tear on the fuel handling equipment decreases, and more power per pound of steam is produced. With less flow per kilowatt, there is less heat rejected in the condenser and corresponding heat-rejection equipment, and ultimately the environment.

When availability and reliability are in good order, planned outages may be postponed if performance is also in good order. The postponement of a planned outage postpones the costs of the outage and the replacement power. Paying close attention to the turbine's thermal performance can aid in early detection of reliability problems.

Measuring the turbine performance before and after an upgrade, modification, or other change to the turbine will allow the owner to quantify the corresponding change in output, efficiency, fuel consumption, throttle flow, and so on. This measurement also determines whether the turbine passed any guaranteed performance criteria, which is often associated with bonuses and/or liquidated damages.

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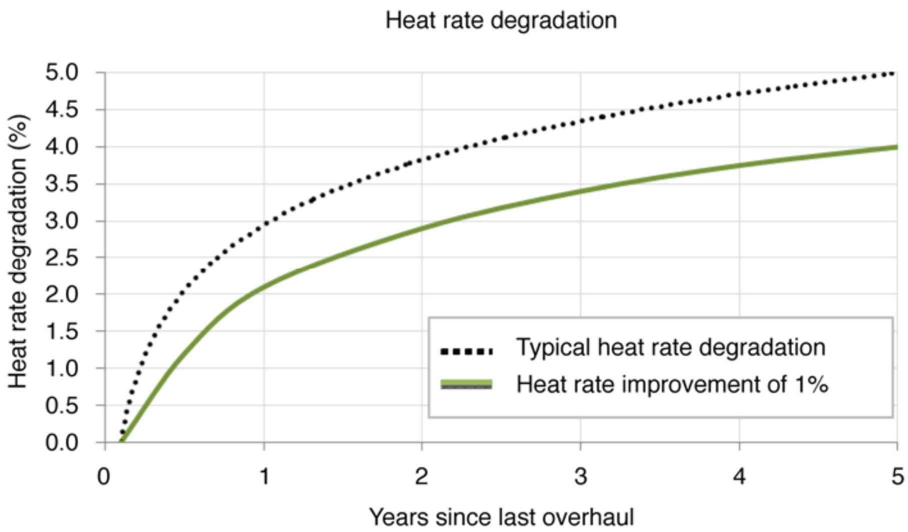

Figure 1-1
Typical heat-rate degradation with and without a heat-rate improvement program

The effect of decreasing heat rate can be tremendous. It can be quantified by the following equation:

Annual Fuel Savings = %ΔHR * 8760 * CF * PWR * FC/10,000,

where

%ΔHR = the percentage of change in heat rate (%)

8760 = the number of hours in a year

CF = the capacity factor (%)

PWR = the rated power output (in kilowatts)

FC = the fuel cost (in dollars per one million/MMBtu)

So, for a 500-MW unit operating at 97.6% capacity with fuel costs of $3/MMBtu, and a design heat rate of 7800 Btu/kWh, the fuel cost savings from a 1% heat-rate improvement is as follows:

Annual Fuel Savings = 0.01 * 7800 Btu/kWh * 8760 h/yr * 0.976 * 500,000 kW * $3/MMBtu

Annual Fuel Savings = $1,000,000/yr

For a sustained heat-rate improvement of 1% per year over the course of five years, the annual fuel savings is $5 million.

Following the heat-rate improvement program described in this guide can result in a heat-rate improvement of 1% per year or more. The troubleshooting from testing and analysis of data, inspections, changes to operation, repairs, and upgrades all are components of the overall program that can increase power output, reduce fuel costs, and decrease emissions.

Establishing a successful heat-rate improvement program does, however, require a commitment of labor and funding.

Engineering Units

English units (U.S. customary) are used for parameters in this report. The following table enables report users to convert to Standard International (SI) units where applicable.

Table 1-1
Metric conversion table

Parameter	U.S. Customary	SI	Conversion Equation
Area	ft^2	m^2	1 ft^2 = 0.0929 m^2
	in^2	mm^2	1 in^2 = 645.2 mm^2
Density	lbm/ft^3	kg/m^3	1 lbm/ft^3 = 16 kg/m^3
Enthalpy	Btu/lbm	kJ/kg	1 Btu/lbm = 2.326 kJ/kg
Heat	Btu	J	1 Btu = 1055 J
Linear dimension	inch	mm	1 in. = 25.4 mm
	foot	meter	1 ft = 0.3048 m
Mass	lbm	kg	1 lbm = 0.4535 kg
Mass flow	lbm/h	kg/s	1 lbm/h = 0.000126 kg/s
	lbm/s	kg/s	1 lbm/s = 0.454 kg/s
Pressure	inches Hg	kPa	1 inHg = 3.386 kPa
	psi	kPa	1 psi = 6.895 kPa
Specific volume	ft^3/lbm	m^3/kg	1 ft^3/lbm = 1/16 m^3/kg
Temperature	°F	°C	1°F = 1.8 * °C + 32
Velocity	ft/min	m/s	I ft/min = 0.0051 m/s
Volumetric flow	ft^3/min	m^3/s	1 ft^3/min = 0.000472 m^3/s

2
PURPOSE AND ROLES OF A TURBINE PERFORMANCE ENGINEER IN A STEAM POWER PLANT

The primary role of a steam turbine performance engineer is to improve and maintain the efficiency and power output of the steam turbine cycle. The measurements of success are the gross turbine heat rate (GTHR) and the gross generator output. The two go hand in hand: a plant may be very efficient, but at the same time be deficient in power output. Conversely, a plant may generate more power than design, yet do so inefficiently. The desired combination, as well as the steam turbine performance engineer's responsibility, is a turbine cycle that produces maximum power at maximum efficiency over the long term.

Figure 2-1 shows the steps involved in a heat-rate improvement program. The process centers on the turbine outage. There is a great deal of effort that goes into planning for the outage so that the time spent off-line is minimized and the cost of steam path repairs is minimized. The figure shown is but one cycle in the continuum that describes a lifetime of maintaining or improving turbine performance.

The first step is to determine the best expected or "new and clean" performance of the unit. This expectation is the basis for comparison during future tests. The next step is to conduct a pre-outage test so the current condition of the turbine cycle can be compared to the best expected performance. Some portion of the difference between the two will be normal wear and tear. Other portions may result from atypical or acute damage to the steam path. During the pre-outage test analysis, the condition of the steam path is estimated. Based on this estimate, plans can be made to address problems during the outage. During this phase, replacement parts and subcontractors are lined up for the upcoming outage. The goal is to enter an outage with a high level of preparedness, an educated expectation of the condition of the steam path, and a plan for restoring the steam path to as close to its new and clean condition as possible. A higher goal is to improve it beyond its new-and-clean condition with steam path upgrades.

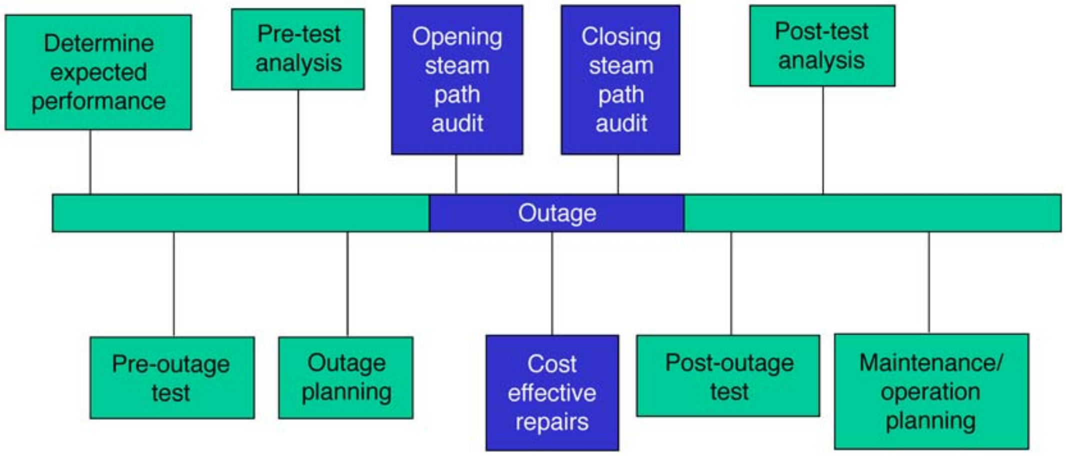

Figure 2-1
One outage cycle for a heat-rate improvement program

Time and resources are of usually in short supply during an outage. An opening steam path audit is conducted as the unit is disassembled to identify and quantify damage to the steam path. Together with the repair costs, the steam path audit allows the steam turbine performance engineer to calculate the cost/benefit ratio for specific steam path repairs or replacements. The list of repairs can be prioritized by return on investment and used as a guide for maintenance decisions during the outage. Typically, the repairs that result in the highest return of Btu/kWh per dollar spent are most economical. Other times, the repairs that result in the largest absolute savings are desirable.

A closing steam path audit is conducted after steam path repairs are complete during the turbine reassembly process. The closing steam path audit serves two purposes: to predict the return-to-service condition of the steam path and to serve as a quality control check on the maintenance conducted during the outage.

Once the turbine is back on-line, a post-outage test allows the determination of the gains in performance as a result of the maintenance conducted during the outage. The resulting analysis shows the gains made by each casing and is used to reconcile the opening and closing steam path audits with the pre- and post-outage tests. With this analysis come recommendations for continued operation with high efficiency. The result of the heat-rate improvement process is to make adjustments to operation and maintenance to maintain a high level of efficiency and power output.

Figure 2-1 shows that steam turbine performance can be improved both while the unit is operating and while it is down for an overhaul or other maintenance. The responsibilities of a steam turbine performance engineer can therefore be divided into two general categories: operation and maintenance.

Operation

Good long-term heat-rate improvement requires understanding of the following three important facets of the steam turbine's operation:

- The unit's historical performance
- The turbine's current performance compared with its baseline
- How operation affects performance

The first is an understanding of the unit's historical performance. When it was first commissioned, the unit was typically performing at its best. This new and clean condition could be the benchmark against which all future operation is judged—or, if the unit undergoes some sort of upgrade or modification, the post-outage test immediately after the steam path improvement might become the new benchmark. Either way, it is important to know the unit's operational history as it made its way through its break-in period, shifts from base loading to cycling and vice versa, upgrades and modifications, major overhauls, and performance problems and their solutions. Knowledge of the unit's history will help in anticipating problems and will allow better prediction of the best expected condition of the unit after future overhauls. For example, if the unit has a history of solid particle erosion (SPE) damage and nothing has changed with the operation or maintenance of the unit since the last outage, it is reasonable to expect SPE damage in the upcoming outage.

The second important facet of the unit's operation is to determine the unit's current condition so that it may be compared with the best expected condition. As the turbine runs, its performance naturally degrades because of normal wear and tear. Some problems might accelerate the degradation process. An event that causes an abrupt decrease in performance might be enough to shut the unit down for an immediate overhaul. Other times, a performance-degrading event might be ameliorated by changing the load, startup procedure, or some other operational change without having to shut the unit down. Either way, knowledge of the current condition and details of unit performance can greatly aid in making an informed decision about operational changes. Knowing the current condition aids in planning for outages when time and maintenance budgets are limited.

The third important facet is an understanding of the effect that operation has on performance. There are many cause-and-effect relationships between how the unit is operated and how the unit performs. As an owner, it is undesirable to be on the wrong end of the "it's how you operated it" versus "it's how you designed it" continuum when negotiating warrantee issues. Following the operating guidelines, documenting operational practices, and understanding how operation affects performance are key to continued high levels of performance.

These three facets are described in detail in the following sections.

Understand the Turbine's Historical Performance

Starting with the design and then the commissioning, the best expected performance of the turbine cycle can be determined using nonproprietary information.

Design Information

The design information that the owner receives from the turbine original equipment manufacturer (OEM) is relevant for the operation and maintenance of the turbine. Understandably, there will be very little information that may be used in reverse engineering parts of the turbine. There is, however, adequate information to determine the turbine's design performance, but the owner must recognize that there is some manufacturer margin built into the design documentation.

Thermal Kit

The thermal kit is a collection of nonproprietary design performance information for the steam turbine. It may contain the following information:

- Extraction-stage shell pressures
- Heat rate as a function of output or loading curve
- First-stage shell pressure versus throttle flow
- Gland or steam seal leakage
- Mechanical losses
- Expansion lines
- Exhaust pressure correction curve
- Generator losses
- High-pressure turbine expansion line endpoints
- Reheat turbine expansion line endpoints
- High-pressure turbine internal efficiency
- Reheat turbine internal efficiency
- Heat balances at various loads
  - Guarantee output
  - Valves wide open (VWO)
  - 5% over pressure (OP)
  - Several part-load points corresponding to valve points
  - Controlled extraction/admission flow on or off

- Exhaust loss curve
- Correction curves, for example, for off-design steam temperatures and pressures

The thermal kit will prove essential to the steam turbine performance engineer in evaluating the turbine's performance. The first step in establishing a heat-rate improvement program is to determine the design or new and clean condition of the turbine. The guarantee heat balance is a logical and a usually readily available place to find this information (see Appendices A and B.)

Vintage

The vintage of the turbine has a great influence on its maximum efficiency. Many events over the decades have resulted in improvements in steam path design and performance.

Domestic large utility turbines built in the 1950s and 1960s were generally workhorses with ample manufacturer margins. Although built with excess capacity, those same turbines by today's standards are very inefficient. Foreign manufacturers had not yet entered the U.S. market, so most large steam turbines in the United States were supplied by either General Electric or Westinghouse. Fuel costs were inexpensive and passed on to the consumers, and many units were designed to be base loaded. Electrical generators were physically large but produced relatively few megawatts because they were limited by air cooling. Power output was limited by generator size; therefore, cross-compound units were developed to use two generators per turbine. The first generation of supercritical units entered the market. They were relatively small and prone to copper deposits. The first nuclear power plants were commissioned in the mid-1950s. In the late 1960s, turbine manufacturers distributed starting and loading instructions to owners in an effort to reduce turbine stresses. Metallurgy was not as sophisticated as it is today, and many rotors contained bores to remove poor-quality material from which cracks could initiate.

With the improvement of materials throughout the 1960s and 1970s, steam inlet temperatures increased, which allowed power output and efficiency to increase as well. The advent of water-cooled generators and other improvements in generator manufacturing in the late 1960s and early 1970s allowed generator ratings to increase; therefore, new units were tandem compound instead of cross-compound. Improvements were made in once-through boilers, and supercritical units were placed in cycles without copper tubing in the feedwater heaters and condenser, and supercritical turbine ratings increased. The turbine-generator units manufactured in the 1970s still had ample margin. With the invention of the microprocessor in 1971, higher level stage design calculations were possible. Vane profiles improved, which increased stage efficiency. Some utilities purchased several identical units, but conducted an acceptance test on only one, leaving many without the instrumentation ports used for routine monitoring.

In the early 1980s, there was a recession and a downturn in demand. The Three Mile Island event in 1979 placed other pressures on the nuclear industry. Turbine owners were concerned with replacing their aging fleet. Older Rankine-cycle plants were changing out their boilers for gas turbines and heat recovery steam generators (HRSGs) to improve overall plant efficiency. Material science continued to improve and design inlet steam temperatures continued to rise, which increased efficiency and ratings. Improvements in the resistance to SPE were made in high-pressure (HP) and intermediate-pressure (IP) turbines. In an effort to control maintenance costs, utilities balanced delaying outages with the risk of incurring greater damage to the steam path. The traditional 5 year overhaul cycle was being extended to 8 years and longer. In the mid-1980s computational fluid dynamics (CFD) allowed solution of the three-dimensional Navier-Stokes equations, which gave designers the ability to refine blade designs to become more efficient. Computerized control systems reduced the number of plant personnel and allowed more efficient operation and response to dispatchers. Advances in instrumentation and software allowed on-line and periodic monitoring for predictive maintenance programs.

In 1990 came a significant amendment to the 1970 Clean Air Act, and with that came significant increases in auxiliary load. The 1990s also saw competition from foreign turbine manufacturers and deregulation of the electric power industry. Demand increased from the slump in the 1980s, and overseas demand increased in emerging markets. All these changes gave incentive to turbine manufacturers to build competitively priced, efficient steam turbines. Gone were the days in the 1970s and 1980 when large steam turbines ran at low loads just to be ready for high demand. Instead, units that were designed for base load were cycling to match demand. The utility modus operandi went from “will run if it can make money” to “will not run if it cannot make money.” With deregulation, performance came to the forefront because fuel costs were now a cost of doing business rather than a pass-through cost. Combined cycle plants with their high efficiency were the preferred design for new plants. Many traditional Rankine plants were updated to combined-cycle plants. Steam turbines were being designed specifically for combined-cycle use with single casings that could respond more quickly to temperature shifts. Demand in natural gas to operate the many new combined-cycle plants drove up the price of natural gas. Increased competition also brought smaller manufacturer margins and increased acceptance testing. More units failed acceptance tests than in the past. CFD analysis and continued improvements in material science allowed longer and more efficient last-stage blades—a significant efficiency improvement because the last stage produces roughly 10% of the turbine’s power.

The new millennium brought with it continued competition to produce efficient steam paths. Many new seal designs were invented and others improved on, lead time for parts decreased significantly, OEMs placed more emphasis on service, and alternative energy came into focus, further increasing the cycling and load after duty of the existing fleet. Combined-cycle plants continue to make efficiency improvements. Metallurgy has improved, and rotor bores are not as common. The current atmosphere is all about upgrades. Units are essentially “gutted” to bring older steam paths up to today’s standards. Turbine suppliers provide parts and entire turbine sections for other manufacturers’ turbines. It is not uncommon to see an upgraded turbine-generator composed of parts from three different turbine manufacturers.

Heat Rate

Heat rate is the inverse of the cycle efficiency in which the purpose of the cycle is to convert the fuel’s thermal energy into electrical energy. There are generally two categories of heat rate: net unit heat rate (NUHR) and gross turbine heat rate (GTHR) (Figure 2-2). In cases in which there is one unit in the plant, the NUHR is sometimes referred to as the net plant heat rate (NPHR).

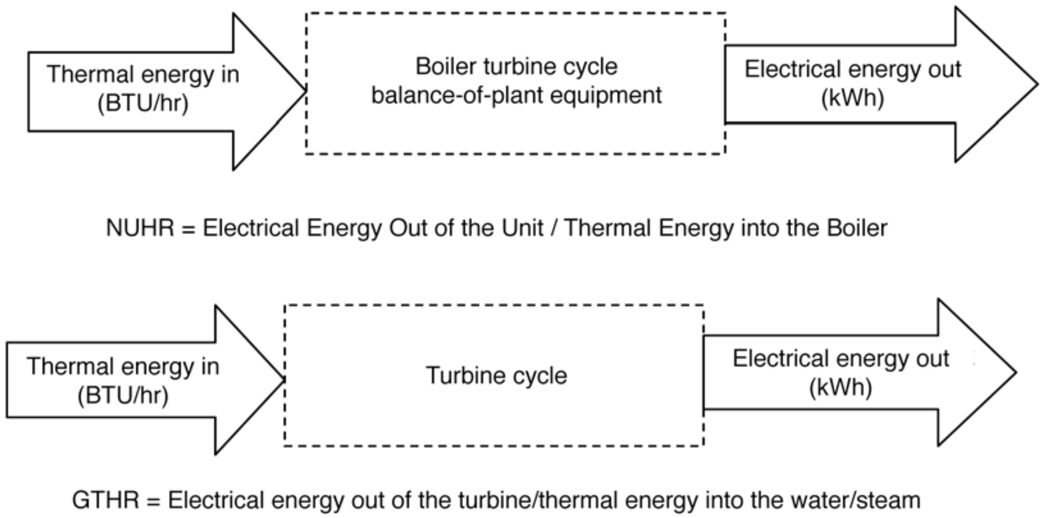

Figure 2-2
Definitions of heat rate

The difference between the two heat rates is where one draws the box around the cycle components. If the box is drawn around the entire power plant, its efficiency is the energy delivered to the electrical grid divided by the energy of the fuel delivered to the plant. The corresponding heat rate is its inverse, commonly referred to as the NPHR.

Now consider all the equipment that expands, condenses, pumps, and heats the steam and water in the turbine cycle from the boiler outlet to the final feedwater location. The efficiency of this cycle is the gross power out of the generator divided by the heat added to the final feedwater and cold reheat steam by the boiler, superheater, and reheater. The GTHR is its inverse. As a steam turbine performance engineer, considering the GTHR is the top priority. For actions described in this guide, that does not include the boiler or auxiliary power.

Heat balances usually show the heat-rate equation and resulting heat rate. If not, there is sufficient information on the heat balance to calculate it. However, because the turbine manufacturer does not include detailed information about the boiler, the heat-rate equation provided by the turbine manufacturer often does not include boiler blowdown and attemperating spray flows, which may be considered essential for delivering steam to the turbine. Appendix A shows an example of a typical heat balance with detailed explanations for many important characteristics.

Although there are many nuances in defining heat rate, a commonly accepted gross turbine cycle heat rate is defined as follows using Figure 2-3 nomenclature:

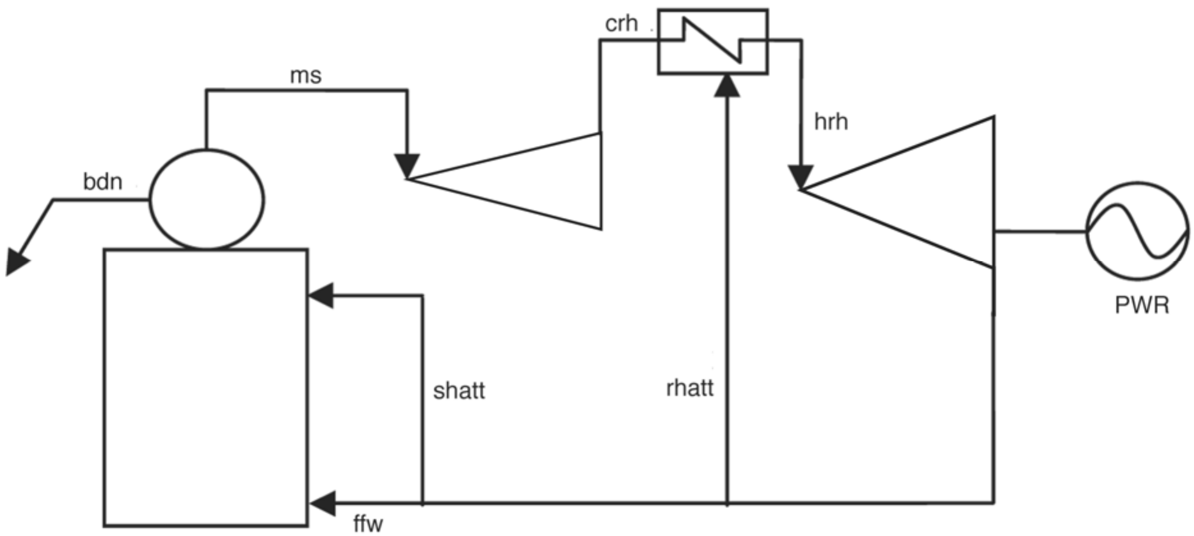

Figure 2-3
Parameters included in the heat-rate equation

$$GTHR = [(W_{ffw} - W_{bdn}) * (h_{ms} - h_{ffw}) + W_{bdn} * (h_{dr} - h_{ffw}) + W_{shatt} * (h_{ms} - h_{shatt}) + W_{crh} * (h_{hrh} - h_{crh}) + W_{rhatt} * (h_{hrh} - h_{rhatt})]/PWR$$

where

GTHR = gross turbine heat rate (Btu/kWh)

W_{ffw} = final feedwater flow (lbm/h)

W_{bdn} = boiler blow down flow (lbm/h)

W_{shatt} = superheat attemperating spray flow (lbm/h)

W_{crh} = cold reheat flow (lbm/h)

W_{rhatt} = reheat attemperating spray flow (lbm/h)

h_{ms} = main steam enthalpy (Btu/lbm)

h_{ffw} = final feedwater enthalpy (Btu/lbm)

h_{dr} = boiler drum enthalpy (Btu/lbm)

h_{shatt} = superheat attemperating spray enthalpy (Btu/lbm)

h_{crh} = cold reheat enthalpy (Btu/lbm)

h_{hrh} = hot reheat enthalpy (Btu/lbm)

h_{rhatt} = reheat attemperating spray enthalpy (Btu/lbm)

The heat balance has the manufacturer’s margin built into it. The actual heat rate will likely be better (lower) than the heat-balance heat rate on commissioning. OEMs are typically conservative about the performance published in the heat balance to compensate for manufacturing tolerances, potential shipping damage, assembly problems, eventful startups, and delays in acceptance testing. Manufacturers’ margins are typically 1% to 2%, meaning that the actual GTHR can initially be 1% to 2% lower than that published on the guarantee heat balance. The true margin is measured as the difference between the performance on the guarantee heat balance and the performance obtained from an acceptance test.

In the absence of an acceptance test, the guarantee heat balance is the best indicator of a new and clean condition of the steam turbine. It can serve as the basis for future performance comparisons in a heat-rate improvement program.

High-, Intermediate-, and Low-Pressure Turbine Efficiency

Design casing efficiencies can be calculated from the heat balance. With any casing, it is important to establish the starting and ending points for the efficiency calculation. Valves and lengths of pipe can add pressure decreases that change the measured casing efficiency. Because the purpose of calculating efficiency is to determine the thermal performance of the turbine, eliminating efficiency changes caused by measurement locations focuses on performance of the turbine itself. Figure 2-4 shows some common turbine configurations and the locations for starting and ending the efficiency calculations.

HP turbine efficiency is usually taken from upstream of the main stop valve to the exit of the HP turbine casing. Think of the plane of responsibility between the plant and the turbine manufacturer. The plant provides main steam piping and the turbine manufacturer provides the main stop valve, the turbine inlet valves and the turbine. Therefore, the starting point is upstream of the main stop valve. On the exit side, the plane of responsibility is where the HP turbine exhaust flange connects to the plant’s cold reheat piping.

The starting point for IP turbine efficiency is usually upstream of the reheat intercept valve. The exit is less clear because the turbine manufacturer provides the crossover piping between the IP and the low-pressure (LP) turbines. Typically, a 3% pressure decrease exists between the exit of the IP turbine blading and the LP turbine bowl, and the exit point can be anywhere along the crossover piping.

If the HP and IP turbines are arranged in an opposed-flow configuration, the internal IP turbine efficiency starts in the IP turbine bowl after the mixing of the hot reheat steam and the HP packing steam. There is typically a 2% pressure decrease across the reheat intercept valve.

The LP turbine efficiency starting point is the crossover, and the exit is the plane of responsibility between the turbine manufacturer and the condenser manufacturer, which is where the turbine exhaust hood flange is bolted to the condenser neck flange. The heat balance will show the LP turbine steam exit flow, condenser pressure, and the used energy endpoint (UEEP), on the condenser icon.

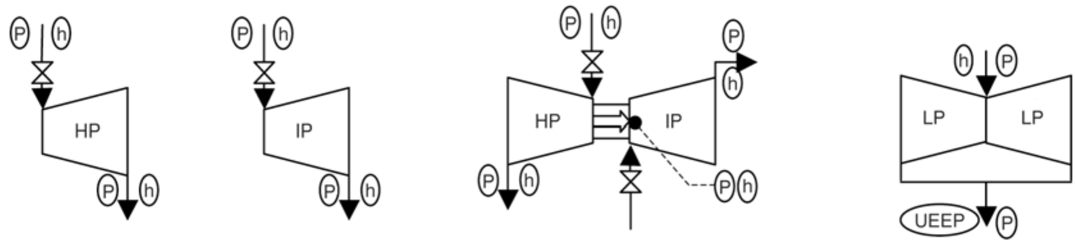

Figure 2-4
HP/IP/LP turbine design efficiency locations

The heat balance shown in Appendix A is used to demonstrate the calculation of the design casing efficiencies in the following:

HP Turbine Efficiency

$$\eta_{HP} = \frac{(h_{ms} - h_{crh})}{(h_{ms} - h_i)} 100 = \frac{(1460.4 - 1316.91)}{(1460.4 - 1290.23)} 100 = 84.32\%$$

where

η_{HP} = HP turbine efficiency (%)

h_{ms} = main steam enthalpy (1460.4 Btu/lbm)

h_{crh} = cold reheat enthalpy (1316.91 Btu/lbm)

P_{ms} = main steam pressure (2414.7 psia)

P_{crh} = cold reheat pressure (605.1 psia)

h_i = isentropic enthalpy ($f(P_{ms}, h_{ms}, P_{crh})$ = 1290.23[1])

IP Turbine Efficiency

Because the HP and IP turbines in Appendix A are in an opposed-flow configuration, the HP packing leakage must be taken into consideration. In this case, the internal efficiency is calculated. Adjustments can be made for the pressure decrease across the inlet valves for comparing the internal efficiency with measured efficiency where the measured efficiency starting point is upstream of the inlet valves.

The most accurate method for determining the design internal IP efficiency is to use the internal efficiency curve provided in the thermal kit. In the absence of the thermal kit curve, the next best resource is the Cotton et al. paper (see the section "What Should Be in Your Library"). Finally, if these two resources are not available, the guarantee heat balance can be used to calculate IP turbine internal efficiency. One reason that this latter method is not as reliable as the former two

[1] Steam properties in this report are calculated using the IAPWS-IF97 formulations, which is the current international standard adopted by the International Association for the Properties of Water and Steam in 1997.

with an opposed-flow HP/IP turbine is the value for HP packing leakage shown on the heat balance. It often corresponds to a sealing clearance that is greater than design. The following calculation shows how to use the heat balance information to calculate internal IP turbine efficiency.

First, the energy equation is used to calculate the mixed enthalpy in the IP turbine bowl:

$$h_b = \frac{\left(h_{hrh} w_{hrh} + h_L w_L\right)}{w_{hrh} + w_L} = \frac{(1518.9 * 3{,}565{,}656 + 1431.7 * 33{,}725)}{(3{,}565{,}656 + 33{,}725)} = 1518.1 \text{ BTU/lbm}$$

where

h_b = IP turbine bowl enthalpy

h_{hrh} = hot reheat enthalpy = 1518.9 Btu/lbm

h_L = enthalpy of the leakage flow (1431.7 Btu/lbm)

W_{hrh} = hot reheat flow (3,565,656 lbm/h)

W_L = leakage flow (33,725 lbm/h)

Next, the internal efficiency, from the bowl to the IP turbine exhaust, is calculated:

$$\eta_{IPint} = \frac{(h_b - h_{IPexh})}{(h_b - h_i)} 100 = \frac{(1518.1 - 1386.8)}{(1518.1 - 1366.4)} 100 = 86.55\%$$

where

η_{IP} = IP turbine internal efficiency (%)

P_b = bowl pressure (533.7 psia)

P_{IPexh} = IP turbine exhaust pressure (179.9 psia)

h_{IPexh} = IP turbine exhaust enthalpy (1386.8 Btu/lbm)

h_i = isentropic enthalpy at the IP turbine exhaust ($f(P_b, h_b, P_{IPexh})$ = 1366.4 Btu/lbm)

LP Turbine Efficiency

Design LP turbine efficiency is calculated from the LP turbine inlet to the UEEP (not the expansion line endpoint). See Appendix A for a discussion of UEEP and expansion line endpoint (ELEP).

$$\eta_{LP} = \frac{(h_{x-o} - h_{LPexh})}{(h_{x-o} - h_i)}100 = \frac{(1386.8 - 1025.5)}{(1386.8 - 959.3)}100 = 84.51\%$$

where

η_{LP} = LP turbine efficiency (%)

$P_{x\text{-}o}$ = LP turbine inlet pressure (179.9 psia[2])

P_{LPexh} = LP turbine exhaust pressure (0.737 psia)

$h_{x\text{-}o}$ = LP turbine inlet enthalpy (1386.8 Btu/lbm)

h_{LPexh} = LP turbine exhaust enthalpy (UEEP) (1025.5 Btu/lbm)

h_i = isentropic enthalpy at the LP turbine exhaust [f($P_{x\text{-}o}$, $h_{x\text{-}o}$, P_{LPexh}) = 959.3 Btu/lbm]

Pump Power

The heat balance does not give detailed information about the pumps. From the turbine manufacturer's perspective, the energy added to the water and the steam necessary to drive a feedpump turbine both affect the GTHR. However, the relationship between the pump, driver, and extraction steam can be calculated using the following methodology, referring to the typical heat balance in Appendix A.

