Steam Turbine Cycle Optimization, Evaluation, and Performance Testing Considerations

James S. Wright
GE Power Systems
Schenectady, NY

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INTRODUCTION

The selection of a thermodynamic cycle for a power plant, which includes steam turbines, requires the definition of a number of design parameters, including steam conditions, turbine configuration, cooling system design, feedwater heating system design and means for accommodating process steam supply and/or admission. Considerable savings in life-cycle cost can be made by careful attention to the optimization of these parameters, and this topic is addressed in the first section of this paper.

In the case of large fossil-fired power generating plants, the plant arrangement, turbine configuration and principal cycle design alternatives available are generally defined by the plant rating. The cycle optimization consists of selecting from among the cycle design alternatives after considering application-specific economic factors. The principal cycle design alternatives available for large fossil-fired plants are discussed and data are provided to facilitate the necessary economic assessments.

Standardizing the plant arrangement and turbine configuration for combined-cycle and cogeneration plants is difficult, due to case-specific variations in key cycle parameters, process steam requirements and expected modes of operation. Consequently, most plants require considerable thermodynamic optimization. In order to facilitate this, the available steam turbine design alternatives (e.g., uncontrolled versus automatic extractions) and key interfaces to other plant components (e.g., condenser, boiler or HRSG, and process steam demand), should be understood and evaluated. Some of the major steam turbine design and interface issues which arise in combined-cycle and cogeneration plant design are described.

The selection of optimum steam-turbine exhaust annulus area and cooling system design is important in the design of all power plants utilizing condensing steam turbines. The key issues are described, together with a suggested optimization approach.

Given defined plant and steam turbine design configurations, the next step for the plant designer is to evaluate specific alternative equipment (e.g., steam turbine) options. The second section of this paper discusses several areas where this can be difficult and provides suggested approaches.

Finally this paper reviews alternatives available for steam turbine performance testing. Different testing procedures are described, together with their associated measurement uncertainties and means for assessing their relative values.

CYCLE OPTIMIZATION AND APPLICATION ISSUES

Fossil-Fired Reheat Units

For reheat cycles, power output is not a reliable basis for comparing cycle effects. For example, as the final feedwater temperature (the temperature of feedwater leaving the highest temperature heater) increases, the overall thermal cycle efficiency improves, even though the power output decreases. Turbine cycle heat rate, rather than power output, is thus used to compare the performance of fossil-fired reheat turbines. Turbine cycle heat rate is defined as the net heat input delivered to the turbine cycle divided by the generator electrical power output, in Btu/kWh. A lower value of heat rate indicates a better overall cycle efficiency. For the parameters which follow, relative turbine cycle heat rate is used as the basis for comparison.

Number of Reheats

Figure 1 shows the heat rate comparison of single- and double-reheat cycles. As the throttle pressure increases, the gain for employing a double-reheat cycle increases. By increasing the first reheat/second reheat temperatures from 1000F/1000F (538C/538C) to 1025F/1050F (552C/566C) and increasing the throttle and reheat temperatures to 1050F/566C, further improvements may be obtained.

The remaining parameter comparisons will focus on single-reheat units. Further information on double-reheat cycles may be obtained from Reference 1.

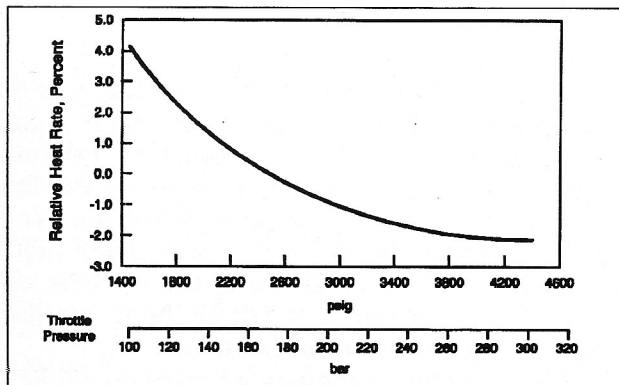#### Throttle Pressure

Figure 2 shows the heat rate variation with
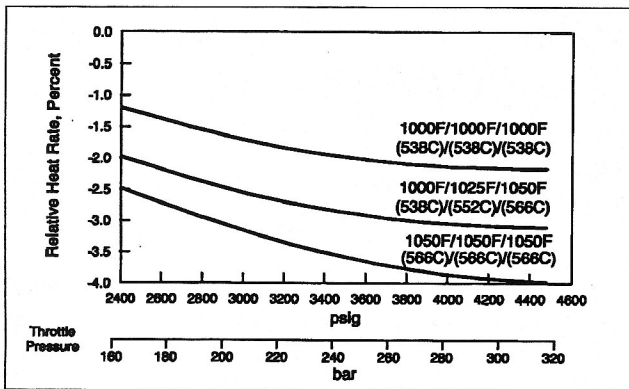
Figure 1. Heat rate variation with throttle pressure for double-reheat cycles (relative to 1000F/1000F (538C/538C) single-reheat cycle)

throttle pressure, assuming constant throttle/reheat temperatures of 1000F/1000F (538C/538C). The curve assumes that the highest-pressure feedwater heater is located at the cold reheat point and that the optimum reheat pressure has been selected for each throttle pressure. The performance difference for locating the highest-pressure heater at other than the cold reheat point is discussed under "Feedwater Heating/Reheat Pressure."
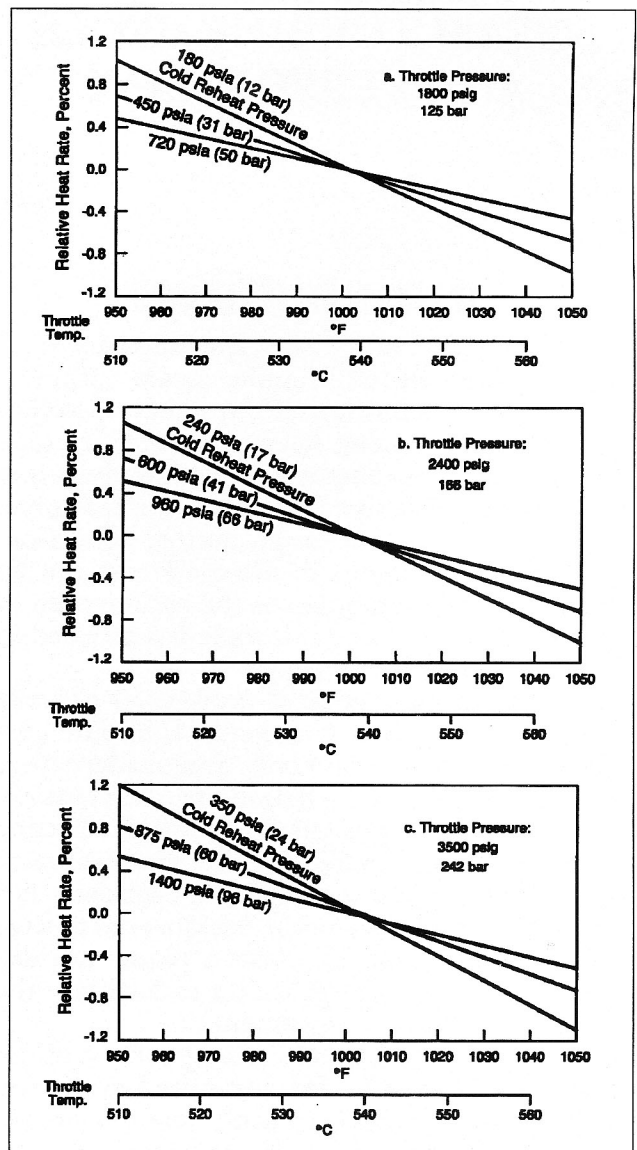
Throttle Temperature

Figure 3 shows the heat rate variation with throttle temperature at throttle pressures of 1800 psig/125 bar, 2400 psig/166 bar, and 3500 psig/242 bar and several cold reheat pressures. Lowering the reheat pressure tends to increase the effect of changing the throttle temperature. This occurs because more of the total power is developed in the high pressure section, which is the section most affected by a change in throttle temperature. A comparison of Figures 3a, 3b,

