Case 1

Steam Turbine Performance Degradation

A private investor-owned power company owns 15 GW of capacity including conventional fossil-fired generation and natural-gas fired combined cycle gas turbine power plants spread throughout the United States. The company competes in several unregulated power markets and takes seriously its ability to provide safe, reliable, low-cost power compared to its competitors while meeting all environmental permit requirements. Quarterly senior management reviews include reports on worker and contractor safety performance, the reliability and efficiency of the facilities, as well as any exceedances of environmental permits. The company spent time and resources establishing guidelines and procedures for regular performance monitoring at its generating facilities, including results analysis. These guidelines are routinely reinforced at every level of the organization with training for new recruits and refresher courses for midlevel management.

The performance-monitoring procedures and guidelines include techniques to analyze the test data based on industry guidelines, particularly ASME PTC Committee (2010) and technical papers from noted industry experts such as Cotton and Schofield (1970). For the company's steam turbines, the condition of the various stages is related to changes in stage pressures at standard conditions knowing how the throttle flow to the machine has changed. The methods are based on the fact that, for a large multistage condensing turbine, all stages, except the first and last, operate with a constant pressure ratio (p_2/p_1.) This allows the general flow equation for flow through a converging-diverging nozzle for stages beyond the first stage to be simplified to equation (1.1)

$$\dot{m} = \Phi \cdot \sqrt{\frac{P}{\upsilon}} \tag{1.1}$$

Case Studies in Mechanical Engineering: Decision Making, Thermodynamics, Fluid Mechanics and Heat Transfer, First Edition. Stuart Sabol.

Companion website: www.wiley.com/go/sabol/mechanical

where

$\dot{m}$, P and υ are the flow rate, absolute pressure and specific volume to the following stage;

Φ is a constant flow function (area).

The flow function Φ includes unit conversions, constants of proportionality, the area of flow, and the coefficient of discharge for the nozzle and blade path. Except for unit conversions it has units of area.

A production engineer at one of the company's coal-fired power plants with three 600 MW subcritical single reheat units has been monitoring the units' performance according to company procedures. In just over 7 months since the last major overhaul one unit has lost 3.4% of its output, and the cycle heat rate has increased 0.6%. Using the guidelines, most of degradation in performance can be explained by changes in the flow-passing capability of the steam turbine and losses in the high-pressure (HP) turbine efficiency.

However, there are changes to characteristics that are not discussed in the corporate standards or the technical papers available in the office. In particular, the intermediate pressure (IP) turbine's extraction temperature has risen noticeably from the expected value. Efforts to explain the symptoms as instrumentation issues have failed. Rather than dismiss or ignore the findings, you, the engineer, are determined to find the cause, its economic value, and to recommend a course of action to address the issue.

1.1 Steam Turbine Types

The variety and application of steam turbines is enormous. It includes the utility tandem compound unit pictured in Figure 1.1, mechanical drives for onshore or marine applications, combined-cycle and single Rankine-cycle units, super critical, single or double reheat units, and nuclear power-plant applications. One way to categorize the various models is by size. Very basically, smaller installations typically serve as variable speed mechanical drives for pumps and compressors. These may be as large as 50 to 75 MW and have inlet conditions up to 750 psi (5.2 MPa) and 700 °F (644 K). Many are located within chemical processing plants or refineries and exhaust into a lower pressure steam header that provides steam for heating, or to drive smaller steam turbines that may exhaust into a surface condenser. The larger varieties will be multistage units with an axial flow exhaust.

Up to about 150 MW, steam turbines typically have an axial flow exhaust with throttle conditions as high as 1500 psi (10 MPa) and 900 °F to 1000 °F (755 K to 810 K). Figure 1.2 shows a drawing of a Siemens axial flow machine. Such turbines may be used in a chemical process plant and have a controlled extraction for process heat or other uses. This size is also common in combined cycle power plants with uncontrolled expansion to the condenser. Occasionally, an axial flow machine will have single reheat as part of the cycle. If it is a condensing cycle, the condenser can be placed on the same elevation as the turbine. Combined cycle units utilize waste heat from a gas turbine to generate steam; thus, steam-turbine extractions for regenerative heating are not employed in a combined cycle. A single Rankine cycle would employ uncontrolled extractions for feedwater heating.

Above approximately 150 MW, the last stage blade (L-0) becomes too long to manufacture and operate reliably. The low pressure (LP) turbine becomes a dual flow design with steam entering the center section and steam traveling in opposing directions to exhaust downward

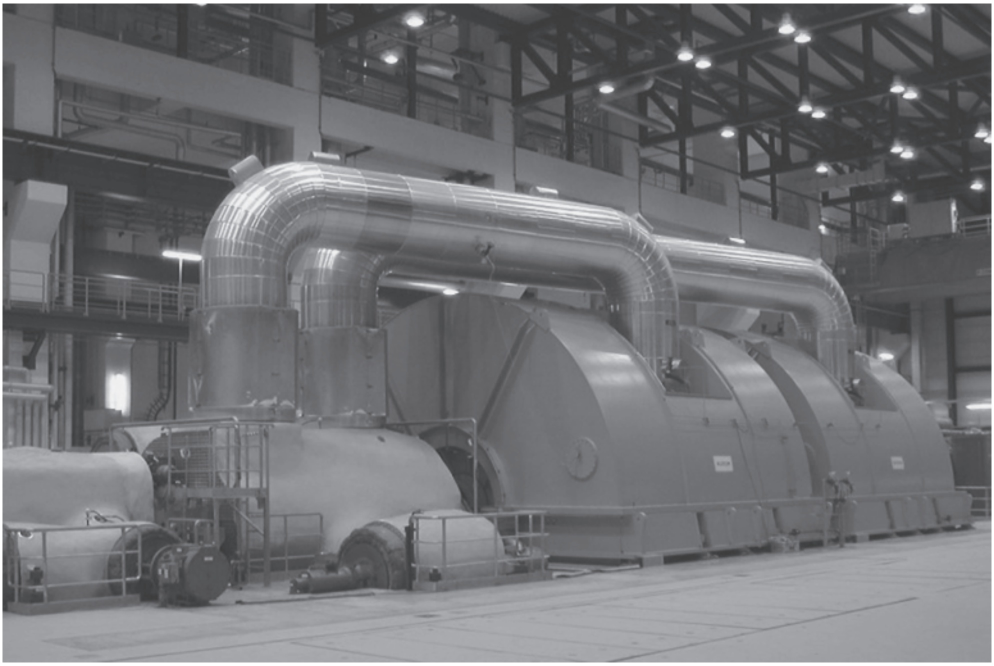

Figure 1.1 Alstom steam turbine. *Source*: Reproduced by permission of Alstom.

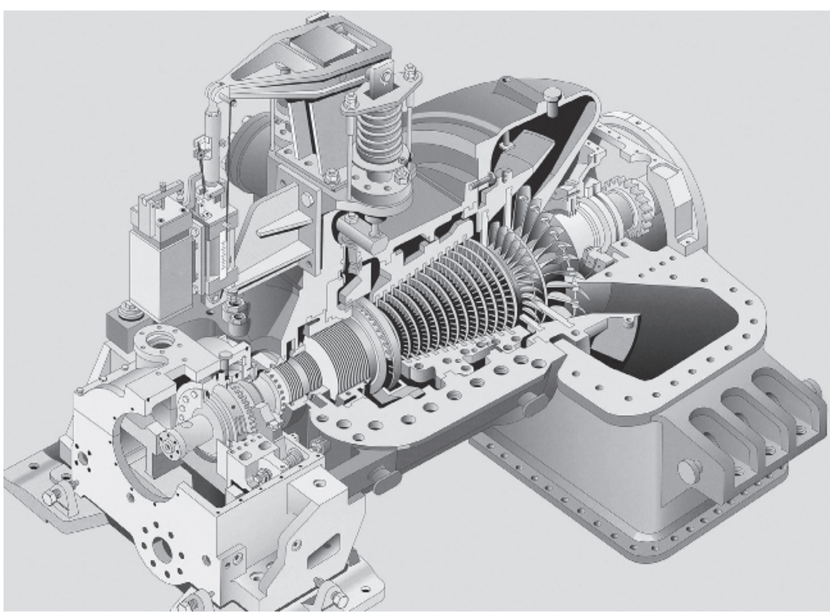

Figure 1.2 Typical axial flow exhaust steam turbine. *Source*: Reproduced by permission of Siemens Energy.