The flow and inlet and outlet water enthalpy and temperature are given for the boiler feedpump (BFP). From here, the inlet and outlet pressures can be calculated iteratively using the steam tables. The steam tables require a pressure and either a temperature or enthalpy. So, by assuming a pressure for a given enthalpy, the temperature can be obtained from the steam tables. If the temperature matches the given temperature, then the pressure is correct. If it does not, choose another pressure and repeat until the two temperatures match. This process can be done alternately by matching enthalpies for a given temperature. This procedure yields the pressures P_{in} = 62.5 psia and P_{out} = 2861 psia. Note that the inlet pressure equals the deaerator pressure, which ignores the additional suction pressure provided by the height of the column of water between the deaerator and the pump inlet.

The next step is to obtain the specific volume of inlet water from the pressure and enthalpy using the steam tables.

$v_{in} = f(P_{in} = 62.5, h_{in} = 265.0) = 0.0175\ ft^3/lbm$

[2] The heat balance in Appendix A shows the same pressure at the exhaust of the IP turbine and the inlet to the LP turbine, which is an obvious error. There is a 3% pressure decrease between the exhaust of the IP turbine and the LP turbine bowl. Therefore, one would expect about a 2% pressure decrease to the LP turbine inlet.

Finally, the power delivered to the water by the pump is calculated as follows:

$$PWR = Wv(P_{out} - P_{in})$$

$$PWR = 3{,}948{,}017\frac{lbm}{hr} * 0.0175\frac{ft^3}{lbm} * (2861 - 62.5)\frac{lbf}{in^2} * \frac{hr}{3600s} * \frac{144in^2}{ft^2} * \frac{kW \quad s}{737.562ft - lbf} = 10{,}486 \ kW$$

where

W = flow through the pump (lbm/h)

v = specific volume of the water entering the pump (ft^3/lbm)

P_{in} = pressure of the water entering the pump (psia)

P_{out} = pressure of the water exiting the pump (psia)

Note that the heat balance diagram shows feedpump power delivered by the BFP turbine as 12,583 kW, which implies the pump efficiency is 10,486/12,583 * 100 = 83.3% at that load.

The inlet and outlet conditions at the boiler feedpump turbine are given on the heat balance and can be used to calculate the shaft power that the feedpump turbine delivers to the pump as follows:

$$PWR = \frac{W(h_{in} - h_{out})}{3412} = \frac{130{,}584(1386.8 - 1058.1)}{3412} = 12{,}580\,kW$$

where

W = flow through the pump (lbm/h)

h_{in} = specific enthalpy of the water entering the pump (Btu/lb)

h_{out} = specific enthalpy of the water exiting the pump (Btu/lb)

Generator Losses

Generator losses are shown on the heat balance in the box representing the generator. There are two types of losses. The fixed losses, also referred to as the mechanical losses, include the bearing losses in the turbine and generator and the oil pump loss. The generator losses include all mechanical and electrical losses in the generator except the generator bearing losses. The generator output plus the fixed and generator losses equals the shaft power of the turbine. This calculation can be checked by summing the shaft power of each turbine section, recognizing that shaft power is the product of mass flow and enthalpy decrease:

$$\text{Shaft Power} = \frac{W(h_{in} - h_{out})}{3412} \text{ in kW}$$

Hydrogen Pressure and Purity

Many large generators use hydrogen cooling because of the desirable characteristics of hydrogen: its low density, high specific heat, and high thermal conductivity. Figure 2-5 shows a simplified diagram of the key components in a hydrogen-cooled generator. The hydrogen circulates throughout the rotor and stator of the generator, then gives up its heat in gas-to-liquid coolers in what is essentially a closed system. However, because the hydrogen atom is so small, leaks occur through paths that would otherwise be airtight. The coolers, joints, flanges, and seals provide opportunities for hydrogen leakage. As the hydrogen leaks out, the hydrogen pressure decreases, and the reduced cooling ability of the remaining hydrogen decreases the generator capacity.

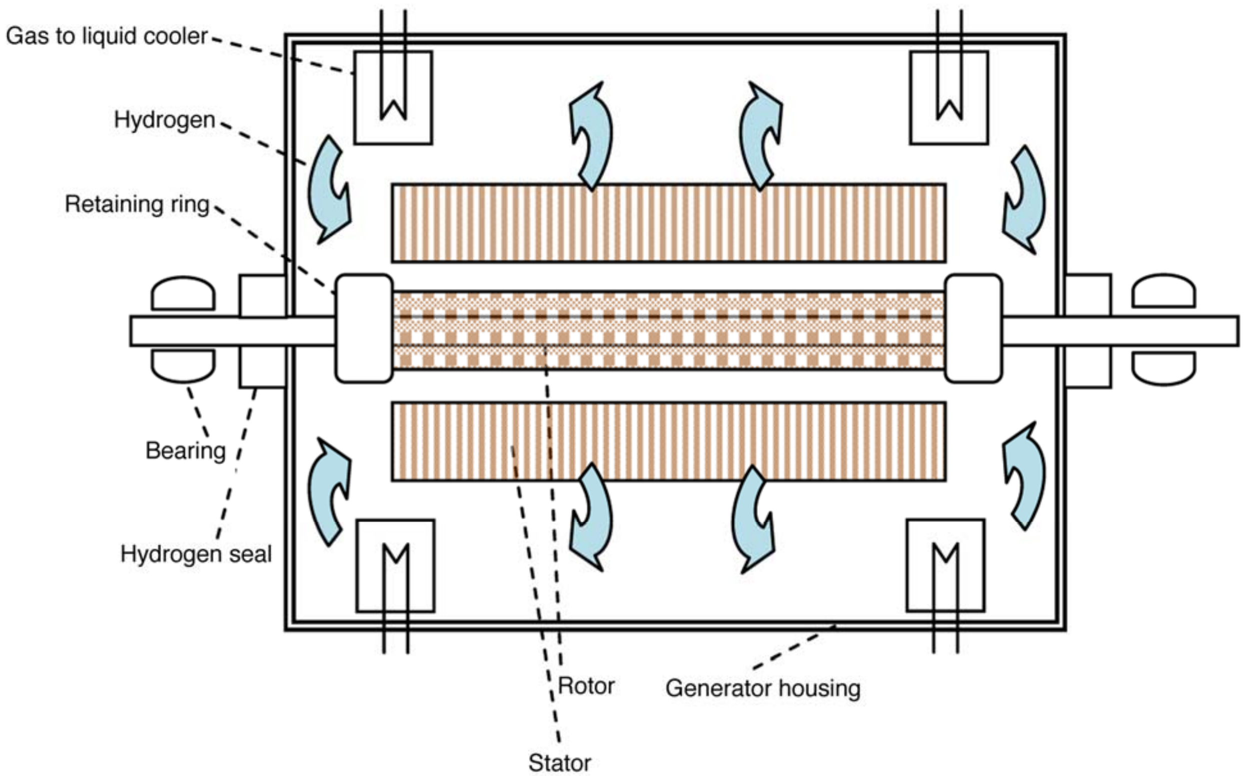

Figure 2-5
Hydrogen-cooled generator

Hydrogen purity is another important parameter. If air or oil contaminates the hydrogen, its purity decreases. The mixture of air and hydrogen can be explosive when the hydrogen purity is less than approximately 90%. The introduction of air increases windage losses because there is more friction between the rotating parts and air than between the rotating parts and hydrogen.

Figure 2-6 shows a schematic of a hydrogen seal. Oil passes through the seal to the surface of the rotor. Most of the oil travels along the rotor toward the air side where it is collected and recirculated. A smaller portion of the oil travels along the rotor to the gas side. The seal prevents hydrogen from leaking out into the room. Because of the small atomic size of hydrogen, some hydrogen is carried away by the oil where it must be separated from the oil. This process depletes the supply of hydrogen in the generator and will decrease its pressure if it is not replenished.

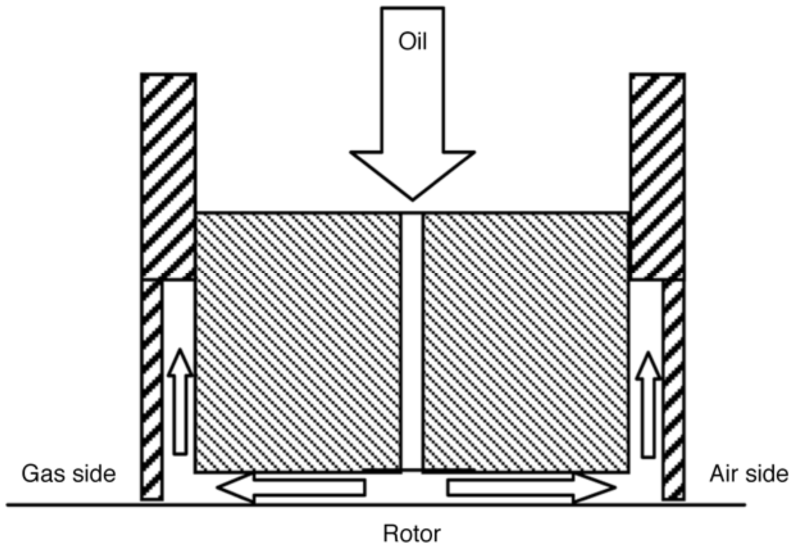

Figure 2-6
Hydrogen seal

Both hydrogen pressure and hydrogen purity affect the power output of the generator. The design calls for rated pressure and purity. If either is off-design during a test, the power output must be corrected for the actual test pressure and purity. The turbine manufacturer often provides correction curves for hydrogen pressure and hydrogen purity.

Auxiliary Power

Auxiliary power is often not shown on a heat balance provided by the turbine OEM. The auxiliary power is used by the balance of plant equipment to operate the power plant. Induced draft fans, forced draft fans, motor-driven condensate, feed and circulating water pumps, mechanical draft cooling towers, fuel-handling equipment, station heating and lighting, and electrification of the station itself are all consumers of auxiliary power.

The difference between gross power output and the net power output is the auxiliary power. Because the auxiliary power comes right off the top of the generated power, minimizing it will increase the net power output—the product that produces revenues.

Sometimes heat-rate calculations include some auxiliary power considered necessary to produce power. For instance, the power consumed by a motor-driven boiler feedpump might be considered in a heat-rate calculation because, without it, there would be no power from the generator.

Turbine Casing Contributions

To perform thermal performance studies on the turbine cycle, it is useful to know the contribution from each turbine casing toward the total power generated. That way, if the efficiency degradation is known in a given casing, the corresponding change in power output can be calculated. This process is done only once using the guarantee or VWO heat balance. Then, casing contributions are used in all future reconciliations of operating data to troubleshoot performance problems. In a typical single reheat fossil plant, the HP turbine contributes 28% of the power, the IP turbine contributes 22%, and the LP turbine contributes 50%.

The power generated by a casing is a function of its shaft power output, the generator/mechanical efficiency, and the loss factor. The heat balance in Appendix A is used in this example to calculate the contribution of the casings.

Generator/mechanical efficiency is taken from the heat balance. The fixed losses, the generator losses, and the generator power add up to the shaft power: 1914 + 7897 + 591,906 = 601,717 kW. The ratio of generator power to shaft power is the generator/mechanical efficiency: 591,906/601,717 = 0.984 or 98.4%. In the absence of given information to calculate generator/mechanical efficiency, a good value to use for most units is 98.5%.

HP Turbine Contribution

Shaft power is the product of mass flow and enthalpy decrease across the expansion. In the HP turbine, consider two separate expansions: one across the first stage and a second across the remainder of the stages within the HP turbine casing. The flow that passes through the HP packing originates at the first-stage shell location (after the first-stage rotating blades and before the second-stage stationary blades).

The mass flow through stage 1 is the main steam flow minus the above seat and below seat valve stem leakage: 3,948,017 − 977 − 3896 = 3,943,144 lbm/h.

The enthalpy decrease across the first stage, or the used energy (UE), is the difference between the main steam enthalpy and the HP packing enthalpy: 1460.4 − 1431.7 = 28.7 Btu/lbm.

Although shaft power is what the turbine produces, power is generally thought of as that produced by the generator. Therefore, the generator/mechanical efficiency must be used to obtain the stage 1 power at the generator terminals.

First-stage generator power output is calculated as follows:

$$PWR_{1st} = \frac{W * UE}{3412}\eta_{G-M} = \frac{3{,}943{,}144\text{lbm/hr} * 28.7\text{BTU/lbm}}{3{,}412\text{BTU/kWh}}0.984 = 32{,}637\text{kW}$$

where

W = steam flow through the stage (lbm/h)

UE = used energy or enthalpy drop (Btu/lb)

Note that the first stage produces 5.5% of the entire power of the turbine, which is more than the average power per stage.

The power from the remaining stages in the HP turbine begins with the second stage and ends at the HP turbine exhaust. The end-packing leakoffs at the HP turbine exhaust have passed through the last stage in the HP turbine. The generator power from the remainder of the HP turbine is calculated as follows:

$$PWR = \frac{(W_{1st} - W_{Leak}) * (h_{1st} - h_{crh})}{3412}\eta_{G-M}$$

$$PWR = \frac{(3{,}943{,}144 - 33{,}725)\text{lbm/hr} * (1431.7 - 1316.9)\text{BTU/lbm}}{3{,}412\text{BTU/kWh}}0.984 = 129{,}431\text{kW}$$

The total power produced by the HP turbine is the sum of the first-stage power and the power from the remaining stages: 32,637 + 129,431 = 162,068 kW. Because the total power generated is 591,906 kW, this HP turbine produces 27.4% of the power.

IP Turbine Contribution

The contribution from the IP turbine is calculated similarly to that from the HP turbine, but in this case, the starting point of the IP expansion is the mixed enthalpy in the IP bowl from the incoming hot reheat flow and the HP packing leakage flow:

$$h_{bowl} = \frac{h_{hrh}W_{hrh} + h_{Leak}W_{Leak}}{W_{hrh} + W_{Leak}} = \frac{1518.9 * 3{,}565{,}656 + 1431.7 * 33{,}725}{3{,}565{,}656 + 33{,}725} = 1518.1\ \text{BTU/lbm}$$

Like the HP turbine, this IP turbine has two sections, but the IP sections are separated by a feedwater heater extraction instead of seal leakage. The power produced by the expansion from the IP bowl to the extraction midway through the IP turbine is calculated as follows:

$$PWR_{IP1st} = \frac{W_{bowl} * (h_{bowl} - h_{extr})}{3412} \eta_{G-M}$$

$$PWR_{IP1st} = \frac{(3{,}565{,}656 \ \ +33{,}725)\ \text{lbm / hr} * (1518.1 - 1443.2)\ \text{BTU/ lbm}}{3{,}412\ \text{BTU/kWh}}\ 0.984 = 77{,}749\ \text{kW}$$

The flow through the second section of the IP turbine is the hot reheat flow plus the HP packing flow minus the extraction flow: 3,565,656 + 33,725 − 142,390 = 3,456,991 lbm/h. The power output is:

$$PWR_{IP2nd} = \frac{W_{IP2nd} * (h_{extr} - h_{x-o})}{3412} \eta_{G-M}$$

$$PWR_{IP2nd} = \frac{3{,}456{,}991\ \text{lbm / hr} * (1443.2 - 1386.8)\ \text{BTU/ lbm}}{3{,}412\ \text{BTU/kWh}} 0.984 = 56{,}229\ \text{kW}$$

where

W_{1st} = steam flow through the first stage (lbm/h)

W_{leak} = leakage flow (lbm/h)

W_{hrh} = hot reheat steam flow (lbm/h)

W_{bowl} = steam flow into the IP turbine bowl (lbm/h)

W_{IP2nd} = steam flow to the first stage of the IP turbine (lbm/h)

h_{leak} = specific enthalpy of the leaking steam (Btu/lb)

h_{1st} = specific enthalpy of the steam entering the first stage (Btu/lb)

h_{crh} = specific enthalpy of cold reheat steam (Btu/lb)

h_{hrh} = specific enthalpy of hot reheat steam (Btu/lb)

h_{bowl} = specific enthalpy of steam in the IP turbine bowl (Btu/lb)

h_{extr} = specific enthalpy of the extraction steam (Btu/lb)

h_{x-0} = specific enthalpy of IP turbine exhaust steam (Btu/lb)

η_{G-M} = generator mechanical efficiency (−)

The power out of the IP turbine is the sum of the power out of the first section and that out of the second section: 77,749 + 56,229 = 133,978 kW. This IP turbine produces 22.6% of the power.

LP Turbine Contribution

The LP turbine power can also be calculated from the flows and UE of each section to the UEEP. One must account for the steam flow rate changes occurring at each of the extractions in LP turbine, or the LP turbine power can be calculated from the difference between the total power and the sum of the HP and IP turbines' power: 591,906 − 162,068 − 133,978 = 295,860 kW, which is 50.0% of the total power output.

Loss Factor

The loss factor is something that is used with the IP turbine when calculating contributions to power loss because it is followed by another power-producing casing. The HP and LP turbines, on the other hand, are followed by the reheater and condenser, respectively, in which no power is produced. Losses in the steam path result in steam emerging from inefficient casings with higher enthalpy than design. The casing that follows can use this increase in thermal energy to produce additional power. Typically, the LP turbine can recover 50% of the losses that occur in the IP turbine. The loss factor is therefore 0.5. Therefore, if IP turbine efficiency is down 4%, the gross generator power output will respond as if only 2% of the IP turbine contribution is reducing power.

The casing contribution and the loss factor are used below in an example. Assume the IP turbine efficiency is down 4% and the IP turbine contributes 22% of the power. How much will power decrease?

$$\Delta PWR = \Delta\eta_{IP} * \text{Casing Contribution} * LF = 4\% * 0.22 * 0.5 = 0.44\%$$

Therefore, when the IP turbine efficiency degrades 4%, the power output will decrease 0.44%, or, in this example, heat balance, 2.6 MW. For losses in the HP and LP turbines, this equation is also valid, but the casing contributions are different and the value of the loss factor is 1.

Heat Balances with the Thermal Kit

The thermal kit will usually contain a family of heat balances. The guarantee heat balance is regarded as the most accurate because that represents the performance that the OEM is willing to guarantee. It is typically at a valve position that is nearly wide open. The steam conditions are rated.

The VWO heat balance represents the cycle when the turbine inlet valves are open as far as is physically possible. The steam conditions are rated.

Another heat balance is the 5% OP diagram. It represents the cycle at VWO with a main steam inlet pressure 5% greater than rated. The other steam conditions (for example, main steam and hot reheat steam temperatures, exhaust pressure) remain at the rated levels.

There are typically several part-load heat balances. If the unit is operated with sequential valves, the part-load heat balances usually represent the cycle at valve points.

Mechanical Losses

Mechanical losses are shown on the heat balance in the box designating the generator. They include friction losses associated with the bearings and couplings as well as windage losses.

Expected Leakages

Whenever there are stationary and rotating elements adjacent to one another and it is undesirable to have metal-to-metal contact, there will be some clearance. The objective of steam-sealing devices is not to eliminate leakage, but rather to control or minimize it. Therefore, there will be some expected leakages throughout the expansion through the steam turbine.

In casings in which the exhaust pressure is greater than atmospheric pressure (the HP and IP turbines), a steam seal leak will result in steam exiting the turbine and coming out into the room (or out to the ambient in outdoor units) where the rotors emerge from casings. This situation is both wasteful and dangerous. It is wasteful because the escaped steam is missing the opportunity to make a contribution in some other location in the steam path such as the LP sealing system. It is dangerous because superheated steam is invisible and can cause severe burns to passersby.

In casings in which the exhaust pressure is less than atmospheric (the LP turbine), a leak will take the form of air leaking into the condenser. This situation will reduce the effectiveness of the heat rejection equipment because the air in the condenser displaces what would otherwise be steam ready to condense into water. Steam condensing into water in a fixed volume (the exhaust hood and condenser) creates a vacuum. Air interferes with this process. The result is an increase in condenser pressure and less energy being extracted from the steam to produce power in the turbine.

Stage Moisture Removal Effectiveness

When the steam expands in the turbine through the saturation line, moisture droplets begin to appear in the steam. In a condensing unit, this two-phase flow begins in the latter part of the LP turbine. The water droplets start out small, but combine to form larger droplets as the expansion continues to downstream stages. As the steam travels through the stages, it is subjected to the centrifugal force from the rotating blades, which causes the water droplets to migrate outward toward the tips of the blading. Water droplets are undesirable in a steam turbine because they move slowly (relative to the steam), which changes their entry angle into the rotating blading, producing turbulence. They also cause impact damage on the blading. Water droplets essentially act like foreign objects in the steam path. Nuclear cycles often have moisture removal stages in the LP turbine to remove water from the turbine and send it off to the feedwater heaters. A fossil unit’s feedwater heater extractions do that also, but to a lesser degree because of the geometry of the turbine casing.

The effectiveness with which the moisture is removed is a function of the stage pressure. The lower the pressure, the more effective is moisture removal. Thus, the L-0 and L-1 stages have more effective moisture removal than the L-2 and L-3 stages. The LP turbine blade rows are labeled from the exhaust toward the inlet. L-0 is the last stage, L-1 is the second to last, and so on.

Extraction Pressures and Heater Shell Pressures

Heat balances have extraction pressures at the turbine and at the heaters (Figure 2-7). Although the enthalpy remains constant in the extraction piping, there is typically a 3% to 8% decrease in pressure along the extraction line to the heater. The extraction pressures at the turbine are typically the pressure at the extraction line flange (the line of responsibility between the turbine manufacturer and the plant), but not always. It is not necessarily clear from looking at the heat balance which pressure is designated. Just because the pressure is typed inside the turbine icon does not mean that it is a stage pressure, such as what is shown in Figure A-1 in Appendix A. There is also a 1% to 3% pressure decrease between the stage pressure and the extraction line flange pressure.

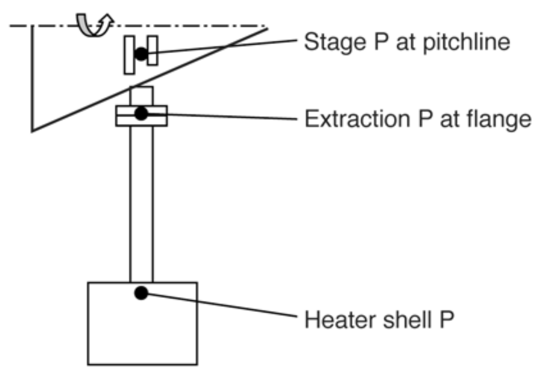

Figure 2-7
Heater extraction pressures

One way to determine whether the heat balance pressures are at the stage or at the flange is to draw or calculate an expansion line from the pressures and enthalpies on the heat balance. One would expect a smooth expansion line. If the expansion line is jagged, the pressures are likely flange pressures and not stage pressures.

The same concept applies to casing exhaust pressures. There is a small pressure decrease between the exit of the last stage in a casing and the casing exhaust where it attaches to the plant piping.

The nomenclature associated with heaters and heater extractions varies. Heaters can be named 1 through N from the lowest pressure to the highest pressure or from the highest pressure to the lowest pressure. Heater nomenclature is usually determined by the plant’s or architect engineer’s preference. Extractions are often referred as the Nth-stage extraction, where the Nth stage is immediately upstream of the extraction port, but not always. Sometimes the Nth stage is immediately downstream of the extraction port. Therefore, in a turbine where there is an extraction port between the 10th and 11th stages, it may be referred to as either the 10th-stage extraction or the 11th-stage extraction. The best way to be certain of the extraction location is to look for the extraction port at the bottom of the turbine casing during an overhaul or to review the turbine drawing.

Full-Arc and Partial-Arc Admission Schemes

Steam can be admitted into the turbine in a number of ways. On one end of the continuum, there is a full-arc, single-admission mode in which all admission valves open and close simultaneously and steam enters the 360° of the control-stage nozzles (Figure 2-8, left). On the other end of the continuum, there is a partial-arc, sequential valve admission mode in which only one valve opens or closes at a time and steam enters a portion of the 360° arc corresponding to each valve (Figure 2-8, right). In between, there are hybrid operating modes in which two or more valves operate simultaneously, and then the remaining valves open sequentially.

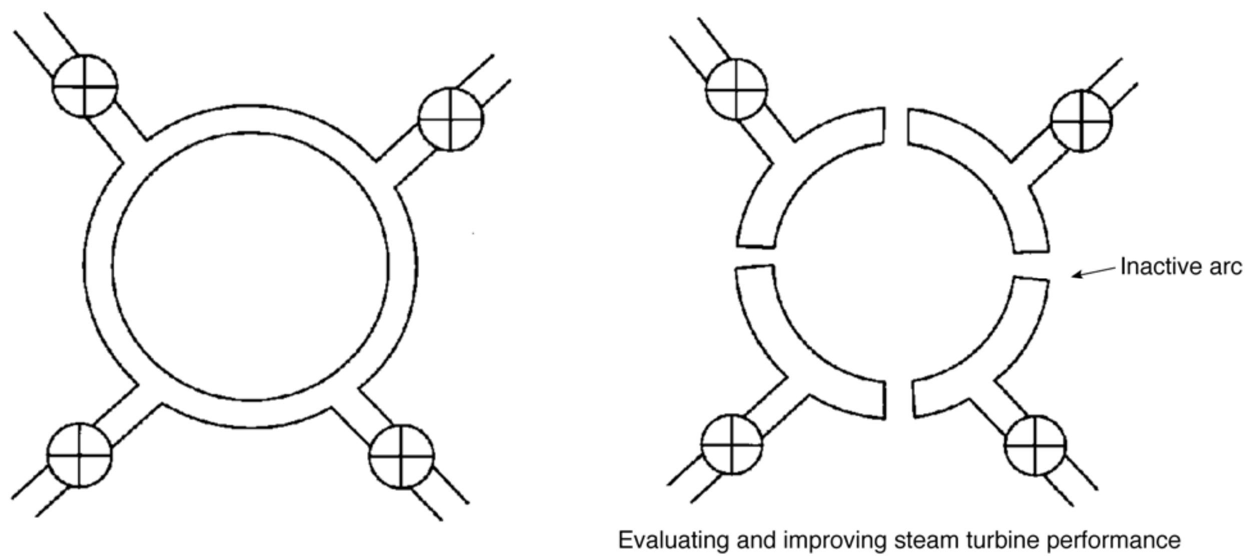

Figure 2-8
Partial-arc and full-arc admission

Sequential valve admission with a partial-arc first stage has the best part-load efficiency because throttling through the valves is minimized. Over the load range only one valve is throttling at a time while the others are either fully open or closed. Except for the first valve to open or the last valve to close, less than 100% of the flow is passing through a throttling valve during load changes.

Single-admission mode on a full-arc first stage, on the other hand, has poor part-load efficiency because at any point during load changes, 100% of the flow into the turbine passes through a valve that is throttling.

Conversely, single-admission mode with a full-arc first stage has the best full-load efficiency. The continuous arc around the 360° without any breaks allows the efficiency to be slightly higher than a partial-arc control stage at full load.

Impulse Versus Reaction

The terms *impulse* and *reaction* refer to the physical design of the stages. By definition, reaction is the portion of the stage energy released on the rotating blade. All stages, whether classified as impulse or reaction, have some reaction in them; there must be some energy decrease across both the stationary and rotating blades, so there is no such thing as a purely impulse turbine blade or one with 100% reaction.

Impulse refers to the force of the steam hitting the rotating blade, causing it to turn. Reaction refers to the high-velocity steam exiting the rotating blade, causing it to turn.

Figure 2-9 shows a typical impulse stage on the left and a typical reaction stage on the right. Impulse stages are built with a wheel-and-diaphragm design. The stationary nozzles are contained in a diaphragm, which has a large pressure decrease across it. Adjacent stationary partitions in a diaphragm form converging nozzles that turn and accelerate the steam. The rotating blades are mounted on wheels and have a relatively small pressure decrease across them. Adjacent rotating blades do not have a converging nozzle shape, but rather a curved passage with similar inlet and outlet cross-sectional areas. The steam seals are designed appropriately for the pressure decrease across them: many teeth for the HP decrease across the diaphragm and fewer teeth for the small pressure decrease across the rotating blades.

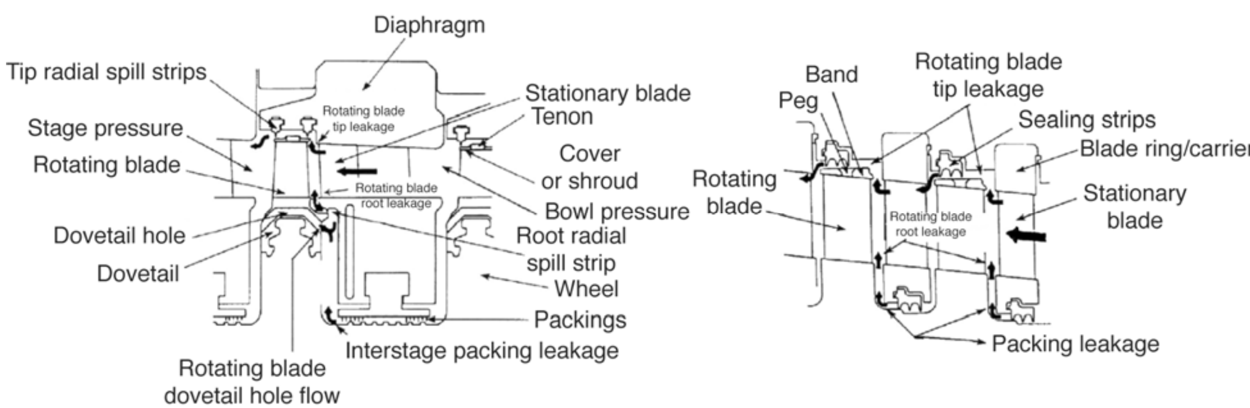

Figure 2-9
Impulse and reaction stages

Reaction stages are built with a drum construction. Several rows of stationary blades are mounted in a blade ring. The rotating blades are mounted on the rotor without the wheels that are part of the impulse design. The shape of the stationary blades is similar to that of the impulse design—a converging nozzle. The shape of the rotating blades is such that adjacent blades within a stage form converging nozzles as well. In a reaction stage, the steam is turned and accelerated roughly equally in the stationary and rotating bladings. Thus, a reaction stage really has 50%

reaction, whereas the stationary and rotating portions each have 50% of the stage energy across them. The steam seals hold back similar amounts of pressure, so their construction is similar on the stationary and rotating components.