Figure 2. Heat rate variation with throttle pressure for 1000F/1000F (538C/538C) single-reheat cycle with top heater fed from cold reheat point
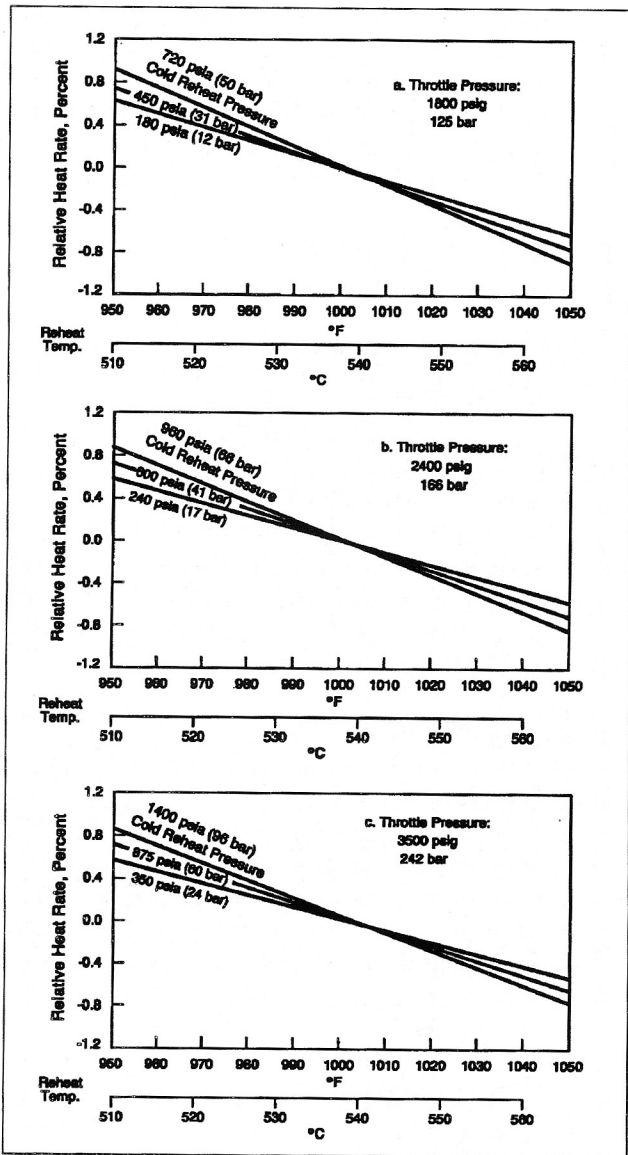

Figure 3. Heat rate variation with throttle temperature for 1000F/1000F (538C/538C) single-reheat cycle at several cold reheat pressures

and 3c also shows a significant increase in the effect of throttle temperature on heat rate at higher throttle pressures.

Reheat Temperature

Figure 4 shows the heat rate variation with reheat temperature at throttle pressures of 1800 psig/125 bar, 2400 psig/166 bar, and 3500 psig/242 bar and several cold reheat pressures. In this case, lowering the reheat pressure tends to reduce the effect of changing the reheat temperature. This occurs because less of the total power is developed after the reheater.
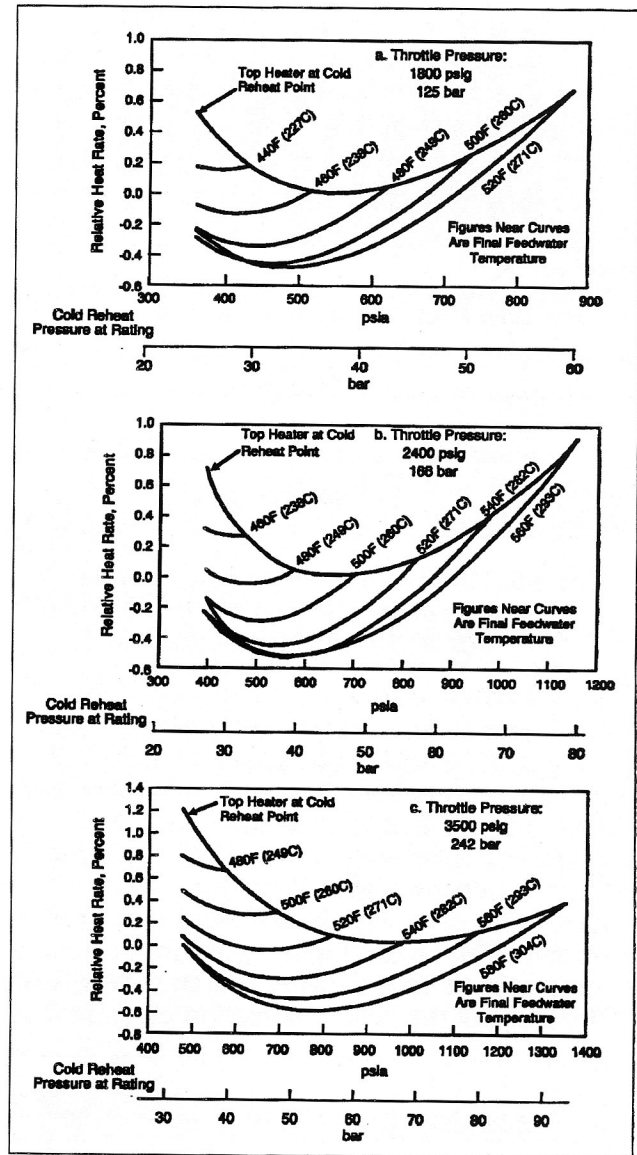
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Figure 4. Heat rate variation with reheat temperature for 1000F/1000F (538C/538C) single-reheat cycle at several cold reheat pressures

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Figure 5. Heat rate variation with final feedwater temperature and reheat pressure for 1000F/1000F (538C/538C) single-reheat cycle

Feedwater Heating/Reheat Pressure

Figure 5 shows the heat rate variation with final feedwater temperature and reheat pressure at throttle pressures of 1800 psig/125 bar, 2400 psig/166 bar, and 3500 psig/242 bar. Figures 5a, 5b, and 5c each show an upper curve which represents a seven-heater cycle with the highest-pressure feedwater heater fed from the cold reheat point. Consequently, the final feedwater temperature is uniquely determined by the cold reheat pressure. The best performance is obtained at the bottom of this upper curve, where the relative heat rate is zero.

In addition to the upper curve, several lower curves have been generated on the basis of constant final feedwater temperature, utilizing an eighth,

higher-pressure heater fed from the middle of the high-pressure turbine. This type of heater is commonly known as a Heater Above the Reheat Point, or HARP. By using a HARP cycle, the heat rate can be improved by up to 0.5% beyond the optimum heat rate obtained with the seven-heater, non-HARP cycle. However, an economic assessment must be made to ensure that the extra hardware associated with the HARP cycle is justified by the heat rate benefit.

Number of Feedwater Heaters

Figure 6 shows the heat rate variation with number of feedwater heaters. This curve was generated on the basis of optimum final feedwa-

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ter temperature. As one would expect, as the number of heaters is reduced, the relative heat rate becomes poorer, reaching a 1.5% penalty with a three-heater cycle.
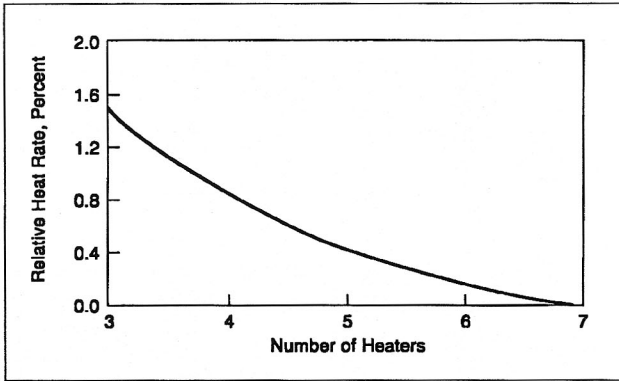

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Figure 6. Heat rate variation with number of feedwater heaters for 1000F/1000F (538C/538C) single-reheat cycle

Feedwater Heater Design Parameters

Figure 7 shows the effect on heat rate of changing a number of design parameters associated with the feedwater heating cycle. The effect of the terminal temperature difference (saturation temperature inside the heater minus feedwater temperature leaving the heater) on heat rate is shown for the two highest-pressure heaters and for the four lowest-pressure heaters. The effect of changing all the heater extraction pressure drops

and drain cooler temperature differences (temperature of drains leaving heater minus incoming feedwater temperature) is shown, as is the effect of removing the bottom heater drain cooler and pumping the drains forward rather than having them drain to the condenser.

Combined-Cycle and Cogeneration Units

Combined-cycle and cogeneration applications introduce considerable case-specific variations in key cycle parameters, process steam requirements and expected modes of operation. The thermodynamic optimization of a combined-cycle or cogeneration plant and selection of an optimum steam turbine configuration requires consideration of the plant's life-cycle operating profile and the life-cycle cost of the alternative steam turbine configurations which are available.