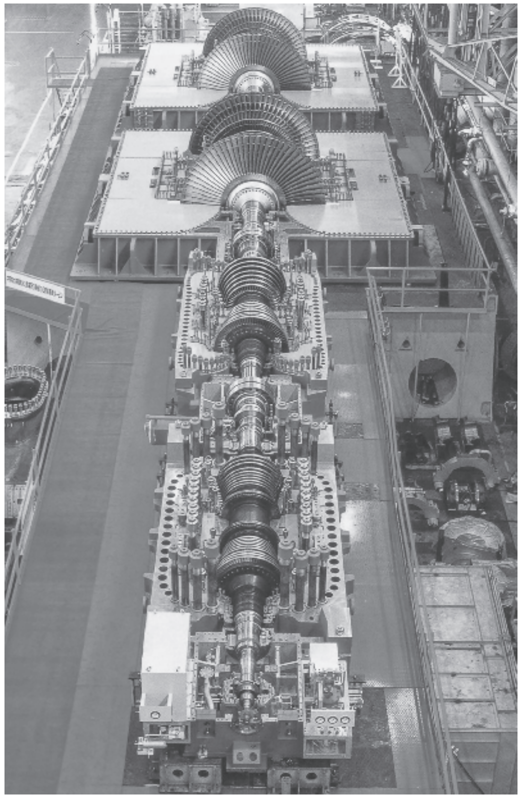

Figure 1.3 700 MW ST Hekinan Unit 3, Chubu Electric Power Co. *Source*: Reproduced by permission of Mitsubishi Hitachi Power Systems America, Inc.

into the condenser. For these machines, the steam turbine must be raised above the condenser, which increases construction costs to include foundations for an elevated turbine. Figure 1.3 is a photograph of the 700 MW ST Hekinan Unit 3, Chubu Electric Power Co. steam turbine. The tandem compound machine has dual flow HP and IP sections in the foreground with two dual-flow LP sections in the background.

Machines as large as 650 to 750 MW usually operate with subcritical steam pressures with a single reheat. Throttle conditions may be as high as 2800 psi (19 MPa) and 1050 °F (840 K) with the reheat temperature matching the throttle temperature. Units in this size range are generally uncontrolled expansion, condensing units used for power generation either in combined cycles or single Rankine cycle units with regenerative heating. The larger single Rankine-cycle units may have two or three dual-flow, down-exhaust LP sections. Combined-cycle steam turbines are limited in size by the gas turbine portion of the combined cycle. As a rule of thumb, the steam portion of the combined cycle plant is about one-third of the plant total electrical output. Most of the single Rankine-cycle units are fossil fired although some may be in nuclear facilities. Combined cycle and fossil-fired units operate at synchronous speed with a two-pole generator. Nuclear units typically have four-pole generators and operate at half synchronous speed.

Above about 650 MW, fossil-fired units begin using supercritical pressures and may include double reheat Rankine cycles with regenerative feedwater heating. Throttle conditions may be above 4000 psi (28 MPa) and 1150 °F (895 K). Reheat temperatures usually match the throttle conditions but cost optimizations may result in the reheat temperatures somewhat above the throttle. Nuclear steam cycle conditions generally do not change much with size. The largest steam turbine at the time of this writing was in the neighborhood of 1800 MW.

1.1.1 Steam Turbine Components

The active components of steam turbines are the rotating and stationary blades. Rotating blades are sometimes referred to as *buckets*, from their shape. Steam to the machine is controlled by multiple throttle valves. In large modern machines there are four hydraulically controlled valves that can close very quickly in the event of an upset. From the control valves, the steam is directed to the first control stage through sets of nozzles. Each set of nozzles accepts steam from one of the inlet throttle valves. The first control or governing stage has impulse or Curtis blading. Beyond the governing stage, the blades take on an increasing amount of reaction, as the pressure diminishes and the pressure ratio increases across each rotating stage.

The rotating blades are mounted on wheels or disks that are fixed to the shaft, or the shaft is machined with integral wheels to accept the blades – see Figure 1.4. The wheels provide increased torque on the shaft. The blades are secured in the wheel by a dovetail or fir tree shaped slot. Each blade is weighted and moment balanced, then ordered so that the assembled rotor is nearly balanced. During assembly of the rotor, the blades are slid into the dovetail slots until the ring is full. The blades are locked in place and the locking mechanism is frequently peened to ensure the security of the blades during operation.

A large steam-turbine generator in a reheat cycle will have a high pressure (HP) rotor, one or more intermediate pressure (IP) rotors, and one or several low pressure (LP) rotors. These may be mounted on a single shaft (tandem compound) or in a dual shaft arrangement with two

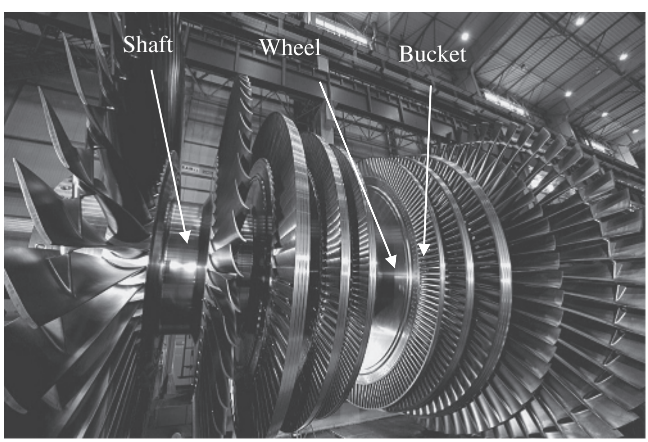

Figure 1.4 An LP section of a large nuclear steam turbine. *Source*: Reproduced by permission of Alstom.

generators (cross compound). Intercept valves are located in the hot reheat line immediately upstream of the IP turbine section. These valves do not control flow but are configured to close quickly in the event of an upset condition preventing the energy stored in the steam lines to and from the boiler reheater from overspeeding the rotor after the generator is disconnected from the electric grid. The pipeline to the boiler from the HP turbine exhaust is referred to as the “cold” reheat line. From the boiler to the IP turbine is the “hot” reheat line.

Once assembled, each rotor is balanced first at slow speed then at high speed. If a field repair requires replacement of worn or damaged blades, the rotor may be removed from the casing and field balanced prior to completing the repair. A trim balance may also be required once the machine is assembled and run at full speed for the first time.

Between rotating blades are stationary vanes, sometimes referred to as nozzles or diaphragms. The blades of the diaphragms are shop assembled in halves between inner and outer blade rings. One half will be fitted in the lower casing and the other half in the top half of the turbine casing. The stationary vanes turn and focus the steam from the exhaust of the upstream rotating blade and direct it to the downstream blade. Each expansion stage is a combination of one set of nozzles and one set of blades.