Historically, General Electric manufactured impulse turbines and Westinghouse manufactured reaction turbines. Other turbine manufacturers who make impulse turbines include Hitachi and Toshiba. Turbine manufacturers who make or made reaction turbines include Mitsubishi, ABB, Siemens, Alstom, Parsons, and Allis Chalmers. Now, many turbine manufacturers have been purchased by other turbine manufacturers, and some manufacturers make both impulse and reaction turbines.

First-Stage Shell Pressure

The first-stage shell pressure is one of the key pressures for determining the flow and condition of the HP stages. The first-stage shell location is between the exit of the first-stage rotating blades and the entrance of the second-stage stationary blades (Figure 2-10).

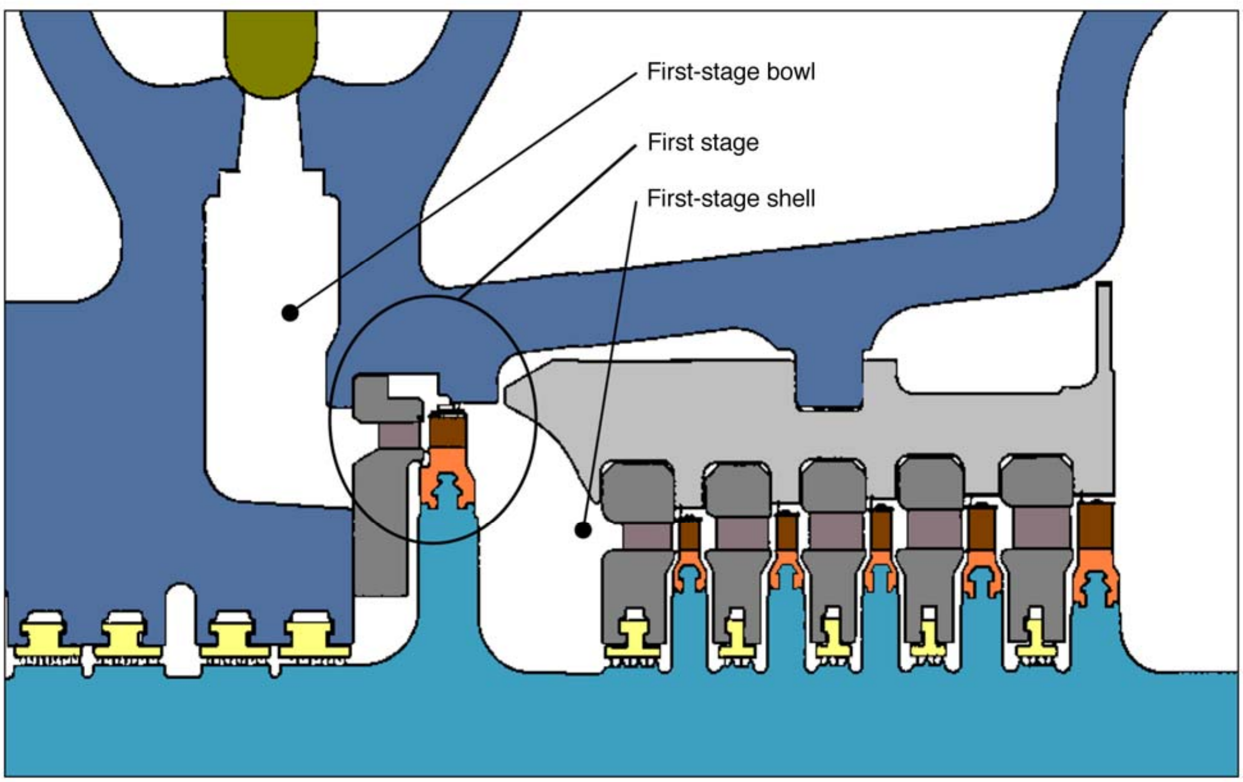

Figure 2-10
The first stage

The turbine's first stage is often a control stage, meaning that it controls the amount of steam entering the turbine. The control-stage stationary nozzles provide the limiting flow area when the inlet valves are wide open. The first-stage shell pressure is the pressure immediately downstream of the control stage and represents the highest pressure contained by the turbine's inner casing or shell.

First-stage shell pressure, as all stage pressures, responds to changes in flow and downstream resistance. Thus, it is a good indicator of changes in flow to the second stage and the area of the stages and seals downstream of the first-stage shell location. However, it is important to recognize that stage pressure responds to **both** flow and downstream resistance; therefore, it is not a reliable indicator of one or the other. For instance, if flow increased 5%, one would expect first-stage shell pressure to increase 5% as well. But, if the stage 2 seals were severely worn, effectively increasing the flow area of stage 2, then the worn seals would cause the first-stage shell pressure to decrease. The net change in the first-stage shell pressure might be a small increase or even a decrease, thereby decreasing the first-stage shell pressure credibility as a reliable indication of flow.

Not all steam turbines have control stages and first-stage shell pressure measurements. Combined-cycle plants, for instance, control steam turbine flow through the gas turbines and HRSG. The first stage of a combined-cycle steam turbine looks very similar to, but slightly smaller than, the second stage. In those cases, first-stage pressure is often the pressure upstream of the first stage.

Flow into the control stage is governed by the general flow equation. The main steam flow is a function of the first-stage nozzle area and the condition of the nozzle, the upstream steam conditions (pressure and specific volume), and the pressure ratio across the first stage:

$$w = CA\beta\sqrt{\frac{P}{v}}$$

$$\beta = \sqrt{\frac{2g\gamma}{(\gamma-1)}\left\{\left(\frac{P_2}{P_1}\right)^{\frac{2}{\gamma}} - \left(\frac{P_2}{P_1}\right)^{\frac{\gamma+1}{\gamma}}\right\}}$$

where

w = mass flow (lbm/h)

C = flow coefficient

A = nozzle area (in.^{2})

P_1 = upstream pressure (psia)

P_2 = downstream pressure (psia)

v = upstream specific volume (ft^3/lbm)

γ = isentropic constant

The general flow equation is an important equation for the steam turbine performance engineer. It gives the relationship between steam conditions and the area and condition of the turbine stages. Becoming familiar with it will allow the steam turbine performance engineer to develop an intuitive sense of how the pressures, temperatures, and flow respond to the turbine condition.

Commissioning

An important piece of information about the historical performance of the turbine is its performance at commissioning. It is this performance that describes the turbine in its new and clean condition. Also, when comparing the commissioning test results with the guarantee heat balance, the manufacturer's margin can be calculated from the difference between the two.

Acceptance Test/Guarantee Performance

Many steam turbines installed in the 1950s to 1970s did not have an acceptance test. The domestic turbine manufacturers built their turbines with plenty of margin during that time. Because the acceptance test is a cost usually borne by the owner, many owners took the risk that the turbine would deliver the promised performance without verification. Still others tested perhaps one out of four, five, or six identical steam turbines, assuming that they would all perform similarly.

The test code for conducting acceptance tests was and still is the American Society of Mechanical Engineers Performance Test Code 6 for Steam Turbines (ASME PTC 6). The Code has been revised many times over the decades to keep abreast with advancements in instrumentation and steam turbine design. Units commissioned in the 1950s used the PTC 6–1949 publication. The next PTC 6 code was published in 1964, then again in 1976. The Code specified a test providing results containing very low uncertainties, but was expensive and time-consuming to plan and conduct.

In 1982, Betz Bornstein and Ken Cotton published a paper entitled "A Simplified ASME Acceptance Test Procedure for Steam Turbines" in the *Journal of Engineering for Power, Transaction of ASME* [1]. This paper was the catalyst for the American Society for Mechanical

Engineers (ASME) to publish an interim test code that described a method to test steam turbines using a technique that was easier, quicker, and less expensive than the Code test without sacrificing test accuracy. The interim code was published in 1984 and given the designation PTC 6.1–1984. It was and is referred to as the alternative test, whereas the original Code test is referred to as the full-scale test.

With the power-generation industry becoming more competitive in the late 1980s and with the advent of the alternative test, more utilities were conducting acceptance tests on their steam turbines.

Now, the current test code is the ASME PTC 6–2004. It is an update of the 1996 publication that combined the alternative test and the full-scale test in a single code.

Other Issues at Commissioning

Usually precision test instrumentation is brought in just for the acceptance test and removed after. Although the temporary installation of precision instrumentation is good for the quality of the acceptance or commissioning test, removal of it increases the uncertainty when future tests using different instrumentation are compared with the acceptance test.

With commissioning sometimes comes rough startups. Unit trips caused by high vibration while the unit is at part load, balance-of-plant equipment malfunctions, and other starting and loading problems can cause unfortunate wear and tear on a new turbine. When researching the historical performance of the turbine, starting and loading events can be helpful in establishing the best expected, new and clean condition.

Determine the Best Expected Performance

There are several elements of a heat-rate improvement program. The first is to determine the best expected performance from the given turbine. Once that performance level is established, all future tests can be compared with it. It is often useful to think of performance changes in terms of percentage of degradation from design. This section describes how to obtain the best expected design performance.

Resources

Performance can be defined as heat rate, power output, steam rate, and/or casing efficiency. There are several resources for obtaining the information to calculate the best expected performance.

Acceptance Test Reports

The best resource for determining the new and clean condition of the turbine is the full-load acceptance test conducted during unit commissioning. This test determines the actual performance without any manufacturer's margin. It is conducted using precision, test-grade instrumentation and likely conforms to the procedures outlined in the ASME Performance Test Code for Steam Turbines. Acceptance test costs are usually borne by the owner and can be quite expensive. As a result, not every unit has a usable acceptance test.

Because acceptance tests are typically done with test-grade instrumentation brought in just for the test, caution must be exercised when comparing recent test results with those of the acceptance test. Sometimes the location of the current station instrument will differ from that of the original acceptance test instrument. If so, the corrections may have to be applied to account for pressure decreases, steam mixing, and so on between the original and current measurement points.

If the acceptance test heat rate is 2% lower than the heat rate on the guarantee heat balance, the manufacturer's margin for heat rate was 2%. If, on the other hand, the acceptance test heat rate is 2% higher than the guarantee, the unit missed its guarantee by 2% and the manufacturer's margin was negative. Either way, the acceptance test gives the most reliable value for the new and clean performance.

Past Outage Reports

Past outage reports are another resource for determining new and clean condition. In the absence of an acceptance test, the combination of an outage report and post-outage test can give a good basis for future performance comparisons. The outage report will give as-left clearances for sealing devices; it will document steam path problems, such as foreign object damage with which the unit went back into service. It may contain a photographic log of the steam path so that any surface, seal, or area that is off-design is documented. The outage report will allow the steam turbine performance engineer to rate how representative the post-outage test is of the best expected condition of the steam path. This process is often used after upgrades or replacements because design changes to the steam path can render the original acceptance test obsolete.

Thermal Kit

The thermal kit contains several heat balances, one of which is usually the guarantee heat balance. In the absence of an acceptance test, the guarantee heat balance is a resource for determining best expected performance. However, it is important to remember that the guarantee heat balance has a manufacturer's margin built into it.

Turbine Instruction Manual

The turbine instruction manual is a good reference for the steam turbine performance engineer. Limits on steam conditions and operating conditions, such as heaters out of service, are discussed. There is likely a good diagram of the steam seal system as well as a cross section or longitudinal drawing of all turbine casings.

Bornstein and Cotton Paper

The Bornstein and Cotton paper [1] is considered an important document because it describes how steam turbine performance is calculated from the OEM's perspective. Originally presented to the ASME in 1962 and revised in 1974, it is now out of print. Although written for General Electric turbines, its principles can be applied to 60-Hz steam turbines with superheated inlet steam from other manufacturers without a significant loss of accuracy. Like the published guarantee heat balances, the results of using the methodology in this paper also contain the manufacturer's margin. Heat-balance software is often based on these methods.

Determining the Turbine's Current Performance

Once the design or new and clean condition of the turbine has been established, the next step in the heat-rate improvement program is to determine the current performance of the turbine. Comparing the current condition with the design condition will result in a percentage of difference from design.

The goal is to measure current performance with as much precision as possible within the bounds of instrumentation, operation, and economical limits. Performance can be defined in many ways, but for the turbine performance engineer, performance will usually be defined as GTHR, casing efficiency, and power output at a given flow (steam rate).

Current Drawings: Piping and Instrumentation Drawings

Piping and instrumentation drawings (P&IDs) are a very useful reference for setting up instrumentation and isolation lists in preparation for a test. They show the locations of pressure gauges, thermal wells, flowmeters, and so on. They show pipe diameters, valves, and balance of plant equipment. When searching for potential isolation breaches, the P&IDs are where the search begins with a focus on the water/steam side of the boiler, the turbine, condenser, and feedwater heaters.

Often the P&IDs are not up to date. Changes in instrument locations, new instrument installations, new bypass lines, elimination of obsolete instrumentation, or equipment may not be reflected in the current set of P&IDs. It is therefore necessary to walk the plant to verify the P&IDs. Although this process can be time-consuming, it serves dual purposes of preparing for the upcoming test and updating the P&IDs for future reference.

The P&IDs are useful for determining which instrumentation will be used for a test and where there might be a need for additional test points. They are also useful in developing an isolation list. Isolation breaches are often the subject of much debate, particularly during an acceptance test. Most isolation problems decrease the performance of the turbine. If the isolation is not verified, an OEM can claim that their underperforming unit missed guaranteed output because of a leak, for which they get credit to apply toward output. This scenario usually presents itself after the test has been completed, and there is no way to verify whether the leak existed during the test.

A preemptive step to avoid this situation is to use the P&IDs as a basis for a plant tour where all parties involved in an acceptance test inspect the potential isolation breaches. For routine monitoring, maintain a list of isolation breaches during operation. Prioritize their importance and make the necessary preparations to fix the leaky valves at the next opportunity, for example, an unscheduled outage.

Instrumentation

The efficiency and health of the turbine are communicated through the instruments measuring pressure, temperature, differential pressure, and power.

Where to Place Precision Instrumentation

The ASME PTC 6 Code for testing steam turbines has guidelines for installing instrumentation. Following the Code results in a test with uncertainty in the 0.25% to 0.35% range. Although that sort of precision is desirable for an acceptance test, it becomes prohibitively expensive for routine monitoring or for tests in which there are not large liquidated damages at stake. Although the adage “garbage in, garbage out” prevails, more casual testing can occur at a fraction of the cost of acceptance testing without completely sacrificing accuracy if a few guidelines are met.

In general, the more influence a parameter has on the overall results, the more precise the instrument should be. For example, if main steam pressure is 1% high, it will increase flow by 1% and will increase power nearly 1%. It is therefore necessary to have a precise instrument measure main steam pressure because of this nearly 1%/1% influence on the final results.

Now consider an LP heater extraction pressure. If the pressure changes 1%, it will change the shell pressure of an LP heater, which will change the temperature of the water emerging from that heater, which will cause the next heater extraction flow to compensate for the change in water temperature entering it. The result is a minor adjustment in the extraction flows and resulting change in the power produced in the relevant stages. Compared with the main steam pressure, the LP extraction steam pressure has a nearly negligible effect on power output. Therefore, the instrumentation chosen for the LP extraction could be of lower quality and cost without sacrificing overall test uncertainty.

In general, pressures should be taken at all locations where steam is entering or exiting the turbine, including main steam, hot reheat steam, casing exhausts, extractions, end-packing leakoffs, and any other steam leaving the turbine for balance of plant equipment or process. Pressure should also be taken along the heater train: heater shell pressures, condensate pressure, and feedwater pressure.

Similarly, temperature should be taken at all locations where single-phase flow (superheated steam or water) is entering and exiting the turbine and feedwater heaters. The steam being measured should have at least 25°F of superheat to guard against condensation collecting on the steam side of the thermal well. Temperatures on lines containing two-phase flow can be used for information, but not for establishing enthalpy. Because condensation is a constant-temperature process, reading the temperature in wet steam will give the saturation temperature. This information may be used to calculate saturation pressure, which could be considered a secondary measurement to validate a primary pressure measurement taken at the same location, but it cannot be used with the pressure to calculate enthalpy.

Differential pressure is used to measure flow in some devices. A flow nozzle in the condensate line or the final feedwater line may have two sets of pressure taps providing a differential pressure corresponding to the mass flow. An excellent resource for information on determining flow is ASME's fluid meters (see the section "What Should Be in Your Library").

How to Determine Where Precision Instrumentation Is Needed

Power output, heat rate, and casing efficiencies can be extremely sensitive to some pressures, temperatures, and flows. There are at least two sources for determining which parameters can influence the measured performance results to such a large degree. The first source is the ASME PTC 6 Code. In it, there are diagrams for various cycles depicting which measurements should be taken with precision instrumentation.

Another source of this information is a heat-balance program. There are several available on the market. A heat-balance program allows the user to model the turbine cycle and then change one parameter at a time. The corresponding change in performance (power, heat rate, casing efficiency, and so on) can be used to conduct a sensitivity study. The result is a list of measurements and the corresponding precision of the instrumentation required to satisfy the overall uncertainty limits of the test.

Work Closely with the Instrumentation and Controls Group

Steam turbine performance engineers depend on the instrumentation and controls (I&C) group for many things. The I&C group will maintain calibrated instrumentation throughout the turbine cycle so that performance can be established and monitored. Often times, instrumentation problems will surface during preliminary tests, and it is the I&C group who usually corrects these problems. Large acceptance tests often have temporary precision instrumentation installed from other vendors, but the I&C group works with the outside vendors to familiarize them with the plant.

Turbine performance engineers work with the I&C group to install instrumentation in locations deemed necessary but none exists, to troubleshoot instrumentation problems when performance results go awry, and to establish valve points, that is, the turbine admission valve locations corresponding to minimum throttling.

Maintain an Inventory of Instruments for Periodic Testing

Instruments do wear out, drift, break, fail, and go out of calibration. It is therefore recommended to measure important parameters in duplicate or triplicate. This duplication does not always result in a similar instrument within close proximity of the first. There are other ways to duplicate measurements. For example, wet steam pressure, such as in the condenser, can be verified by measuring the hot well temperature, which should be the saturation temperature corresponding to the condenser pressure. Another way to verify temperature is to calculate an energy balance around, for example, a desuperheating station. The exit steam temperature should match that calculated from the desuperheating station inlet steam and attemperating spray pressures and temperatures.

Inevitably, calibration problems will be uncovered while measuring parameters in duplicate and triplicate. The I&C group will prove invaluable when correcting these problems.

Setting up, maintaining, and using the data acquisition system is a large part of both the turbine performance engineer's responsibilities and those of the I&C group. The turbine performance engineer's role is usually to communicate which measurements are needed to carry out the desired steam turbine cycle testing and/or monitoring. It is here where the duplicate and triplicate measurements can be compared easily. Data can be captured on a nearly continuous basis, quickly filling up memory allocated for it. Data on the steam turbine performance engineer's instrumentation list can be downloaded to text files that can be read by spreadsheet programs for further manipulation and calculation of results.

It is often helpful to include some redundant data in the collection process as a secondary or tertiary check of the data collected by the automated data acquisition system. Data collected routinely using station instrumentation and data collected from station pressure gauges and thermometers, control room charts, and so on, can all serve as a sanity check when something goes awry in the data acquisition system or if a primary instrument goes out of calibration or fails.

Instrumentation Audit

Because instrumentation is such a critical element in a heat-rate improvement program, consideration may be given to conducting an instrumentation audit. This audit can be done in-house or by an outside consultant. During the audit, malfunctioning, uncalibrated, and missing instruments are identified. Water legs can be calculated and improper water legs can be corrected. In some cases, an uncertainty analysis can be conducted for upcoming tests.

Uncertainty Analysis

If the Code is followed for a full-scale test, the test uncertainty is 0.25%. If the Code is followed for an alternative test, the test uncertainty is 0.33%. Both tests are rigorous and expensive to conduct, primarily because of the cost of installing and dismantling a high volume of calibrated precision instrumentation. Routine monitoring often makes use of station instrumentation, which may be considerably less precise than that used for acceptance testing. In between these two ends of the instrumentation continuum are an infinite number of combinations of precision and lesser grade instrumentation. Regardless of the instrumentation used, an uncertainty analysis will be an indicator of the quality of the test: the lower the uncertainly, the higher the quality of the test.

The uncertainty analysis reveals the contribution of each measurement to the overall test uncertainty. Thus, the analysis serves as a useful tool for improving the quality of the test.

The ASME publishes "Test Uncertainty, Instruments and Apparatus—Supplement to ASME Performance Test Codes" (ASME PTC 19.1–2005). It is an exhaustive explanation of the many nuances of testing uncertainty, complete with examples and models for unique and complicated situations. This Code is the standard by which all steam turbine test uncertainties should be calculated.

There are two types of uncertainty: systematic and random. Together, they compose the total uncertainty. Figure 2-11 illustrates the difference between the two.

Systematic uncertainty is the portion of the total measurement that remains constant in repeated measurements. It is sometimes referred to as bias, but the Code uses the term systematic. An example of systematic uncertainty would be the difference in the average pressure readings between a perfect test-grade pressure gauge (Figure 2-11, left) and an uncalibrated station pressure gage (Figure 2-11, right).

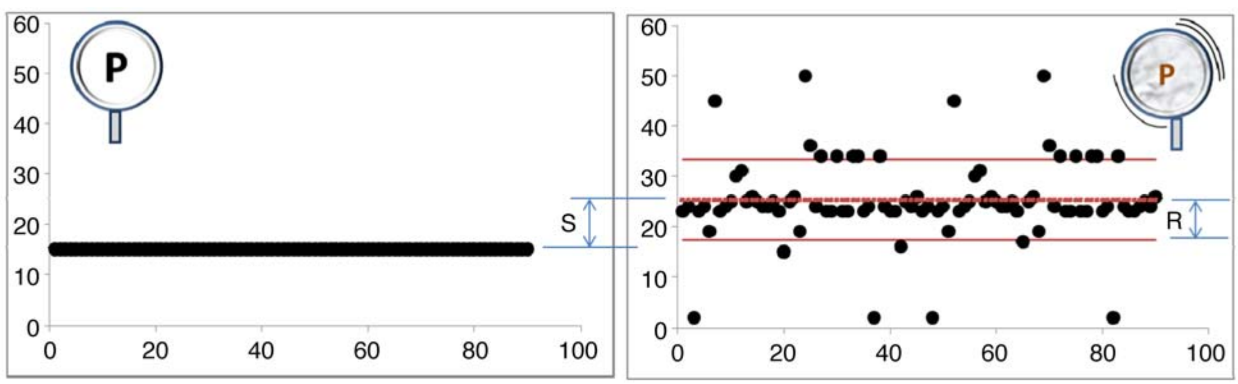

Figure 2-11
Systematic uncertainty (S) and random uncertainty (R)

Random uncertainty is the portion of the total measurement that varies in repeated measurements. It is evaluated by the measurement of scatter. Previously, this uncertainty was referred to as precision, but the Code now uses term random. An example of random uncertainty is the standard deviation around the average of several measurements.

What follows is a simple example of calculating HP turbine efficiency uncertainty. Assumptions consistent with the 19.1 Code result in a total uncertainty with a 95% confidence level. In other words, there is a 95% probability that the true value falls within the total uncertainty, which includes the systematic and random uncertainty. This statement is written mathematically as follows:

$$U_{95} = [\ S^2 + (2R)^2\ ]^{½}$$

where

U_{95} = uncertainty (%)

S = systematic uncertainty (%)

R = random uncertainty (%)

Calculating the enthalpy decrease efficiency of the HP turbine involves a seven-step process.

1. The first step is to collect the data. For this example, the test data consist of pressure and temperature in and out of the HP turbine. Data were collected every minute over the course of 2 hours. Table 2-1 shows the data.

Table 2-1
HP efficiency uncertainty example

Time	Pressure in	Temperature in	Pressure out	Temperature out
1	2400	1000	535	625
2	2405	998	532	624
3	2398	1002	538	626
⬇				
120	2402	999	536	627
Mean (psia or F)	2400	1000	535	625
Standard deviation (psia or F)	3.2	4.1	1.9	3.7
Systematic uncertainty (psia or F)	2.0	1.0	1.5	1.0
Sensitivity (%/psia or %/F)	0.04	0.29	0.09	0.31

2. The second step is to calculate the mean and standard deviation for the data. The units are in pounds per square inch and degrees Fahrenheit, respectively, for pressure and temperature. This step can be quick and simple by using existing statistical functions in spreadsheets.
3. The third step is to estimate the systematic error. This portion of the uncertainty analysis has the most subjectivity. The Code gives guidelines for assessing the systematic error. Ideally,

one can compare the test instrument with a calibrated standard, similar to what is illustrated in Figure 2-11. Less ideally, engineering judgment can be used to estimate the combined errors from the calibration process, transducer errors, environmental effects, and so on. Note that a highly accurate instrument can still have high uncertainty if the steam is stratified or if the thermo well is not clean, among others.

4. The fourth step is to determine the sensitivity, which is a measure of how much the efficiency changes with a unit change in each parameter. Using the mean pressures and temperatures to look up enthalpies, the HP turbine efficiency is:

$$\eta = \frac{hin - hout}{hin - hi}100 = \frac{1461.11 - 1309.81}{1461.11 - 1280.79}100 = 83.91\%$$

   If the inlet pressure was increased by 1 psi, the resulting inlet enthalpy would be 1461.08 and the isentropic enthalpy would be 1280.72. The new efficiency would be:

$$\eta' = \frac{hin - hout}{hin - hi}100 = \frac{1461.08 - 1309.81}{1461.08 - 1280.72}100 = 83.87\%$$

   The sensitivity, θ, is calculated from the difference in efficiency for a 1-psi change in inlet pressure:

$$\theta = \frac{\eta - \eta'}{\Delta \mathrm{P}in} = \frac{83.91 - 83.87}{1} = 0.04\% / psi$$

   The sensitivity for the remaining pressure and temperatures is calculated in a similar manner.

5. The fifth step is to calculate the random uncertainty, *R*, of the efficiency calculation, which is a function of the individual sensitivities and standard deviations:

$$R = \sqrt{\sum_{1}^{4} (\theta isdevi)^2}$$

$$R = [(0.04 * 3.2)^2 + (0.29 * 4.1)^2 + (0.09 * 1.9)^2 + (0.31 * 3.7)^2]^{½} = 0.36\%$$

   When necessary, the random uncertainty can be improved by evaluating the scatter from a curve fit of the data. In other words, an instrument with zero scatter will erroneously be calculated as having scatter if the measured value slowly varies or drifts during a test.

6. The sixth step is to calculate the systematic uncertainty, *S*, in a similar manner from the individual sensitivities and systematic uncertainties:

$$S = \sqrt{\sum_{1}^{4} (\theta i S i)^2}$$

$$S = [(0.04 * 2.0)^2 + (0.29 * 1.0)^2 + (0.09 * 1.5)^2 + (0.31 * 1.0)^2]^{½} = 0.45\%$$

7. The seventh and final step in calculating HP turbine efficiency uncertainty is to combine the random and systematic uncertainties:

$$U_{95} = [0.45^2 + (2 * 0.36)^2]^{½} = 0.85\%$$

Therefore, the HP turbine efficiency is 83.91 ± 0.85%.

Isolation

Isolation can be one of the greatest points of contention in the reconciliation of test results.

Isolation Checks

Cycle isolation is extremely important in assessing the performance of the turbine cycle. The power out of the turbine depends on the mass flow going through it. There is nearly a 1% per 1% ratio of power output to mass flow. When an isolation breach occurs, the flow passing through the turbine is in question, as is the performance of the turbine. When conducting acceptance tests, what seems like an insignificant leak can make the difference between pass and fail.

A classic example of how isolation affects acceptance test results occurs when the unit appears to have failed the test, but close attention to isolation was not paid before the test. After review of the data, the owner states that the unit failed by, say, 1%. The supplier, on the other hand, states that there was a 2% leak at the boiler and if that leak was not there, the turbine would have passed by 1%. Although both parties point toward cycle isolation as the one thing that would change the results, the two parties were not in agreement about the state of isolation immediately before and during testing. Unfortunately, after the test when operation of the plant is restored to normal, it is impossible to determine the actual state of isolation during the test retroactively.

There are some steps to take to avoid this unfortunate situation. These steps are not reserved for just acceptance testing. Accurate assessment of turbine performance requires careful attention to cycle isolation at all times. Proper cycle alignment provides the flows for optimal turbine performance.

Use the P&IDs as a Guide

Up-to-date P&IDs are the starting point for an isolation check. In reviewing them, anticipate where possible leaks may occur. Identify bypass lines, heater, economizer and turbine drain lines, steam traps, shutoff valves for process lines, auxiliary equipment, startup steam sealing, and so on. From the P&IDs, an isolation list can be developed and used to walk the plant.

Measure Pipe Temperature

One technique for determining whether a valve is leaking is to measure the temperature of the pipe downstream of a valve that is supposed to be closed. It is a good idea to have a baseline temperature because conduction along the pipe walls and the surrounding heat will influence the pipe temperature. It is also useful to review the heat balance to know the steam conditions upstream of the valve in question. It is important to realize, however, that measurement of the temperature will not give the quantity of flow leaking through the valve. It will only be an indicator of a leak. Therefore, this method is useful only on pipes where no flow is expected.

Measure Pressure Upstream of Condenser

Often the quantity of the leak is desired. In leaks that go to the condenser, there is a method that will allow estimation of the quantity of the flow entering the condenser. This method provides a good approximation and makes use of the large pressure ratio across the leaking valve that accelerates the steam to sonic velocity (Figure 2-12).

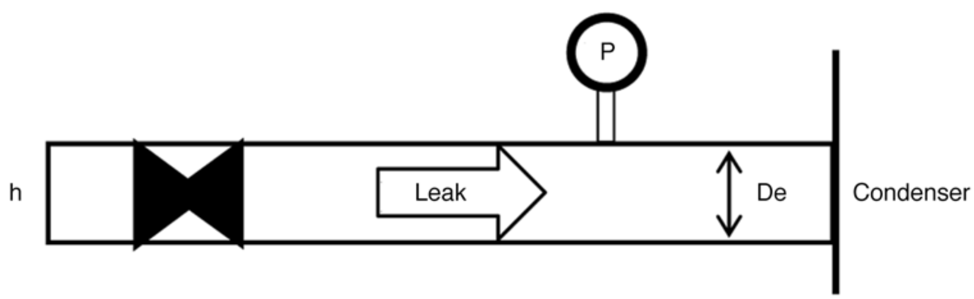

Figure 2-12
Schematic for determining mass flow rate to the condenser from a leaking valve

According to the law of continuity, mass flow is a function of flow area, specific volume, and velocity. With the described setup, the flow area can be determined from the pipe diameter, the specific volume from the enthalpy, and measured pressure. Because there is no enthalpy decrease across a throttling valve, the enthalpy of the steam just upstream of the condenser is that of the steam upstream of the valve. The heat-balance diagram gives the enthalpy at this location.

The critical pressure ratio is the pressure ratio that accelerates a fluid-to-sonic velocity. The critical pressure ratio is a function of temperature, but is approximately 2:1. Therefore, any valve holding back more than twice the condenser pressure, which is most, will work with this method. Sonic velocity is also a function of temperature as follows:

$$Vs = \sqrt{\frac{2gPv144\gamma}{(\gamma+1)}}$$

where

V_s = sonic velocity (ft/s)

g = 32.174 lbm-ft/lb_f-s^2

P = pressure in the pipe (psia)

v = specific volume in the pipe (ft^3/lbm)

γ = isentropic constant (1.3 for superheated steam)

Now, with the specific volume from pressure and enthalpy and the diameter of the pipe, the mass flow is calculated as follows:

$$W = 25 * \pi * D^2 * V_s / (4 * v)$$

where

W = leakage mass flow (lbm/h)

D = diameter of the pipe entering the condenser (in.)