Life-Cycle Operating Profile

The first step to be taken when performing a thermodynamic optimization is to define the boundary of the system being optimized. The operating profile can then be described in terms of the variation in quantities which flow across the boundary (e.g., fuel consumption, power output, throttle flow and/or process extraction flow) and variation in boundary conditions (e.g., ambient

Feedwater Heater Design Parameter	Relative Heat Rate
Top Heater	
Change Terminal Temp. Difference From 0F (0C) to -3F (-1.7C)	-0.05%
Change Terminal Temp. Difference From 0F (0C) to +5F (+2.8C)	0.08%
2nd From Top Heater	
Change Terminal Temp. Difference From 0F (0C) to -3F (-1.7C)	-0.02%
Change Terminal Temp. Difference From 0F (0C) to +5F (+2.8C)	0.04%
4 Lowest Pressure Heaters	
Change Terminal Temp. Difference 5F (+2.8C) to 10F (+5.6C) on 4 Heaters	0.17%
All Heaters	
Change Piping Pressure Drop From 3 to 5 Percent	0.10%
Change Drain Cooler Temp. Difference From 10F (+5.6C) to 15F (+8.3C)	0.02%
Bottom Heater	
Pump Bottom Heater Drains Forward With No Drain Cooler	-0.02%

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Figure 7. Heat rate variation with changes in feedwater heater design parameters for 1000F/1000F (538C/538C) single-reheat cycle
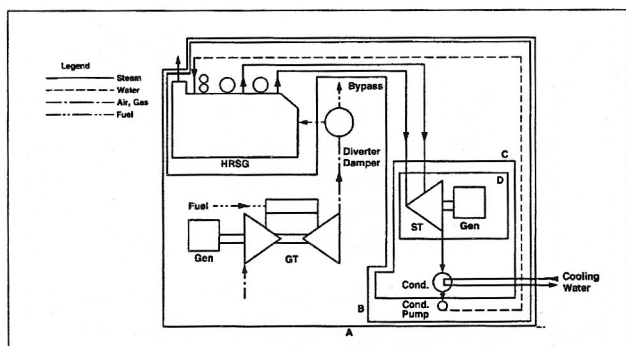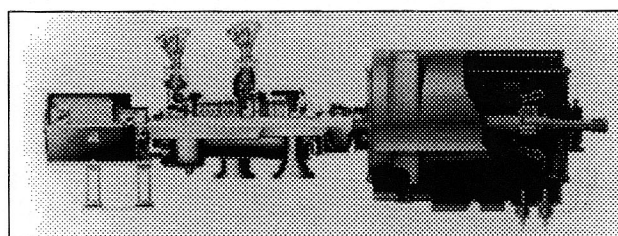
Figure 8. Two pressure combined-cycle diagram illustrating typical system boundaries for cycle optimization

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Figure 9. Cross section of a single automatic extraction non-condensing steam turbine

temperature and/or cooling water temperature).

Figure 8 shows a typical combined-cycle system, including the gas turbine, HRSG, steam turbine and feedwater system. Four common system boundaries are shown (A through D). Boundary A includes the entire combined-cycle system (gas turbine, HRSG, steam turbine and condensing system) and is of interest to the plant designer. Boundary B includes the steam side of the combined-cycle system (HRSG, steam turbine, and condensing system). Boundary C includes the steam turbine and condensing system. Boundary D includes only the steam turbine.

The system being optimized is simplified by considering only the steam turbine (Boundary D). Although this simplifies the evaluation of alterna-

tive steam turbine designs, it makes it more difficult to evaluate interactions between the turbine and other plant components, which, if considered, could lead to a better (i.e., lower life-cycle cost) overall plant design. Several of these interactions will be discussed in order to illustrate the importance of this point.

Process Extraction Flows

Two alternatives should be considered for supplying process flow from a steam turbine extraction: the controlled or automatic extraction and the uncontrolled extraction.

The automatic extraction uses multiple control valves located within the turbine to vary the flow-passing capability of the turbine section downstream of the extraction point. See Figure 9 for a cross section of a typical single automatic extraction non-condensing turbine. The use of multiple valves minimizes throttling losses in the turbine section downstream of the extraction point over a wide
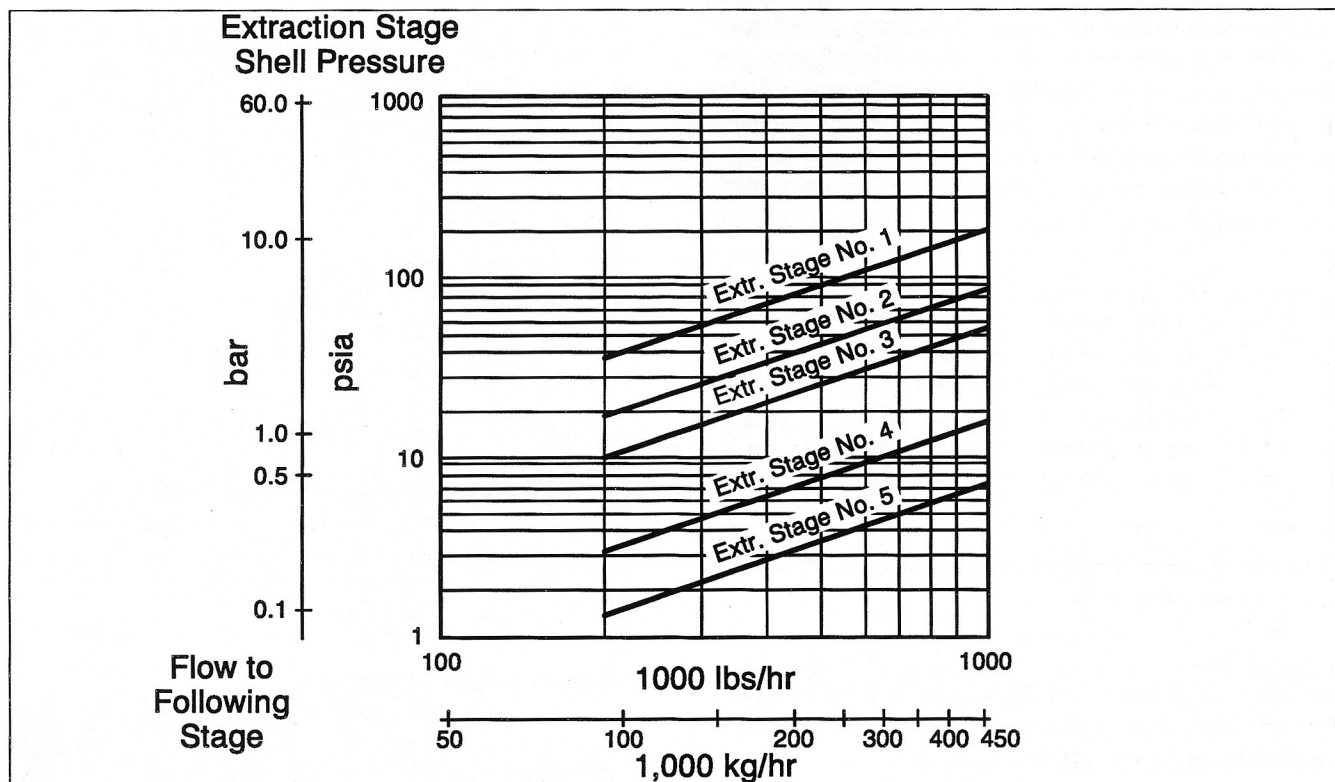

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Figure 10. Typical curve of available uncontrolled extraction stage pressure versus flow to following stage

range of operating conditions and provides a smooth pressure control characteristic.

Uncontrolled extractions are simply openings in the turbine casing at which the available extraction pressure varies directly with the flow to the following stage (Figure 10). The available extraction pressure must be throttled externally to maintain the desired process pressure.

The uncontrolled extraction is usually applied with extraction flows from 5% to 10% of the flow to the following stage, where the external throttling losses are small relative to the gains of lower cost and improved internal turbine efficiency. The automatic extraction is usually applied with extraction flows above about 15% of the flow to the following stage, where the external throttling losses become greater than the losses of higher cost and reduced internal turbine efficiency.

Automatic Extractions

The principal optimization issue for the automatic extraction turbine is the sizing of each turbine section. To illustrate this, consider the single automatic extraction non-condensing turbine shown in Figure 9. The process flow requirements will determine the turbine size in terms of throttle flow and power output. In the case of a condensing unit, however, either throttle flow power output or extraction flows could be specified. The process flow requirements need to be considered together to establish turbine sizing. The usual approach is to specify several operating conditions and an associated number of operating hours per year, as shown in Figure 11a. It is important to ensure that this data includes the entire envelope of anticipated operation.

The turbine section flows for the high pressure (HP) and low pressure (LP) sections are then

Operating Condition	165 psig (11.4 bar) Extraction Flow	65 psig (4.5 bar) Exhaust Flow	Hours per Year
A	550,000 (249,480)	250,000 (113,400)	3300
B	300,000 (136,080)	300,000 (136,080)	3100
C	450,000 (204,120)	200,000 (90,720)	1400
D	400,000 (181,440)	100,000 (45,360)	900
E	300,000 (136,080)	500,000 (226,800)	60

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Figure 11a. Process flow requirements, lb/hr (kg/hr)

determined, as shown in Figure 11b. In many cases, the flows from Figure 11b can be used directly for turbine sizing. However, two additional items need to be considered:

1. Unusually large section flows may exceed the design capability of the automatic-extraction

Operating Condition	High-Pressure Section Flow	Low-Pressure Section Flow	Hours per Year
A	800,000 (362,870)	250,000 (113,400)	3300
B	600,000 (272,160)	300,000 (136,080)	3100
C	650,000 (294,830)	200,000 (90,720)	1400
D	500,000 (226,800)	100,000 (45,360)	900
E	800,000 (362,870)	500,000 (226,800)	60

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Figure 11b. Turbine section flows derived from process flow requirements, lb/hr (kg/hr)

| Turbine Section	Maximum Section Flow
High-Pressure	800,000 (362,870)
Low-Pressure	300,000 (136,080)

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Figure 11c. Recommended section flows for turbine sizing, utilizing external pressure-reducing station for Condition E, lb/hr (kg/hr)

control valves. Alternatives, such as multiple turbines or uncontrolled extractions, should then be considered.

