



GE Power Generation

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Klaus M. Retzlaff is a senior steam turbine product design engineer in GE's Power Generation group. He has worked in GE's steam turbine design engineering organization for over twenty years. Before joining GE, Klaus worked in Germany for two German steam turbine suppliers.

Prior to assuming his present position, Klaus was a technical leader in various mechanical and thermodynamic steam turbine design functions. He has co-authored several technical papers, some on the subject of ultrasupercritical steam turbine designs. He has received a U.S. patent for the design of a single-shaft combined cycle steam turbine.



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STEAM TURBINES FOR ULTRASUPERCRITICAL POWER PLANTS

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INTRODUCTION

The history of steam turbine development can be described as an evolutionary advancement toward greater power density and efficiency. Power density is a measure of the amount of power that can be efficiently generated from a steam turbine of a given physical size and mass. Improvements in the power density of steam turbines have been driven largely by the development of improved rotor and bucket alloys capable of sustaining higher stresses and enabling the construction of longer last stage buckets for increased exhaust area per exhaust flow. Improvements in efficiency have been brought about largely through two kinds of advancements. The first type of advancement is improvement in mechanical efficiency by reduction of aerodynamic and leakage losses as the steam expands through the turbine. The second type of advancement is improvement in the thermodynamic efficiency by increasing the temperature and pressure at which heat is added to the power cycle. The focus of this paper is predominantly on the latter type of efforts to advance the state-of-the-art in steam turbine technology.

EXPERIENCE

Efforts to increase the efficiency of the Rankine cycle by raising steam pressures and temperatures are not new. Early steam turbines produced at the turn of the century were designed for inlet pressures and temperatures of approximately 200 psi, 500 F (13.7 bar and 260 C), respectively. As time progressed and average unit size increased, main steam temperatures

and pressures also increased. The 1950s was a period of rapid growth in average power plant size with the average unit shipped by GE increasing from 38 MW in 1947 to 156 MW in 1957. During this period, the reheat cycle became well established commercially and maximum steam conditions were raised from 2400 psi / 1000 F (165 bar / 538 C) up to those of the experimental units at the Philo power station with inlet conditions of 4500 psi, 1150 F / 1050 F / 1000 F (310 bar, 620 C / 566 C / 538 C). This effort provided the basic knowledge that led to placing in service, in 1960, several large capacity cross-compound units with modest, but still for the time challenging, steam conditions of 3500 psi, 1050 F / 1050 F / 1050 F (241 bar, 566 C / 566 C / 566 C). At this time a 325 MW 2400 psi, 1100 F / 1050 F / 1000 F (165 bar, 593 C / 566 C / 538 C) unit was also commissioned.

By 1969, a simpler tandem-compound double reheat design was placed into service that combined 3500 psi, 1000 F (242 bar, 538 C) high pressure and 1025 F/552 C first reheat turbine sections in a single opposed-flow casing. The second reheat flow section at 1050 F/566 C was designed in a double-flow configuration to provide adequate volume flow capability and to confine the highest temperature conditions to the middle of the casing^[1]. The cross section in Figure 1 illustrates this design, which has experienced exceptionally good reliability while exceeding performance expectations.

In addition to units with double reheat, during the 1960s and 1970s GE placed into service numerous supercritical units with single reheat and nominal steam conditions of 3500 psi, 1000