V_s = sonic velocity (ft/s)

v = specific volume (ft^3/lbm)

25 = conversion factor (ft^2-s/in^2-h)

The flow calculated using this method ignores the flow coefficient associated with the physical configuration of the pipe. The actual flow will be somewhat less than what is calculated here, depending on the flow coefficient, which varies from approximately 0.7 to 0.9.

Maintain an Isolation To Do List

During normal operation, maintain an ongoing list of isolation breaches and leaky valves to correct during planned or unplanned outages. Prioritize the list in terms of overall effect on performance results and the time required to fix the problem. Fixes might include inspecting and maintaining valves, installation of instrumentation (such as the pressure taps installed in the lines to the condenser discussed previously), or adding valves in series to existing valves that are suspect. A short outage, such as that caused by a boiler tube leak, might become a perfect opportunity to correct an isolation problem that takes a day or two.

For those fixes that will take more than a couple of days, a major overhaul may be the only opportunity to address the problem. Entering an outage with a prioritized list, the supplies, and the support of the I&C department will facilitate this process.

On-Line Monitoring

Periodic testing essentially keeps a finger on the pulse of the turbine's health. However, the potential is there to monitor thousands of steam turbine data points every second to the point where one might be overwhelmed with data. From the steam turbine performance engineer's perspective, keeping track of turbine cycle performance by tracking a few major parameters is time- and cost-effective. Once a major parameter shows a red flag, collecting additional data or collecting the data more frequently might be warranted. The major parameters to watch are the following:

- Corrected gross power output
- Corrected gross turbine cycle heat rate
- HP and IP casing efficiencies
- Corrected stage pressures
- Final feedwater flow

All the major parameters should be compared with the basis for comparison, which may be the original acceptance test, the last turbine overhaul, or perhaps the last turbine upgrade. If they are all presented as a percentage of change from the expected, correlations between the different parameters are easy to reconcile.

Corrected Gross Power Output

The gross generator output, before the auxiliary load is taken off, is corrected to rated conditions. The output should follow flow closely: if flow increases 5%, then output should increase approximately 4.9%. If it does not, further investigation is necessary. Over time, the power output may decrease for a given flow. Plotting the gross power output versus time will give an indication of whether any change in power occurred gradually or abruptly.

Corrected Gross Turbine Cycle Heat Rate

The gross turbine cycle heat rate is the heat input to the final feedwater and the cold reheat steam divided by the gross generator power output. It is the reciprocal of the cycle efficiency. Once corrected to rated conditions, it is compared with the basis for comparison (acceptance test, last retrofit, and so on). Heat-rate monitoring is generally more complicated and uncertain than power output because it involves the measurement of flow in the calculation.

HP and IP Turbine Efficiencies

Turbine casings that have exhaust steam at least 25°F in the superheated region are candidates for enthalpy decrease efficiency. In most cycles, that means the HP and IP turbines. Unlike power, heat rate, and stage pressure, the efficiencies are not corrected to rated conditions. The uncorrected pressures and temperatures are used to determine enthalpy (although the measured pressures are compensated for water legs). For the reconciliation process, it is useful to know the contribution of each casing so that the corresponding effect on power output and heat rate can be determined.

Corrected Stage Pressures

Stage pressures include first-stage shell pressure, cold and hot reheat pressure, crossover pressure, and the heater extraction pressures. These pressures are corrected to rated conditions because plant conditions can influence stage pressures. Pressure is a function of two things: flow and downstream resistance. The percentage of change in a stage pressure from its design level can indicate changes in flow through the following stage and in the resistance of downstream stages affected by deposits, erosion, or mechanical damage.

Corrected Flow

Mass flow into the turbine, corrected for main steam pressure and temperature, is another useful parameter to monitor. Flow, power output, and stage pressures generally follow one another. When they do not, that is cause for concern. Leaks and damage to the steam path are likely causes when these parameters do not follow one another.

Testing

With on-line monitoring systems becoming more sophisticated and automated, the line between monitoring and testing is beginning to fade. With calibrated instrumentation installed and performance calculations integrated into the digital control system, testing can be a matter of reaching steady load and collecting a set of data over a period of a few hours.

Turbine testing can serve different purposes. Acceptance testing measures the performance of a new turbine, turbine section, or turbine part (such as a new last-stage blade). The results of the test are used to determine whether the supplier's guarantee is met. Bonuses or penalties are usually associated with the acceptance test results.

Periodic Testing

Periodic testing keeps track of performance changes caused by turbine overhauls, load changes, operation changes, and other changes to the turbine.

There are several types of tests that can be conducted on a periodic basis to keep track of turbine performance. They can also vary from fairly casual to expensive and complicated. The more casual testing can be considered an extension of the on-line monitoring, whereas the expensive and complicated testing might be considered in an upgrade. The type of test, the instrumentation grade, and the frequency of testing is something that the steam turbine performance engineer must decide depending on the situation.

Several different tests might be part of the overall heat-rate improvement program. Maximum power output tests might be completed using the on-line monitoring system once per week. Enthalpy decrease efficiency tests might be conducted on a monthly or quarterly basis. Heat-rate tests might be conducted quarterly or annually. Several different factors go into deciding the frequency of testing including budget, currently known or suspected performance problems, changes in operation such as from base-loaded to cycling, and so on.

Repeatable Valve Points

When the turbine changes load at a constant inlet pressure, it does so by changing the position of the inlet valves. When the valve position changes, the corresponding throttling losses also change. The throttling has an effect on power output, heat rate, stage pressures, and HP turbine efficiency. This measured loss, however, is not a function of the turbine's condition. It is merely a matter of how much the inlet valves are throttling. Because the purpose of turbine testing is to determine the turbine's condition, filtering out any throttling loss would improve the accuracy of the testing. The best way to eliminate changes in throttling losses is to conduct tests at repeatable valve points. That way, the throttling losses are minimized and the difference between throttling losses from one test to another is minimized as well.

Whether the control stage is a partial-arc or full-arc design, the VWO position is generally the best repeatable valve position. At part load in a full-arc design, all the valves throttle and it is difficult to get a repeatable valve position. At part load in a partial-arc design, the repeatable valve points occur at the points of minimum throttling, corresponding to some valves wide open, while others are fully closed. Of course, the control system usually requires some overlap in the valve opening and closing sequence.

Always include the valve position when collecting test data. The data might include some readout in the control room, or it might involve measuring the valve stem travel. It might include the pressure decrease across the valves. No matter how the valve position is ascertained, this frequently overlooked parameter should be included in the data set for tests.

Testing Over the Load Range

Much of the testing is conducted at full load, but it is useful to conduct tests over the load range as well. The results will help verify the data. Some parameters change over the load range such as HP and LP turbine efficiency, power output, heat rate, and stage pressures. It is useful to determine whether the shapes of the curves meet expectations. If they do not, there is likely a problem with the test data. Figure 2-13 shows the expected shapes of several performance curves over the load range. The plots of HP turbine efficiency and heat rate contain both the sequential valve admission (scalloped) and the full-arc admission (straight line) curves, although the relative magnitudes of those two curves vary per machine and are not representative of any one turbine or all turbines. All are plotted against load except LP turbine efficiency, which is plotted against annulus velocity.

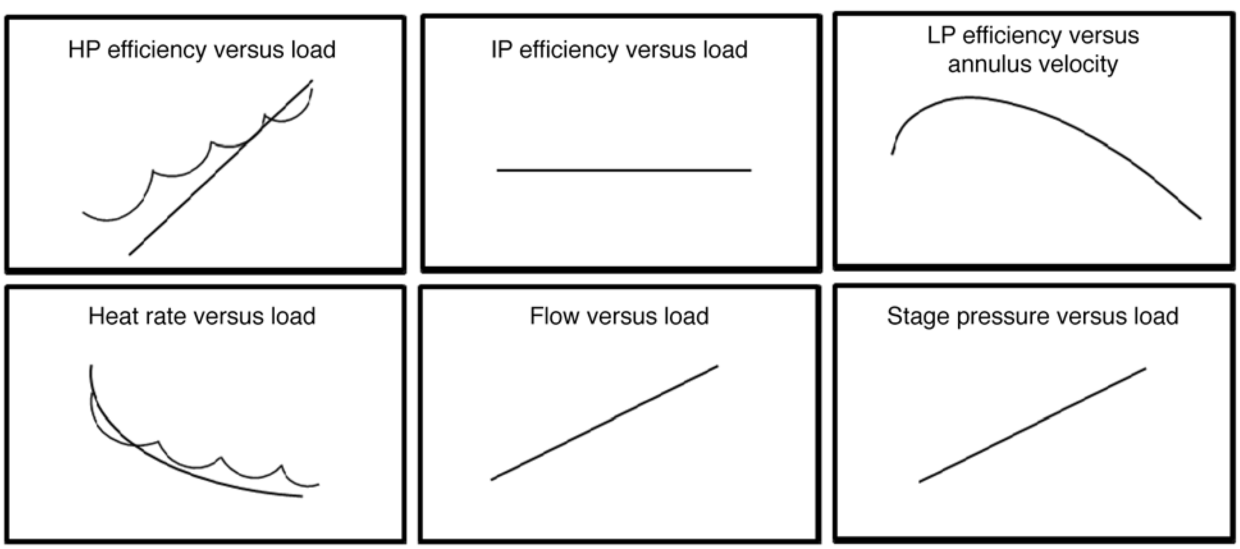

Figure 2-13
Expected shape of the performance curves over the load range

Other parameters that should remain constant over the load range include the percentage of pressure decrease across the reheater, pressure ratios between stage pressures, and, for a check of valve position, pressure decrease across the control valves.

Specialized Testing

The periodic testing described may identify some performance problems. However, before making recommendations to management that represent time and expense invested in improving performance, the steam turbine performance engineer may want to increase the certainty level with which these decisions are being made. Certain specialized tests can help reinforce, or refute, the findings of the more general periodic testing.

HP-IP Internal Leakage

Where the HP and IP turbine casings are in an opposed-flow configuration in a single outer casing, there will be internal leakage from the front end of the HP turbine to the IP turbine bowl. Often this sort of leakage is identified because measured IP turbine efficiency (taken from the hot reheat point to the crossover) is higher than expected or increases over time. The measured efficiency increases because the lower enthalpy incoming leakage steam enters the IP turbine after its inlet temperature has been taken. The resulting leakage decreases the IP exit temperature, giving the illusion that the IP turbine is becoming more efficient. This anomaly is a red flag for HP-IP internal leakage, but there are several tests to confirm whether this leakage situation exists.

A distinction must be made between measured IP turbine efficiency and internal IP turbine efficiency. The measured IP turbine efficiency is taken from the hot reheat conditions, usually upstream of the reheat intercept valves. It uses pressures and temperatures in and out of the IP turbine to calculate IP turbine efficiency (Figure 2-14).

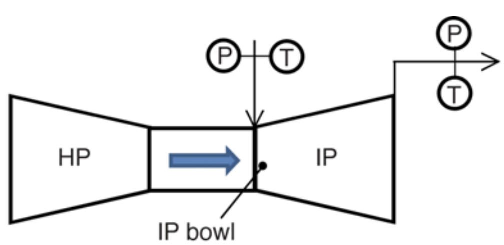

Figure 2-14
Opposed-flow HP-IP turbine casing with instrumentation for measured IP turbine efficiency

The internal IP turbine efficiency is calculated from the IP bowl to the IP exhaust. The bowl conditions are obtained from the mixing of hot reheat steam and the internal leakage entering upstream of the IP turbine first stage. The energy equation is used to calculate the mixed enthalpy in the IP turbine bowl as follows:

$$H_{hrh} * w_{hrh} + h_l * w_l = h_b * w_b$$

where

h_{hrh} = enthalpy of the hot reheat steam, a function of measured hot reheat temperature and measured hot reheat pressure

w_{hrh} = calculated hot reheat flow rate from the measured final feedwater flow, calculated top heater extraction flow, and other flows between the feedwater flowmeter and the IP inlet such as boiler blowdown, superheat attemperating spray, and end-packing or gland seal leakoffs

h = leakage enthalpy taken from the design heat balance

w_l = assumed leakage flow rate

h_b = calculated IP turbine bowl enthalpy from the equation

w_b = IP turbine bowl flow rate = $w_{hrh} + w_l$

The paper by Booth and Kautzmann (see the section “What Should Be in Your Library”) is the resource for conducting specialized tests for estimating HP-IP turbine internal leakage. Warren Hopson from Southern Company has published some useful articles on the application of the Booth-Kautzmann method (see the section “What Should be in Your Library”). The temperature shift test and the IP slope over the load range test are two common ways to measure this leakage. Both tests make use of the fact that internal IP efficiency does not change with changes in turbine inlet conditions or changes in load. From the IP bowl to the IP exit, the efficiency is expected to remain constant.

In the temperature shift test, inlet temperature to the HP turbine and IP turbine are changed one at a time. Data are collected for internal IP efficiency for each temperature shift. To calculate the efficiency, an assumption must be made about the amount of leakage flow. The real IP turbine internal efficiency does not change with inlet conditions. Therefore, where the IP turbine efficiency for one temperature equals that for the shifted temperature, the corresponding leakage flow is the correct flow from the HP turbine to the IP turbine. Figure 2-15 shows the resulting graph.

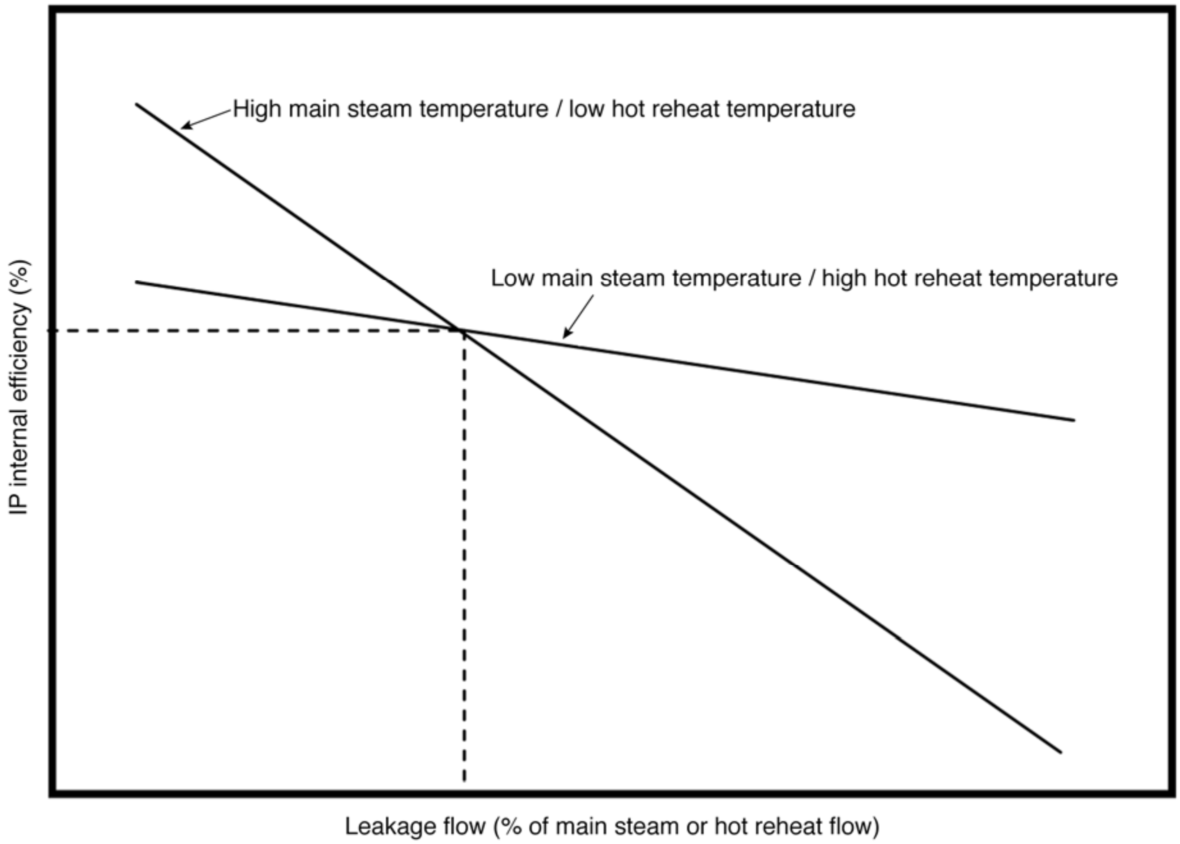

Figure 2-15
IP turbine internal efficiency using the temperature shift method

IP Efficiency Slope over the Load Range

Another method to determine HP-IP internal leakage discussed in the paper by Booth and Kautzmann takes advantage of the fact the IP turbine internal efficiency remains constant over the load range. The IP efficiency slope over the load range method makes use of the slope of the internal efficiency versus load for a series of assumed leakage flows.

The method involves taking measured IP turbine efficiency readings at valve points over the load range. At each valve point, the unit should reach steady-state conditions for at least an hour before data are collected. Once all the data are collected, the calculations can begin.

The internal IP turbine efficiency is calculated using an assumed internal leakage for all load points and plotted across the load range. If the slope is not flat, then the assumed leakage is wrong. A new assumed leakage will yield a new internal efficiency over the load range slope. This process is repeated until the slope is flat, which corresponds to the correct internal leakage and the correct IP turbine internal efficiency (Figure 2-16).

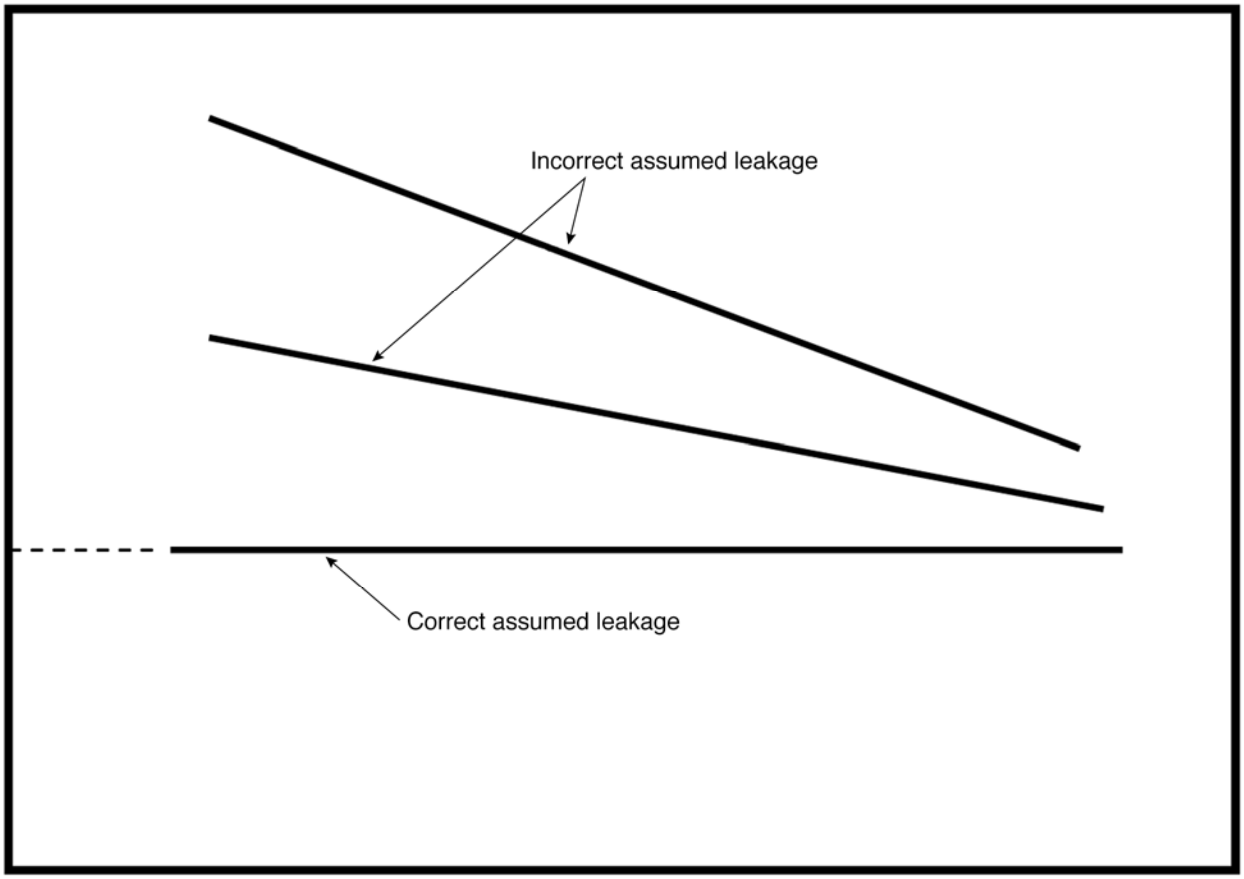

Figure 2-16
IP turbine internal efficiency using the slope over the load range method

SPE Erosion in the HP Control Stage

When routine testing, control-stage SPE will show up as increased flow, possibly increased power output, and decreased HP turbine efficiency. All these changes happen gradually over time, but there is another specialized test that will give better confirmation of control-stage SPE.

SPE occurs frequently in partial-arc control stages operated in sequential valve admission mode. The first valve to open has the largest pressure decrease across it and therefore the highest steam velocity. There may be a high concentration of solid particles in the boiler tubes and main steam piping after an outage caused by oxygen entering the system. Conditions are very favorable for SPE of the nozzles under the first valve to open when the unit first starts up after an outage. This test for the existence of SPE makes use of this fact.

The HP turbine efficiency is taken over the load range. Unlike the IP turbine efficiency discussed previously, there is no differentiation made between measured efficiency and internal efficiency. The data are taken at valve points at which the unit has reached steady-state conditions. The data are then plotted as HP efficiency versus load where a line is drawn through the efficiencies at the valve points. This curve is plotted along with and compared with the design curve. At low loads, there will be a greater percentage difference between the two than at high loads. This difference arises because at low load, most of the first-stage flow passes through the damaged nozzles. At high loads, a smaller portion of the first-stage flow passes through the damaged nozzles. Figure 2-17 shows the resulting plot that confirms the existence of control-stage SPE. There is a larger decrease in HP efficiency at low loads than at high loads.

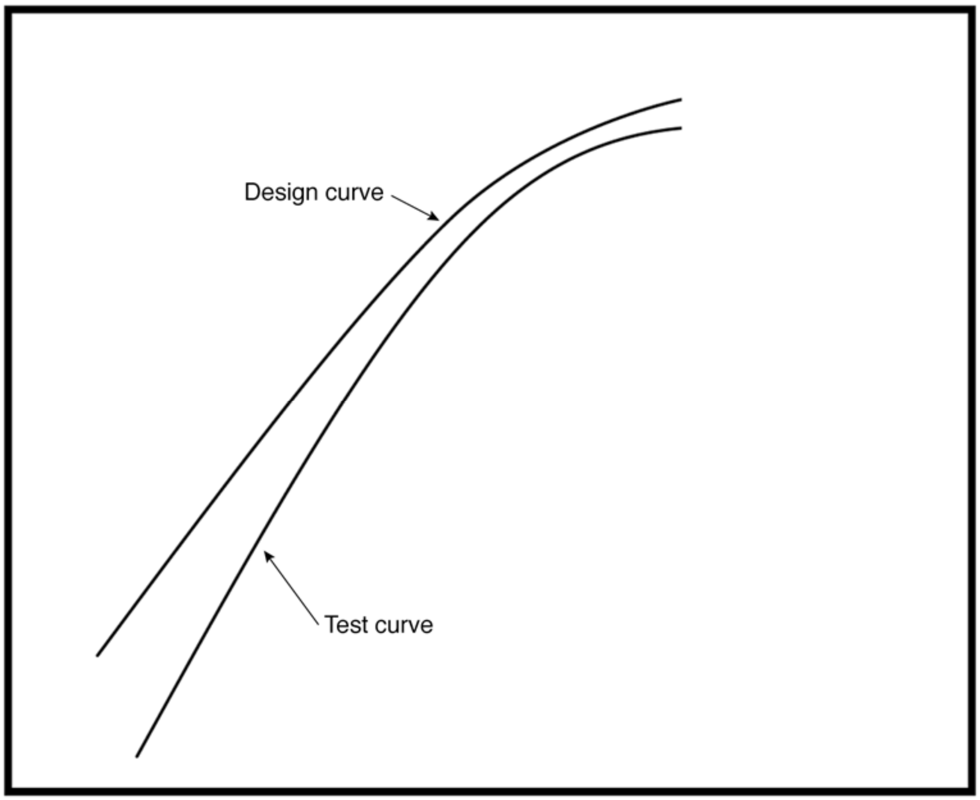

Figure 2-17
HP turbine test for control-stage SPE

Correcting Test or Operating Data

Purpose of Correcting Data

The purpose of correcting data is to calculate how the turbine would perform if it were provided rated conditions from the plant. As an example, consider a condenser pressure that is too high because of warm ambient conditions. The turbine's power output will decrease, but this decrease is not from a problem with the turbine. Correcting the test power to the power corresponding to the rated condenser pressure will allow the steam turbine performance engineer to measure steam turbine performance deterioration from turbine problems. The corrections filter out plant influences on turbine performance.

What to Correct

The following performance parameters should be corrected to rated conditions to filter out plant influences:

- Stage pressures
- Power output
- Flow
- Heat rate

What to Correct for

The PTC Code describes group 1 and 2 corrections for test data. Group 1 corrections are mainly for cycle parameters that are off-design such as condenser subcooling. They generally have less magnitude than the group 2 corrections. Group 2 corrections are for the steam conditions entering and exiting the turbine that are the responsibility of the plant. Steam conditions such as main steam pressure and temperature, reheat temperature, reheater pressure decrease, and condenser pressure are included in group 2 corrections. When any one of these parameters is off-design, the turbine performance changes. Group 2 corrections remove this influence from the test result so that the analysis can focus on turbine performance changes that are caused by the turbine itself.

How to Correct Test Data

There are sources available to the turbine performance engineer for determining the equations used to correct test power and heat-rate data. The ASME PTC 6 Code gives some generic curves, but these curves are based on a single reheat steam turbine with inlet conditions of 2400 psig/1000°F/1000°F. Other turbines not fitting this description will have different corrections.

The OEM will typically have group 2 correction curves for power and heat rate included in the thermal kit. If they are not in the thermal kit, the steam turbine performance engineer is well within bounds to ask the OEM for the group 2 corrections.

It is a good idea to review the correction curves carefully. Sometimes there are generic curves, such as in the ASME Code, that should not be applied to a specific unit. A good way to validate the curves is by using a heat-balance program that can model the turbine cycle. Then a single change to a steam condition can be used to see the resulting change in power output and heat rate. Corrections to stage pressure can also be ascertained in this manner. Corrections to stage pressures are found in Figure 2-18.

Formula	Type of Unit	Pressures to be Corrected
$P_C = P_o\sqrt{\frac{P_D\upsilon_T}{P_T\upsilon_D}}$	Non-reheat	P_{1st}, P_{extr}
$P_C = P_o\sqrt{\frac{P_D\upsilon_T}{P_T\upsilon_D}}$	Reheat	P_{1st}, P_{HARP}
$P_C = P_o\sqrt{\frac{P_D\upsilon_T}{P_T\upsilon_D}}\sqrt{\frac{\upsilon_{DR}}{\upsilon_{TR}}}\left[\frac{\omega_{RH}}{\omega_{RH}+\omega_S}\right]$	Reheat	P_{crh}, P_{hrh}, $P_{x\text{-}o}$, P_{extr}
$P_C = P_o\sqrt{\frac{P_D\upsilon_T}{P_T\upsilon_D}}\sqrt{\frac{\upsilon_{DR}}{\upsilon_{TR}}}\left[\frac{\omega_{RH1}}{\omega_{RH1}+\omega_{S1}}\right]$	Double Reheat	P_{crh1}, P_{hrh1}, P_{IPextr}
$P_C = P_o\sqrt{\frac{P_D\upsilon_T}{P_T\upsilon_D}}\sqrt{\frac{\upsilon_{DR}}{\upsilon_{TR}}}\left[\frac{\omega_{RH1}}{\omega_{RH1}+\omega_{S1}}\right]\left[\frac{\omega_{RH2}}{\omega_{RH2}+\omega_{S2}}\right]$	Double Reheat	P_{crh2}, P_{hrh2}, $P_{IP2extr}$, $P_{x\text{-}o}$, P_{LPextr}

Figure 2-18
Corrections to test stage pressures

Parameter definitions for Figure 2-18 are as follows:

P_C = corrected pressure for plotting (psia)

P_o = test stage or shell pressure (psia)

P_T = test throttle pressure (psia)

P_D = design or reference throttle pressure (psia)

υ_D = design or reference throttle-specific volume (ft^3/lbm)

υ_T = test throttle-specific volume (ft^3/lbm)

υ_{TR1} = specific volume at **test temperature** and **test pressure** at the inlet to intercept valves of the first IP section (ft^3/lbm)

υ_{DR1} = specific volume at **design reheat temperature** and **test pressure** at the inlet to intercept valves of the first IP section (ft^3/lbm)

υ_{TR2} = specific volume at **test temperature** and **test pressure** at the inlet to intercept valves of the second IP section (ft^3/lbm)

υ_{DR2} = specific volume at **design second reheat temperature** and **test pressure** at the inlet to intercept valves of the second IP section (ft^3/lbm)

w_{RH} = reheat flow rate (lbm/h)

w_S = reheat spray flow rate (lbm/h)

Troubleshooting

KCC Signature Method to Analyze Data

Ken Cotton is the author of *Evaluating and Improving Steam Turbine Efficiency* [3]. He spent an illustrious career improving steam turbine efficiency. While at the General Electric Company, he supervised GE's Fluid Mechanics Advanced Engineering Program and the Aerodynamic Development Laboratory. He taught Fluid Mechanics at Union College and is the author of several papers and the recipient of numerous ASME awards. He served on the ASME and international test code committees for steam turbines for 30 years. His contributions to the industry are very much appreciated and will be long remembered.

Ken Cotton honed his skills at analyzing steam turbine performance data and troubleshooting turbine problems. His signature method starts with verifying the data, then trending them, and then reconciling them before moving on to the analysis and recommendation portion.

Verify

The verification process essentially filters out bad test data. Much valuable time can be wasted trying to make sense out of bad test data. The verification process allows the turbine performance engineer to focus on the relevant data.

Understand the data historian's capabilities, for example, compression limits. Sometimes data are collected only when the parameter has drifted outside a preset range. If the range is too large, the quality of the data is compromised.

It is a good idea to conduct tests over the load range rather than solely at full load or some other single-load point. Many expectations for data involve how certain parameters respond with changes in load. IP turbine internal efficiency, for instance, should remain constant over the load range. If it does not, there is likely a problem with the data. If only one load point is tested, the IP efficiency is unknown at other loads, and a potential problem may not be identified.

After collecting data, it is useful to plot the data to eliminate outliers. The Code gives guidelines for acceptable stability and allowable deviations from expected conditions.