2. A large, but seldom required, section flow, such as Operating Condition E in Figure 11b, is often better handled by using an external pressure-reducing station than by oversizing the turbine section to pass the excess flow. This can be seen by looking at a typical curve of LP efficiency versus flow (Figure 12). LP section outputs for Conditions A through E are shown in Figure 13 for two alternative maximum section flows (300,000 lb/hr (136,080 kg/hr) and 500,000 lb/hr (226,800 kg/hr). The improvement in efficiency for Conditions A through D more than offsets the loss due to throttling a portion of the flow for Condition E. In addition, there may be turbine hardware cost savings associated with the reduction in maximum LP section flow. The recommended section sizing flows are shown in Figure 11c.

Uncontrolled Extractions

One or more uncontrolled extraction openings can be an effective means of supplying process steam requirements. The most important consideration in making this assessment is the

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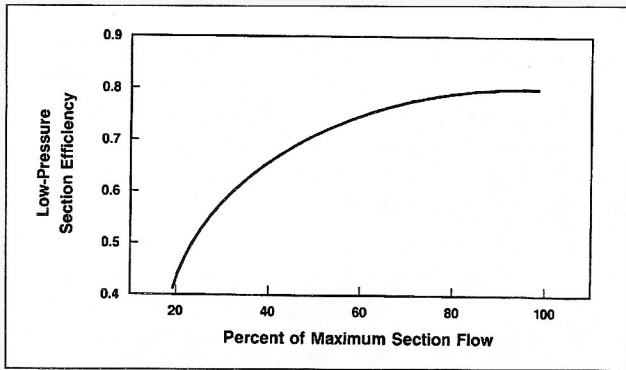
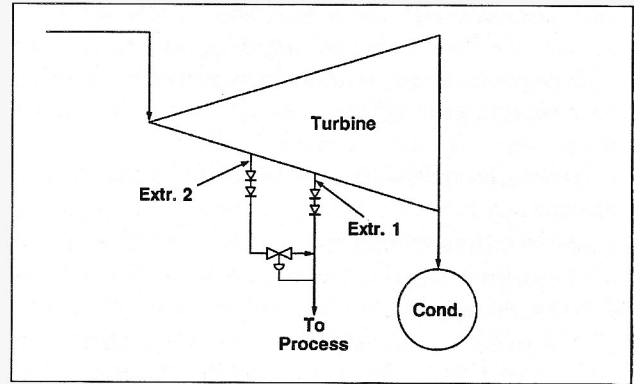Figure 12. Typical variation of low-pressure section efficiency with section flow for conditions of Figure 11

Figure 13. Low-pressure section outputs for two alternative design cases for conditions of Figure 11 (initial enthalpy 1276.1 Btu/lb, 2967.7 kJ/kg)

Figure 14. Flow schematic of condensing steam turbine using two uncontrolled extraction openings to provide process steam

variation in the flow to following stage, which sets the available extraction pressure, as shown in Figure 10. Flow to following stage is defined as the flow through the preceding turbine stage minus the extraction flow. If it is desired to supply process steam at a fixed pressure from an uncontrolled extraction opening over a range of operating conditions, the variation in flow to following stage will determine the available extraction pressure and, therefore, the external throttling losses, which will have to occur at all but the minimum required flow to following stage. For example, consider a condensing application (Figure 14) where the maximum inlet throttle flow is 700,000 lb/hr (317,520 kg/hr) at steam conditions of 1450 psig/101 bar and 1000F/538C with no other extractions or admissions. It is desired to supply 200,000 lb/hr (90,720 kg/hr) to process at 200 psig/14.8 bar. For simplicity, we will neglect the need to desuperheat to maintain the desired process steam temperature.

The principal optimization issue for an uncontrolled extraction is the location of the extraction opening. If optimum efficiency is desired at 200,000 lb/hr (90,720 kg/hr) extraction flow with maximum LP section flow, the opening should be

located at a stage with pressure as close as possible to 200 psig/14.8 bar, plus any extraction pressure drop, when the flow to following stage is 500,000 lb/hr (226,800 kg/hr) (maximum load). This is shown at point A in Figure 15, and utilizes Extr. 1 in Figure 14. The problem with this design is that whenever the extraction flow increases, or the throttle flow decreases, the flow to following stage at Extr. 1 falls below 500,000 lb/hr (226,800 kg/hr), and the opening is no longer able to supply steam at 200 psig/14.8 bar. In this case, an alternate opening, perhaps two or three stages upstream (Extr. 2 in Figure 14), would be utilized. The stage extraction pressure will then trace out line BC in Figure 15 as the flow to following stage is reduced.

To supply steam at 200 psig/14.8 bar, the preceding would be one approach. A second would be to supply steam from a stage with pressure at 300 psig/21.7 bar, at maximum load. As the flow to following stage decreases, the available extrac-
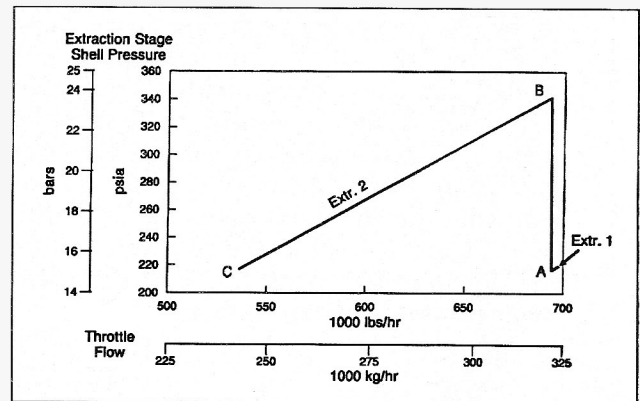
Figure 15. Uncontrolled extractions located to optimize points A and C (for extraction flow of 200,000 lb/hr (90,720 kg/hr))

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tion pressure will decrease until, at about 70% of maximum flow to following stage, it drops below 200 psig/14.8 bar. Only then would it become necessary to utilize an alternate opening upstream.

Several conclusions can be drawn from this discussion:

1. It is important to understand the actual range of operation, so the uncontrolled extraction can be located to minimize overall output losses, rather than losses at a single operating point. Data on the expected duration of operation at different operating conditions can be used to properly evaluate alternative extraction locations and methods.

2. Combined-cycle plants may utilize a source of steam to inject into the gas turbine. If this steam is supplied by an uncontrolled extraction located to optimize efficiency at full load, switchover to a higher stage extraction will occur almost immediately. This can be especially undesirable for plants designed with multiple gas turbines feeding the steam turbine. If one gas turbine is shut down, the steam injection extraction will be taken from a higher stage, with resulting throttling losses and poorer cycle efficiency. An alternate approach would be to supply the steam injection flow from a dedicated HRSG pressure level.

3. When comparing an uncontrolled extraction turbine design with an automatic extraction design, it is important not to limit the evaluation to one or two operating points, unless that accurately represents the expected operation. In our example, at points A and C in Figure 15, the uncontrolled extraction design will supply the required 200 psig/14.8 bar with a minimum of external throttling. A plot of turbine output versus throttle flow for 200,000 lb/hr (90,720 kg/hr) extraction flow is shown in Figure 16a, which compares the uncontrolled extraction design with an automatic extraction design. A considerable improvement in output is achieved with the automatic extraction design whenever the uncontrolled extraction design is not operating at point A or C. Figure 16b shows the relative output of the uncontrolled extraction versus the automatic extraction turbine design.