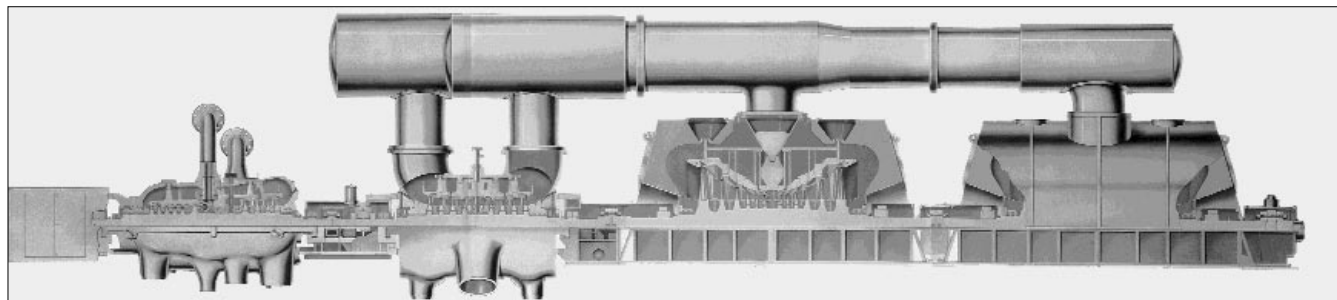


Figure 1. Tandem-Compound Double-Reheat Supercritical Steam Turbine

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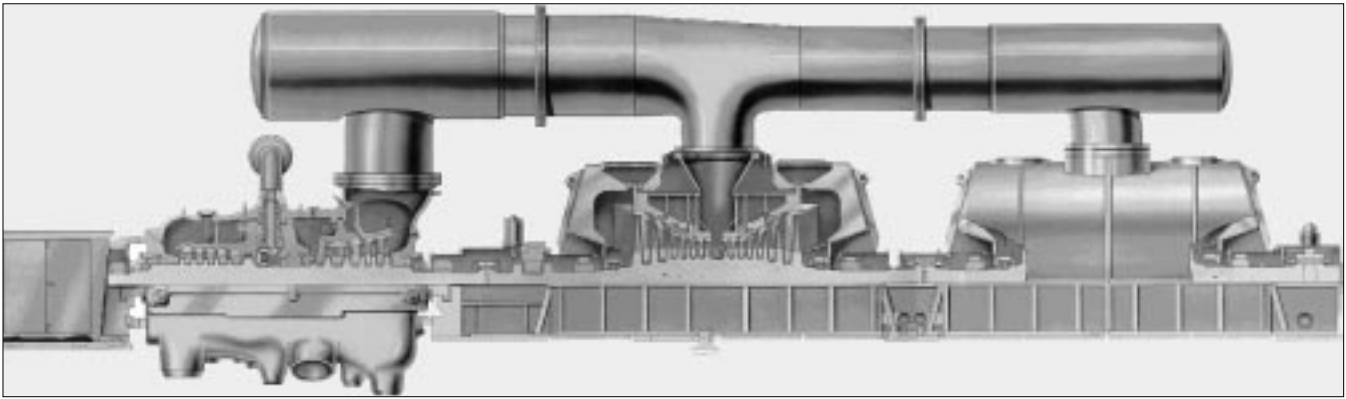


Figure 2. Tandem-Compound Single-Reheat Supercritical Steam Turbine

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F / 1000 F (241 bar, 538 C / 538 C) as shown in Figure 2. These units ranged in size from 350 MW to 1103 MW. Included were units of tandem-compound design ranging in size between 350 MW and 884 MW.

The combination of experience with single and double reheat units, together with the knowledge gained on the advanced steam condition designs of the 1950s, served as the basis for several Electrical Power Research Institute (EPRI) studies conducted during the 1980s of double-reheat turbines designed for operation at the advanced steam conditions of 4500 psi, 1100 F / 1100 F / 1100 F (310 bar, 593 C / 593 C / 593 C). Such designs have been offered for a number of years and although there appears to be little interest in the United States for advanced steam conditions, other countries, most notably in Asia and northern Europe, have pursued this option. An example of a recent advanced steam turbine generator recently designed by GE is a single-reheat cross-compound unit for operation with main steam conditions of 3626 psi, 1112 F (250 bar / 600 C) and reheat steam temperature of 1130 F/610 C. This unit is being executed in a four-casing design with separate high-pressure and intermediate-pressure sections on the full speed shaft and two double-flow LP turbines on the half-speed shaft.