During assembly of the rotors and casings, sealing strips are mounted to the rotor between the wheels, and in the casing between diaphragms. Mating seal strips are placed on the blade tips, or are made as an integral part of the blade, to match with the casing seals. The inner rings of the diaphragms have seals that match with the rotors. Figure 1.5 shows blade tip seals from US Patent 6926495. The sharp edges are an inefficient flow path reducing leakage between the rotating and stationary components. The sealing components are adjusted during assembly so that there is proper clearance throughout the circumference to prevent the rotating seals from striking the stationary components in the cold and operating conditions.

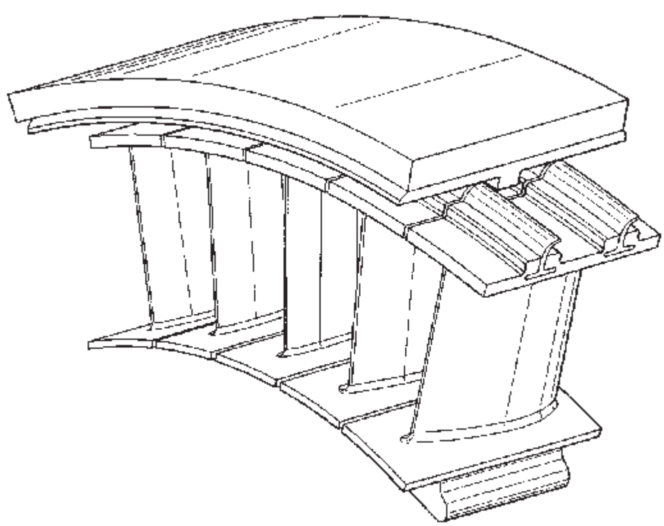

Figure 1.5 Blade tip-seals (US Patent 6926495 Ihor S. Diakunchak). *Source*: Ihor S. Diakunchak, Siemens Westinghouse Power Corporation.

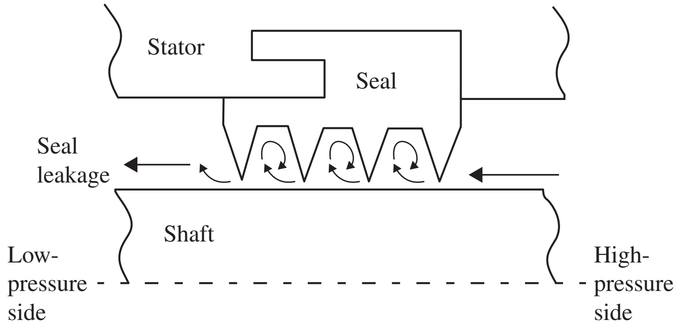

Figure 1.6 Shaft seal leakage. *Source*: Reproduced by permission of Dresser-Rand Company.

The ends of each shaft have a series of labyrinth seals to reduce the amount of high-pressure steam leaking out and air leaking in at the ends of the LP rotors – see Figure 1.6. Very often the HP and IP turbine sections are built on a single shaft and separated by a labyrinth seal between the HP exhaust and IP inlet. High-pressure steam-seal leakage is collected and used as sealing steam in lower pressure sections. It is also injected into the LP shaft seals to prevent air in-leakage.

Once the lower half casing is loaded with diaphragms, seals and bearings; the rotors are lowered into place onto the bearings. The rotors are positioned and aligned with the casing, and mated to the other tandem shafts and the generator rotor. Once all clearances are checked, the upper half bearing halves installed, and the upper half diaphragms and seals in place, the upper casing is lowered onto the bottom casing, and the two casing halves are bolted together. Each bolt is torqued, then heated and torqued again so that it provides the proper tension at operating temperature.

Figure 1.7 shows a tandem compound machine with a dual flow LP casing and the rotor in place inside the lower casing. Stationary vanes in the lower casing can be seen upstream of the latter stages. There are multiple steam extraction points along the flow path.

Steam and auxiliary piping, and instrumentation are assembled prior to insulating the casings and piping segments around the turbine. If there are turbine housings, these are assembled for normal operation.

1.1.2 Startup and Operation

Once assembled, the turning gear motor maintains a slow shaft rotation of about 4 to 5 rpm to prevent bowing. Bowing of the rotor under its own weight during assembly or by differential cooling when the machine is temporarily off line can cause high vibrations during startup. Slow rotation of the shaft prevents differential cooling from bowing the rotor and reduces bowing due to maintenance enough to allow a safe roll up to operating speed. Prior to startup, any water in the casing from steam condensation must be drained to prevent impact damage on the rotating blades.

The high-pressure casing of a large steam turbine can be 10 to 12 inches thick; thus, the rotor will reach operating temperature much faster than the shell. To avoid differential thermal expansion between the rotor and shell, steam turbines must be warmed slowly to prevent

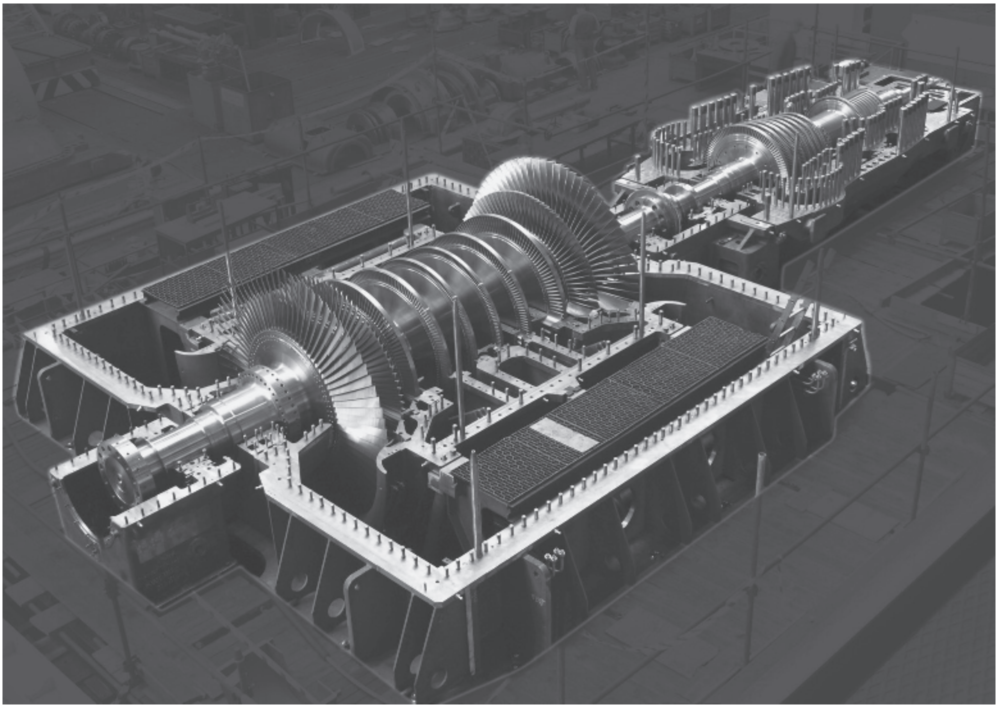

Figure 1.7 Tandem Compound steam turbine with extractions. *Source*: Reproduced by permission of Doosan Skoda Power.

contact between the rotating and stationary components. The manufacturer specifies heating rates and allowable ranges for differential expansion, which are continuously measured and maintained within allowable limits during startup and operation.

In recent years, manufacturers have devoted considerable efforts to decrease the warm-up time for steam turbines in combined cycles. Fast starting units on a single shaft with a gas turbine driving a single generator are less expensive to build and can provide backup generation for renewable power sources. See the web sites below for more information:

https://powergen.gepower.com/services/upgrade-and-life-extension/heavy-duty-gas-turbine-upgrades-f-class/power-flexefficiency.html (accessed January 25, 2016).

http://www.energy.siemens.com/hq/pool/hq/power-generation/power-plants/gas-fired-power-plants/combined-cycle-powerplants/Fast_cycling_and_rapid_start-up_US.pdf (accessed January 13, 2016) (Baling, 2010).