The differences in output shown in Figure 16b can be compared on an economic basis. Two alternative modes of operation are considered, based on a value of $2000/kW for incremental power:

1. Assume that operation will be equally distributed over the range of throttle flows from 550,000 lb/hr (249,480 kg/hr) to
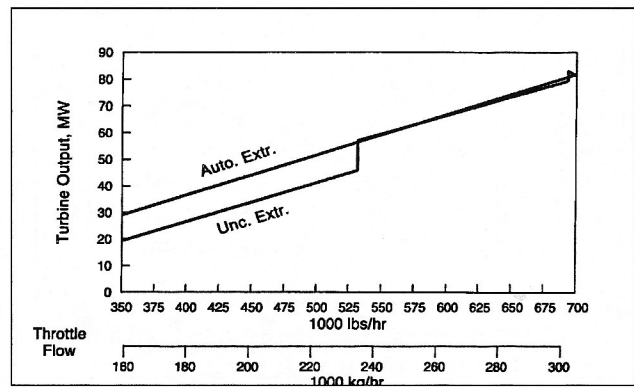
Figure 16a. Output of uncontrolled extraction design and automatic extraction design (for extraction flow of 200,000 lb/hr (90,720 kg/hr))
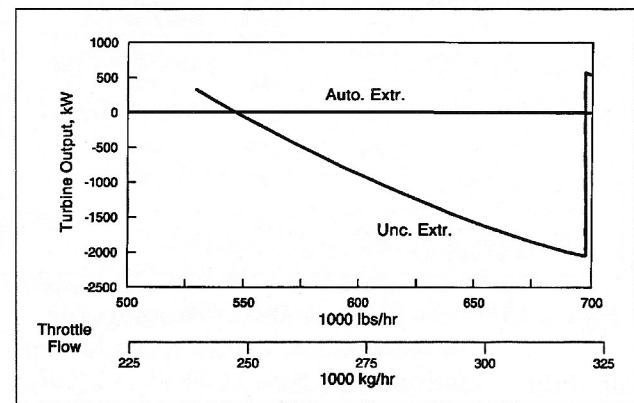
Figure 16b. Output of uncontrolled extraction design relative to automatic extraction design (for extraction flow of 200,000 lb/hr (90,720 kg/hr))

700,000 lb/hr (317,520 kg/hr). Using the data from Figure 16b, the automatic extraction turbine provides the greatest output over this range, by an average of 1134 kW. At $2000/kW, the value of this advantage is $2.27 million.

2. Assume that the steam turbine will always operate with a throttle flow of 700,000 lb/hr (317,520 kg/hr). Using the data from Figure 16b, the uncontrolled extraction turbine now provides the greatest output, by an average of 523 kW. At $2000/kW, the value of this advantage is $1.05 million.

The optimum steam turbine configurations for the two evaluated modes of operation are thus totally different, demonstrating the importance of properly considering anticipated operation in selecting a steam turbine design.

Other Extraction Approaches

Other approaches to providing process extrac-

tions are sometimes used. The concepts which have been presented for the automatic and uncontrolled extractions, can also be applied to assess these approaches. One such approach uses an external valve, which is located in the steam line between two turbine casings and maintains extraction pressure by throttling the main steam flow downstream of the extraction point (Figure 17). Conceptually, this is the same as an automatic extraction with only a single valve. At the operating condition where this throttling valve is wide open, this design is quite efficient. However, at all other operating conditions, the main steam flow downstream of the extraction point will be throttled, with a substantial loss in performance. As with the uncontrolled extraction versus automatic extraction assessment, consideration of the operating range of the turbine is required to select the optimum cycle and steam turbine configuration.

Extraction Stage Pressures: Interface Issues

It is evident from the foregoing discussion that proper location of extraction openings, as well as selection of turbine configuration, is critical to developing an optimum thermodynamic design. An important interface between the steam turbine designer and the plant designer is
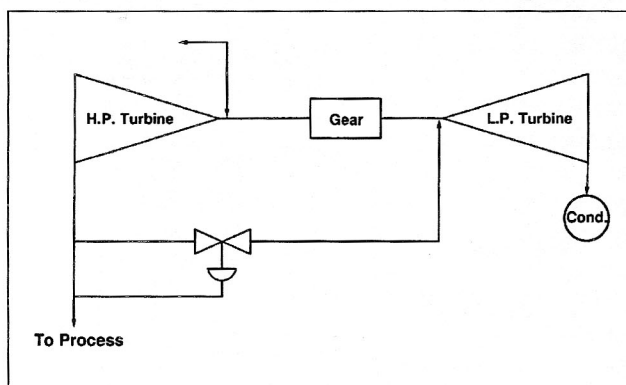

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Figure 17. Two-casing steam turbine using main flow throttling to hold process extraction pressure

matching plant design heat balances with actual turbine stages. Some useful principles follow:

1. Ground rules for locating uncontrolled extraction openings and evaluating alternative configurations for providing process steam, such as external valves versus automatic extractions, should be clearly described and should reflect the anticipated operation in enough detail to permit proper evaluation. Alternatively, one could specify that an uncontrolled extraction be capable of providing the required pressure at a specific operating point or minimum flow to the following stage.

2. Process extraction pressures are generally set by process requirements and are thus not flexible. However, this is not true of feedwater heater extraction pressures, since they are internal to the steam turbine cycle. Sufficient flexibility should be provided to permit alternative steam path designs to be used without compromising the turbine stage design or introducing undue external throttling losses. An example of this would be where the stage pressures available at rated flow are 25 psia/1.7 bar and 65 psia/4.5 bar and a plant specification calls for a deaerator pressure of 30 psia (2.1 bar). The preferred approach is to modify the deaerator pressure to match one of the available stage pressures, rather than to introduce throttling to make the alternative steam turbine designs supply steam at the same deaerator pressure.

Uncontrolled Admissions and Sliding Pressure Operation

Steam turbines, particularly for combined-cycle applications, are frequently designed to operate in a sliding pressure mode at the throttle and at low pressure admission(s). This can make design point optimization and evaluation and off-design performance calculations difficult. Both problems arise because the HRSG steam flows and steam turbine throttle and admission pressures are interdependent.

In establishing the design point performance, especially for low pressure admissions, the same issues arise as for extractions: only certain admission pressures will match steam turbine stage pressures; and it is desirable to avoid introducing throttling losses. The effect of admission pressure on combined-cycle performance is generally small within the range of variation needed to match the nearest turbine stage pressure. The HRSG drum pressure should be designed to be consistent with the turbine admission stage pressure, with an allowance for the necessary line pressure drop. This may necessitate a different HRSG design, or at least a different performance calculation, for each steam turbine alternate in order to make a proper comparison. Since the flows involved are often several times larger than those associated with feedwater heating extractions, proper handling of these calculations becomes even more important. Once the design point performance has been properly optimized, the simplest way to prepare off-design data is to use

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a calculation model which includes the HRSG, the steam turbine and the condensing system. This approach facilitates the iteration of temperatures and flows in the HRSG depending on steam turbine flow-passing characteristics. There are very real practical difficulties in creating a sufficiently accurate combined calculation model, particularly when the plant and steam turbine are being designed by different parties. In practice, iterations must often be handled by other means.

Some suggestions to simplify these iterations follow:

1. The following flow formula can be derived from one-dimensional compressible flow theory and is quite useful in understanding steam turbine stage flow-passing characteristics:

where:

F = mass flow rate

A_e = effective area

= A C,

where A = physical area

C = flow coefficient

N = restriction factor (1.0 for critical pressure ratio)

ω/p = critical mass flow rate for an isentropic process,

= function of p (pressure) and h (enthalpy)

$$F = A_e N \frac{\omega}{p} p$$

p = pressure

Figure 18 shows the parameter ω/p as a function of pressure and enthalpy, as well as a table of restriction factor, N, as a function of pressure ratio.

In the case of steam turbine stages having constant pressure ratio, the effective area (A_e) and restriction factor (N) are also constant, so the quantity A_eN is constant. A_eN is also constant for condensing last stages, despite variations in pressure ratio, so long as the pressure ratio is greater than critical (1.83). For stages with constant A_eN, the flow formula provides a direct relation between the mass flow rate, pressure and enthalpy at the inlet to a stage.

2. In the case of a steam turbine with sliding throttle pressure and no admissions or extractions, the throttle A_eN is constant. The flow formula can then be used by the plant designer to calculate HRSG steam flows and temperatures, which properly match the steam turbine throttle flow-passing capability.