THERMODYNAMIC CYCLE OPTIMIZATION

Effect of Higher Steam Conditions on Unit Performance

As the first step in the optimization of cycle steam conditions, the potential cycle efficiency gain from elevating steam pressures and temper-

atures needs to be considered. Starting with the traditional 2400 psi / 1000 F (165 bar / 538 C) single-reheat cycle, dramatic improvements in power plant performance can be achieved by raising inlet steam conditions to levels up to 4500 psi/310 bar and temperatures to levels in excess of 1112 F/600 C. It has become industry practice to refer to such steam conditions, and in fact any supercritical conditions where the throttle and/or reheat steam temperatures exceed 1050 F/566 C, as “ultrasupercritical”. Figure 3a illustrates the relative heat rate gain for a variety of main steam and reheat steam conditions for single-reheat units compared to the base 2400 psi, 1000 F / 1000 F (65 bar, 538 C / 538 C) cycle.

Double Reheat vs. Single Reheat

It has long been understood that improved

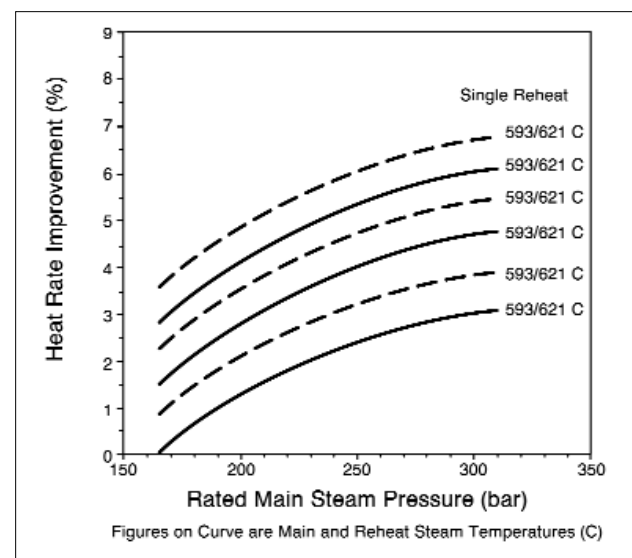


Figure 3a. Heat Rate Improvement from Steam Cycle with Ultrasupercritical Steam Conditions

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plant performance is possible by employing a double, rather than single, reheat cycle. These performance benefits were recognized by utilities in the 1960s and, as a result, many double-reheat machines were built by GE [1]. The benefit of using the double reheat cycle is further enhanced by the feasibility of using ultrasupercritical pressures and temperatures. During the mid-1980s, an extensive development project under the auspices of EPRI led to the design of large ultrasupercritical 4500 psi, 1100 F / 1100 F / 1100 F (310 bar, 593 C / 593 C / 593 C) double reheat units with gross output of 700 MW and below [2,3]. Figure 3b demonstrates the performance gains possible by utilizing a double reheat cycle at various steam conditions.

For any particular application, the heat rate gain possible with the double reheat cycle will have to be evaluated against the higher station costs attributable to greater equipment complexity in the boiler, piping systems and steam turbine. The result of this trade-off will depend

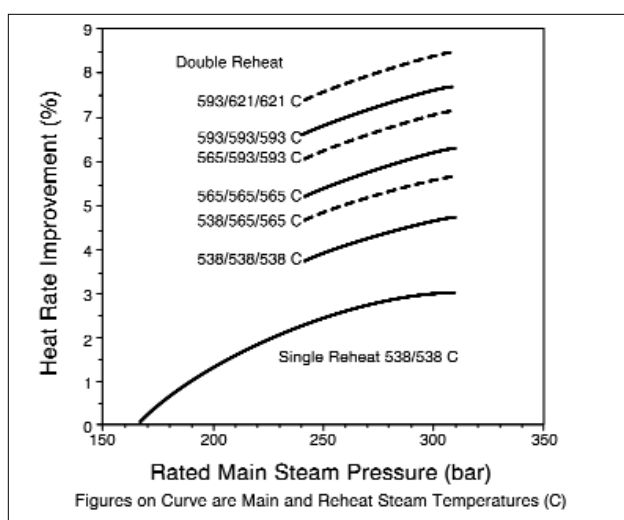


Figure 3b. Heat Rate Improvement from Steam Cycle with Ultrasupercritical Steam Conditions GT25591

heavily on local site conditions, fuel costs and environmental requirements.