Much of the verification process includes checking to make sure the data meet expectations. One expectation is that certain parameters should remain constant with load and changes in inlet steam conditions. The percentage of pressure decrease across the reheater, pressure ratios across turbine sections (other than the first and last stages), and IP turbine internal efficiency are all examples of parameters that should remain constant with flow. If they do not, this serves as an alert that there may be a problem with the data.

Another expectation is that HP and IP turbine efficiency will remain less than 100%. If a measured efficiency between the inlet and outlet of a casing exceeds 100%, there is clearly a problem with the data. It is not so obvious, however, when calculating efficiencies between extractions. The extraction steam temperature can be hotter than the average stage temperature in the turbine from which it originates because of a radial thermal gradient along the turbine blades and tip leakage. Therefore, the section efficiencies between extractions may have unreasonably high or low values. If that extraction temperature is falsely high, the individual section efficiencies, when calculated using that value, may not reflect reality. Figure 2-19 shows the effect on those calculated section efficiencies caused by a falsely high extraction steam temperature.

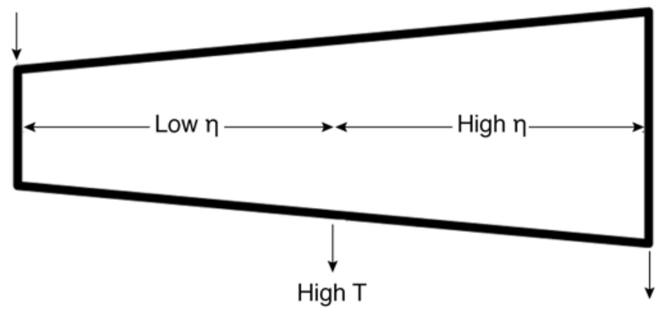

Figure 2-19
Expected section efficiencies with a mid casing extraction

Trend

The next step in Ken Cotton's signature analysis method is to trend the data. Trending helps to determine when a problem occurred and whether it occurred abruptly or gradually. A snapshot of data might show that performance is currently down, but trending helps determine whether the current problem is ongoing or whether it occurred at some point in the past.

The parameters to trend are relatively few. The Ken Cotton methodology focuses on getting 80% of the information needed from 20% of the available data. Then, if further testing is required, the remaining data can be used. The parameters to trend are the following:

- Corrected power output
- Corrected turbine cycle heat rate
- Corrected throttle flow
- Key pressures
- Casing efficiencies

It is useful to plot all casing efficiencies on one graph; power, flow, and heat rate on another graph; and flow and pressures on another graph.

The Ken Cotton technique used to trend the data is very helpful in viewing the whole picture and in communicating results to others. Guidelines in trending the data are as follows:

- Plot the percentage of deviation from design or expectation. Having already calculated the new and clean condition of the turbine, it can be used as a basis for comparing test data. Present the data on graphs showing the percentage of change from the basis of comparison, not absolute values. A 5% decrease in power is more meaningful than 5 MW when looking at the percentages of changes in pressures, flows, and casing efficiencies.
- Use the same scale on all graphs. The Y axis will be the percentage of change from design and the X axis will be time.
- Do not smooth out the curves. An abrupt change in the data with time is useful diagnostic information. It is useful to superimpose other relevant events that correspond with the abrupt changes such as startups, high vibration events, boiler tube repairs, and so on.

Reconcile

The reconciliation process involves adding up the changes in power from known casing efficiency changes and changes in flow and comparing them with measured power. If the change in, for example, HP efficiency is known and the contribution from the HP to the overall power output is known, then the effect on power from a change in HP efficiency can be calculated.

An example of setting up the reconciliation equations follows for a typical reheat turbine. The HP/IP/LP split in their contribution toward power output is 28%/22%/50%, respectively. Assume that HP turbine efficiency is down 3%, IP turbine efficiency is down 4%, and main steam flow is down 2%. The measured power output is down 5% from design. The reconciliation process is as follows:

First, the HP turbine efficiency: The efficiency is down 3% and the HP turbine contributes 28% of the power. The loss factor for the HP turbine is 1.0 because what follows it is the reheater. Therefore, the effect on power from the HP turbine efficiency being down 3% is:

$$\%\Delta \mathrm{PWR} = \%\Delta\eta_{\mathrm{HP}} * \text{Casing Contribution} * \mathrm{LF} = 3.0 * 0.28 * 1.0 = 0.84\%$$

Now the IP turbine efficiency: The efficiency is down 4% and the IP turbine contributes 22% of the power. The loss factor for the IP turbine is 0.5 because the following LP turbine stages recover roughly half of the IP turbine stage losses. Therefore, the effect on power from the IP turbine efficiency being down 4% is:

$$\%\Delta \mathrm{PWR} = \%\Delta\eta_{\mathrm{IP}} * \text{Casing Contribution} * \mathrm{LF} = 4.0 * 0.22 * 0.5 = 0.44\%$$

Finally, the flow: Power output is proportional to flow. For every 2% decrease in throttle flow, one would expect almost 2% decrease in power output. It is not exactly a 1%-to-1% relationship, however. For every change in throttle flow, there is a corresponding change in LP turbine exhaust flow. In a typical Rankine cycle, the percentage of change in inlet flow equals the percentage of change in exhaust flow. When the exhaust flow changes, the exhaust losses also change. One of Ken Cotton's rules of thumb [3] is for every 1% change in throttle flow, there is a 0.94% change in power output. (This rule of thumb is corroborated by review of the total exhaust loss curve in which total exhaust losses are proportional to the annulus velocity, which, in turn, is proportional to turbine exhaust flow.) The contribution from the 2% decrease in flow is:

$$\%\Delta \mathrm{PWR} = \%\Delta W * 0.94 = 2.0 \ * 0.94 = 1.88\%$$

Putting it all together, the reconciliation for a 5% decrease in measured power is as follows:

Flow: down 2%: −2% * 0.94 = −1.88%

HP turbine efficiency: down 3%: −3% * 0.28 * 1.0 = −0.84%

IP turbine efficiency: down 4%: −4% * 0.22 * 0.5 = −0.44%

Total: −3.16%

For parameter definitions, refer to the sections on LP Turbine Contribution and Loss Factor on page 2-19.

Therefore, the reconciliation shows that 3.16% out of 5% can be reconciled. The remaining power output loss that is not accounted for (1.84%) is likely found in the LP turbine or the cycle, recognizing that LP turbine efficiency is not usually measured during routine testing. Because the LP turbine contributes 50% of the power, its efficiency would have to be down 1.84/0.50 = 3.68%. A scenario such as this seems reasonable because the HP and IP turbine efficiencies are down 3% and 4%, respectively.

It is rare to get a perfect reconciliation. If so, the power changes from the HP and IP turbine efficiencies and the flow would equal the measured power output and the LP turbine efficiency would be unchanged. The previous example demonstrates an underreconciliation, which means the effect of the efficiencies and flow totaled a value less than the measured power change. If the measured power in this example decreased 1% instead of 5%, then it would be an example of an overreconciliation. Overreconciliation means bad test data. Consider the previous example in which the reconciled power output is a decrease of 3.16%. With measured power output down 1%, the LP turbine efficiency would have to increase (3.16 − 1.00)/0.50 = 4.32%. There is no good reason why an LP turbine's efficiency would increase over time, so if the data show an overreconciliation, the data are highly suspect.

Although trending shows **when** a problem occurred, the reconciliation process shows **where** in the cycle a problem occurred. If the largest portion of the reconciliation is from HP turbine efficiency, then the problem is in the HP turbine. Likewise, if the largest portion of the reconciliation is in the IP turbine efficiency, then the problem is in the IP turbine. Because flow is set by the control stage and is a function of pressure ratio across the first-stage blading, a change in flow indicates a change in the front end of the HP turbine. Therefore, if the measured power output is reconciled with flow and HP turbine efficiency, the problem is in the HP turbine.

Pressure Is a Function of Two Things

Stage pressures are very useful diagnostic indicators of problems in the steam path. It is important to understand that stage pressure is a function of two things: flow and downstream resistance.

Other than the control stage, the turbine stages have a fixed area. When more flow passes through a fixed area, the pressure upstream of that area begins to increase. This trend is evident when opening the turbine inlet valves at constant main steam pressure and temperature: the flow increases and first-stage shell pressure increases as well, usually directly proportional to flow.

Under normal circumstances, the stage areas remain fixed. Occasionally, however, foreign object damage, erosion, or deposits may alter the stage areas. These off-design conditions decrease the performance of the machine. Using the stage pressures to identify the location of the damage is very useful. For constant flow, when the area of a stage opens up, perhaps caused by SPE, the pressure immediately upstream of that stage decreases. Conversely, for constant flow, when a stage area decreases potentially because of the formation of deposits, the pressure immediately upstream of that stage increases.

Consider the first stage in the IP turbine, which is susceptible to SPE from reheater carryover. As the stage erodes, the nozzle area opens up. Therefore, the pressure immediately upstream of it, the hot reheat pressure, decreases. The inlet to the IP is far enough away from the front end of the HP turbine that this decrease in pressure will not affect first-stage shell pressure. Therefore, flow into the HP turbine, and therefore the IP turbine, will remain constant. This example shows how pressure is a function of downstream area; with constant flow, upstream pressure decreases as area increases.

Thus far, the two effects on stage pressure have been presented independently. But what happens when there is **both** a change in downstream resistance **and** a change in flow? This situation is common when some sort of flow path damage occurs in the HP turbine.

Consider thick deposits accumulated on the second stage in the HP turbine. The deposits cause the nozzle area to decrease and the pressure immediately upstream, the first-stage shell pressure, to increase. An increase in first-stage shell pressure will decrease the pressure ratio across the control stage; the main steam pressure remains constant, but the first-stage shell pressure increases. Because flow through the turbine is a function of the pressure ratio across the first stage according to the general flow equation (see previously under the heading First-Stage Shell Pressure), the flow will decrease. In essence, the increase in first-stage shell pressure **caused** the turbine inlet flow to decrease.

The example with deposits in the HP turbine decreasing flow refutes a popular misconception—that first-stage shell pressure is a reliable indication of flow. In the example, first-stage shell pressure increased whereas flow decreased, contrary to the relationship between pressure and flow alone. Sometimes plants use first-stage shell pressure as an indication of flow in the absence of a flowmeter. Using a flow based on first-stage shell pressure for diagnostic purposes introduces a high degree of uncertainty in the analysis because the practice ignores the possibility that changes in area can cause changes in pressure.

How to Use the Three Key Pressures

In a typical reheat cycle, the following three readily available pressures can be used for diagnostic purposes:

- First-stage shell pressure
- Cold or hot reheat pressure
- Crossover pressure

The relative change between these three key pressures can be indicative of changes in stage areas and flow. Figure 2-20 shows how the three key pressures respond to a variety of problems throughout the steam path.

Condition	P_{1st}	P_{hrh}	$P_{x\text{-}o}$
Increase $A1^{st}$ HP stage	⬆	⬆	⬆
Increase A IP stages	—	⬇	—
Decrease A_{1st} HP stage	⬇	⬇	⬇
Decrease A IP stages	—	⬆	—
Decrease A_{2nd} HP stage	⬆	⬇	⬇
Increase A_{2nd} HP stage	⬇	⬆	⬆
Decrease A LP stages	—	—	⬆

Figure 2-20
How key pressures respond to off-design conditions

Effect of Operation on Performance

Variable Pressure Operation

Variable pressure operation (VPO), also known as sliding pressure, is using the boiler pressure to control flow rather than the turbine inlet valves. It is usually used below the 50% load point. There are many benefits to operating with VPO at low loads.

VPO is often used to reduce SPE. In a partial-arc control stage operated with sequential valve admission, most of the erosion occurs on the arc under the first valve to open because of the large pressure ratio across it during constant pressure operation. When the unit first starts up in constant pressure operation, the inlet pressure is rated, for example, 2400 psi, and the first-stage shell pressure is proportional to flow. Just before the first valve opens, the first-stage shell pressure may be slightly in vacuum. The pressure ratio across the first stage is therefore tremendous. In VPO, the inlet pressure is much lower, so the pressure ratio across the first stage is much smaller and the velocity of the particles is correspondingly smaller. The resulting particle velocity is below the threshold velocity at which the particles begin eroding the blading material. In VPO, the solid particles may still pass through the control stage, but they are not moving fast enough to erode it.

Another benefit of VPO is a better (lower) heat rate in VPO than constant pressure operation. With VPO, the main steam temperature remains higher than it would be at constant pressure as the unit comes down in load. Therefore, the HP turbine has more available energy to work with at low loads. The cold reheat temperature is higher at lower loads than with constant pressure operation, and therefore less heat is needed in the reheater.

Yet another benefit of VPO is the reduced throttling losses across the turbine inlet valves. This benefit increases HP turbine efficiency and improves turbine cycle heat rate.

Feedwater Heaters Out of Service

Occasionally, feedwater heaters are taken out of service for maintenance without shutting down the turbine. Other times, heaters are taken out of service to increase the power output at the expense of heat rate. When a heater is taken out of service, the steam that used to be extracted now flows through the remaining stages in the turbine, producing additional power. The feedwater, however, remains unheated and the final feedwater entering the boiler may be significantly lower than design, requiring a significant increase in fuel input to the boiler.

The turbine instruction book has guidelines for taking heaters out of service. It details the number and order of multiple heaters that can be taken out of service. The risk of exceeding the allowable heaters out of service is the damage caused by increased flow through the back end of the turbine. The last stage, in particular, is susceptible to flow-induced stresses that can damage the blading. The turbine instruction book often requires a reduction in throttle flow to compensate for the increased back-end flow that comes from shutting off heater extractions.

Low-Load Operation and Reduced Minimum Load

Operating at low load is generally very inefficient. The stages operate significantly off their design points, throttling losses may occur through inlet valves, the last stage may experience recirculation, and proportionally more heat may be required in the boiler and reheater because of low cold reheat steam pressures and low final feedwater temperatures. Sometimes at very low loads, automatic turbine drains meant to remove condensation from startup open up. If the unit sits at a low load, these drains are not needed and otherwise cause leakage from the turbine. One way to improve low-load performance is to close these drain valves if it will not interfere with successful operation of the unit.

Over Pressure (OP) Operation

OP operation occurs when the turbine inlet pressure exceeds the rated pressure. A 5% OP is a common condition. There is typically a 5% OP heat balance in the thermal kit. The turbine instruction book also addresses OP operation with guidelines for the amount and duration of OP operation. The risk in not adhering to the manufacturer's guidelines is the potential damage caused by an increase in turbine inlet flow.

With a constant turbine inlet valve position, the flow into the HP turbine is directly proportional to the pressure in a typical Rankine cycle. When the inlet pressure increases 5%, the inlet flow increases 5%. Other parameters follow suit: the power output increases almost 5%, and stage pressure increases 5%.

Increased Ramp Rates

Ramp rates are part of the guidelines used for starting, loading, and shutting down the turbine. The turbine instruction manual will have recommended and maximum ramp rates for temperature changes per time, power output changes per time, among others. It is extremely important to remain within the ramp rate limits set by the OEM. If a ramp rate is exceeded, the chances are high that the elastic limit of the turbine material has been exceeded, resulting in plastic deformation. Also, ramp rates that are too high can result in differential expansion between the stationary and rotating parts, resulting in direct contact. The plastic deformation can result in casing warpage, which may open up leakage paths within the turbine such as horizontal joint leakage. Differential expansion can result in labyrinth packing teeth being sheared off, which increases steam seal leakage.

High ramp rates can cause much more than performance deterioration. They can become a reliability issue and can reduce the remaining life of the steam path components.

Turbine Follow Mode of Operation

Turbine follow mode is a way to operate the plant such that the power generation from the turbine-generator set is constant so the turbine must respond appropriately to maintain a steady power output. Power output is proportional to flow, and the general flow equation describes the relationship between the flow, control stage inlet steam conditions, and the control-stage area. With constant valve position, the pressure ratio across the control stage remains constant. The inlet steam temperature is controlled by the attemperators. Therefore, the variables in the general flow equation are the flow, inlet pressure, and pressure ratio across the control stage and control-stage area. For load following, the flow through the turbine varies to match the load:

$$w = CA\beta\sqrt{\frac{P}{v}}$$

where

> w = steam flow through turbine (lbm/h)
>
> A = flow area (tt^2)
>
> C = flow coefficient (−)
>
> β = pressure ratio (−)
>
> P = inlet steam pressure (psia)
>
> v = specific volume of the steam (ft^3/lb)

Starting with the general flow equation and the ideal gas law, at constant inlet temperature, the flow through the turbine is proportional to the control-stage area, the pressure ratio across the valves, and the upstream pressure. Power output is directly proportional to flow.

At constant valve position, the pressure ratio across the control stage and the control-stage area is constant. Flow and therefore power are directly proportional to inlet pressure. Inlet pressure is also a function of the turbine inlet valve position because the valves are downstream of the boiler outlet. Inlet pressure is difficult to control because it responds to both how hard the boiler is fired and the turbine inlet valve position.

At constant pressure, the flow and power are directly proportional to the product of area and β, the pressure ratio function. As the control valves open, the flow area increases, but the inlet pressure decreases. To increase inlet pressure, the flow increases, which increases first-stage shell pressure and decreases the pressure ratio across the control stage.

Obviously, maintaining a steady power output involves a delicate balance between the firing rate of the boiler and the valve position.

Maintenance

Outage Planning

The outage is the event during which many efforts of the steam turbine performance engineer come to fruition. Much work goes into planning for the outage so that the performance needs of the plant are met without jeopardizing the schedule or budget of the outage. Before the outage, the steam turbine performance engineer has ideally quantified the pre-outage performance of the unit, has an expectation for the type of deterioration that will be uncovered, has a list of leaky valves to be fixed and instrumentation to be installed or maintained, and has lined up contractors to address the anticipated tasks during the outage.

Know the Pre-Outage Condition of the Turbine Cycle

To quantify the performance benefit of the outage, the pre- and post-outage condition of the turbine must be known. Ideally, this determination comes from similar-type tests before and after the outage using the same calibrated instrumentation. Knowing the pre-outage condition of the turbine can do more. It can help prepare for what otherwise might be unexpected. Through the use of on-line monitoring and periodic testing, performance problems can come to light. Through specialized testing, the causes of specific problems may surface. Proper interpretation of the test data may lead to a high confidence level that unique problems exist in the turbine and can therefore be planned for and addressed quickly during the outage.

Anticipate Problems Uncovered During Outages

Pressure ratios are useful in interpreting test data. Because stage pressures are a function of both flow and downstream resistance, and one would expect the downstream resistance to remain unchanged, then as flow increases, so do the stage pressures by the same percentage. Therefore, pressure ratios within the turbine should remain constant with time and with load. For example, if the pressure ratio across the IP suddenly increased after a short outage to repair reheater tubes, one might expect damage in the front end of the IP that decreased the nozzle area. This situation would likely be corroborated with a low IP turbine efficiency, high hot and cold reheat pressure, and a sudden decrease in power. The anticipated problem would likely be either debris caught in the blades or rolled-over trailing edges in the first reheat nozzle. The fix would likely involve a weld repair of the damaged areas.

Flow functions are also useful in interpreting test data. The flow function is derived from the general flow equation and is defined as the CAβ portion. It is the portion of the general flow equation that is supposed to remain constant if there is no change in downstream resistance, in other words, if there is no damage. The flow function is as follows:

$$FF = CA\beta = W\sqrt{\frac{v}{P}}$$

where

FF is flow function

w is the flow rate

v is the specific volume

P is the pressure

The flow function can be calculated at the HP turbine inlet, the IP turbine inlet, the LP turbine inlet, or at any location where the pressure, temperature, and flow are known. One would expect the flow function to remain constant over time; therefore, the units are not as important as the percentage of change over time. If the HP flow function gradually increased over time, one might expect the area of the first-stage nozzle to have increased such as what would exist with SPE. In planning for the outage, a new nozzle with different geometry might be considered, or the existing nozzle might be weld repaired and then sprayed with an erosion-resistant coating.

Casing efficiencies are, of course, very useful in anticipating whether there might be deterioration in a specific casing. The reconciliation process helps to determine whether the casing in question is contributing to most of the heat-rate deterioration. Whether a casing efficiency deteriorates suddenly or gradually will give a clue as to the type of damage. SPE and deposits occur gradually, whereas mechanical and foreign object damage happens suddenly. Steam seal rubs may occur shortly after a startup and level off, showing the same sort of trend in the casing efficiency.

Monitoring flow into the turbine can be useful in troubleshooting turbine problems. The flow may increase from an opening up of the stages in the front end of the HP. It is not only the first or control stage that affects the flow; it is also the area of the stages downstream of the first stage. Any damage that causes the first-stage shell pressure to change will change the pressure ratio across the first stage and therefore the flow, as is evident in reviewing the general flow equation, or the flow may decrease from debris or deposits blocking the front end of the HP. The change in flow may come gradually (deposits or SPE), or it may change abruptly (foreign object damage or mechanical damage). No matter how flow changes, it is usually a result of a change in area or the condition of the stages in the front end of the HP.

Diagnose Performance from Testing/Operating Data

After collecting a good set of data from calibrated instrumentation and after verifying, trending, and reconciling them, the analysis can begin. Certain parameters can help in seeing problems quickly.

Pressure Ratios

Turbine stage pressures include the first-stage shell pressure to the L-0 inlet pressure. The main steam pressure and turbine exhaust pressure are set by the plant or by ambient conditions; they are not a function of the turbine's physical condition as the stage pressures are. One would expect pressure ratios between stages to remain constant over the load range. If they are not, they will change according to the magnitude and location of the damage. Remembering that pressure is a function of flow and downstream resistance, if a pressure ratio increases, it may mean that there is a blockage in the stages, or it may be a decrease in pressure at the downstream location from an increase in an area further downstream. If a pressure ratio decreases, it may indicate that the area of the section has increased, or it may be an increase in pressure at the downstream location.

A pressure ratio across the control stage is a check on the valve position. For several tests conducted together, the pressure ratio across the control stage serves as an indicator for valve position. Ideally, the valves do not change position, and the pressure ratio across them remains constant.

Flow Function

The flow function described earlier gives an indication of changes in downstream resistance. Like the pressure ratio, it will respond to changes in area or condition of the stages. Normally, the flow function should stay at its design value, but if deposits or erosion occurs in the steam path, the flow function will change gradually. Conversely, it will change abruptly with foreign object damage or mechanical damage. The flow function does not mean too much by itself, and its units are not particularly descriptive. The value of using the flow function comes when trending it over time using the percentage of change from the baseline on the Y axis and time on the X axis.

Casing Efficiency

When damage occurs in a casing, the casing efficiency usually decreases, but not always. There are two common examples in which casing efficiency does not decrease. The first is the end-packing leakage from an HP turbine or from a separate span IP turbine (where it is not in an opposed-flow configuration with the HP turbine). When excessive steam leaks out of the end packings or glands, the casing enthalpy decrease in efficiency is not affected. The casing may produce less power because less flow is going through it, but the flow that is going through it is doing so at the efficiency before the leak. Consider the diagrams in Figure 2-21. The measured efficiency is a function of the condition of the stages **and** the condition of the steam in the piping where the instrumentation is installed. The turbine on the left side of Figure 2-21 has the HP leakoff going somewhere else in the cycle, likely a heater extraction. The excess leakage at the HP packing does not affect the HP turbine exhaust. Therefore, the efficiency measured from the turbine inlet to the location shown by the temperature gauge does not change when there is an HP packing leak.

Now consider the middle diagram in Figure 2-21 where the HP packing leakoff steam returns to the HP exhaust line. It returns after the exhaust temperature is taken. Again, this leak will not affect the measured efficiency.

Finally, consider the diagram on the right of Figure 2-21. The excess leakage returns to the HP turbine exhaust line before the exhaust temperature is measured. This configuration is more likely than the middle diagram because the leakoff is often through a pipe located adjacent to the underside of the casing, enclosed within the turbine insulation. The temperature at the exhaust now reads higher than the previous two scenarios because the high-temperature leakoff steam is mixing with the exhaust steam. Consequently, the measured HP efficiency will be lower. The lower efficiency can be deceiving because the HP stages have not been affected by this leak.

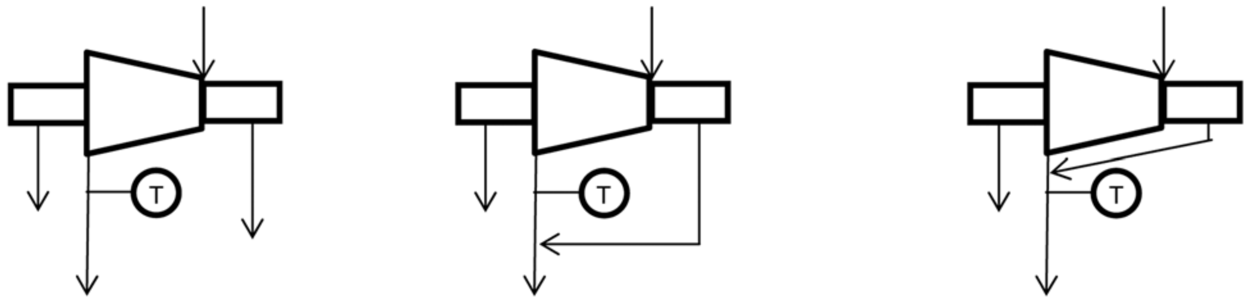

Figure 2-20
Temperature measurement location affects measured casing efficiency

The situation shown by Figure 2-21 on the right can be deceiving or it can be a useful diagnostic tool. If, for example, the measured HP turbine efficiency was holding steady for some time, and then there was a vibration event at the bearing adjacent to the HP packing that caused rubbing in the packing, the amount of excess leakage can be calculated from the change in measured HP turbine efficiency. Using the steam seal diagrams and Martin's formula, the seal clearance necessary to pass that additional flow can be calculated, which would help determine whether the packings should be sharpened or replaced.

Flows

Unexplained changes in HP turbine inlet flow are usually a result of area changes in the front end of the HP turbine. Gradual changes are usually caused by deposits or erosion, which occur gradually over time. Abrupt changes are usually caused by mechanical damage, such as broken blades and covers and foreign object damage.

Flow changes in other parts of the turbine cycle can be useful in diagnosing performance problems. Often these changes manifest as leakage bypassing turbine stages or entire turbine casings. If so, the pressure–flow relationship results in stage pressures decreasing proportionally with the amount of the decrease in stage flow. If 5% of the hot reheat flow bypassed the IP turbine and reentered the steam path at the crossover, then the pressures into the IP turbine and the mid-IP turbine heater extraction would decrease by 5%.

Key Pressures

Recognizing that pressure is a function of both flow to the following stage and downstream resistance, the key pressures can be extremely useful in diagnosing turbine performance. If flow increases, pressure will increase; if flow decreases, pressure will decrease; if downstream resistance increases, pressure will increase; and if downstream resistance decreases, pressure will decrease. The challenging diagnosis comes when **both** flow and downstream resistance change. Figure 2-22 shows the two scenarios.

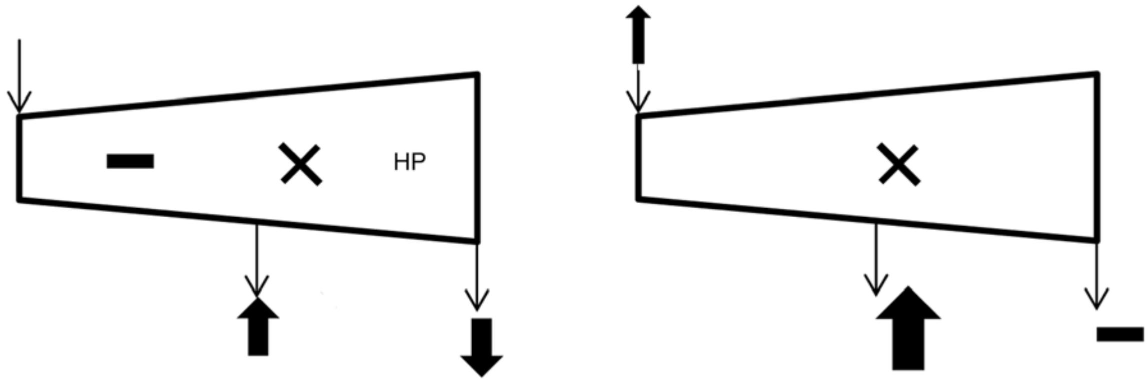

Figure 2-21
Pressures responding to changes in flow and downstream resistance

Consider first the diagram on the right side of Figure 2-22. It is the IP turbine and changes in area and pressure are considered too far downstream from the HP turbine control stage to affect the HP turbine inlet flow (and therefore the IP turbine flow). Assume a blockage occurred in the stage immediately downstream from the mid-IP turbine heater extraction point. The heater extraction pressure would increase in response to the increase in downstream resistance. The IP turbine inlet pressure would increase as well because the blockage is downstream of it also. However, the magnitude of the pressure increase at the inlet would be less than that at the extraction point because the stages between the two points absorb some of the change in pressure. The IP turbine exhaust pressure remains unchanged because its flow and downstream resistance remains unchanged.

Now consider the same situation in the HP turbine. The blockage immediately downstream of the mid-HP extraction increases the extraction pressure. This increase in pressure feeds back to the first-stage shell location. The magnitude of the first-stage shell pressure increase is less than that at the extraction, but it changes the pressure ratio across the first stage nevertheless. According to the general flow equation, when the control-stage pressure ratio decreases, the flow into the turbine decreases. The stage pressures respond to the decrease in flow by decreasing. Therefore, the pressure at the extraction increases because of the downstream blockage, and it decreases because of the decrease in flow. The result is still an increase in pressure, but smaller than what would occur if flow did not change. The first-stage shell pressure likely remains unchanged because the two opposing pressure changes would likely be equal in magnitude. The HP turbine exhaust pressure decreases by the amount that the flow decreases.

Specialized Testing

In preparing for what to expect during an outage, the results of specialized tests can increase the likelihood that there will be few expensive or time-consuming surprises when the turbine is open. HP turbine efficiency over the load range may confirm or refute the presence of control-stage SPE on a partial-arc, sequential valve machine.

Water washes may identify a deposit problem. A water wash can be as simple as a cold startup. Otherwise, there are operation techniques available to allow wet steam to form in the turbine shortly after the control stage. Immediately before and after the water wash, the enthalpy decrease efficiency is measured across the casing or casings suspected to contain water-soluble deposits. If there is a step improvement in efficiency after water washing, then that is confirmation that water-soluble deposits exist. The water wash, however, is not a long-term fix. It is merely an indicator of water-soluble deposits. The deposits will return if their source is not found and eradicated.

Specialized tests for leaks include the HP-IP internal leakage, discussed in the paper by Booth and Kautzmann. Other leaks include bypass flow that ends up in the condenser that can be quantified by knowing the leak enthalpy and pipe diameter and measuring the pressure into the condenser. Still other leaks can be detected with changes in turbine stage pressures and by walking the plant using thermography leak detection equipment.

Turbine inlet valves can have problems; for example, valve stems can break or stick and disks and seats can erode. Valve stroke tests and plotting power over the load range can help determine whether a valve is malfunctioning. In an eight-valve turbine, one would expect roughly one eighth of the power with one valve open, one fourth of the power with two valves open, and so on. If a valve is malfunctioning, the power over the load range will have discontinuities in it.