After steam is raised in the boiler, sealing steam provided to the LP shaft seals allows evacuation of air from the condenser. Pulling vacuum permits introduction of steam through the throttle valves, disengaging the turning gear, and accelerating the shaft up to rated speed. During the rollup, condensing steam from contact with cool turbine parts, and that which condenses through the expansion path, must be continually drained from the casing to avoid damage that can be caused by sudden vaporization or impact with rotating components.

There will also be several critical speeds during rollup when the shaft resonates, causing spikes in vibration. These speeds are anticipated, and the operator will accelerate the shaft through the critical speeds rather quickly. If vibrations exceed allowable limits, the startup is aborted and the shaft allowed to coast down before being placed on turning gear. If this happens, the unit may remain on turning gear for a manufacturer-recommended period prior to a second attempt to start. If the situation warrants, a trim balance weight may be calculated and placed on the shaft.

As the shaft rotation approaches full speed, the operator will begin preparations to engage the driven equipment or place the process in normal operation. In the case of a large steam turbine generator, the operator will prepare to close the generator breaker. Once the breaker is closed, load on the generator is rapidly increased by increasing steam flow, pressure and temperature through the throttle valves. At specified temperatures, the casing drains are closed, and the unit begins normal operations. Normal operation may continue without interruption for years!

During normal operations, the staff must diligently maintain feedwater and steam purity. Several elements or compounds can vaporize at subcritical boiler pressures, most notably silica. Silica present in sufficient concentration will vaporize in the HP boiler drum and condense in the HP turbine as the steam temperature drops through expansion. Copper can vaporize as well and carry through to the HP turbine. Spray water used to control main steam and reheat temperatures will also inject whatever is in the water directly into the downstream turbine sections.

In supercritical units, there is no boiler drum; thus feedwater quality is more critical. Anything in the water will be carried directly into the HP turbine section.

Reading

Read Cofer *et al.* (1996). To test your understanding of this article, answer the following questions. An answer key is provided at the end of this case study.

1. What components of the blade path design account for up to 90% of the efficiency losses?
   (a) Aerodynamic design and tip leakage.
   (b) Rotational losses.
   (c) Shaft packing leakage.
   (d) Inlet nozzle losses.
2. What is a logical first step in the design of an improved blade path?
   (a) Laboratory wind-tunnel tests.
   (b) Computational fluid dynamic calculations.
   (c) Full-scale wind-tunnel tests.
   (d) Customer field testing in commercial applications.
3. Why is the last stage bucket design the most important contributor to performance?
   (a) From the First Law: work is equal to $\int P\,dv$.
   (b) The last stage bucket operates in the wet region.
   (c) The last stage bucket produces 10% to 15% of the steam turbine output.
   (d) Answers a and c above.

1.1.3 Performance Monitoring and Analysis

Due to the large pressure drop from the throttle to the condenser (1.7 to 12 kPa) the general equation for flow through a converging-diverging nozzle can be simplified considerably. The general flow equation for subsonic nozzle flow requires knowledge of the upstream and downstream pressures. For a large steam turbine with uncontrolled expansion, except for the first and last stages, the pressure ratio across the stages is constant throughout the load range. Therefore, the downstream pressure is proportional to the upstream pressure, and the general flow equation can be reduced to equation (1.1).

The first stage of the turbine is affected considerably by changes in flow, and the last stage is affected by flow and condenser pressure.

With the simplified flow equation, much can be determined about the condition of a turbine by monitoring feedwater flow, and pressures and temperatures into and out of the various turbine sections (HP, IP and LP inlet). The HP turbine first-stage pressure is measured downstream of the governing row or rows.

Not long ago, scheduled and carefully controlled performance tests with precision instruments were the best way of monitoring steam-turbine performance. Thermocouples calibrated against standards in precision furnaces measured with precision potentiometers and dead weight gages for pressure measurement were the instruments of choice. With the advances in digital computing and in temperature and pressure transmitters, online monitoring has become the norm. Regular calibrations of the transmitters and use of transmitters that self-correct for ambient temperature can yield sufficiently accurate results. Automated collection of instrument readings and computer storage systems retain data for long periods permitting research into performance trends that can help determine the cause or causes of performance deterioration. Furthermore, the data are available at the site, in a corporate office or easily made available to consultants. Acceptance tests can still require special test instrumentation, but even these are more often conducted with mostly plant instrumentation calibrated just prior to the test.

1.1.4 Analyzing Performance Data – Corrected Pressures

With the data, there are two basic methods for analyzing performance trends. The first relies on corrected stage pressures with knowledge of changes in throttle flow. This method requires the engineer to correct the measured output and measured pressures to standard conditions. Manufacturer curves provide the means to correct the unit output to reference or design conditions. The first-stage pressure correction is shown in equation (1.2) from Cotton and Schofield (1970):

$$P_C = P_O \cdot \left(\frac{P_{dt}}{P_{Ot}} \right) \tag{1.2}$$

Extraction pressures, if present in the HP expansion path, may be corrected with the same correction without loss of accuracy.

Pressures beginning with the Hot Reheat Intercept Valve and downstream are corrected with equation (1.3), again from Cotton and Schofield (1970):

$$P_C = P_O \sqrt{\frac{P_{dt}}{P_{Ot}} \cdot \frac{\upsilon_{Ot}}{\upsilon_{dt}}} \sqrt{\frac{\upsilon_{dr}}{\upsilon_{tr}}} \tag{1.3}$$

where:

P_C = corrected pressure (absolute);
P_O = observed pressure (absolute);
P_{Ot} = observed throttle pressure (absolute);
P_{dt} = reference throttle pressure (absolute);
υ_{dt} = reference throttle specific volume;
υ_{Ot} = observed throttle specific volume;
υ_{tr} = test specific volume at the intercept valves; and
υ_{dr} = reference specific volume at intercept valves.

Use of corrected pressures to analyze the condition of the steam turbine is best explained by example. In a particular case, the following changes in observed parameters occurred over a 22 month period:

Example 1.1

Throttle flow −17.2%
Output −16.5%
First-stage pressure +21.2%
High-pressure efficiency −12.2% (Cotton and Schofield, 1970)

The lower throttle flow together with the high first-stage pressure indicates loss of ability to pass steam in the second or latter stages of the HP turbine. As the degradation in performance occurred over a period of 22 months, the cause was suspected to be deposits rather than mechanical damage or components that became lodged on blades or vanes. The inspection revealed heavy deposits throughout the HP turbine.

When throttle flow is not available, analysis is a bit more complicated. However, logical analysis of the data generally yields accurate interpretations. For example, again from Cotton and Schofield (1970), the following data from a turbine efficiency test showed significant changes from a previous test eight months earlier.

Example 1.2

Output −2% (approximate)
First-stage pressure +0.3%
Hot-reheat pressure +6.3%
Low-pressure inlet +2.2%
HP turbine efficiency −1.3%
IP turbine efficiency −6.9%

The large increase in the hot-reheat pressure without a corresponding increase at the first stage indicated that there was a flow restriction at the intercept valves or downstream in the IP turbine. Some plugging might be indicated in the LP turbine as well. The intercept valves were eliminated by a quick measurement of the pressure loss across the valve; therefore, pluggage in the IP turbine was suspected. Deposits in the IP turbine but not in the HP turbine would be possible, but not likely. Since the first-stage pressure changed very little, mechanical damage in the IP turbine was suspected.

Upon inspection, distortion of the IP casing was found, which caused severe rubbing at the ninth stage, which liberated blade covers, and resulted in failure of ninth stage blades. The failed blades further damaged the ninth-stage vanes, which were removed from the diaphragm, and caught between the rotating blades and the vanes of the ninth stage. The distortion was determined to be caused by water induction into the operating unit that suddenly cooled a portion of the shell.