Heater Selection and Final Feedwater Temperature

In order to maximize the heat rate gain possible with ultrasupercritical steam conditions, the feedwater heater arrangement also needs to be optimized. In general, the selection of higher steam conditions will result in additional feedwater heaters and a economically optimal higher final feedwater temperature. In many cases the selection of a heater above the reheat point (HARP) will also be warranted. The use of a separate desuperheater ahead of the top heater for units with a HARP can result in additional gains in unit performance.

The use of a HARP and the associated higher final feedwater temperature and lower reheater pressure have a strong influence on the design of the steam turbine and will be discussed in more detail below.

Other cycle parameters such as reheater pressure drop, heater terminal temperature differences, line pressure drops and drain cooler temperature differences have a lesser impact on turbine design, but should also be optimized as part of the overall power plant cost/performance trade-off activity. Table 1 shows typical gains for different heater configurations associated with a 4500 psi, 1100 F / 1100 F (310 bar, 593 C / 593 C) single reheat cycle and a 1100 F / 1100 F / 1100 F (593 C / 593 C / 593 C) double reheat cycle. Figure 4 shows a typical single-reheat cycle featuring eight feedwater heaters including a HARP.

Reheater Pressure Optimization and Use of a HARP

The selection of the cold reheat pressure is an integral part of any power plant optimization

Cycle	No. of Feedwater Heaters	HARP	Heat Rate Benefit
Single Reheat	7	No	Base Case
	8	No	+0.2%
	8	Yes	+0.6%
	9	Yes	+0.7%
Double Reheat	8	No	Base Case
	9	No	+0.3%
	9	Yes	+0.2%
	10	Yes	+0.5%