Study Cross Section for Potential Leaks

The cross section or longitudinal drawing can be useful in anticipating where leaks other than those through sealing devices might occur in the steam path. Common leakage paths include the following:

- Horizontal joints between upper and lower blade ring or diaphragm halves
- Horizontal joints between upper and lower inner casings
- Where packing boxes fit into casings
- Axial clearances where blade rings or diaphragms fit into casings
- Snout rings or bell seals

When the outage begins, it is helpful to be present as soon as the suspected leak is exposed. Otherwise, the disassembly process can eliminate telltale signs of a leak. For example, the shell horizontal joint is often an area where deposits form trails along the leakage paths. Once the top half is lifted during an outage, workers step on the bottom half horizontal joint to gain access to the turbine for the remainder of the disassembly process. In doing so, they inadvertently remove deposits that show the source and destination of the leak.

List of Isolation Breaches

As the outage approaches, the steam turbine performance engineer should develop a list of isolation breaches discovered during previous testing, routine monitoring, and plant walkdowns. The process of eliminating the breach should be determined with input from the I&C and maintenance departments. Some fixes might involve repairing or replacing valves. Others might include replacing steam traps or replacing feedpump seals. The P&IDs are useful in this process and should be updated if any changes are made.

The turbine outage can be an ideal time to eliminate isolation problems, but time is usually in short supply during the outage. A heat-balance program can be used to determine the effect each leak has on turbine cycle performance. In doing so, the relative importance of fixing the leak can be ascertained so the outage to do list can be prioritized.

List of Instrumentation Problems

Like the isolation problems, the instrumentation problems can be fixed during the outage, and like the isolation problems, if the instrumentation problems are prioritized is such a way to address those that have the greatest effect on performance first, then the outage can offer a very productive opportunity to improve the overall quality of future testing and monitoring. The I&C department will be involved, and the P&IDs should be updated.

In some cases, instrumentation can be installed where it was never installed or is missing. Some measurement points may warrant installation of redundant instrumentation. Other instrumentation may simply be out of calibration. Some instrumentation, such as final feedwater nozzles, must be shipped off-site for calibration. In all cases, an instrumentation list should be prioritized to get the highest quality test within the time and budget confines of the outage.

Line Up Contractors/Suppliers

Anticipating the types of steam path degradation through pre-outage analysis of test data allows the steam turbine performance engineer to contract with maintenance contractors and suppliers of turbine parts. A typical open/clean/close turbine overhaul will undoubtedly involve some minor weld repairs to restore dinged blades, but it may not include major weld repairs. Major weld repairs may involve mobilizing on-site equipment and personnel or shipping the damaged parts off-site to a repair shop. Both options might come at a premium, particularly if it is “outage season.” Many overhauls are conducted during the low heating and cooling periods of the calendar year such as early fall and late spring.

The same is true with suppliers of turbine parts. Their lead time to deliver parts can often be very short, but if they have a backlog during outage season it is difficult to fill last-minute orders in a timely fashion.

When the outage schedule is known, it is a good time to line up a steam path audit. If the steam path audit is to be conducted with in-house personnel, they must be given adequate time to model the unit, order any tools that they might be lacking, and in general be a dedicated participant for the steam path audit. Having other responsibilities during the disassembly and reassembly processes will compromise the quality of both the steam path audit and the other responsibilities.

If, on the other hand, the steam path audit is to be subcontracted out, give the subcontractor as much notice of the outage as is practical. Subcontractors realize that the schedule may vary somewhat, but at least they can anticipate their upcoming workloads and have ready their necessary personnel and tools. The engineer conducting the steam path audit must be on-site when the steam path is exposed (that is, when the turbine is opened and the inner casing removed, exposing the rotating blades). They must stay at least one day after the rotor or spindle is removed from the casing.

Grit blasters can be lined up ahead of time because the steam path must be cleaned up before any nondestructive evaluation (NDE) work, but ordering the most appropriate type of grit blasting material is something that can be done in advance. Most grit blasting is done with 220 aluminum oxide grit. It is coarse enough to remove stubborn deposits, yet smooth enough to leave an acceptable finish. However, if the steam path is relatively new or has a history of excellent water chemistry control, there may not be deposits in the steam path that require a coarse grit. A finer grit, such as 300 aluminum oxide, will leave a smoother finish. The goal is to choose the finest grit that will remove whatever deposits formed on the blading in a reasonable amount of time.

If the unit can go back on-line with reasonable assurance that deposits will not be a problem in the future, then polishing the blading may be warranted. The blades and specific areas to polish are shown in Figure 2-23. In general, only HP turbine and possibly IP turbine blades benefit from polishing. The portion with the highest steam velocity is polished because friction losses are proportional to the velocity squared. Therefore, the convex side on the exhaust side is polished. Stationary blades in an impulse design and both stationary and rotating blades in a reaction design benefit from polishing.

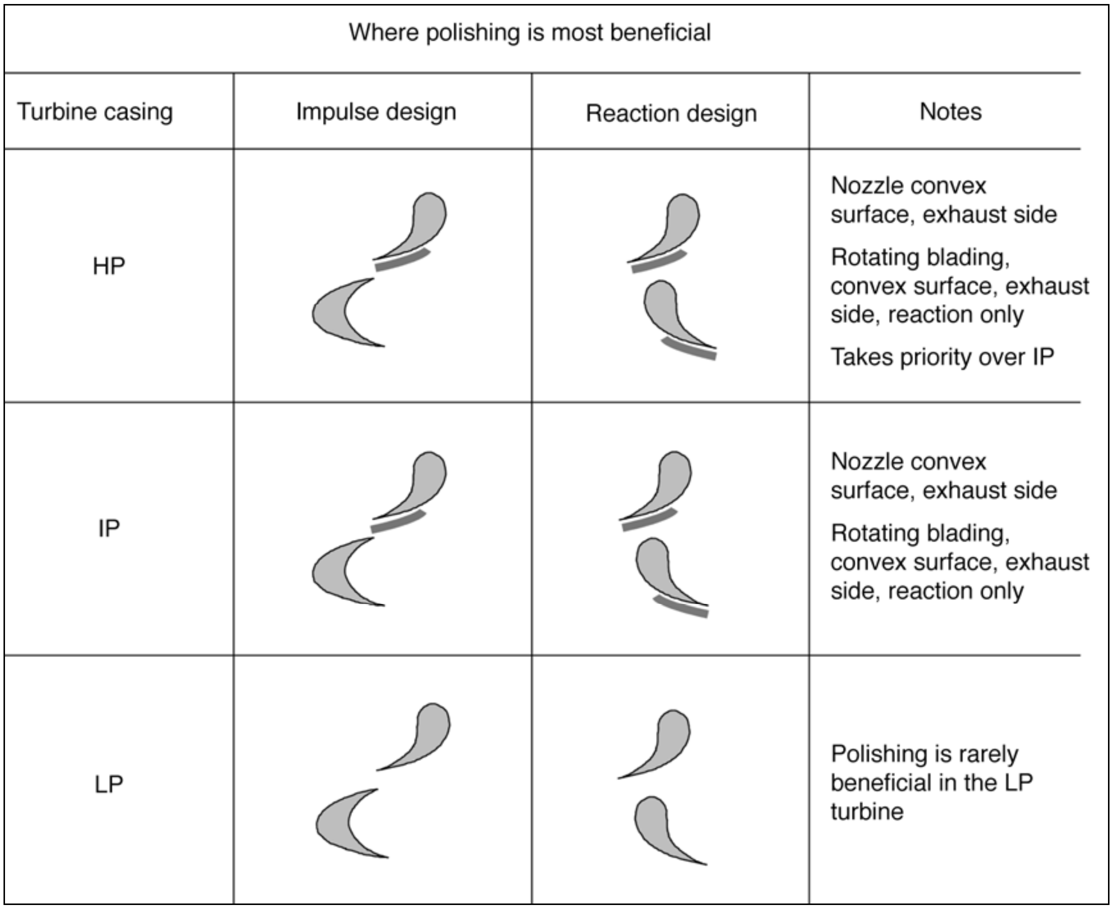

Figure 2-22
Areas where blade polishing is most beneficial

Outage Schedule

The steam turbine is often on the critical path of the outage schedule. A tremendous amount of preparation goes into a well-planned outage. Time is a precious commodity during the outage, and success or failure is often measured in terms of effect on the original outage schedule. The activities of a steam turbine performance engineer are often not on the critical path, but rather mesh with critical path activities.

Figure 2-24 shows a typical high-level Ghant chart for an open/clean/close turbine overhaul. More detailed outage Ghant charts can be 20 pages or more. Superimposed on the high-level Ghant chart in Figure 2-24 are opening and closing steam path audits that occur simultaneously with the disassembly and reassembly of the turbine casings, respectively. Depending on the number of casings and rows of blading, the opening steam path audit may take five to eight days. The closing audit, during reassembly, typically takes three to five days. The closing steam path audit is shorter than the opening steam path audit because steam path geometry is measured during the opening and does not change for the closing, and the surfaces are uniform after grit blasting for faster surface roughness evaluation. The chronology of a steam path audit is detailed in Appendix C.

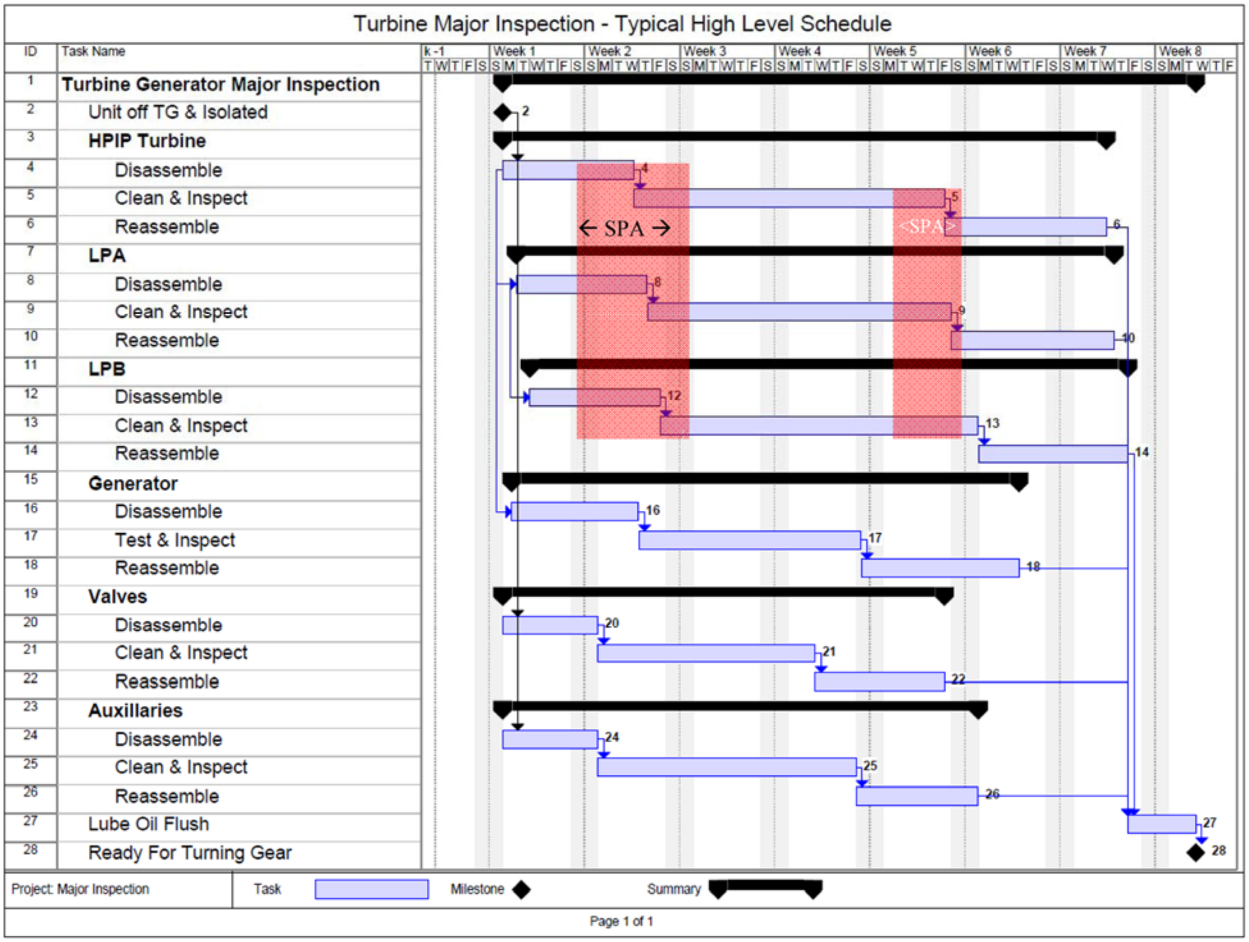

Figure 2-23
Typical high-level turbine overhaul Ghant chart

The deliverable of an opening steam path audit is a list of maintenance items to perform during the outage to improve steam path efficiency. Each maintenance item shows the expected heat-rate improvement associated with that action. They are prioritized so the turbine cycle can gain the greatest heat-rate return on the maintenance investment.

The gap in time between the opening and closing steam path audits provides an ideal time to focus on other performance activities such as fixing isolation problems and installing and calibrating instrumentation.

Reverse Engineering Data-Gathering Opportunities

The outage provides an opportunity for reverse engineering. Steam path upgrades come in a variety of combinations. An entire outer casing with new contents may be replacing an older casing; a new rotor may be placed in an existing casing; a new set of stationary blades may go into an existing outer shell, and so on. In all these cases, on-site measurements are usually needed by the supplier of the new parts. If the new supplier is the same as the OEM, then design drawings with dimensions might be available for the supplier at their offices. However, over time the turbine can warp and go out-of-round. If these conditions are not to be fixed before the planned upgrade, then the upgrade must take the deviation from design dimensions into consideration.

Other times a supplier is chosen who is not the OEM; therefore, design dimensions are not available in drawings. The supplier must make the necessary measurements on-site during an outage. This extra step can put the non-OEM supplier at a disadvantage because of time constraints, which can limit the owner's options when it comes to upgrade suppliers. When planning for upgrades several years in advance, the owner can invite potential upgrade bidders to take measurements during an outage that precedes the outage in which the upgrade is to take place.

Why Consider an Upgrade or Modification?

The current fleet of steam turbines is aging while technology and heat rate continue to advance. When a 30- or 40-year-old turbine is nearing its end of life, options include replacing the turbine with an upgrade, redesigning the plant to, for example, a combined cycle, and, finally, decommissioning the plant.

For steam turbines that are not close to their end of life but contain obsolete technology, their life can be extended and their performance improved by considering upgrading certain components.

Low Efficiency

Turbines built with older technology have less aerodynamic blades than those built today. Their materials were less resistant to solid particle and water erosion. Their construction was limited by the metallurgy of the day. Not as much was known about seal leakage, erosion mechanisms, secondary flows, and three-dimensional two-phase flow as is known today. Top HP turbine efficiencies in the 1960s and 1970s were in the low 80% range, whereas now they are in the 90% range. LP turbine efficiencies have increased significantly as well, especially with more efficient last-stage blades.

Last-Stage Blades

A turbine's last stage produces approximately 10% of the power of the entire turbine. Therefore, an efficiency improvement in the last stage has the greatest effect on heat rate and capacity of all individual stages. Because of its long length, large mass, and supercritical pressure decrease, the last stage presents many design challenges. Older technology last-stage blades can have multiple tie wires for damping, metallurgy that is susceptible to water droplet erosion, lack of rotating blade covers, and shapes that are aerodynamically substandard compared with today's technology.

Excessive Pressure Decreases

For every 1% pressure decrease in the steam path, there is approximately a 0.1% decrease in performance [3]. When steam gives up potential energy across a valve or length of pipe, no power is generated. Pressure decreases across partially throttling valves, bulky casing exhausts, through nonstreamline crossover piping, and boxy LP turbine exhaust hoods all represent losses in energy and efficiency.

Double-Flow and Curtis Control Stages

There are several ways to design control stages to minimize the large forces on the blading from low-load operation. In a partial-arc design, sequential valve admission scheme, the control-stage bowl pressure can be 1000 psi, whereas the shell pressure might be 10 psi at startup, producing tremendous forces on the blading. There is an inverse relationship between the structural reliability of a stage and its aerodynamic efficiency: the stronger and bulkier a stage, the less efficient it will be. From an efficiency perspective, the ideal nozzle trailing edge is razor thin, but it would never hold up to the large forces normally found in the control stage.

Older turbines were designed with Curtis stages and double-flow control stages to reduce the forces on the rows of blading (Figure 2-25). A Curtis stage has two sequential rows of blading that break down pressure decrease across the blades. A double-flow control stage has two parallel rows of blading, each passing one half of the stage flow so the blade height can be half that of a single-flow stage.

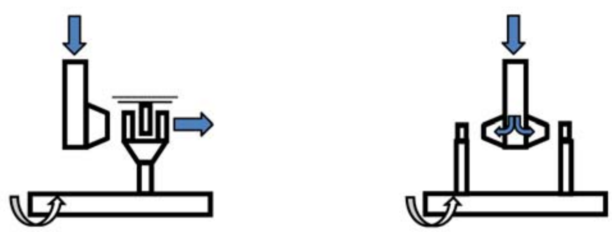

Figure 2-24
Two-row and double-flow control stages

The shorter blades in the double-flow design have greater end-wall losses than the longer blades in the single-flow design.

Exhaust Hood Geometry

The LP turbine exhaust hood carries the steam from the last-stage rotating blades to the condenser neck. The flow is two phase and has an extremely complicated three-dimensional flow field. The exhaust hood’s geometry is boxy with struts, expansion joints, seal system piping, and bearing cones obstructing the flow. The steam must turn, in most cases, 90° before traveling down the condenser neck to the condenser tubes. It is a very arduous path to take after passing through a cascade of aerodynamic blades.

Aerodynamic Shapes of Blades

Throughout the decades, advances have been made in the aerodynamic shapes of both stationary and rotating blading. These advances can be attributed to CFD, which allows designers to make use of the Navier-Stokes equations, and improvements in numerically controlled manufacturing processes. Older style blades have more losses associated with them because of high entrance and exit coefficients and nonstreamlined flow passages (Figure 2-26).

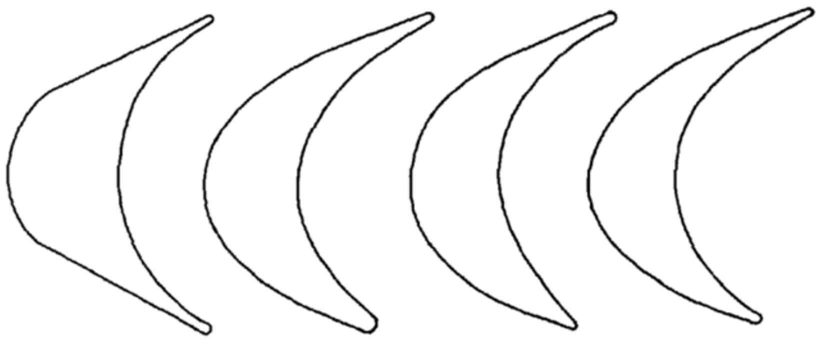

Figure 2-25
Blade profiles

Known Reliability Problems

Despite the best intentions of the OEMs, some problems surface after a turbine of a particular design has been operating for a while. These problems might include a susceptibility toward erosion, valve flutter problems, rotating blade covers lifting, bolts loosening, rotors bowing, thermal distortion, or any number of problems that affect an entire fleet of turbines.

The General Electric Company issues technical information letters (TILs) to notify owners about know problems. Siemens-Westinghouse issues technical bulletins. Both usually contain the root cause of the problem, recommendations for fixing the problems, and operation restrictions that may need to be put in place for reliable operation until a fix.

Although the TILs and technical bulletins address reliability issues, some may address performance as well. Part of the preparation process for an upcoming outage should include review of the relevant reliability information from the OEM. For example, if information from the OEM states that the HI/IP turbine rotor tends to vibrate excessively during startup, then consideration might be given to installing HP packing rings designed to withstand that sort of vibration or perhaps antiswirl packing.

Understanding the Performance Effect of Upgrades and Modifications

Many OEMs and suppliers of turbine parts offer upgrades and modifications. Steam turbine performance engineers are often asked to evaluate differing bids for improving performance. This section gives explanations for why some of the upgrades and modifications improve performance.

Twisted Blades

Twisted blades improve stage efficiency because they allow better matching of the steam inlet angle to the angle of the rotating blade. When there is a mismatch in angles, there is increased turbulence at the rotating blade inlet. The steam gives up its energy to turbulence instead of producing power.

Velocity diagrams (Figure 2-27) at the root and tip of the blade demonstrate how the relative velocity of the steam angle changes in response to the changing tangential velocity of the rotating blade, which is a function of distance from the rotor centerline.

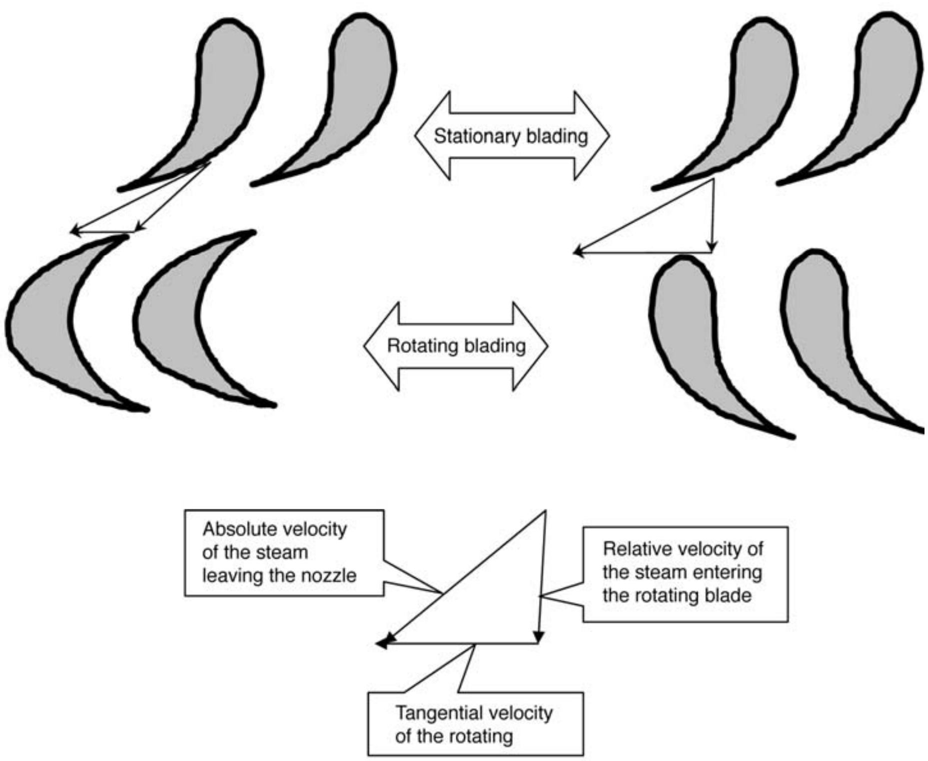

Figure 2-26
Velocity diagrams of twisted blades

Notice how the relative velocity vector angle changes from the root to the tip. Also note how the inlet of the twisted rotating blades is angled such that it matches the incoming steam vector angle. Without twisted blades, the leading edge of the rotating blade would be a constant angle from root to tip and the steam angle would match at only one infinitesimally small point, usually at the pitch line. This mismatch, common in older designs, creates turbulence at the rotating blade inlet. Newer twisted blades reduce the turbulence and increase efficiency.

Additional Stages

Turbine manufacturers are offering steam paths with additional stages because they have a higher efficiency than fewer stages expanding steam through the same available energy range. The Mollier diagrams in Figure 2-28 demonstrate the effect of adding a stage to an expansion.

On the left side of Figure 2-28 is a Mollier diagram showing an expansion through one stage. The available energy is h1-hi. The UE is h1-h2. By definition, enthalpy decrease efficiency is the ratio of UE to available energy.

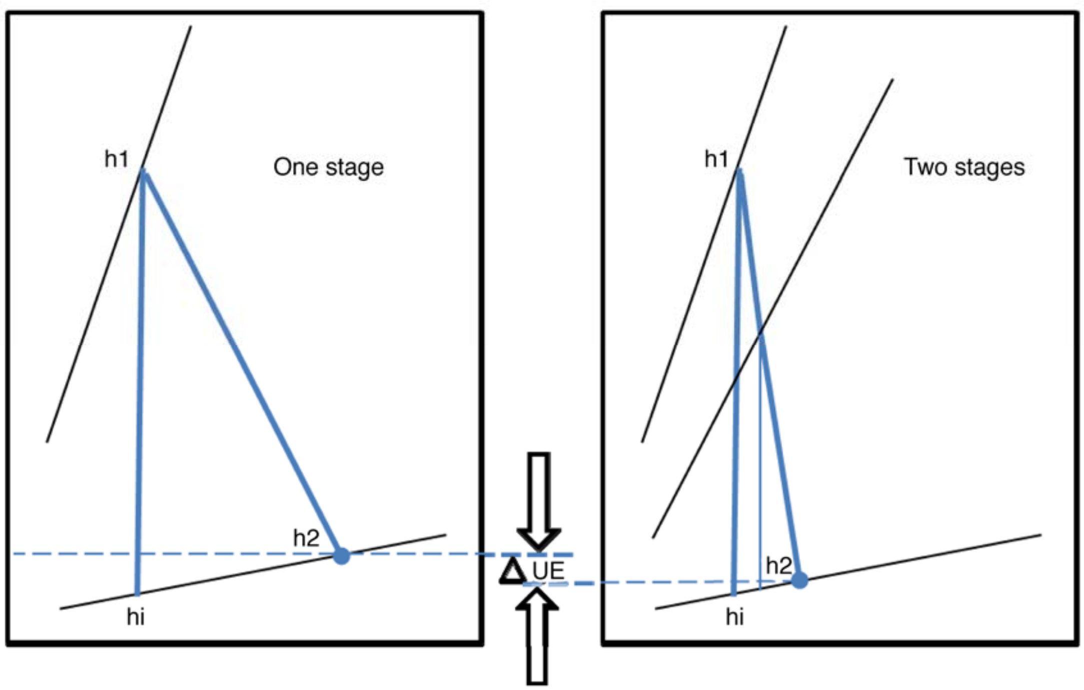

Figure 2-27
Mollier diagram of one-stage and two-stage expansions

Now consider the right side of Figure 2-28. It is an expansion through the same available energy range with two stages, both with the same efficiency as the single stage on the left side of Figure 2-28. An interim isobar (line of constant pressure) is drawn between the original two to represent the inlet pressure to the second stage. The isobars diverge in going from left to right, and the exhaust pressure of the first stage has a much steeper slope in the two-stage expansion than the isobar at the exit of the one-stage expansion. The first stage expansion line in the two-stage expansion is much closer to vertical than in the original expansion.

The exit of stage 1 marks the beginning of the expansion through stage 2. If expanded at the same efficiency as that of the first stage (and the single stage in the one-stage example), the endpoint will fall on the exhaust pressure isobar at an enthalpy lower than the one-stage expansion. The available energy across the entire energy range has not changed, but the UE has increased in the two-stage expansion. Because efficiency is UE/AE, the expansion with the greater number of stages has the higher efficiency.

Double-Flow to Single-Flow Control Stages

Figure 2-25 shows a double-flow first stage. The double-flow design allows the blade heights to be half of what they would be in a single-flow design. The stresses on the blades are less than a single-flow design. Steam path upgrades often include a switch from a double-flow first stage to a single-flow first stage. The benefits in making this change are threefold.

First, the pressure decrease between the first and second stages is reduced. In the double-flow design, the flow traveling away from the HP turbine exhaust had to turn 180° to reunite with the other half of the first-stage flow. The turn results in a pressure decrease from which no useful work results. For every 1% pressure decrease in the steam path, there is a 0.1% heat-rate penalty [3]. A single-flow first stage does not have this extra pressure decrease.

Second, the end-wall losses are less with a single-flow design because the blades are longer. The newly designed blades have been upgraded to withstand the forces of a single-flow first stage. An end-wall loss exists because of the geometry and surface roughness at the root and tip of the blade where the blade is held in place. The turbulence at the root and pitch extend a relatively small distance radially at the root and tip no matter what the blade length. On a shorter blade, the end-wall loss zone makes up a greater portion of the blade height. As blades become longer, the percentage of their height affected by end-wall losses decreases and the blade efficiency increases.

Third, the elimination of the double-flow first stage allows more room in the casing for an extra stage downstream without affecting the size of the outer casing. Reusing the outer casing in a steam path upgrade can mean considerable savings. The extra stage in the HP turbine increases efficiency (see the Additional Stages section on the previous page).

Supersonic Tips on Last-Stage Rotating Blades

The last stage has a pressure decrease across it that exceeds the critical pressure ratio. The pressure upstream of the last stage is a function of flow and the pressure downstream is condenser pressure, which is largely a function of the condenser's circulating water cooling method. At full load, the pressure ratio across the last stage may be 6:1 or more, where the critical pressure ratio is roughly 2:1.

The reaction in a stage varies from the root to the tip of the blading. Whether the stage design is reaction or impulse, the reaction at the tip of the last stage is approximately 70%. In other words, 70% of the stage energy is across the rotating blade. Stage energy can be defined using pressure. Therefore, if the pressure upstream of the last stage is 6 psia and the condenser pressure is 1 psia, the pressure decrease across the stage is 5 psia and the pressure decrease across the tip of the last-stage rotating blade is 70% of 5, or 3.5 psi. Because the critical pressure ratio is 2:1 and the pressure ratio across the tip of the last-stage blade is 3.5:1, the tip of the last-stage blade has sonic velocity steam passing through it.

A converging nozzle can accelerate steam up to sonic velocity where a shock wave forms at the nozzle exit, which acts as a blockade. The shock wave causes turbulence and reduces stage efficiency. A converging/diverging nozzle can accelerate steam up through sonic velocity and into the supersonic velocity range without the shock wave as the steam goes from subsonic to supersonic. Instead, after the steam leaves the blading and decelerates, shock waves form further downstream of the blades than those formed with a converging nozzle. The decrease in turbulence with a supersonic tip increases stage efficiency.

Eliminated Tie Wires

Tie wires are used in the latter stages in the LP turbine to aid in damping. However, from an aerodynamic perspective, they serve as bars in cross-flow—an obstruction to flow that causes turbulence and inefficiency. Some last-stage blades have three tie wires. Some L-1 and L-2 blades have tie wires. Some newer stages have eliminated the need for some or all the tie wires by addressing blade damping needs through the rotating blade cover treatments.

Increased Set Back on the First Reheat Stage

SPE in the first reheat stage is a common problem. The IP turbine erosion mechanism is different from the HP turbine erosion mechanism. In the HP turbine, high-velocity particles impinge on the concave side of the nozzle at the exhaust (Figure 2-29, left). The solid particles entering the first stage of the IP turbine move much more slowly than those in the HP turbine because the stage pressure ratio is lower. The pressures on either side of the first reheat stage are like all the other stage pressures: they are a function of flow and downstream resistance. The pressure upstream of the control stage, on the other hand, is controlled. Therefore, although the control-stage pressure ratio can approach very high levels at low load, the pressure ratio across the first reheat stage remains at approximately 1.3 over the load range.

The erosion in the first reheat stage occurs from the slowly moving particles (relative to the steam and to those particles passing through the control stage) passing through the nozzles into the space between the exit of the stationary nozzles and entrance of the rotating blading. Here, the tangential speed of the rotating blading exceeds that of the particles. The impact of the rotating blade hitting the particles sends the particles upstream, against the direction of flow, until they come into contact with and erode the nozzle trailing edges (Figure 2-29, center).