Example 1.3: Check Knowledge
What do the following symptoms indicate from the corrected conditions shown below?

Power output −7%
First-stage pressure +2%
Hot-reheat pressure −3%
LP inlet pressure −9%
HP turbine efficiency −7%
IP turbine efficiency −1.5% (Cotton and Schofield, 1970)

Analysis
The rise in first-stage pressure indicates either an increase in flow or decreased flow-passing ability in the remainder of the HP turbine. If flow had increased, due to increased first-stage area, the increased flow would have been expressed throughout the remainder of the flow path. The IP and LP inlet pressures decreased, so an increase in flow is eliminated.

Therefore, the most probable cause is some type of blockage in the second and subsequent stages of the HP turbine. Since the LP inlet pressure fell substantially more than the IP inlet, there is likely additional blockage in the IP turbine, which is supported by the drop in IP turbine efficiency.

The information in Example 3 does not specify the time over which the changes took place. Nevertheless, there should be some common connection between the simultaneous changes in the HP and IP turbine performance. This could be due to deposits from the steam or another mechanism, such as scale that abrades the blades and vanes of the HP and IP turbines. Deposits generally lead to increased stage pressures. Only the first stage pressure increased here, so a mechanism of mechanical abrasion or mechanical damage to the blades should be suspected. Inspection of the machine showed that solid particle erosion had removed considerable material from the blades in the HP and IP turbines. Figure 1.8 shows an example of vanes heavily eroded by solid particle abrasion.

1.1.5 Analyzing Performance Data – Flow Function

The second method of analyzing the turbine performance is to go directly to a calculation of Φ, a method that does not require corrections to standard conditions. Instead thermodynamic states are determined at the inlets to main turbine sections, and an energy and mass balance determines flows throughout the cycle. Changes in the value of Φ imply physical changes inside the turbine such as flow area, surface condition, or shape of the blades in the following

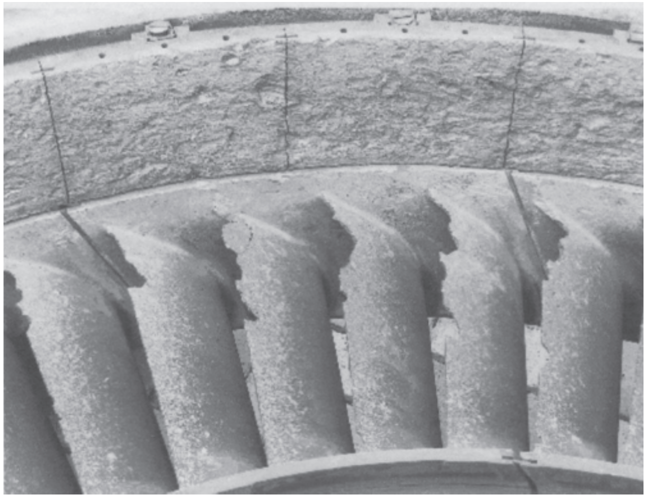

Figure 1.8 Solid particle erosion. *Source*: Wilcox *et al.* (2010).

stages. If deposits clog the area, Φ will decrease. If the blade surface becomes rough, if a turbine component or foreign object blocks the flow path, or if mechanical damage changes the shape of the blades or vanes, Φ will decrease. An increase of the downstream area due to erosion or mechanical damage will show an increase in the value of Φ.

Example 1.4

Example 1.1 above, suppose the steam turbine has the following design conditions:

Turbine inlet condition:
pressure (psia / MPaa): 3514.7 / 24.233
throttle temperature (°F /K): 1050 / 838.7
Governing stage exit condition:
first-stage pressure (psia / MPaa): 2612.1 / 18.01
first-stage enthalpy (Btu/lb / kJ/kg): 1431 / 3329

At these conditions, the apparent governing stage efficiency from the throttle valves is 72%. The specific volume at the first stage is 0.281 ft^3/lb (0.0175 m^3/kg). From the data in the example, the first-stage pressure increases 21.2% to 3151 psig (21.725 MPa). Assuming the first-stage efficiency falls at the same rate as the overall HP turbine efficiency yields enthalpy and specific volume values of 1451 Btu/lb (3375 kJ/kg), and 0.2397 ft^3/lb (0.0150 m^3/kg), respectively, at the corrected test conditions. This assumption is not necessarily accurate but is a fair approximation.

A change in the flow function can be calculated from equation (1.4):

$$\frac{\Phi_{2O}}{\Phi_{2d}} = \sqrt{\frac{P_{1d}}{\upsilon_{1d}}}\left(\frac{\dot{m}_{1t}}{\dot{m}_{1d}}\sqrt{\frac{\upsilon_{1O}}{P_{1t}}}\right) \tag{1.4}$$

where:

Φ_{2d} = design second-stage flow function;
Φ_{2O} = observed second-stage flow function;
$\dot{m}_{1d}$ = design flow to the first stage;*
$\dot{m}_{1t}$ = observed flow to the first stage;*
P_{1d} = design first-stage pressure (abs);
P_{1O} = observed first-stage pressure (abs);
υ_{1d} = design first-stage specific volume; and
υ_{1O} = observed first-stage specific volume.

For this example, the ratio of test to design flow function to the stage following the governing stage is 0.694 indicating a diminished flow passing ability of second or subsequent stages of the HP turbine. As with the original analysis, the steady decline over a 22-month period would indicate deposits on the blades of the second or subsequent stages of the HP turbine.

The flow function at the HP turbine inlet (at the throttle valves), in this example, changes in the same proportion as the throttle flow (−17.2%). A calculation of Φ at the first stage exhaust shows that there is a progression of the deposit buildup through the steam path. In other words, the deposits should be expected to become greater and greater as the steam passes through the HP turbine.

Similar analysis is possible at the IP and LP turbines provided there is energy balance information available showing the flows to the following stages.

1.2 Refresher

1.2.1 Steam Turbine Efficiency

Steam turbine efficiency is calculated from the ratio of the actual change in enthalpy between the inlet and outlet to the isentropic change in enthalpy between the same two states. Turbines operating in the superheat region, dry steam, have the thermodynamic states determined by measured pressure and temperature. In the wet region, the thermodynamic states must be determined by heat balance or another means of determining the thermodynamic state. When calculating turbine section efficiency, the inlet thermodynamic state is usually downstream of the throttle or intercept valve, if there is one.

As an example, consider an HP turbine with throttle conditions of 31.02 MPa, and 866.5 K, exhaust conditions of 7.76 MPA and 650.8 K and a 4% pressure drop through the throttle values. The ideal and actual expansions are shown in the Mollier diagram of Figure 1.9. The section efficiency is 86% between an inlet enthalpy of 3416 kJ/kg and an exit of 3081 kJ/kg.

1.2.2 Example

Determine the exhaust temperature of an HP turbine section with throttle conditions of 16.68 MPa, 810.9 K, a 4% throttle valve pressure drop, an exhaust pressure of 4.97 MPa, and a section efficiency of 84%.

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*Throttle valve stem leakage and seal leakage from the first-stage yields a flow to the second and subsequent stages of the HP turbine that is less than the throttle flow. For the examples in this case study, throttle flow is used as a reasonable approximation.