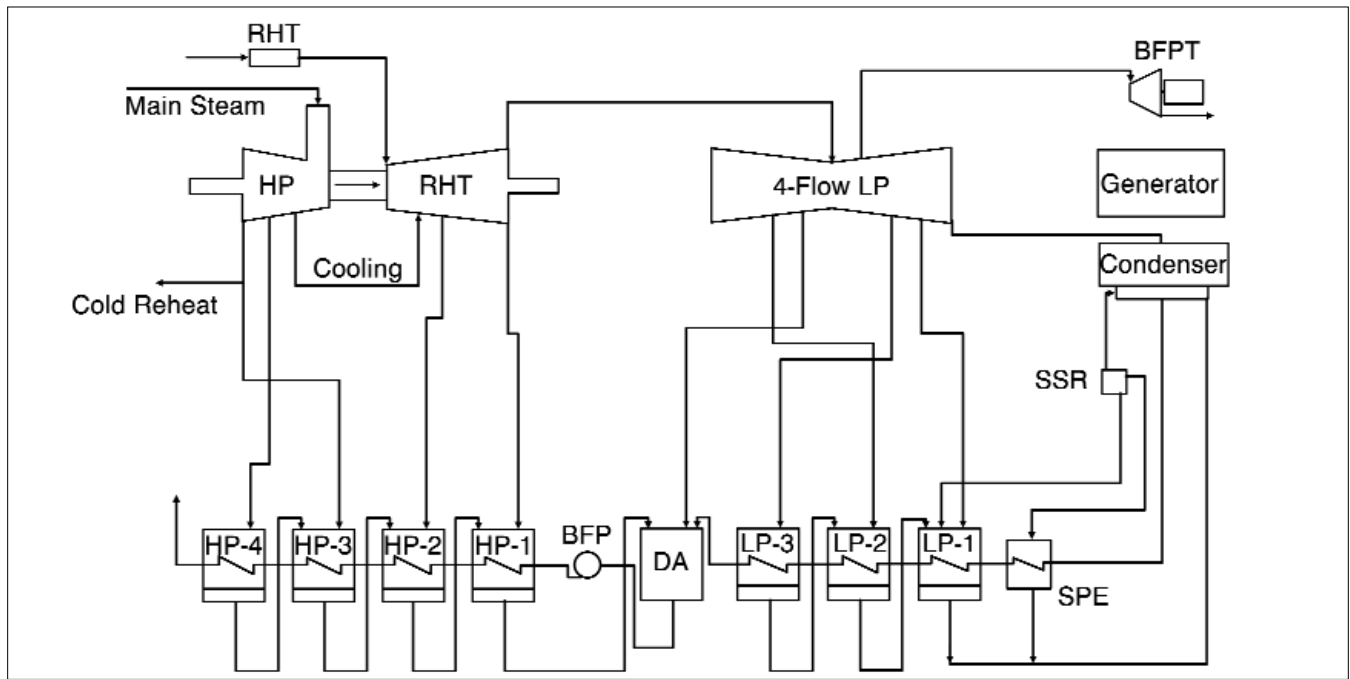
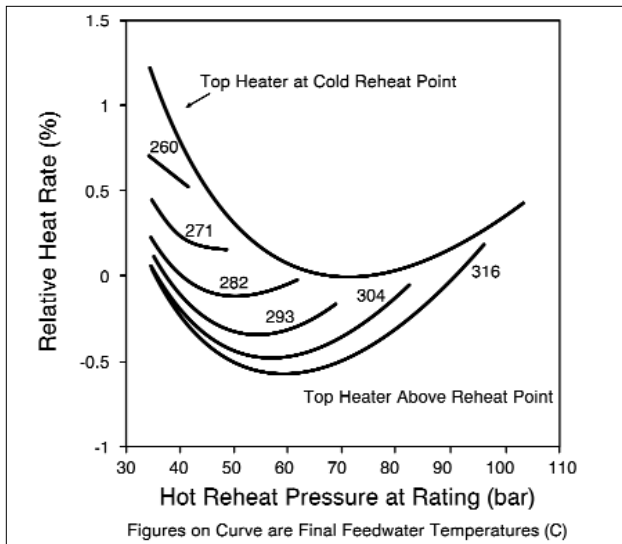


Figure 4. Typical Single Reheat Heater Cycle with Heater Above Reheat Point

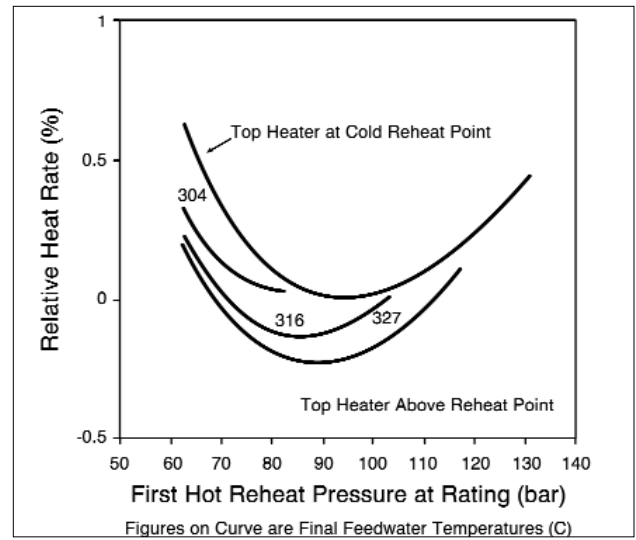
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Figure 5a. Effect of Final Feedwater Temperature and Reheat Pressure on Turbine Net Heat Rate

process, but becomes more important for plants with advanced steam conditions. Figure 5a shows the heat rate impact of different final feedwater temperatures for single-reheat units with advanced steam conditions. Comparing the heat rate at the thermodynamic optimum, the improvement resulting from the use of a HARP amounts to about 0.5%. However, economic considerations of the boiler design without a HARP will tend to favor a lower reheat pressure at the expense of a slight decrease in cycle



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Figure 5b. Effect of Final Feedwater Temperature and Reheat Pressure on Turbine Net Heat Rate

performance. Therefore, the resulting net heat rate gain is usually larger, approaching 0.6 - 0.7%.