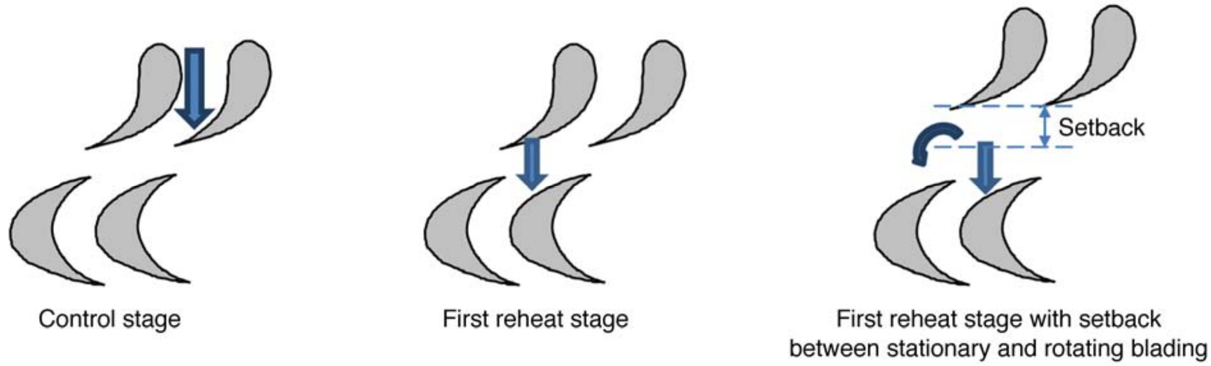

Figure 2-28
SPE in the control stage and first reheat stage

To reduce the force with which the particles impact the IP turbine nozzles, these nozzles are moved upstream to increase the distance that the particles must travel to hit the nozzles. The further away the nozzles are, the more apt the particles are to be overcome by the steam flow. Before the opportunity to damage the nozzles, the particles turn in the direction of steam flow and continue downstream. They may bounce around between the stationary and rotating blading, but eventually the centrifugal force sends them out radially where they do little damage.

Contoured End Walls in the Control Stage

End walls in the control-stage stationary components can offer incoming solid particles a streamline path to the vulnerable nozzle trailing edges. To reduce the speed of the particles, newer designs have contoured end walls such that the solid particles encounter more turns and friction, which slow the particles down.

Seal Designs

There are many options for upgrading steam seals. A good source of information is the work of Cotton [3]. The primary considerations in choosing a seal design are its performance during high-vibration events and its ability to maintain sharp teeth and a small clearance.

Some seal designs address the high vibration that normally occurs at startup and shutdown when the turbine rotor goes through its critical speed. The critical speed is the speed corresponding to the resonant frequency of the rotor. There is often more than one speed between 0 Hz and 60 Hz. When a rotor's rotational speed hits its resonance frequency, any vibration stimulus is accentuated. So, if there is some minor rotor imbalance, it can become quite extensive during a critical.

Another phenomenon of rotor dynamics is the movement of the rotor when subjected to a rub. When a rotor is less than its critical speed, the local heating caused by the rub will cause the rotor to move into the rub. Conversely, when the rotor is higher than its critical speed, the local heating from a rub will cause the rotor to move away from the rub. Therefore, a rub at low speeds can continue to intensify and wipe out entire seals. It is quite unfortunate and costly to install new seals during an outage only to have them wiped out during the first startup.

Conventional Packings

Conventional packings along the shaft (diaphragm packings, shaft packings, end packings, interstage packings) maintain a clearance between the rotor and the packing teeth using a spring that pushes the packing in radially toward the rotor. When a rotor vibrates excessively, the conventional springs give somewhat, but often not before the packing teeth are rubbed.

Retractable Packings

Retractable packings have a different spring configuration. Instead of radial springs pushing the packing rings in toward the rotor, they have tangential springs between ring segments that hold the packing rings away from the rotor. They also have a path for steam to flow behind the packing ring that allows the pressure of the steam to push the packing ring down toward the rotor. When a steam turbine starts up, the packing teeth are held away from the rotor. As the steam pressure increases in the turbine, the pressure behind the packing rings eventually overcomes the spring force and the sliding friction force to close the packing rings. The rings are designed to close after the critical speeds have occurred, when the rotor is at decreased risk of vibrating excessively.

Guardian Packings

Guardian[3] packings are designed to reduce the damage caused by high-vibration events and to preserve the integrity of the packing teeth. A Guardian packing has several conventional teeth and usually two Guardian posts. The posts are roughly 5 mils longer than the other teeth, they have a square tip rather than one that comes to a near-point and are made of a different material from that of the other teeth. The height of the Guardian teeth ensures that during a rub, the rotor would come in contact with the Guardian posts before the conventional teeth, thereby preserving the integrity of the conventional teeth. The Guardian post material has heat transfer properties such that friction with the rotor produces less heat. During a rub, some of the Guardian post material adheres to the rotor so future rubs will occur between Guardian post material on both the packing and the rotor. The springs behind the packing rings are radial, like conventional packing, but their spring force is less, which allows the packing teeth to move out of the way during a rub.

Vortex Shedders

Vortex Shedders[4] are tip spill strips or shroud seals located at the outer diameter of the rotating blades. They are designed to decrease flow around the tips of the rotating blades primarily by reducing the tangential flow component (Figure 2-30). Most of the flow through the turbine

[3] Guardian is a registered trademark of Turbo Parts, LLC.
[4] Vortex Shedders is a registered trademark of Turbo Parts, LLC.

travels axially, parallel with the centerline of the rotor. However, some of the flow is subjected to the tangential movement of the rotating blades, so the resulting flow through the turbine has both axial and tangential, or circumferential, components to it. Vortex Shedders have discontinuities that create vortices that decrease the tangential flow.

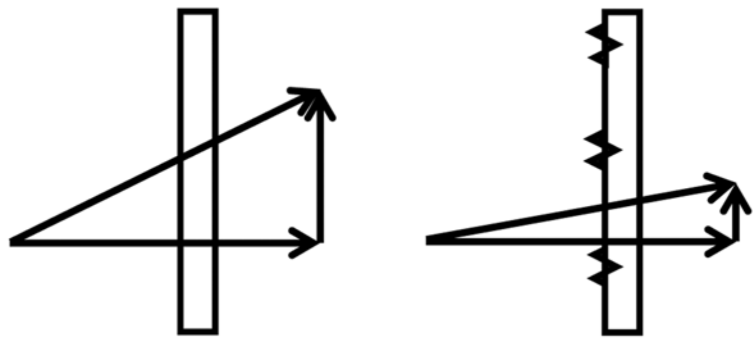

Figure 2-29
Steam velocity vectors through conventional seals and Vortex Shedders

The vortices also reduce pressure upstream of the seal, which reduces the pressure decrease across the seal. Think of potential energy (pressure) given up as turbulence (vortices). Because flow is a function of pressure ratio, the flow across the seal decreases.

Brush Seals

Brush seals can be used for shaft packings and tip seals. Shaft seals typically have one tooth out of the eight or so in a ring that is a brush seal. The brush has metal bristles angled in the direction of flow and maintains a small clearance with the rotor. The benefit of a brush seal comes from the small clearance and the lack of damage to the bristles during a rub. If the turbine has a history of SPE damage, however, the brush seals might capture the hard iron oxide particles in the bristles and machine the rotor surface during a rub.

LP Exhaust Hood Design and Size

The aerodynamics of an exhaust hood can affect the overall efficiency of the turbine cycle. The steam encounters a comparatively arduous path after leaving the last-stage rotating blade on its way to the condenser tubes. The losses between the last-stage blading and the condenser tubes are the total exhaust losses and consist of the three main types shown in Figure 2-31.

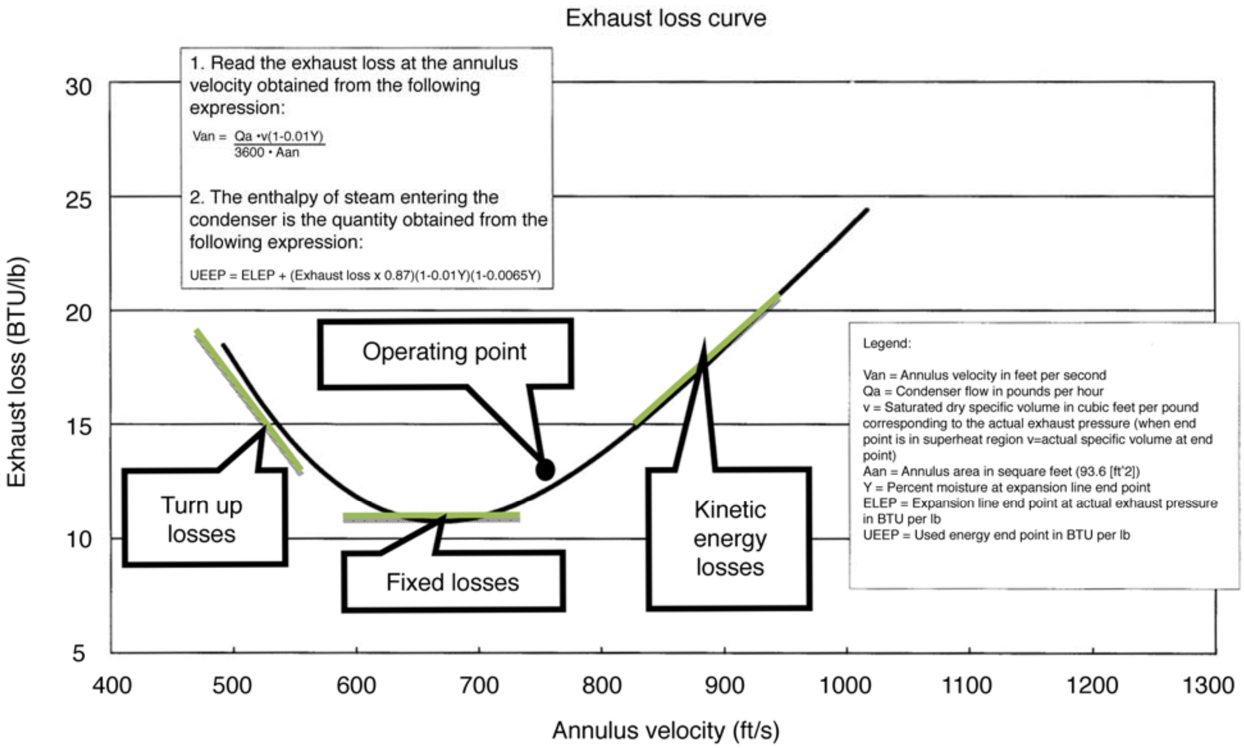

Figure 2-30
Total exhaust loss curve

The placement of the operating point allows the turbine to vary slightly around the rated load point while remaining on the right side of the total exhaust loss curve. The minimum point on the curve corresponds with the fixed losses, which are a function of the geometry of the exhaust hood. An exhaust hood with a smooth exhaust annulus, guide vanes, and few obstructions will have lower fixed losses than a hood with a nonstreamline exhaust annulus, no guide vanes, and struts in cross-flow immediately after the exhaust annulus. The turn up losses result from recirculation in the last stage when low-load operation causes the L-0 bowl pressure to fall lower than the exhaust pressure. It is here that the turbine blade acts like a pump, adding energy to the steam instead of extracting energy from it.

The right side of the total exhaust loss curve shows a slope that is proportional to annulus velocity squared. The losses result primarily from friction losses. At this point in the steam expansion, the specific volume of the steam is large, and the exhaust hood geometry offers many opportunities for turbulence and friction.

Annulus velocity is a function of annulus area, the mass flow, and the specific volume. Exhaust losses decrease with a decrease in annulus velocity in the operating point region. Therefore, increasing annulus area, decreasing mass flow per exhaust end, and decreasing specific volume all contribute to decreasing annulus velocity. Longer last-stage blades increase the annulus area because the annulus area is a function of the blade length and the pitch diameter. For a given turbine inlet flow, increasing the number of exhaust ends decreases the mass flow per end. However, going from a single-flow LP turbine to a double-flow LP turbine, for example, is an uncommon upgrade. The specific volume is a function of the exhaust pressure. As turbine exhaust pressure decreases, specific volume increases, which increases friction losses in the hood. However, the increase in exhaust losses is offset by the increase in power from a lower back pressure. Of the three contributors to reducing annulus velocity, increasing the length of the last-stage blade is the most beneficial.

Exhaust hood upgrades improve performance by offering fewer fixed losses and shallower turn-up and kinetic energy curve slopes. Often, newer LP turbines have a lower operating point, which can be a sustained benefit in a base-loaded unit that does not spend much time on the left side of the curve.

Increasing the Length of the Last-Stage Blade

As stated in the previous section, increasing the length of the last-stage blade reduces total exhaust loss by increasing the annulus area and decreasing the annulus velocity. Because exhaust losses include friction losses, and friction losses are a function of velocity squared, reducing the annulus velocity has a significant effect on total exhaust losses.

In addition to decreasing total exhaust losses, longer last-stage blades are more efficient than shorter ones because of smaller end-wall losses. However, the difference in end-wall loss between, for example, a 30-in. and a 33.5-in. last-stage blade is small. Larger improvements come from the other design features of newer last-stage blades such as elimination of one or more tie wires, addition of blade covers so a tip sealing device can be used, and the modification of the blade tip to accommodate supersonic flow.

Valve Position

The position of the turbine inlet valves will influence the amount of throttling of steam at the inlet of the HP turbine. When a valve is neither fully open nor fully closed, there is an excessive pressure decrease across it that robs the steam of the available energy that would otherwise be used across a turbine stage to produce power. If the turbine is forced by dispatchers or mechanical limitations to sit at a particular load point that involves throttling for extended periods of time, the performance may be improved by adjusting the valves such that the required load point corresponds to a valve point.

HP Turbine Stage Areas

Flow into the turbine is controlled by the front end of the HP turbine. The control-stage area and the first-stage shell pressure are very influential in the general flow equation. Often, when a plant has excess boiler capacity, it may be beneficial to increase the flow capacity of the turbine to generate more power. Sometimes this increase is accomplished by a new HP turbine steam path of larger capacity. Other times it is accomplished by increasing the existing areas through shop repairs. Either way, the following are several things to consider:

- The remainder of the plant will have increased duty. Balance-of-plant equipment from fuel handling to emissions to feedwater pumps and heaters to heat rejection will have a higher load.
- An increase in power rating may require significant upgrades to the emissions cleanup equipment.
- The turbine stages, particularly the last-stage blade, will be subjected to higher forces and will be at higher risk of deformation, cracking, and failure.
- The generator output will increase as will its cooling requirements.
- Increased flow increases the annulus velocity and the total exhaust losses. If the flow increases too much, the back end of the turbine might enter an undesirable choked flow condition.
- Replacing the HP turbine with a new, upgraded turbine will result in a better heat rate than opening the existing stage areas.

In some circumstances, heavy deposits in the HP turbine are a recurring problem. Although it is most desirable to eliminate the source of deposits, some owners have opted to live with the deposits, but open the turbine stage areas to regain the original flow capacity. In this situation, operating at a reduced load is necessary until the deposits build up enough to not exceed the maximum allowed flow.

Continuously Covered Blades

Older LP turbines often contain rows of rotating blades without covers. An upgrade to this type of blade is one that has a cover so that a sealing device can be installed at the tip of the rotating blade. Some OEMs offer continuously covered blades that also offer some damping characteristics. As such, often one or more of the tie wires can be eliminated, improving stage efficiency.

The last stage has approximately 70% reaction whether it is an impulse or reaction design. The last stage also has a very large pressure decrease across it and therefore a large energy range. That is the primary reason that the last stage produces approximately 10% of the power of the turbine. With the large energy range and the high reaction, there is a large pressure decrease across the tip of the rotating blade. Finally, the tip of the last-stage blade has a very large diameter. So, the potential for large tip leakage is great. Covered blades offer a landing for a sealing device at the tip of the last-stage blade.

Coatings

Coatings on blades can protect the base metal from damage caused by SPE and foreign object damage. They may be considered to extend the life of certain rows of blading between outages. Coatings do, however, increase the surface roughness of the blades. Therefore, weighing the benefits of erosion reduction against the cost of increased surface roughness must be part of the decision to upgrade with coatings.

Smaller Rotor Diameters

For the same blade lengths, a smaller rotor diameter will result in a smaller leakage area at both the root of the stationary blades and the tips of the rotating blades. A smaller diameter rotor with longer blades will result in smaller leakage area at the root of the stationary blades and lower end wall losses.

Typical Outage Repairs and Their Effect on Turbine Performance

A typical open/clean/close turbine overhaul includes some routine maintenance that also has an effect on performance. Below are some of the routine maintenance items.

Grit Blasting

Grit blasting the steam path is required before conducting NDEs for detecting cracks, inclusions, and other potential problems. The grit blasting operation typically occurs as soon as the steam path parts are removed from the casing. If a steam path audit is conducted, the surface roughness evaluation must be done before the grit blasting operation. Therefore, close attention to the outage schedule must be paid by the individual conducting the steam path audit.

Grit blasting removes scale and deposits that accumulated on the stationary and rotating blading from impurities in the steam, but it is considered a temporary measure. Unless the source of the deposits is identified and eradicated, the deposits will return on startup. It is therefore necessary to take samples of the deposits found on the blades so that the chemistry department can identify the impurities in the steam.

A typical grit blasting material is 220 aluminum oxide grit, but there are other size grits and blasting media available. The factors involved in deciding which material to use involves the cost and availability of blasting media, the ability to remove steam path deposits, blast times, recyclability, and the resulting surface finish. The surface finish left behind by grit blasting with aluminum oxide is rougher than a new and clean surface. To leave a smoother finish, consideration might be given to glass bead, walnut shells, and other media that will leave a smooth finish.

If there is reasonable assurance that deposits will not be a problem in the upcoming operation of the turbine, polishing the blades after blast cleaning them may restore the surface roughness to the new and clean condition. The HP and IP turbines have the greatest performance benefit from polishing.

Chemical Analysis of Deposits

Impurities in the condensate, feedwater, boiler drum, and throughout the system can find themselves deposited on the steam turbine blading. These deposits roughen the surfaces and, if thick enough, can reduce the flow passing area. Corrosive deposits can damage the integrity of the blading when accumulated on blade attachments, covers, and other structural elements of the steam path. When a steam turbine is opened up for an overhaul, a representative from the chemistry department should be ready to take samples of any deposits found in the steam path.

Samples should be taken at each stage because adjacent stages can have very different material deposited on them. The precipitate pressure is a characteristic of a compound, and the L-1 stage might have very different deposits than the L-2, and so on.

Depending on the type of deposits, recommendations may include making changes to the boiler chemistry, the condensate polishing system, the makeup water, heat transfer tube material in the condenser and feedwater heaters, and other processes affecting the purity of the steam entering the turbine.

Minor Weld Repairs

Minor weld repairs are often a standard maintenance item during an open/clean/close outage. Dings, cracks, and foreign object damage can often be repaired with a minor weld repair. The repairs improve performance by making the flow path more streamline by reducing turbulence.

Special care must be given to the trailing edges of nozzles if performance is to be maximized. The highest steam velocities in the stage occur at the exit of the stationary blading. Any weld repair, minor or major, that leaves the trailing edges of the nozzles thicker than design is going to decrease stage efficiency because of the increased turbulence and wake at the nozzle exit. The purpose of the nozzles is to turn and accelerate the steam before entering the rotating blades. If a portion of the kinetic energy of the steam is given up for turbulence, then there is less energy to produce power in the rotating blading and stage efficiency decreases.

Seals

The outage serves as an opportunity to improve the effectiveness of the steam sealing devices. A steam path audit is very useful in guiding seal decisions based on benefit-cost analysis. The benefit of replacing a seal comes from the steam path audit where clearances and tooth condition data are used to determine the extra annual fuel costs associated with the excess leakage through that seal. The cost of replacing a seal comes from the seal vendor. The ratio of the benefit in annual fuel savings to the cost to purchase gives a benefit/cost ratio that can be prioritized along with the other sealing devices.

Often the required return on a seal investment is one year, or a benefit/cost ratio of 1. So, if a seal costs $5000 and the excess annual fuel costs from the rubbed seal are $5000, then the return on the new seal will be realized in one year. If, on the other hand, the excess fuel costs are $2000, the benefit/cost ratio is 2:5, and it would be more cost-effective to spend $500 sharpening the tooth. The benefit of sharpening the seal teeth will be less than that of a new packing, but if the benefit was, for example, $1000, the resulting benefit/cost ratio is $1000/$500 = 2 and the return on investment is six months.

Figure 3-32 shows a sharp tooth on the left and a rubbed tooth on the right. Sharpening the rubbed tooth is typically done with a die bit tool that will peel the ridges away from either side of the tip of the tooth. The result is a sharp tooth but at the cost of some additional clearance. If the sharpened clearance is near the design clearance, it will be likely that sharpening is more cost-effective than replacing the sealing device.

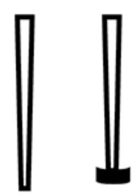

Figure 2-31
Sharp and rubbed steam seal tooth

The types of steam seals include the following:

- Interstage, diaphragm, or shaft packings at the inside diameter of the stationary blading. The seal controls leakage around the inside diameter of the stationary blading along the rotor. The excess leakage flow misses the opportunity to turn and accelerate to the nozzle exit conditions before releasing its kinetic energy in the rotating blade. These packings are very often hi-lo labyrinth packings in HP and IP turbines in impulse designs. They are often integral to the stationary blades in reaction designs.
- Tip spill strips or shroud seals control leakage around the outer diameter of the rotating blading. Excessive leakage here bypasses the rotating blade and the opportunity to produce power. These seals are usually composed of one or two rows of teeth, although there are designs with several teeth. There is not too much difference between the tip or shroud seals between impulse and reaction machines.

- Root spill strips are usually found only in impulse designs. They control the leakage between the flow into the rotating blade and the flow between the interstage packing and balance hole.
- End packings and gland packings are found in both impulse and reaction design and control the leakage where the rotor or spindle emerges from the casings. They are usually hi-lo labyrinth packing rings, although the LP turbine sometimes has teeth all of one height to avoid shearing off high teeth in a rotor long or rotor short event.

Horizontal Joints

The horizontal joint, where the upper and lower halves of steam path components come together, is often the path of steam leaks. Sometimes the leaks leave behind a trail of deposits that can be collected and analyzed for their content. Other times the steam cuts the base metal, which needs to be filled in with weld material and ground to a smooth, level finish during an outage.

Warping is another problem that may prevent the horizontal joint from making a tight seal. Warping can take several forms. Casings can spring in or out depending on the type of residual stress that may reside in the upper and lower halves while they are bolted together.

The outage may be an opportunity to measure the amount of warpage in a casing or other steam path part. Often tops on bolted/tops off unbolted vertical and horizontal diameter measurements are made to determine both ellipticity and the amount of thermal strain once unbolted.

Severe warpage may require an off-site solution where parts are welded, machined, and heat-treated to relieve stress.

Horizontal joint leakage losses are typically small. The magnitude of the loss depends on the specific volume of the steam, the area of the leak, and the portion of the steam path bypassed. A leak across a diaphragm horizontal joint in the LP turbine might have surprisingly small losses associated with it because the specific volume is large and only a small amount of mass flow can fit through a small leak area. The leak bypasses the energy range of only one stage. A horizontal leak of similar cross-sectional area in the HP turbine inner casing, on the other hand, will have a much larger performance effect. The leakage path may bypass several stages, the specific volume is small, and therefore a large amount of mass flow will miss the opportunity to produce power in several stages.

The horizontal joints should be clean, level, and tight-fitting on reassembly.

Responsibilities During a Major Turbine Overhaul

A steam turbine performance engineer typically spends much time and effort in planning for an outage. Once the outage arrives, time is at an utmost premium. Although the performance items on the schedule are not usually on the critical path, they must be done such that they do not interfere with the critical path. The Ghant chart in Figure 2-24 shows where the opening and closing steam path audit might occur in the schedule. Other performance activities can occur between the opening and closing steam path audits.

Opening Steam Path Audit

The opening steam path audit is a visual inspection of the steam path conducted while the unit is being disassembled. The collection of data and subsequent performance calculations result in a list of recommended maintenance items prioritized by return-on-investment or benefit/cost ratio. A detailed description of a steam path audit is given in Appendix C.

The steam path audit results help to guide the maintenance decisions during the outage and can identify the root cause of several types of damage. For instance, a steam path audit will identify the amount of excessive leakage through a seal and the corresponding fuel costs associated with that leak. However, it will also shed light on the root cause of that leak. The seal might be rubbed from imbalance or misalignment; the seal might be broken from foreign object damage or the reassembly/disassembly process; corrosive deposits might have weakened the sealing teeth allowing them to break easily; a retractable packing might be stuck in the open position; the as-left clearances from the previous outage may have been too large, and so on. Knowing the root cause is necessary to solve the problem.

The steam turbine performance engineer may conduct the steam path audit, or it may be subcontracted out to a contractor who performs steam path audits routinely. Either way, the steam path audit results must be ready within a day or two of lifting the rotor out of the casing so that the results can be used in guiding maintenance decisions.

The steam turbine performance engineer works closely with the chemistry department, the maintenance department, and the outage foremen to complete the audit in a thorough and timely manner. The results and maintenance recommendations are typically delivered to interested personnel on-site within two days after lifting the rotor.

Cycle Isolation

Ideally, the steam turbine performance engineer goes into an outage with a complete list of cycle isolation issues to fix, with the full backing of the I&C and maintenance departments.

Instrumentation

Like the cycle isolation fixes, the steam turbine performance engineer ideally goes into an outage with a complete list of instrumentation problems to fix with the full backing of the I&C department.

The outage is the time to calibrate instruments that could not be calibrated in place while the unit was operating. Some instrumentation might need replacing or upgrading. Still other locations might need some instrumentation where none existed before. Water leg problems can be corrected if there are incorrect slopes in the pressure-sensing lines.

The outage might also be a good time to conduct an instrumentation audit, especially if a precision test is planned.

Photographic Record of the Steam Path

Whether the steam path audit is conducted by in-house personnel or by an outside contractor, it is a good idea to include a photographic record of the steam path. Photographs of the inlet and exhaust side of each stage and photographs of the horizontal joint with the top half off before the rotor or spindle has been removed provide a detailed look at the steam path. The documentation is useful in comparing the condition of the steam path from one outage to the next. It also serves as a record for the returned-to-service condition of the steam path during the closing steam path audit.

Photographs taken during the opening steam path audit document the as-found condition of the steam path, whereas photographs taken during the closing steam path audit document the as-left condition of the steam path.

Labeling of the steam path components should be done in chalk, not grease or paint pen, before taking the photograph. Alternatively, a card with a label on it can be placed within the photograph. Needed steam path repairs are often marked with a grease or paint pen by the NDE personnel, so it is best to not add marks that might cause confusion in the repair process. It is recommended to label the subjects in all photographs rather than to rely on a perceived ability to remember the photograph subject later on.

Reverse Engineering Data-Gathering Opportunities

If reverse engineering of steam path parts is to be done in the future, outages are often the only opportunities to gather the necessary dimensional measurements. The steam turbine performance engineer may be involved in taking these measurements for in-house reverse engineering, or potential outside bidders for future upgrades may ask to visit the site during the outage to collect dimensional data.

Closing Steam Path Audit

A closing steam path audit takes place during the reassembly of the turbine. The measurements taken are very similar to those taken during the opening steam path audit. The closing steam path audit serves two purposes: it is a quality control check on the maintenance conducted during the outage, and it predicts the return-to-service condition of the unit.

Unexpected Damage/Deterioration

Despite the best pre-outage planning and analysis of test data, there can still be surprises once the steam path is exposed. The steam turbine performance engineer should carefully document and photograph all expected and unexpected damage that might result in performance deterioration.

The outage is an ideal time to determine the root cause of unexpected damage or deterioration because the ultimate fix might require an outage. Better to fix the problem during the current outage than the next outage several years later. Determining the root cause might involve bringing in other in-house personnel, or it may require bringing in outside experts in steam turbine performance and maintenance repairs.

It is a good idea to have an expert on call and authorized by the purchasing department to show up on-site at a moment's notice. Table 2-2 provides a list of things to do or not do during an outage.

Table 2-2
Rules of thumb during an outage

Do	Don't
Communicate with the technical director about the upcoming schedule.	Delay the outage.
Empty your pockets before climbing up on the turbine.	Drop anything in the turbine.
Tie off everything before inspecting the turbine.	Go underneath parts being lifted by the crane.
Make safety a priority, attend safety orientation, wear all safety equipment.	Use a grease or paint pen on turbine parts.
Bring unusual findings to the attention of the technical director.	

Constant Communication

The steam turbine performance engineer must be in constant communication with several people and departments during an outage. Figure 2-33 shows a typical organizational chart during an outage. The on-site person with overall responsibility for the outage is the outage manager. Reporting to the outage manager are managers from maintenance, I&C, chemistry, safety, and the outage services project manager, who may be an outside contractor. The steam turbine performance engineer is in this group. The outage services project manager might be a representative from the turbine OEM, contractor, or engineering company who serves as a liaison between the owner and the on-site personnel hired specifically for the turbine overhaul. This project manager might be responsible for several outages. On-site, the technical director is the highest-level representative from the OEM, contractor, or engineering company. The steam turbine performance engineer should be in close contact with this individual.

Many coordinated efforts take place between the performance engineer, chemists, the maintenance manager, I&C, the technical director, shift supervisors, and the other subcontractors.

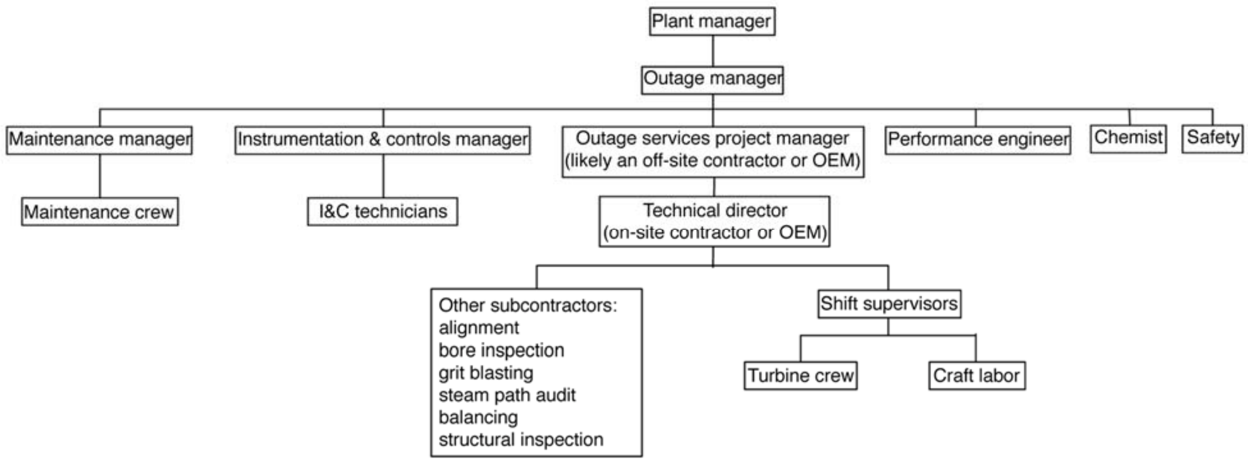

Figure 2-32
Outage organizational chart

The steam turbine performance engineer's responsibilities are usually off the critical path. Daily communication with shift supervisors about the schedule will help to keep it that way.