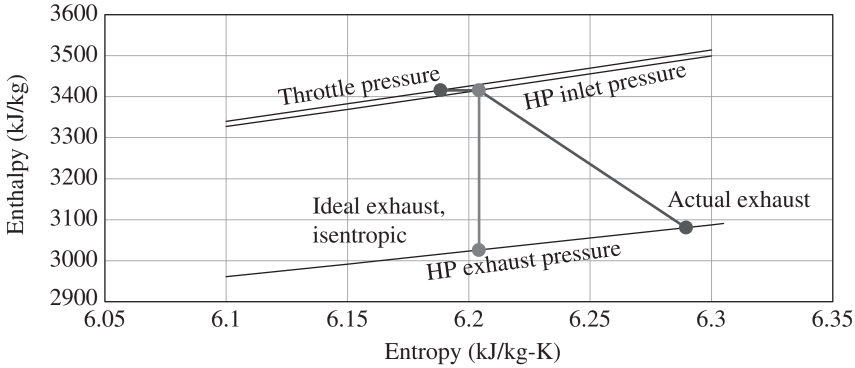

Figure 1.9 Example: HP expansion.

Solution:

Throttle enthalpy: 3398.2 kJ/kg
HP inlet pressure: 16.01 MPa
HP inlet entropy: 6.432 kJ/kg-K

Isentropic exhaust enthalpy: 3077.5 kJ/kg
Exhaust enthalpy: 3110.4 kJ/kg
Exhaust temperature: 638.7 K

1.3 Case Study Details

A nominal 600 MW fossil-fired subcritical, single reheat steam power plant has seven stages of regenerative feedwater heating. All extractions are uncontrolled, governed only by the heat transfer to the feedwater. The first steam extraction point (in the direction of steam flow) is taken from the HP turbine exhaust, the cold reheat line. The second is taken from an extraction port about midway through the IP turbine and the third at the IP turbine exhaust. The remaining four extractions are taken from extraction ports in the two identical dual-flow, downward exhaust LP turbines, which exhaust to the condenser at about 1.5 ″Hga (5.08 kPa) at design conditions.

1.3.1 Performance Trend

In a little over 7 months, load has dropped 3.4%. Regular turbine performance tests on the unit with the throttle valves wide open (VWO) show the following changes after seven months:

Throttle flow: −6.31%
First-stage pressure: +3.04%
HP turbine efficiency: −1.25%
IP turbine efficiency: slightly less than expected

A performance test after 7 months showed the uncorrected pressures and temperatures given in Table 1.1.

Table 1.1 Turbine performance data – Part 1.

	Design	Test
Throttle flow (kpph / kg/s)	3641 / 458.8	3423 / 431.3
Barometric pressure (psia / kPaa)	14.7 / 101	14.7 / 101
Throttle pressure (psig / MPag)	2400 / 16.547	2405 / 16.582
Throttle temperature (°F / K)	1000 / 810.9	998 / 809.8
First-stage pressure (psig / MPag)	1795 / 12.378	1854.2 / 12.784
First-stage enthalpy (Btu/lb / kJ/kg)	1431.5 / 3329.7	
HP exhaust pressure (psig / MPag)	706.8 / 4.873	672.7 / 4.638
HP exhaust temperature (°F / K)	682.2 / 634.4	671.6 / 628.5
Hot reheat flow (kpph / kg/s)	3304 / 416.3	3142 / 335.8*
Hot reheat pressure (psig / MPag)	634.6 / 4.376	603.9 / 4.164
Hot reheat temperature (°F / K)	1000 / 810.9	1003 / 812.6

Assignment 1
From energy and mass balance calculations following the test:

1. Calculate the corrected first-stage and hot-reheat pressures and determine a likely scenario for the lost output.
2. Calculate the first-stage thermodynamic state at test conditions. Assume the HP turbine section efficiency from the first stage to the exhaust degrades proportional to the degradation of the overall HP turbine efficiency.
3. Use the design and measured conditions to calculate the items below for design and test conditions:
   (a) Turbine inlet flow function.
   (b) First-stage turbine flow function.
   (c) IP turbine inlet flow function.
4. What do the parameters from step 3 above indicate about the condition of the HP and IP turbines?

1.3.2 IP Turbine Enthalpy Drop

In addition to the parameters shown above, Table 1.2 provides data to determine the IP turbine enthalpy drop.

Assignment 2

1. Plot the IP turbine expansion line on a Mollier (h-s) diagram. Use a 2% pressure loss through the intercept valves and a 3% pressure loss from inside the turbine shell to the extraction pressure measurement.
2. Calculate the overall IP turbine efficiency.
3. Calculate the two IP turbine section efficiencies – from the Hot Reheat to the second extraction and from the second extraction to the IP Turbine Exhaust.
4. Are the two IP section efficiencies from 3. above reasonable?

Table 1.2 Turbine performance data – Part 2.

	Pressure (psig / MPag)	Temperature (°F / K)
Hot reheat	603.9 / 4.164	1003 / 812.6
Second extraction	369.8 / 2.550	902.4 / 756.7
IP-LP crossover	211.6 / 1.459	745.4 / 669.5
	Design (kpph / kg/s)	Test (kpph / kg/s)
Expected extraction flow rate	140 / 17.6	131 / 16.5

5. The IP turbine expansion line is usually drawn as a straight line on the Mollier diagram from the point downstream of the intercept valves to the crossover. Calculate the temperature difference between the expected and measured extraction temperatures.
6. What are the most likely components or operating parameters that lead to the high extraction temperature?

Read Pastrana *et al.* (2001).

1.4 Case Study Findings

The observed difference between the expected and measured extraction temperatures were quickly verified as accurate by comparing the measured extraction temperature at the turbine extraction port to the temperature measured downstream at the feedwater heater. The test thermocouple was checked against another test thermocouple, both measured with a laboratory potentiometer. Given the slight change in IP turbine efficiency, liberation of parts was discounted. Therefore, the high extraction temperature was most likely due to seal leakage between the casing and rotating blades.

The machine in question was occasionally washed to remove suspected deposits on the HP turbine blades. Washes were conducted under low load conditions with the main steam pressure reduced and the main steam temperature controlled to just above saturation conditions via spray water. As the steam expanded through the HP turbine, the latter sections would operate in the wet region and much of the deposit could be removed. Casing drains carried the condensate and deposits to waste.

As the steam condensed during the short wash period, the rotor could cool much faster than the shell under condensing heat transfer causing high-differential expansion / contraction between the shell and rotor. While operators watched the parameters very closely, differential expansion resulting in contact of the sealing surfaces was a likely cause for the high seal leakage after just a few months' operation. Other causes may have included: normal wear and tear, high vibration during a startup or wash, or improper installation during the last major overhaul. Normal wear and tear was eliminated given the short operating period during which the seal leakage increased. Therefore, the three most probable causes for the seal leakage were high differential expansion, high vibrations, and improper installation.

Measurements through the LP turbine expansion combined with offline computer-based heat and material balances of the full steam cycle showed that the LP turbine suffered similar differences between the expected and measured extraction temperatures. The temperature differences were not systematic; that is, they did not progress in any pattern through the machine and were often negligible. Therefore, installation errors may have been the initial cause for the high seal leakage but this could not be determined at the time.

As identified earlier, the turbine inlet flow function changed proportionally to the change in throttle flow. This indicated that the deposits were present on the first-stage nozzles and blades. Given that the steam to the HP turbine was slightly superheated for the washes, the first stage did not receive the full benefit of wet steam washing. Therefore, the turbine washes were less and less effective leaving the company little choice but to remove the unit from service for a complete overhaul.

Inspection revealed heavy brown deposits throughout the HP turbine. Analysis of the deposits showed a high concentration of copper, which was the material of choice, at the time the unit was constructed, for heat exchanger tubes within the condenser and low pressure feedwater heaters. Highly pure condensate slowly dissolved the copper tubes and transferred the ions to the boiler where they vaporized with the steam. While minute quantities were dissolved from each tube, the great number of tubes and the effect of boiling led to concentrations in the boiler drum high enough to vaporize copper, which deposited in the turbine. Most of the copper was removed from the system in the HP turbine with little effect on the IP or LP turbines.