The use of a HARP results in a lower optimal reheat pressure and a higher optimal final feedwater temperature. Both of these considerations significantly impact the design and cost of the boiler. As a result, careful plant-level cross-optimization needs to be done, in considering the use of a HARP, to ensure an economically optimal cycle selection is made.

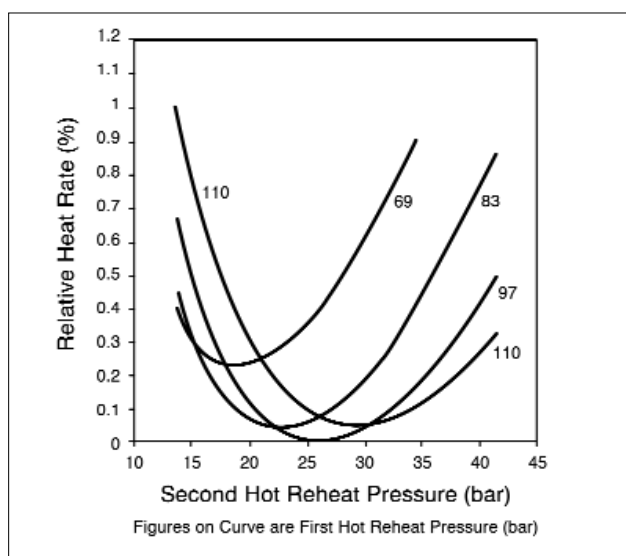


Figure 6. Reheat Pressure Cross Optimization for Double Reheat Units

Reheater Pressure Optimization for Double Reheat Units

For double reheat units, the above described optimization of various design parameters is more involved and has to include a cross-optimization process in order to properly select the first and second reheat pressures. For double reheat units without HARP, the best performance would be achieved with the first reheat pressure of approximately 1450 psi/100 bar. However, economic considerations associated with the boiler and piping systems would typically favor reducing this to a lower level. As with

single reheat units, the use of a HARP can significantly improve unit heat rate. This relationship is shown in Figure 5b.

An example of the cross-optimization of first and second reheat pressures is shown in Figure 6. The typical outcome is that the first reheat pressure is chosen below the thermodynamic optimum while the second reheat pressure is generally selected slightly above to reduce the LP inlet steam temperature. As shown in Table 1, the double reheat cycle can be further improved by using an additional low pressure and/or high pressure heater. A typical double reheat cycle with ten feedwater heaters, including a HARP, is shown in Figure 7.

Crossover Pressure Optimization

The use of advanced reheat steam conditions strongly affects the inlet temperature to the low pressure (LP) turbine section. An increase in hot reheat temperature translates into an almost equal increase in crossover temperature for a given crossover pressure. However, the maximum allowable LP inlet temperature is limited by material considerations associated with the rotor, crossover and hood stationary components. Of these, the rotor material temperature limits are usually reached first.

Two basic parameters can be varied to adjust the LP inlet temperature for a given hot reheat temperature: reheater pressure and crossover pressure. To lower the crossover temperature, the reheater pressure has to be increased or the crossover pressure has to be decreased. As shown in Figure 5a, there is a direct correlation