3
INTERRELATIONSHIPS WITH OTHER PLANT OR CENTRAL OFFICE DEPARTMENTS

The sites with the most successful performance programs are those that include all employees on the periodic performance reports, ensuring that all are aware of their individual effect on heat rate. Communicating the activities and goals for plant performance makes it a sitewide effort, multiplying the capability of the steam turbine performance engineer by the number of employees aware of and engaged in the program.

Operations

Operators in the control room can be a very valuable resource for steam turbine performance engineers. They often keep meticulous records of equipment problems, load changes, and plant response. They are on the front line when a pressure, temperature, or flow goes awry. Control room operators also have a good feel for the cause-and-effect response of the plant, which can be very useful in troubleshooting isolation and performance problems. The steam turbine performance engineer should always consult with the operators before recommending changes to operation or making performance conclusions from test data analyses.

The steam turbine performance engineer also works closely with control room operators during performance tests. Although there is sometimes a somewhat contentious relationship between performance engineers and operators, the performance engineers must respect the responsibility of the operators to respond to the dispatchers with rapid, safe, and smooth delivery of the required power.

Chemists also typically serve in the operations department and offer valuable information for performance. Surface roughness from poor water chemistry control is typically one of the greatest losses found during steam path audits.

Maintenance

The maintenance manager is ultimately the person responsible for carrying out maintenance recommendations coming from the performance department. Conversely, the maintenance department may make repairs that affect performance. In most maintenance cases, there is a component of reliability and a component of performance. The steam turbine performance engineer must realize that the scale will usually tip in the direction of reliability.

Instrumentation and Controls

Technicians in the I&C department will prove invaluable when preparing for a test and while conducting routine monitoring. They may conduct in-house instrumentation audits or be involved in outsourced instrumentation audits.

Scheduling and Outage Management

Recognizing that performance issues are not usually on the critical path of an outage, the steam turbine performance engineer may have to advocate for a slot on the Ghant chart.

The following are what the outage manager can do to facilitate the steam path audit process without sacrificing outage time:

- Lay down stationary turbine blades (diaphragms or blade rings) with the exhaust side up, which will allow easier access to important dimensions on the nozzles.
- Do not remove steam packing rings from the turbine until they have been inspected.
- Place casings on the turbine deck with the steam path side up for easy access for inspection.
- Keep the audit engineer informed of changes to the disassembly and reassembly schedule.

Purchasing and Legal

Recognize that hiring subcontractors requires finite lead time to get through the proposal and contract steps. Final agreement on terms may involve a certain amount of iteration and time. A blanket contract might be beneficial if there are subcontractors that are used often. That way, one phone call might be all that is necessary to get someone on-site immediately.

Other Engineering Staff

Predictive Maintenance

Turbine performance can be measured by more than steam and water pressures, temperatures, and flows. Vibration, oil purity, bearing temperatures, and so on can help to anticipate problems and corroborate that a suspected thermal performance problem might exist.

System Engineers

The turbine is but one of many machines operating in the many interdependent systems and cycles in the power plant. Knowledge of boiler efficiency, auxiliary loads, heat rejection performance, and feedwater heater performance can aid in diagnosing problems with the steam turbine.

Dispatch

Communicating with the dispatcher can help in planning tests and short outages. Some specialized testing requires part-load operation, which needs to be scheduled appropriately.

Table 3-1
Who should be in your speed dial

Control Room
EPRI
Flow nozzle calibration facility
I&C department manager
Instrumentation rental and calibration
Local trucking company
Local turbine shop
Maintenance manager
Plant chemist
Turbine crew on short notice
Turbine performance consultant
Turbine testing company

4
TRAINING

The half-life of an engineer's knowledge is approximately seven years. Ongoing training is necessary to keep current with continually evolving technology.

Some of the basic educational requirements of a steam turbine performance engineer are a bachelor of science degree in mechanical engineering and a working understanding of thermodynamics and fluid mechanics. However, having graduated with the aforementioned credentials, many new steam turbine performance engineers have not been exposed to the steam turbine performance field. Postgraduate studies may include some courses in the power-generation field, but the opportunity to take university level courses on steam turbine cycle efficiency is rare.

Many steam turbine performance engineers are also licensed professional engineers or engineers-in-training. Many states are now including ongoing education as a requirement to maintain the professional engineer license.

Some of the best informal training can come from on-the-job experience. This opportunity is enhanced when the new steam turbine performance engineer can work with someone who might be considered a mentor. Utilities often have programs for newly hired engineers in which they rotate through various jobs, including performance.

More formal training can come from a variety of sources that present seminars and workshops. Many of these offer professional development hours required by the state professional engineering licensing boards. One such training course is presented by Deborah Cioffi, the principal investigator for this guide: Mechanical Dynamics & Analysis's *Evaluating and Improving Steam Turbine Performance*, based on Ken Cotton's textbook [3] of the same name. General Physics Corporation offers many plant performance–related courses including heat-rate awareness training, which was developed under EPRI auspices to fulfill the needs of plant performance engineers. Other training companies offer seminars in the steam turbine performance field as well.

The ASME offers many relevant courses through their training and development program (www.ASME.org). EPRI, of course, holds many seminars, workshops, conferences, and other opportunities to learn about steam turbine performance (www.EPRI.com).

Another source of learning is the training that accompanies the purchase and use of a heat-balance program. There are several on the market, and many of the companies that offer the software also offer training. Because the heat-balance software is so focused on the thermodynamics of the steam turbine cycle and the intricacies of the components within the cycle, the software training often involves a large amount of background information on the steam turbine cycle and its components beyond the use of the software.

5
WHAT SHOULD BE IN YOUR LIBRARY

ASME Appendix A to PTC 6, The Test Code for Steam Turbines, PTC 6A, American Society of Mechanical Engineers, New York, 2000.

ASME Flow Measurement, PTC19.5, American Society of Mechanical Engineers, New York, 2004.

ASME Performance Test Code 6 on Steam Turbines, PTC 6, American Society of Mechanical Engineers, New York, 2004.

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Sanders, W. P., *Turbine Steam Path, Volume 3a - Mechanical Design and Manufacture*. Tulsa, OK, 2004.

Sanders, W. P., *Turbine Steam Path, Volume 3b - Mechanical Design and Manufacture*. Tulsa, OK, 2004.

Turbine instruction manuals

Your college fluids and thermodynamics textbooks

EPRI’s three-day biannual heat-rate improvement conference has been the industry gathering with a focus on power plant performance. The proceedings listed below contain examples of how companies have reduced their heat rates, justified those investments, and analyzed the results.

EPRI's Twelfth Heat Rate Improvement Conference Proceedings (2001)	1001328
Proceedings: 2003 EPRI Heat Rate Improvement Conference, January 28–30, 2003	1004118
2005 EPRI Heat Rate Improvement Conference	1010321
2007 EPRI Heat Rate Improvement Conference Proceedings	1014799
2009 EPRI Heat Rate Improvement Conference Proceedings	1017546

6
REFERENCES

1. B. Bornstein and K. Cotton, "A Simplified ASME Acceptance Test Procedure for Steam Turbines." *Journal of Engineering for Power, Transactions of ASME*, Vol. 104, No. 1, pp. 224–230 (January 1982).
2. *Replacement Interstage Seals for Steam Turbines*. EPRI, Palo Alto, CA: 2005. 1010214.
3. K. C. Cotton, *Evaluating and Improving Steam Turbine Performance*, 2nd edition. Cotton F.A.C.T., Rexford, NY, 1998.

A
ANATOMY OF A FOSSIL HEAT-BALANCE DIAGRAM

The heat balance is typically developed by the turbine OEM. It contains a vast amount of information about the turbine and the requirements of the balance-of-plant equipment. The heat balance shows the thermal energy entering and exiting each major piece of equipment in the turbine cycle. It is a schematic drawing and not to scale. The thermal kit generally contains a family of heat balances: guarantee load, VWO load, 5% OP, and several part-load heat balances that sometimes correspond to valve points.

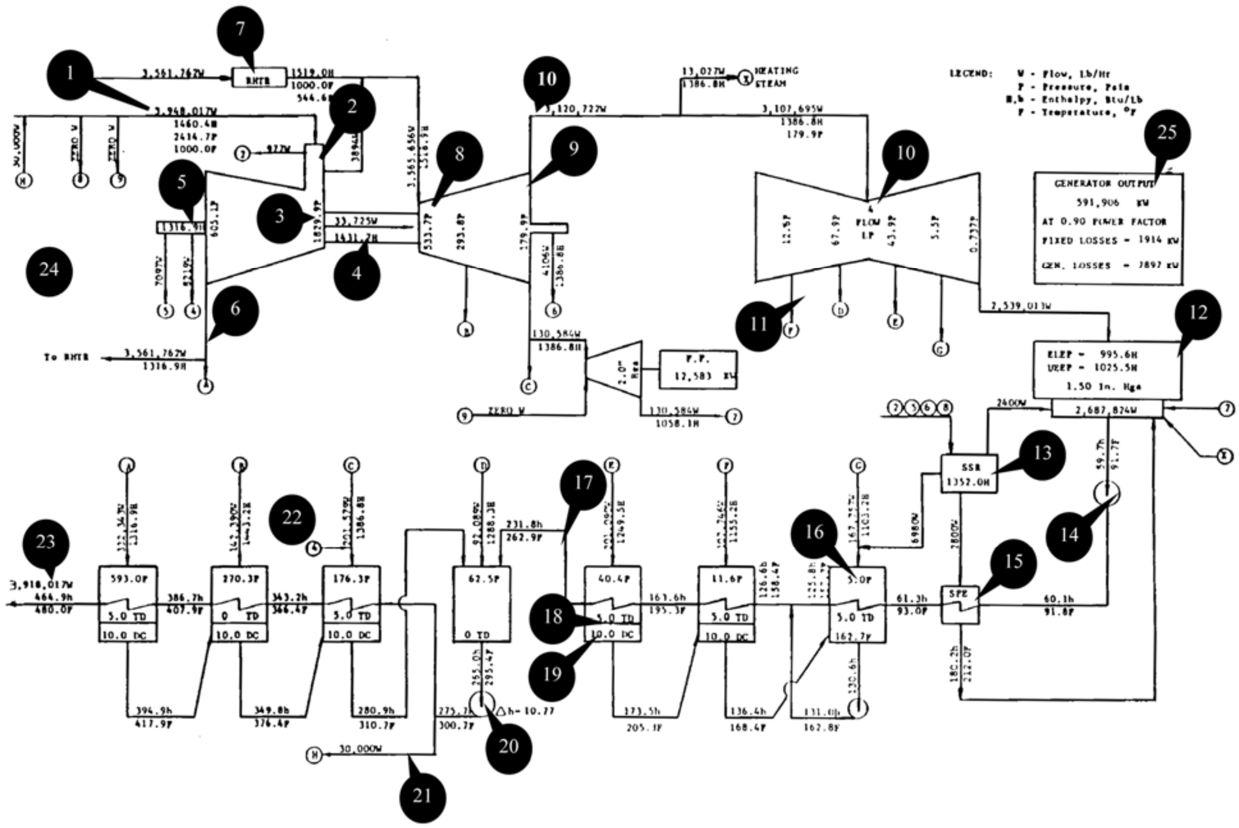

Figure A-1
Typical fossil heat balance

The following notes correspond to the callouts shown in Figure A-1:

1. **Main steam**. The main steam conditions (pressure, temperature, enthalpy, flow) entering the turbine. Auxiliary steam is shown leaving the main steamline before reaching the turbine. Often auxiliary steam from the main steamlines will have zero flow at full-load conditions. The main steam valves and turbine inlet valves are not usually shown on the heat balance.
2. **Above and below seat drains**. Drains flow rates from the turbine inlet valves.
3. **First-stage shell pressure**. This pressure is not usually shown on the heat balance. It is usually shown on a separate first-stage shell pressure versus flow curve contained in the thermal kit. The first-stage shell pressure location is after the control-stage rotating blades and before the second set of stationary blades.
4. **HP end packing**. This packing is located closest to the front end of the HP turbine and has the highest energy associated with it of all the end packings. Its purpose is to minimize the leakage out of the HP turbine casing. Excessive leakage from this packing can have very high losses associated with it. The source of this leakage is the first-stage shell. In an opposed-flow HP/IP configuration shown in Figure A-1, the endpoint of the leakage is in the IP turbine bowl. In a separate span HP turbine, the HP end packing usually has three leakoffs: one for a heater extraction, one for the steam seal supply system, and one for the stealing steam exhauster/condenser.
5. **HP exhaust end packing**. The purpose of this packing is to minimize leakage emerging from the exhaust end of the HP turbine casing along the rotor. The source of steam is the HP turbine exhaust, and the steam usually leaks off to the steam seal supply system and the steam seal condenser/exhauster system.
6. **Cold reheat**. The exhaust steam from the HP turbine is referred to as the cold reheat steam. It includes the steam from the exit of the HP turbine to the entrance to the reheater. Typically, the highest pressure heater extraction steam originates from this location. Occasionally, a steam cycle will have a heater above the reheat point (HARP), where the extraction steam originates within the HP turbine upstream of its exhaust.
7. **Reheater**. Cold reheat enters the reheater and emerges as hot reheat steam. Heat balances are typically developed by the turbine manufacturer and offer nominal values for reheater pressure decrease. Typically, the percentage of pressure decrease across the reheater is $(P_{crh} - P_{hrh}) * 100/P_{crh} = 10\%$.
8. **IP bowl**. The location just upstream of the first-stage stationary blades in the IP turbine is referred to as the IP turbine bowl. There is typically a 2% pressure decrease between the hot reheat location and the IP turbine bowl, across the reheat intercept valve. In an opposed-flow HP/IP turbine, the IP turbine bowl is where the HP end packing leakage steam enters the IP turbine and mixes with the incoming hot reheat steam. The IP turbine bowl is the starting point for the expansion through the reheat turbine (IP and LP turbines—everything downstream of the reheater), and approximately 70% of the power is generated in the reheat turbine.

9. **IP exhaust**. The location at the exit of the IP turbine. It is often the location from which auxiliary flows originate (power house heating steam, boiler feedpump turbine flow, air preheater flow). It is also the origin of the IP turbine exhaust end-packing flow. Most of the IP turbine exhaust flow enters the crossover piping that carries the IP turbine exhaust steam into the LP turbine.
10. **LP inlet**. The LP turbine may have one, two, three, four, or more parallel flows, but the heat balance often shows it schematically as a single or double casing with a note for the number of actual parallel flows. There is typically a 3% pressure decrease from the exit of the IP turbine to the LP turbine bowl, just upstream of the first stationary blades in the LP turbine.
11. **LP heater extractions**. There are usually several extractions in the LP turbine that provide heating steam to the LP heaters. Although the steam conditions of the extractions can be obtained from the heat balance, the cross section must be used to determine the exact location of the extractions. Sometimes the extractions on one end, for example, the turbine end, are mirror images of the extractions on the other end—the generator end. Other times, the turbine end may have extractions at stages L-1 and L-3 in the turbine end, whereas the generator end has extractions at L-2 and L-4, and so on.
12. **Condenser**. It is here that the turbine manufacturer conveys the information to the condenser manufacturer as to how much heat must be rejected from the exhaust steam. The product of the mass flow leaving the turbine and its enthalpy represent the thermal energy of the steam leaving the LP turbine headed for the condenser. On the condenser schematic, there will be at least one enthalpy. Figure A-1 shows two turbine exhaust enthalpies: ELEP and UEEP. If a heat balance shows only one unlabeled enthalpy, it is the UEEP. The UEEP is the physical endpoint of the expansion to condenser pressure. If the turbine exhaust pressure and moisture were measured, the resulting enthalpy would be the UEEP. The ELEP is a fictitious point, used by the turbine designers when designing the last-stage blading. The ELEP is the endpoint enthalpy if there were no total exhaust losses. The total exhaust loss is the UEEP minus the ELEP. Sometimes the circulating cooling water temperature and flow rate are shown on the condenser (it is not shown in Figure A-1).
13. **Steam seal regulator or gland steam regulator**. This box represents the steam sealing system that takes leakoff flows from the HP end packings and supplies sealing steam to the LP end packings so air does not leak into the condenser at the exit of the LP turbine where the rotor emerges from the casing. The HP seals usually supply more than enough sealing steam for the LP turbine seals, so the schematic of the steam seal regulator/gland steam regulator also shows where the overflow goes. In Figure A-1, the overflow goes to the extraction to the lowest pressure heater.
14. **Condensate pump or hot well pump**. There are likely two or more pumps that pump the condensate from the hot well to the dearator, but the heat balance schematically represents all of them as one.
15. **Steam packing exhauster or gland steam condenser**. This heat transfer device condenses the steam and removes the air from the air/steam mixture entering the end packing outboard leakoff lines. It provides a small amount of heat to the condensate line before the condensate enters the feedwater heater train.

16. **Lowest pressure feedwater heater**. The lowest pressure heater is the first feedwater heater through which the condensate passes. Sometimes the heater drain flow is pumped forward into the condensate line before the second heater (as in Figure A-1). Other times the drain flow cascades to the condenser. The steam seal regulator overflow line also may enter the steam extraction line before the heater; otherwise the overflow may go to the condenser. The heat balance represents parallel heater trains as one.
17. **Condensate into deaerator**. The water entering the deaerator is referred to as condensate. Once the flow emerges from the feedwater pump after the deaerator, it is referred to as feedwater. The condensate line entering the deaerator is often an ideal location for a condensate flowmeter.
18. **Terminal temperature difference (TTD)**. The TTD is a design characteristic of the heater and is a measure of the heater's heat transfer capability. It is defined as the difference between the saturation temperature corresponding to the heater shell pressure and the condensate or feedwater outlet temperature. TTDs can be zero or negative if the heater has a superheating section. Typical values range from 5°F to −5°F. If, during service, the TTD increases, it is a sign that heater performance is decreasing. When a heater's performance decreases, it and subsequent heaters often call for more extraction steam. The power and heat-rate losses result from extra extraction steam leaving the turbine, bypassing the opportunity to produce power.
19. **Drain cooler approach temperature difference (DC)**. Like the TTD, the DC is a design characteristic of a portion of the heater and provides a measure of the subcooling of the drain flow, indicative of how much heat is removed from the extraction steam. It indicates poor heater performance (typically caused by level control problems) if it increases during service, which causes an increase in extraction flow. It is defined as the difference between the drain temperature and the approach temperature.
20. **BFP**. Often there is more than one BFP, but the heat balance shows one schematically. There may be a collection of motor-driven and turbine-driven BFPs. The purpose of the BFP is to provide adequate turbine inlet pressure after overcoming pressure decreases in the HP heaters, economizer, boiler, superheater, valves, and piping. Often the superheater attemperating spray originates at the BFP.
21. **Attemperating spray**. Main steam temperature and hot reheat steam temperature are usually controlled by attemperation. The attemperating sprays often originate at the BFP, the highest pressure in the cycle. The heat balance in Figure A-1 shows the attemperating spray entering the main steamline because this heat balance does not include a boiler.
22. **HP feedwater heaters**. The heaters after the BFP are referred to as the HP heaters. Their drains cascade back to the deaerator. Performance deterioration in the HP heaters has a greater effect on turbine cycle efficiency than the same performance deterioration in the LP heaters.
23. **Final feedwater**. The final feedwater emerges from the highest pressure heater and subsequently enters the economizer. It is an important location for monitoring turbine cycle efficiency. The final feedwater flowmeter is placed in this location. Heat-rate calculations are based on the feedwater flow and enthalpy.

24. **Boiler**. Figure A-1 does not include a boiler, but many times a heat balance will include one. When it does, the schematic may show attemperating spray entering the superheater, and it may also show blowdown flow leaving the steam drum.
25. **Generator**. The generator on a heat balance is represented by a box containing the electrical output, power factor, and electrical and mechanical losses.

B

ANATOMY OF A NUCLEAR HEAT-BALANCE DIAGRAM

There are many more similarities between fossil and nuclear heat balances than differences. Therefore, the notes pertaining to the generic nuclear heat-balance diagram focus on what is unique in a nuclear cycle. Those wishing to study the anatomy of a nuclear cycle heat balance are encouraged to study the fossil heat balance in Appendix A as well.

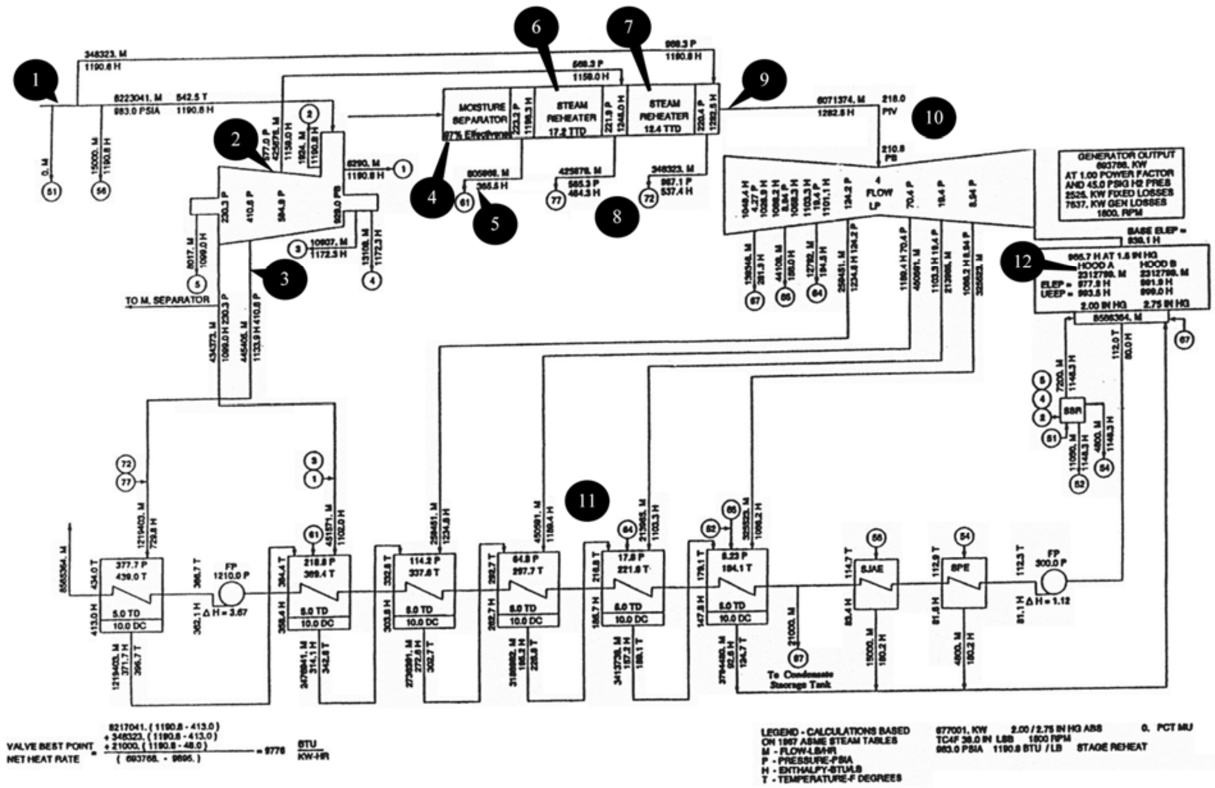

Figure B-1
Typical nuclear heat balance

The following notes correspond with the callouts shown in Figure B-1:

1. **Main steam**. The main steam conditions (pressure, moisture, enthalpy, flow) entering the turbine. Auxiliary steam is shown leaving the main steamline before reaching the turbine. Often auxiliary steam from the main steamlines will have zero flow at full-load conditions. The stage 2 reheating steam comes from the main steamline.
2. **HP reheater extraction**. Stage 1 reheater steam comes from an extraction part way through the HP turbine.
3. **HARP extraction**. It is common for a nuclear cycle to have a HARP. The extraction steam and the main steam entering the reheater stages compose the reheating steam.
4. **Moisture separator**. When the steam emerges from the HP turbine, the moisture content is approximately 10%. The moisture separator removes the water and sends the dry steam to the reheater (in a reheat cycle). The effectiveness of this moisture separator is nominally 97%. In other words, it removes 97% of the water from the wet steam.
5. **Moisture separator drain**. The water removed from the HP turbine exhaust flow is sent to another feedwater heater in the cycle.
6. **Stage 1 reheater**. The steam that emerges from the moisture separator enters the shell of the stage 1 reheater. The HP extraction steam (the reheating steam) enters the tube side of the reheater and condenses as it gives up its heat to the reheated steam. The TTD is written on the heat-balance diagram and is the saturation temperature corresponding to the reheater stage 1 pressure minus the temperature of the reheated steam.
7. **Stage 2 reheater**. Similar to the stage 1 reheater, the stage 2 reheater has reheated steam on the shell side and reheating steam on the tube side. The reheating steam originates from main steam. The TTD is the difference between the saturation temperature of the reheating steam at the stage 2 pressure minus the temperature of the reheated steam emerging from the reheater. Not all nuclear cycles have two-stage reheaters. Some have no reheat and others have one-stage reheaters where the reheating steam usually comes from main steam.
8. **Reheater drain**. High-pressure condensate from the reheating steam leaves the reheater stages and enters the top feedwater heater shell where it flashes to steam.
9. **Hot reheat steam**. In a nuclear reheat cycle, the hot reheat steam is usually the only superheated steam in the turbine cycle with enough superheat to get a good temperature measurement. Elsewhere throughout the turbine, the steam is wet and a temperature measurement would give saturation temperature.
10. **LP inlet**. Nuclear cycles do not have IP turbines. The phrase LP inlet is preferred over hot reheat. Nuclear cycles often have two or three double-flow LP turbines. The large number of back ends is required to pass the large amount of mass flow. The available energy range of a nuclear cycle is less than that of a fossil cycle, but the mass flow is much greater.
11. **No deaerator**. Nuclear cycles do not normally have deaerators. They scavenge oxygen and other impurities chemically.
12. **Condensers in series**. This type of condenser is found in both nuclear and fossil cycles. The circulating water enters hood A, then continues on to hood B in series. The condenser pressure and UEEP of hood A are less than those in hood B.

C
DESCRIPTION OF AN EFFICIENCY STEAM PATH AUDIT

Description of an Efficiency Steam Path Audit

A steam path audit is a visual inspection of the steam turbine while the unit is open. The purpose of a steam path audit is to quantify the observed damage in terms of power output, heat rate, and fuel costs. The losses are compared with the necessary repair or replacement costs to determine whether the cost/benefit ratio of the fix results in a desirable return on investment. The result of using a steam path audit to guide turbine steam path repairs is to realize the greatest return on the maintenance dollars invested during the outage.

A steam path audit begins with modeling the turbine cycle on a stage-by-stage basis. The power output, efficiency, and steam thermodynamic properties are calculated for each turbine stage. Nonproprietary information such as the VWO heat balance, cross section, seal diagram, and first-stage shell pressure curve are used to develop the stage-by-stage computer model.

The on-site portion of the steam path audit begins when the audit engineer arrives on-site just as the upper casing is lifted off the rotor. The audit engineer makes several measurements of seal clearances, surface roughness, miscellaneous leakage areas, area changes caused by deposits and/or mechanical damage, and any other anomaly in the steam path. The inspection begins where steam enters the outer casing upstream of the first-stage nozzle and ends at the exhaust annulus where the steam leaves the turbine for the condenser. Each stationary and rotating blade, each shaft and tip seal, each end packing, the horizontal joint, and all casing fits are inspected and appropriately measured where a deviation from design conditions is observed.

The measurements taken by the audit engineer do not interrupt the scheduled outage. The work is done simultaneously with the disassembly of the unit. The process of taking the measurements is flexible; for example, while the outage crew is working on the left side of the turbine, the audit engineer is working on the right side. While the crew is working on the HP turbine, the audit engineer is working on the LP turbine, and so forth. The steam path audit does not prolong the outage.

The measurements taken by the audit engineer are input into the steam path audit calculations spreadsheet. The calculations compare the as-found condition of the steam path with the design condition. Power output, heat rate, and annual fuel costs are assigned to each loss. For example, if the rubbed tip spill strip on the third stage had a clearance of 40 mils and the design clearance was 30 mils, the extra 10 mils of clearance would represent, for example, a 200-kW loss in power, a 5-Btu/kWh gain in heat rate, and $12,500 per annum in additional fuel costs. If a new tip spill strip cost $7000, the return on investment would be less than seven months. In this case, it would be desirable to replace the tip spill strip.

If, on the other hand, the rubbed packing had a clearance of 32 mils, which increased fuel costs by $1400, it would not be cost-effective to replace the tip spill strip. Instead, it would be cost-effective to sharpen the tip spill strip and put it back into service. Knowing what not to replace can be just as beneficial to the maintenance budget as knowing what to replace.

It takes roughly a week to complete the on-site portion of a steam path audit. The actual time on-site depends largely on the number of stages in the turbine. Before leaving the site, the audit engineer presents preliminary findings to interested personnel. This exchange allows personnel to review and fully understand the steam path audit results. Often the owner requests that photographs be taken to document the condition of the steam path.

What has been described thus far is an opening steam path audit. There is also a closing steam path audit that is conducted simultaneously with the reassembly of the steam turbine after the repairs have been made. Like the opening steam path audit, the closing steam path audit does not interfere with the outage schedule, and like the opening steam path audit, the preliminary results are presented to interested personnel before the audit engineer leaves the site. The closing steam path audit serves two purposes: a quality control check on the maintenance performance during the outage and an estimated return-to-service condition of the unit. The closing steam path audit is often considered optional by customers.

Shortly after delivery of the preliminary steam path audit results, a final report is published that contains the stage-by-stage performance results, photographs, calculation procedures, and references. Recommendations for repairs and replacements are based on the annual fuel costs and the desired cost/benefit ratio. If a closing steam path audit is conducted, the results are also contained in the final report. The final report contains results for the following loss categories:

- Interstage packings (diaphragm packings)
- Tip spill strips (shroud seals)
- End packings
- Miscellaneous leakages
- Surface roughness
- Solid particle erosion
- Deposits
- Foreign object damage
- Other miscellaneous losses

Steam Path Audit Toolbag
6-in. steel ruler
Ball gauges
Caliper with depth gauge
Camera
Chalk
Clipboard with pencil attached
Feeler gages
Flashlight with extra batteries
Magnifying glass – illuminated
Micrometer
Mirror
Radial packing gauges
Snap gauge
Steel measuring tape
String
Surface roughness comparitor
Taper gauges

D
IMPULSE AND REACTION NOMENCLATURE

Table D-1
Impulse and reaction nomenclature

Impulse	Reaction
Main stop valve	Throttle valve
Intercept valve	Interceptor valve
Control valves	Governor valves
First stage	Control stage
Diaphragms	Blade rings
Buckets	Blades
Erosion shields	Stellite strips
Tie wires	Lashing lugs
Bucket covers	Shroud bands
Packings	Glands
N-X packing	Dummy packing
Spill strips	Seal strips
Packing	Spring backs
Rotor	Spindle
Shell	Cylinder
Steam seal regulator	Gland steam regulator
H2 seal casing	Gland bracket
Turbine end/generator end	Governor end/exciter end
Upper shell/lower shell	Cover/base
First-stage shell	Impulse chamber
First-stage bowl	High-pressure nozzle chamber
Snouts/snout rings	Bell seals

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