When the machine was removed from service several years later for its major overhaul, the IP turbine seals were found to be worn, and there was evidence of a rub over a portion of the arc.

1.5 Decision Making and Actions

1.5.1 Value

Prior to the machine inspection, there were several unanswered questions and uncertainties:

- Was the damage to the seals caused by high vibration or differential expansion?
- Was there an installation error?
- What is the value of the seal leakage?
- How can it be prevented in the future?
- What should be done now and who should do it?

In general, these types of questions appear over and over during a career. Basically these can be summarized below:

- Do uncertainties or unknowns need to be eliminated or reduced in order to make a decision?
- Is a decision necessary now?
- How does the condition affect the company’s bottom line?
- Is there value in making a change?
- How should the change be implemented?

The first step in the process of answering these types of questions is to identify the value of the loss or losses. Perfection is generally not required – a Pareto analysis (80 : 20) may often be enough. Once determined, the value will help establish how strongly to pursue corrective actions. It will also help prioritize your own work schedule.

Constant pressure lines have just a slight upward curvature on the Mollier diagram in the superheat region, so they can be very accurately approximated by straight lines over small distances. With this assumption, the expected turbine shell enthalpy at the extraction point can be located by the intersection of two straight lines – one the line drawn from the IP inlet downstream of the intercept valve to the IP turbine exhaust, and the other a line connecting the entropy / enthalpy points along a constant pressure line at the turbine extraction shell pressure. The hot reheat entropy and the turbine exhaust entropy are usually close enough to one another for an accurate calculation of the extraction shell entropy and enthalpy. The intersection of these two lines yields an enthalpy generally less than 0.1 Btu/lb (0.2 kJ/kg) from the true intersection of the constant pressure line and the turbine expansion line. Using the foregoing assumption, just a few steam table calls in the turbine monitoring program allow a fast and efficient method of monitoring the expected and measured temperatures continuously at the second extraction point.

Assignment 3

1. With the above approximation, develop a simple model of the leakage around the turbine section and calculate the flow and lost output due to the leakage.
2. For the following inputs, calculate the present value of the identified seal leakage assuming the second segment of the IP turbine leaks at the same rate as the first:
   (a) Wholesale power price: $50/MWh.
   (b) Operating capacity factor of 96%.
   (c) Discount rate of 11%.
   (d) Period between major overhauls of five years.

1.5.2 Decision Making and Actions – Alternatives

Once a monetary value is determined, the next step in a resolution process would be to identify actions that could be taken to either correct a situation or reasons to delay a decision until a future event makes corrective actions possible. Immediate corrective actions are not warranted every time that issues are identified. In the case of maintenance on a large steam turbine, overhaul expense plus the value of lost production during maintenance easily overwhelm the seal steam-leakage losses. Since the lost output due to seal leakage is minimal, with a relatively small cost, a maintenance decision can be postponed allowing more time to evaluate alternatives and reduce uncertainties.

Reaching a decision to do nothing or postpone a decision requires the same level of involvement and analysis as decisions to take immediate action. Both courses of action are equally as valuable to an owner. If an engineer is employed directly by the owner, or has a position with an outside contractor or consultant, helping the management team reach a decision with clear presentations of results is a value-generating proposition for the engineer.

Though the losses related to steam seal leakage in this case were small compared to the cost of an overhaul, there were several actions that the owner could have pursued to help resolve the uncertainties identified above. First, an in-office review of vibration and differential expansion data from the plant's computer historian could identify if a single event led to contact between the rotating and stationary components. Second a monitoring program could be implemented to observe the difference between measured and expected extraction temperatures or enthalpies between major overhauls.

The third option, or task, would be to review the last outage report. The field service engineer should have recorded "as-found" and "as-left" seal clearances, and these would indicate the quality of the work performed, and condition of the unit following the last overhaul. These findings might indicate that closer supervision of the contractor would be justified. Knowing the cost of the seal leakages could support hiring an outside expert during reassembly of the turbine to help avoid installation problems and future performance losses.

The fourth option would be for the owner to consider advanced designs to improve long-term performance. Again, knowing the value of the losses would help determine if the advance technology parts would be of value and improve the company's returns to shareholders.

1.5.3 Decision Making and Actions – Making a Plan

Knowledge of issues with a thorough analysis of their cost and potential causes is the first step toward improving performance. The next step is to develop a plan, and persuade management that changes in behavior are justified and necessary. At times there are actions within an engineer's control to correct a problem. In most cases, however, resources or approvals will be necessary prior to completing the actions. Even if an individual could easily change a computer program or drawing, change control and QA/QC protocols might require reviews, checks, and approvals. The company's authority limitations, policies, and procedures should be consulted prior to taking individual action. Likewise, within a consulting firm, quality checks and peer reviews need to be completed prior to finalizing a recommendation to the owner or purchaser of services.

Assignment 4

Consider the foregoing and create a short presentation that includes the following related to seal leakage:

- What was found from the testing?
- The value of the lost production due to seal leakage.
- The most probably causes for the leakage.
- Outline a recommended plan to reduce seal-leakage losses in the future.
- Identify resources that would be necessary to carry out the recommended action plan.

1.6 Closure

This case study addresses technical and nontechnical aspects of engineering. On the technical side, seal leakage in an axial flow turbine was observed, corporate engineering standards were improved, and plans were laid for long-term improvements to performance with advanced technology parts.

On the nontechnical side, the case study was a situation that was outside normal practices, giving the engineer an opportunity to demonstrate independence and the ability to work without supervision. As an engineer, take advantage of these opportunities. When you become a supervisor, seek out and reward those who volunteer and who you can depend to work without close supervision. These are the individuals from whom you will learn the most, and who will provide you the opportunity to leverage your position of authority to improve the company's competitiveness.

All rotating equipment that is operated by or operates on a working fluid has leakage between rotating and stationary components. On rare occasions, the leakage can be observed and monitored with common plant instrumentation. This is not always the case, but future developments may provide additional tools to engineers and manufacturers of pumps, compressors, fans, and turbines that could improve their energy efficiency.

Standards and practices are everywhere in engineering. Standards organizations regularly update industry codes; but that is not always the case with practices or guidelines developed within a company. Engineers should be alert to cases in which observations do not necessarily agree with practices in their company manuals. Occasionally take time to research new developments. Read related articles in trade journals and attend conferences sponsored by standards organizations to stay current with your field of engineering. The improvement in corporate standard practices provides new information that could assist with improvements in operating procedures and prevention of future efficiency losses.

When presenting information to management, be clear on the objectives, and use the time you have wisely. If the meeting is for information only, you might expect new assignments on items you have not anticipated. Therefore, think through the possible outcomes before the meeting. If you make a request for help solving a problem or for resources, be specific on the request and show you have a plan to bring the matter to a conclusion. Listen to recommendations and be eager to take on new responsibilities.