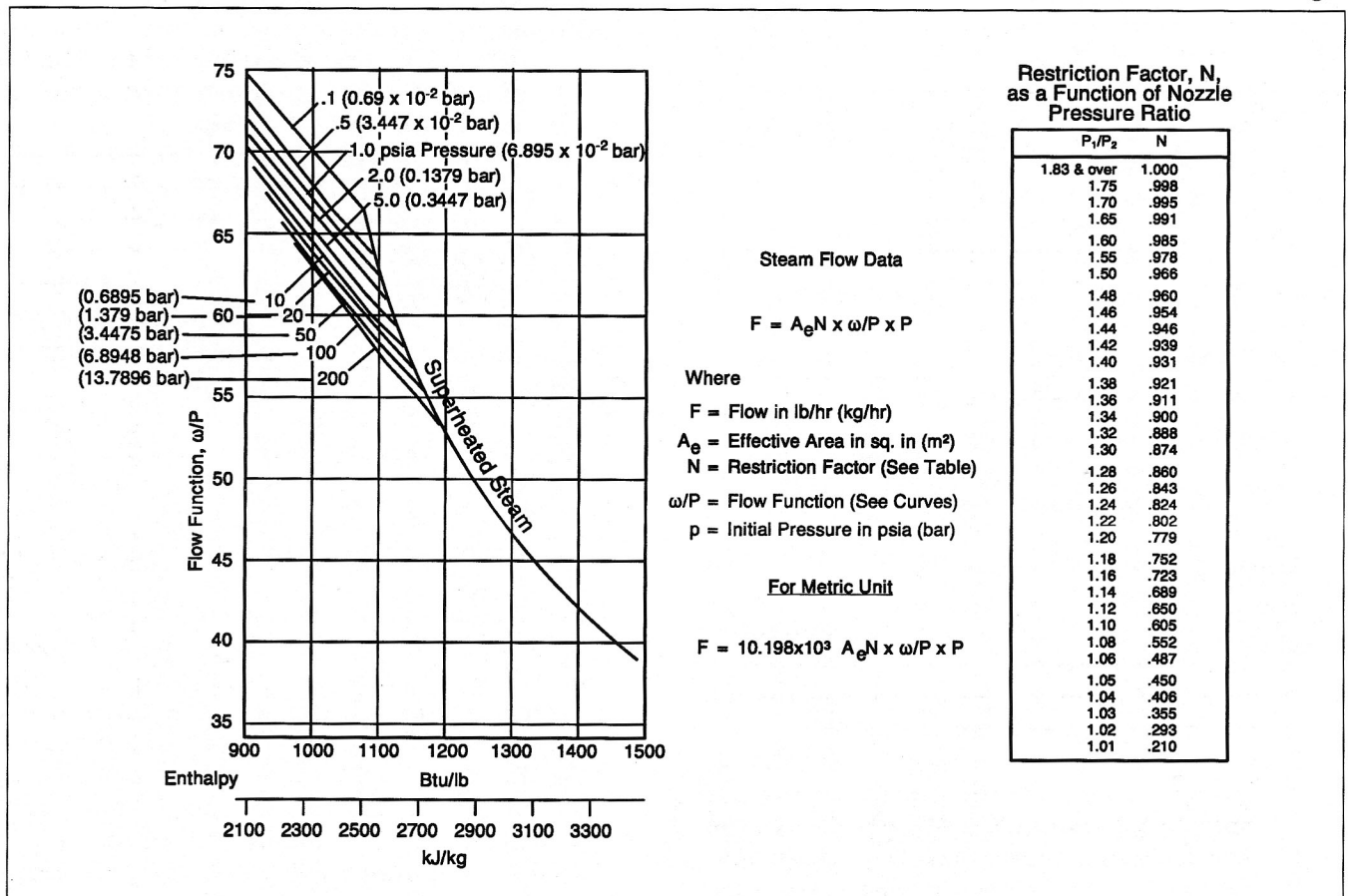3. If admissions or extractions are far enough

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Figure 18.   Critical mass of flow rate steam for an isentropic process, ω/p, and restriction factor, N

downstream from the throttle, or are in reasonably constant proportion to the throttle flow, the effect on the first-stage pressure ratio, and thus on the throttle A_eN, of variations in admission or extraction pressure can be neglected and the same procedure applied.

4. If all admissions and extractions are in reasonably constant proportion to the throttle flow, are small (not greater than about 5% of throttle flow), or occur in low-pressure stages where the pressure ratios are at or above critical, the flow formula can be used, based on flow to following stage. This is used to establish the A_eN of turbine stages immediately downstream of the admissions or extractions, as well as the throttle.

5. In the case of sliding admission pressure, the plant designer will need to estimate the turbine efficiency from the throttle to the admission point in order to establish the mixed enthalpy needed to determine ω/p for the flow entering the turbine stage immediately downstream of the admission. One approach is to assume the same turbine efficiency from the throttle to each of the admission or extraction points as exists at the design point. Alternatively, an approximate steam turbine model can be set up based on the design point heat balance to facilitate this iteration.

6. For applications having large variations in admission or extraction flows (e.g., combined-cycle applications with wide variations in steam injection flow required), the steam turbine stage pressure ratios vary significantly over the operating range, and the stage A_eN values cannot be assumed constant. These applications can only be modeled with a stage-by-stage efficiency calculation and require that the plant and turbine designers work closely together to perform the iterations needed to establish the performance.

Condenser and Last-Stage Bucket Optimization

The cooling system design and steam turbine last-stage bucket selection are critical parts of steam power plant optimization, with the decisions made having multi-million dollar cost and performance ramifications.

The example to be discussed here will assume direct water cooling, with a constant cooling water temperature. The emphasis will be on the condenser and last-stage bucket sizing aspects of plant optimization. If the cooling system included a cool-

ing tower and/or an air condenser, their cost/performance characteristics would also be assessed and the plant optimization, though more complex, would be handled similarly. Likewise, seasonal variations in cooling water temperature, as shown in Figure 19, and daily or other variations might need to be considered. One approach used in practice is to define a discrete number of operating points which include all the significant variations in cooling medium conditions and operating conditions.

The example will be discussed in the context of the combined-cycle power plant of Figure 8, where the steam turbine throttle and admission flows are assumed constant. Consider first the system defined by Boundary D. Steam turbine out-
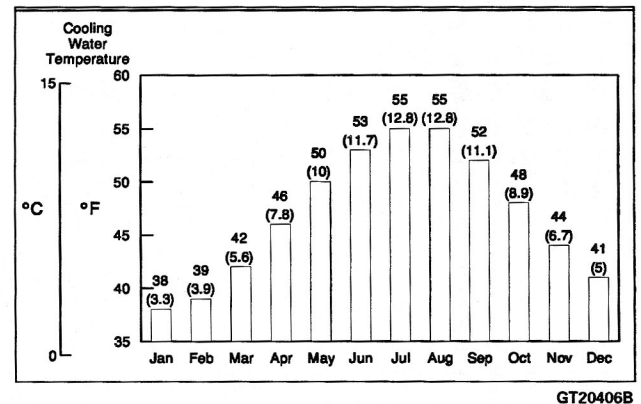

Figure 19. Representative annual distribution of cooling water temperature

puts can be estimated for alternative last-stage bucket configurations at the condenser pressure(s) given for the specified operating condition(s). The value of differences in output can be compared using steam turbine and associated plant cost differences and the optimum last-stage bucket selected. This approach can also be applied to conventional steam turbine cycles with feedwater heating, where heat rate at constant power output is used as the basis for evaluation, provided that heat rate is used in place of power output as the parameter of value.

A more complex case involves the system defined by Boundary C. The fundamental objective is to establish the cost and performance of the available alternatives. Assuming a constant cooling water temperature of 75 F/24 C, the principal design variables are condenser surface capital cost, circulating water pump capital cost and power requirements, and steam turbine capital cost and power output. Assuming a constant condenser terminal temperature difference of 5 F/15 C, each of the principal design variables may be estimated as a function of condenser pressure.

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The steam turbine power output can be calculated for each of three candidate last-stage bucket configurations as a function of condenser pressure. A typical curve is shown in Figure 20, and includes the 2 x 26 inches (double-flow 26 inches) (2 x 660mm), 2 x 33.5 inches (2 x 851mm), and 2 x 42 inches (2 x 1067mm) last-stage buckets (all 50 Hz designs).

The required circulating water pump power can also be calculated as a function of condenser pressure. The net power output, defined as steam turbine power output minus circulating water pump power,
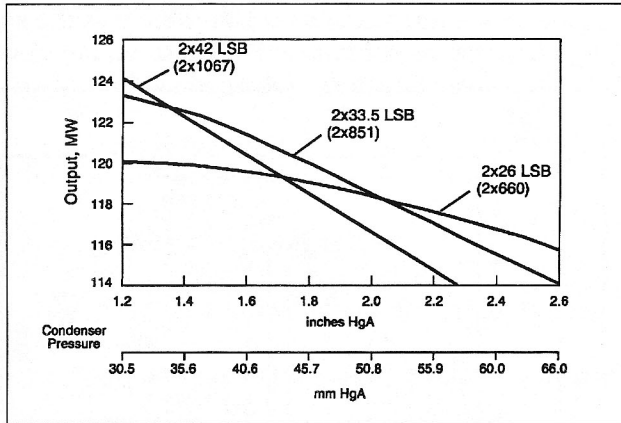
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Figure 20. Steam turbine output versus condenser pressure for alternative last-stage bucket configurations (50 Hz)

can then be calculated for each of the last-stage bucket configurations, and is shown in Figure 21.

The condenser surface and circulating water pump capital cost can be estimated as a function of condenser pressure. The steam turbine cost can be estimated based on the last-stage bucket
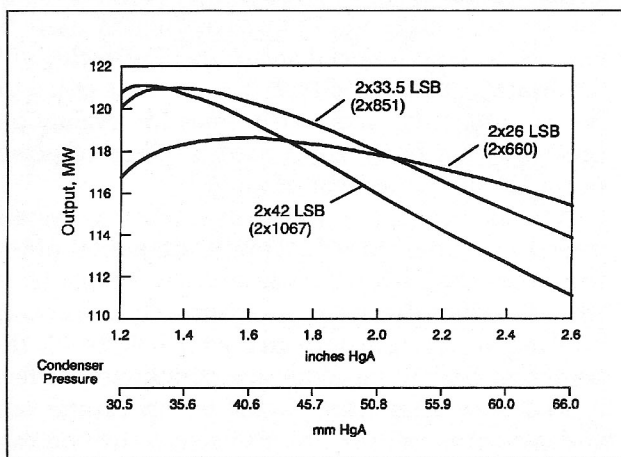
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Figure 21. Steam turbine output (less circ. pump power) versus condenser pressure for alternative last-stage bucket configurations (50 Hz)

configuration and other hardware differences, including foundation, building and other plant design differences. The total installed cost can then be plotted versus condenser pressure for the different last-stage bucket configurations. A typical curve is shown in Figure 22, in which the cost values shown are relative to a 2 x 26 inches/2 x 660mm last-stage bucket at 2.6 inches HgA/66.0mm HgA condenser pressure. The values shown are for illustrative purposes and should not be interpreted as representative of actual designs.

The data from Figures 21 and 22 can be combined to prepare a useful set of curves relating cost and performance. For each of the alternative
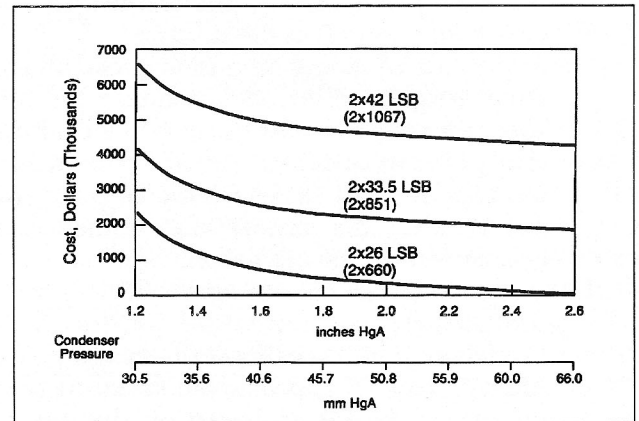

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Figure 22. Relative plant installed cost (typical) versus condenser pressure for alternative last-stage bucket configurations (50 Hz)

last-stage bucket configurations, the cost and output are known over a range of condenser pressures. These values are used to prepare the curves shown in Figures 23a, 23b and 23c. For a given evaluated worth of output ($2000/kW is illustrated), lines tangent to each of the last-stage bucket curves are drawn. Each point of tangency then identifies the optimum condenser pressure for that last-stage bucket.

The outputs and relative costs at the optimum condenser pressure point for each last-stage bucket configuration are then compared as shown in Figure 24. The 2 x 33.5 inches/2 x 851mm last-stage bucket at 1.4 inches HgA/35.6mm HgA is the optimum, since it achieves an incremental performance improvement over the 2 x 26 inches/2 x 660mm which costs only $966/kW, while a further performance improvement with the 2 x 42 inches (2 x 1067mm) would cost $25,918/kW, well above the evaluation value of $2000/kW.

Definitions

Before closing the discussion of steam turbine
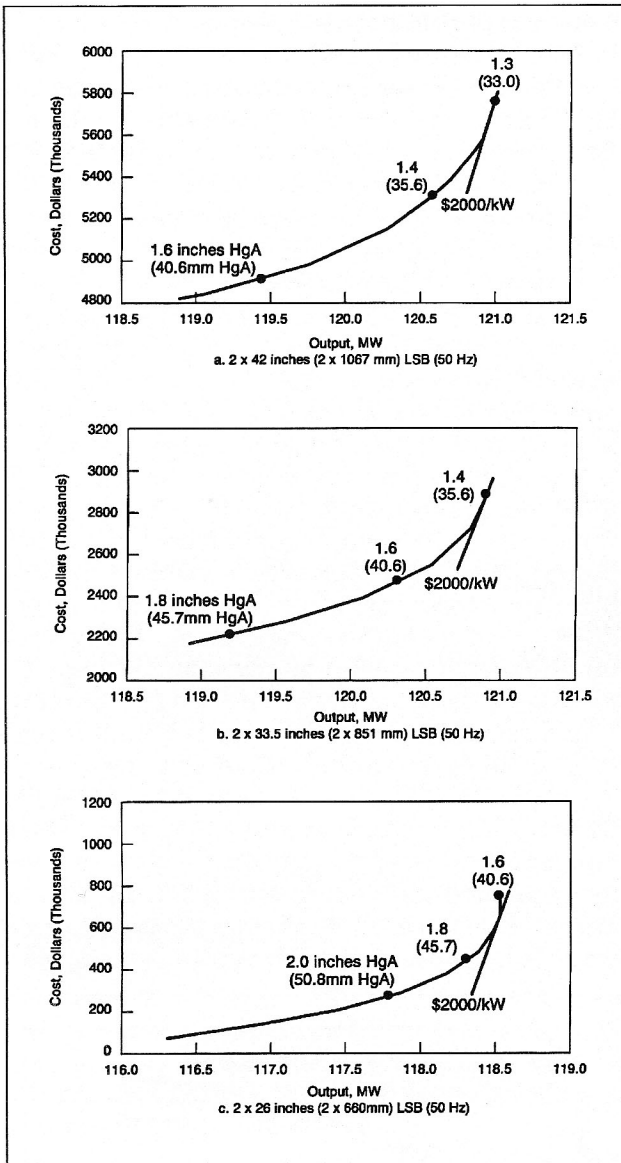
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Figure 23. Last-stage bucket and condenser cross-optimization for evaluation of $2,000/kW

LSB Inches (mm)	Optimum Cond. Press. Inches HgA (mm HgA)	Relative Plant Installed Cost ($000)	Output (MW)	$/kW vs Next Smaller LSB
2 x 26 (2 x 660)	1.65 (41.9)	Base	118.51	-
2 x 33.5 (2 x 851)	1.40 (35.6)	2318	120.91	966
2 x 42 (2 x 1067)	1.30 (33.0)	5169	121.02	25918

GT21493A

Figure 24. Comparison of optimum condenser pressure points for alternative last-stage bucket configurations

application interface issues, it may be helpful to review several concepts which arise frequently and for which clear definition is important.

Maximum Guaranteed Throttle Flow

This is the maximum throttle flow-passing capability required at rated steam conditions. Steam turbines are generally designed with additional flow margin to allow for uncertainties in flow coefficients and manufacturing tolerances on nozzle areas, etc. The maximum guaranteed throttle flow may or may not correspond to a performance evaluation point.

Valves-Wide-Open Throttle Flow

This is the steam turbine's expected design throttle flow-passing capability at rated steam conditions and includes any flow margin. The normal approach is to leave the choice of the amount of flow margin to the turbine designer, who is most familiar with the uncertainties associated with flow-passing capability, rather than mandating a specific level of flow margin or none.

Pressure Margin

In some cases, such as combined-cycle steam turbines intended to operate in a sliding-pressure mode, no flow margin may be provided. Instead, the steam turbine (and HRSG) are designed to withstand a somewhat higher pressure than the expected value, in order to protect against uncertainties in flow coefficients, etc. This allowance for increased pressure is termed pressure margin.

Maximum Continuous Rating (MCR)

This term is sometimes used to describe the steam turbine output at some maximum flow condition (e.g., the boiler design point). In cases where it is to be guaranteed, MCR should be defined at or below maximum guaranteed throttle flow. Where the expected (not guaranteed) maximum output is desired, MCR should be defined at valves-wide-open throttle flow.

EVALUATION OF DESIGN ALTERNATIVES

In evaluating steam turbine design alternatives, the objective is to obtain a consistent comparison of life-cycle cost. The relative performance levels of the design alternatives are important in making this comparison, due to the impact of electrical output, process energy and fuel consumption on operating revenues and costs.

The concepts of the previous section can be applied to select a reasonable steam turbine configuration and to properly optimize the plant

cycle and match it to actual turbine hardware. However, the fact that the plant cycle has been optimized around a specific turbine configuration can make it difficult to evaluate alternatives.

This section will discuss some of the more difficult issues which arise in evaluating steam turbine design alternatives. Some basic concepts and principles are presented first, followed by specific examples.

Concepts and Principles

The following are some suggested principles to follow in evaluating steam turbine design alternatives:

1. Each design alternative should be evaluated based on the same cycle boundary (e.g., Figure 8) and boundary operating conditions.
2. Each design alternative should be evaluated based on its optimum plant cycle for the application. This will often require the cycle boundary for evaluation purposes to be larger than the boundaries of the equipment guarantee, or scope of supply, in order to allow proper cycle optimization. For example, steam turbine performance is commonly quoted in the form of a heat balance which includes feedwater heaters, pumps, etc.
3. The cycle boundary for evaluation purposes should be clearly communicated to the steam turbine designer. Boundary operating conditions and evaluation parameters should also be clearly communicated to the steam turbine designer.
4. The performance of plant components outside the boundaries of the steam turbine equipment guarantee or scope of supply, but inside the cycle boundary, should be calculated on the same basis for all design alternatives (exception: see item 5, next).
5. Where differences in steam turbine hardware require changes to other plant components for proper cycle optimization (e.g., condenser surface differences as in Figure 23), these changes should be made, along with the necessary adjustments to cost for evaluation purposes.
6. A list of the plant components for which such changes will be permitted (per item 5) should be provided to the steam turbine designer and should allow the changes necessary to properly optimize the cycle for each steam turbine design alternative.