1.7 Symbols and Abbreviations

$\dot{m}$:	mass flow
Φ:	flow function of a steam turbine stage from equation (1.1)
P:	pressure
v:	specific volume
″Hga:	inches of mercury, absolute pressure

Subscripts

1:	1st stage conditions
C:	Corrected
O:	observed
t:	throttle
d:	design or reference
r:	reheat intercept valves

1.8 Answer Key

Section 1.1.2

1. (a)
2. (b)
3. (d)

Section 1.3.1, Assignment 1

1. Corrected first-stage pressure = 1865.0 psia (12.859 MPa); +3.04%. Corrected hot-reheat pressure = 600.7 psia (4.142 MPa); −8.27%. Lower throttle flow indicates loss of flow passing ability in the governing stage. The combination of lower throttle flow and an increase in first-stage pressure indicates loss of flow function in the second or subsequent stages of the HP turbine. The hot reheat pressure has fallen more than the change in throttle flow, indicating a slight increase in the flow area of the IP turbine or continued degradation of HP turbine stages beyond the first stage. As the changes occurred gradually over 7 months, deposits in the HP turbine are suspected. A small increase in the flow area of the IP turbine may have occurred.
2. Conditions at the first stage: P = 1868.9 (12.886 MPa) psia, T = 930.4 °F (772.3 K) h = 1434.3 Btu/lb (3336.3 kJ/kg), entropy = 1.539 Btu/lb-R (3.579 kJ/kg-K), specific volume = 0.396 ft^3/lb (0.025 m^3/kg). Conditions at the first stage are solved by trial and error to achieve the desired second HP section turbine efficiency.
3. Flow functions are given in Table 1.3.
4. The conclusions from 1. above are supported by the flow functions – suspected deposits in the HP turbine beginning at the governing stage. Since the change in flow passing ability to the HP turbine second stage is greater than at the throttle, deposits are suspected to be heavier in the later stages of the turbine.

Section 1.3.2, Assignment 2

1. See Figure 1.10. The second extraction measurements show an elevated temperature.
2. Overall IP turbine efficiency: 88.38%.
3. IP section 1: 73.76%, IP section 2: 98.16%.
4. The efficiencies of 3. above are not reasonable. The case study states that the IP efficiency changed very little. A 10% improvement in the second section to near ideal efficiency is unrealistic. If the first section dropped by 15%, the second section would have a similar drop.
5. Expected turbine shell temperature = 887.3 °F (475.2 K). ΔT = 15.1 °F (8.4 K).
6. The most likely cause for the high extraction temperature is blade-tip leakage between rotating and stationary components. Causes for the increased leakage include mechanical damage to the seals caused by high vibration, or differential expansion between the rotor and shell, or an installation error. The seals should be expected to perform

Table 1.3 Turbine flow function results.

	Design customary (SI)	Test customary (SI)	Change (%)
Turbine inlet	41 891 (15.882)	39 254 (14.883)	−6.3
First stage	54 615 (20.707)	49 831 (18.893)	−8.76
Hot reheat	147 878 (56.066)	147 870 (56.062)	−0.01

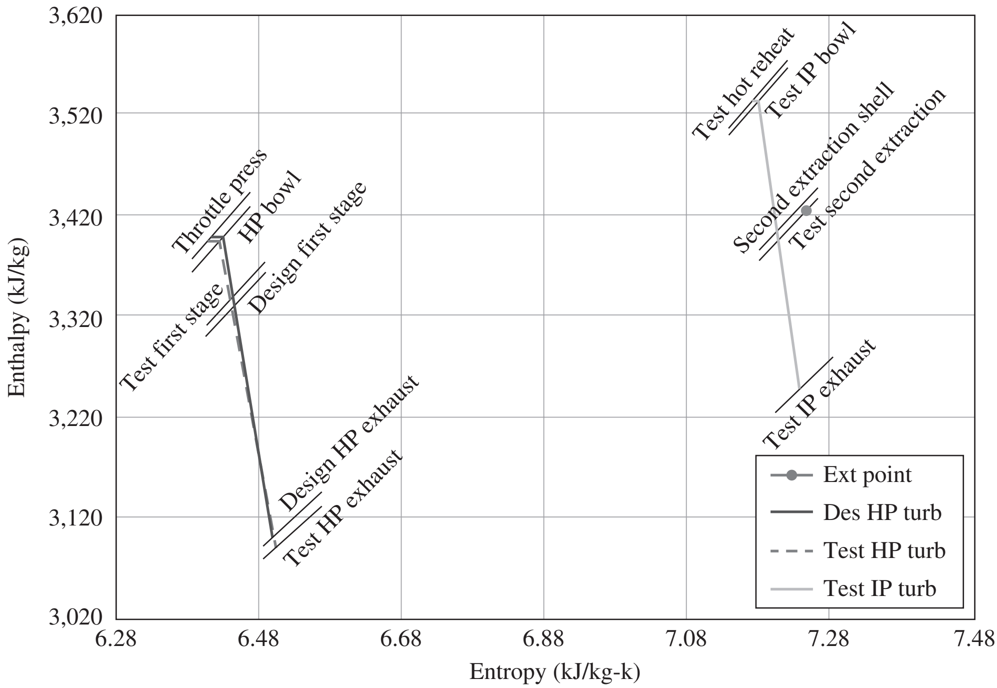

Figure 1.10 Answer Assignment 2, number 1.

reasonably well over a 5-year period between major overhauls supporting the conclusion of mechanical damage or an installation error.

Section 1.5.1, Assignment 3

1. A model similar to the one in Figure 1.11 is expected. Extraction is taken from the periphery of the rotating blades through slots in the turbine casing. Therefore, tip leakage can have a pronounced effect on observed extraction temperature. Leakage flow per section from the model is 20.0 kpph (2.51 kg/s). The lost output from the two sections is 0.72 MW.
2. $1.1 million present value.

Section 1.5.3, Assignment 4

The students should determine if the report is for information or to present a recommendation. If there is a recommendation, it should be accompanied by a request and a plan to implement what has been requested.

The lost output from the seal leakage is far too small to recommend an immediate overhaul. The lost output from the overhaul would overwhelm the 0.72 MW lost from seal leakage. Since something new was discovered that was not included in the company's monitoring program, a recommendation to include a new calculated parameter, shell to extraction temperature difference for example, would be reasonable. An upgrade to better seals would be valuable if the cost of the seals were less than the losses identified, and this too would be a reasonable recommendation.

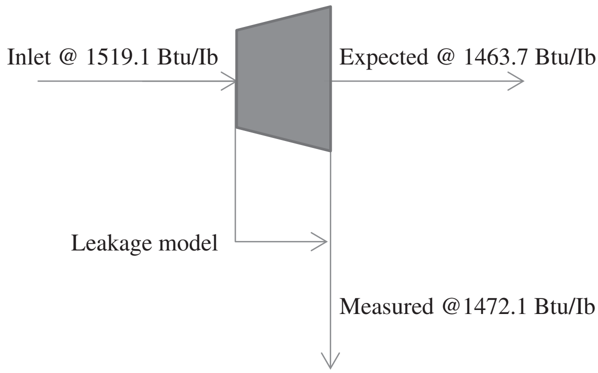

Figure 1.11 Answer: Assignment 3, number 1.

Improved monitoring of the turbine service provider during maintenance outages would also be a reasonable recommendation. The students may estimate the costs for a consultant and compare that to the value of the lost output as a way to improve their scores on the report. This could require some extra research to determine an appropriate hourly or daily rate to pay the consultant, estimates for travel, and so forth.

The recommendations should be accompanied by a request for resources. In the monitoring case, the engineer can initiate a peer review or similar exercise. Requesting sponsorship from a supervisor would be a helpful addition and could from the basis for the report. A product upgrade suggested by the second outside reading assignment would also be a reasonable recommendation that would require additional resources at the time of the next overhaul. Suggested reviews of actual installations of the newly designed components, which require resources, would be helpful support for a management decision.

An effort to spend additional time studying records, the vibration history for example, should not be included as a recommendation unless additional resources are required. Simple studies should be mentioned in an appendix after the main body of the report, or as something that will be done as a matter of course by the engineer. A good follow-up question would be to inquire why the study has not already been completed. Including results of a fictitious study would be reasonable and could be used to support the student's conclusions.

Following the recommendations, the report should include the steps necessary to bring the matter to closure. Mention of a meeting that has already been scheduled with the turbine manufacturer to discuss product upgrades would show that the engineer has already acted independently and that would improve the value of the report. An outline of the monitoring procedure, as well as supporting cost / benefit calculations can be included in an appendix or as a backup to the main report.

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