Examples

Several examples of issues which arise in evaluating steam turbine design alternatives follow.

Combined-Cycle Pressure Optimization

An important part of the task of optimizing a combined-cycle system is the proper selection of HRSG pressure levels. Many parameters influence the selection of HRSG pressure levels, including:

- heat transfer surface costs
- pressure vessel and piping material costs
- cycle thermodynamic efficiency
- steam turbine efficiency

Steam turbine efficiency enters into the equation because of its reduction which occurs with reduced volume flow, due to stage leakage and endwall losses, as well as shaft end leakage. This is important to understand because different steam turbine design configurations may have different relationships between efficiency and volume flow. Shown in Figure 25 are curves illustrating efficiency trends versus volume flow for a high-speed geared steam turbine and a direct-drive steam turbine configuration. The geared unit efficiency is better at low volume flows, due to smaller leakage areas and longer blading. The direct-drive unit efficiency is better at high volume flows, due to the lack of gear losses.

As a result of such differences, different steam turbine design configurations may have different optimum throttle and admission pressures for combined-cycle operation. Referring back to item 5 from page 14, this is a case where it would be necessary to develop two or more
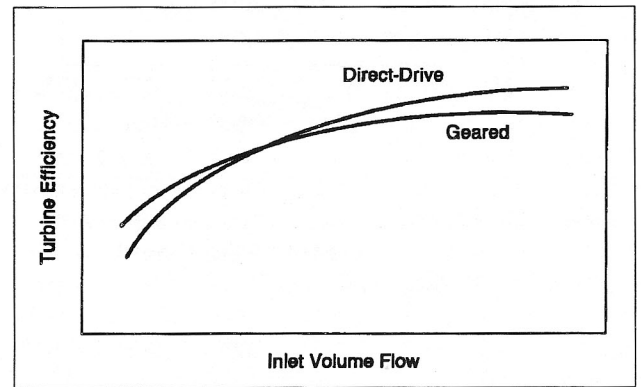

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Figure 25. Relative efficiency trends for direct-drive and geared steam turbines

alternative plant designs with one corresponding to the optimum pressure for each steam turbine design configuration, in order to properly evaluate the design alternatives.

Condenser and Last Stage Bucket Optimization

As shown in Figure 23, the optimum condenser pressure and surface area depends on the exhaust annulus area of each steam turbine

design alternative. Thus, applying item 2 from page 14, the cycle boundary for evaluation purposes should include the condenser system.

If the steam turbine manufacturer is also responsible for specifying the condenser, this optimization can be readily performed.

The choice of Boundary D (Figure 8) for evaluation of steam turbine design alternatives conflicts with item 2, and thus does not directly ensure an optimum plant design. For example, in Figure 23, if the condenser pressure were specified as 1.8 inches HgA/45.7mm HgA, the 2 x 26 inches/2 x 660mm last-stage bucket would be selected.

In order to help obtain an optimum plant design when using Boundary D, an estimate of the cost of the condenser system as a function of condenser pressure can be made by the plant designer and given to the steam turbine designer. This reduces the number of design alternatives which would need to be prepared and evaluated. And, as noted in item 5, the steam turbine designer would know, at the outset, the evaluation to be applied for variations from the specified condenser pressure. And as noted in item 5, the steam turbine designer should know, at the onset, the cost of the condenser for differing condenser pressures. This procedure makes the cycle boundry, boundry C, but allows the turbine designer to be concerned only with the turbine slope of supply.

Cycle Detail Differences

Quite often, steam turbine heat balances are prepared for design alternatives which assume inconsistent performance or cycle design for components outside the steam turbine scope of supply, but within the cycle boundary. Some common examples are:

- feedwater heater temperature differences and line pressure drops
- reentry location and enthalpy for make-up water
- pump efficiencies and discharge pressures
- process extraction enthalpy
- desuperheating water enthalpy

Differences of this type are generally not necessary to optimize a cycle around different steam turbines. Accordingly, adjustment for these differences should be made before comparing the steam turbine designs.

Different Basis of Guaranteed Performance

Sometimes, steam turbine performance for design alternatives is presented on a different basis. For example, some losses may not be included (e.g., valve pressure drops, exhaust and leaving losses, generator excitation power). Also, performance may be conditioned on different acceptance testing procedures.

Differences of this type are again not related to steam turbine or cycle optimization, their adjustment should be made as part of any evaluation of steam turbine design alternatives.

PERFORMANCE TESTING ALTERNATIVES AND ISSUES

When a turnkey power plant or complete power island is guaranteed by one supplier (e.g., a GE STAG plant), it is only necessary to accurately measure overall plant output, heat rate, and export process energy, if any, to verify satisfactory plant performance. The supplier may, of course, choose to install additional instrumentation on component equipment (e.g., the steam turbine) in order to confirm that equipment is operating within normal design limits.

However, when a steam turbine is being supplied on an equipment-only basis, the selection of the performance test that verifies its performance becomes an important part of the plant design and specification process. There are costs directly associated with designing, planning and executing the test. In addition, it is important that alternative steam turbine designs be evaluated on the basis of the same anticipated performance test.

The two principal factors involved in the selection of a steam turbine performance test are: (1) the cost of executing the test; and (2) the measurement uncertainty of the test. The reason for running an accurate test is to verify that the steam turbine is meeting its guarantee.

A suggested procedure for determining the optimum accuracy of an acceptance test on a cost-effectiveness basis was presented by Cotton, et al. (Reference 2). The most accurate test available, the ASME PTC 6 test (Reference 3), has an uncertainty of ± 0.25%, and is generally justified for steam turbines over 200-300 MW. However, as the steam turbine rating decreases, an assessment of the cost-effectiveness of different test alternatives is recommended. Alternative approaches available for steam turbine performance testing and measurement uncertainty associated with each will be discussed. Measurement uncertainty for a steam turbine performance test can be determined using the methods of Reference 4.

The following steam turbine performance testing procedures are commonly used in the industry.

ASME PTC 6 (Reference 3)

This is often referred to as a precision or full-scale test, and is the most accurate test that can be performed consistently with the best engi-

neering knowledge and practice currently available. The associated measurement uncertainty for a reheat or nonreheat condensing unit without process steam extraction is ± 0.25%. Uncertainties for units with process steam extraction may be as high as ± 1.0%.

ASME PTC 6 Report - 1985 (Reference 4)

This is a procedure to determine performance test uncertainty due to instrumentation errors.

ASME PTC 6.1 (Reference 5)

This is often referred to as the alternative test, and is intended to allow high accuracy, but without the complexity and cost associated with the PTC 6 test. The principal difference from the PTC 6 test lies in the use of a calibrated high pressure flow nozzle with an inspection port in the feedwater line and a consequent reduction in the number of precision pressure and temperature measurements required. The associated measurement uncertainty (on the same basis as PTC 6) is ± 0.37%.

Both of the above test procedures are based on the principal of lowest practicall uncertainty and are commonly used without allowance for measurement uncertainty.

ASME PTC 6S (Reference 6)

This test code was written with the intent of providing guidance for routine performance testing during operation (e.g., for performance monitoring), rather than use as an acceptance test. However, it is frequently cited in specifications as a lower cost alternative to be used on steam turbines below 200 MW. Considerable latitude exists as to test instrumentation, and the associated measurement uncertainty must be determined on a case-by-case basis. Typically, it ranges from ± 1.5 to 3.0%.

DIN-1943 (Reference 7)

This testing code was developed by the German standards organization. It is similar in terms of test instrumentation to the ASME PTC 6 tests. However, specific allowances are allowed for such items as:
- measurement uncertainty
- aging
- unaccounted-for leakages during test
- fluctuations during test

These are specifically allowed by the DIN-1943 code. These allowances can easily result in the application of "corrections" of ± 1.5% or more to test results.

CIE/IEC 953-1 (Reference 8)

This testing code is very similar to ASME PTC 6. The drafting committee was apparently unable to reconcile the disparate philosophies of ASME PTC 6 and DIN-1943 and decided to publish two alternative procedures, 953-1 and 953-2.

CIE/IEC 953-2 (Reference 8)

This testing code is very similar to DIN-1943, including the numerous tolerance allowances.

Station Instrument Tests

Testing with station instruments is sometimes used to verify the performance of small steam turbines. The lack of calibration and inspection of instrumentation generally leads to high measurement uncertainties (greater than ± 5.0%). With such an inaccurate test, it is impossible to demonstrate conclusively performance deficiencies even as great as several percent.

The wide range of possible measurement uncertainties and other tolerances allowed by the different performance testing codes shows the importance of selecting the performance test procedure to be specified and ensuring that quoted performance is provided on the basis of the same test procedure.

CONCLUSION

A number of the concepts and issues associated with steam turbine cycle optimization, evaluation and performance testing have been discussed. The topics discussed and suggestions put forward should help plant and steam turbine designers work together more effectively in optimizing and evaluating steam turbine design and cycle alternatives. The importance of an effective technical interface to proper optimization and evaluation should also be evident.

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LIST OF FIGURES

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