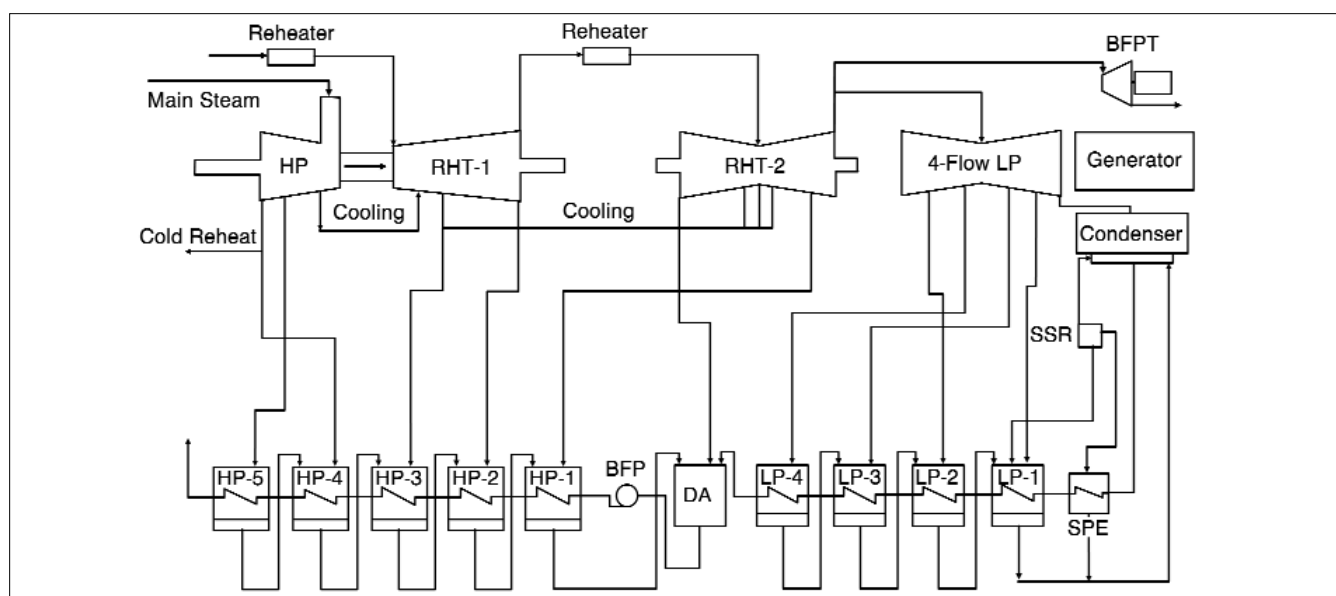
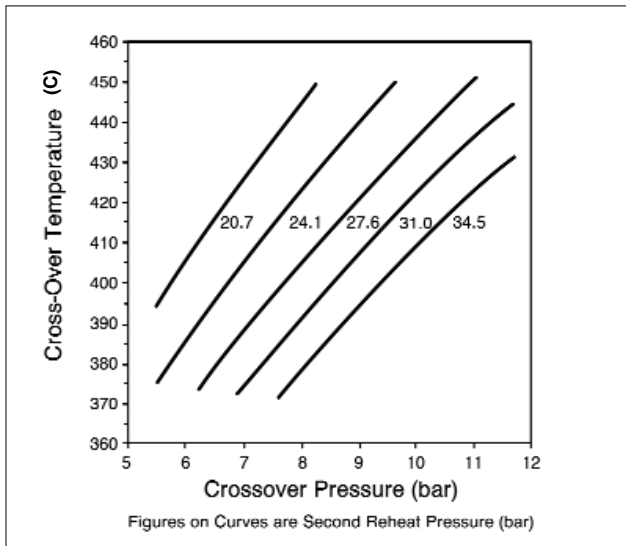


Figure 7. Double Reheat Cycle with Heater above Reheat Point

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Figure 8. Crossover Temperature vs. Second Reheat Pressure

between reheat pressure and unit performance. Since the use of a HARP is likely to be the economic choice for most ultrasupercritical cycles, the reheater pressure will be lower to maximize the heat rate gain from the HARP. This, unfortunately, will result in increased crossover temperatures.

This effect can be offset by lowering the crossover pressure by an equivalent pressure ratio. However, this tends to increase the energy on the reheat section which, in turn, increases the number of stages and results in longer bearing spans. Also, the crossover volume flow increases and could present a limitation for very large ratings. The correlation between crossover inlet temperature and second reheat pressure is

shown for double reheat units in Figure 8. The relationship is similar for single reheat units.

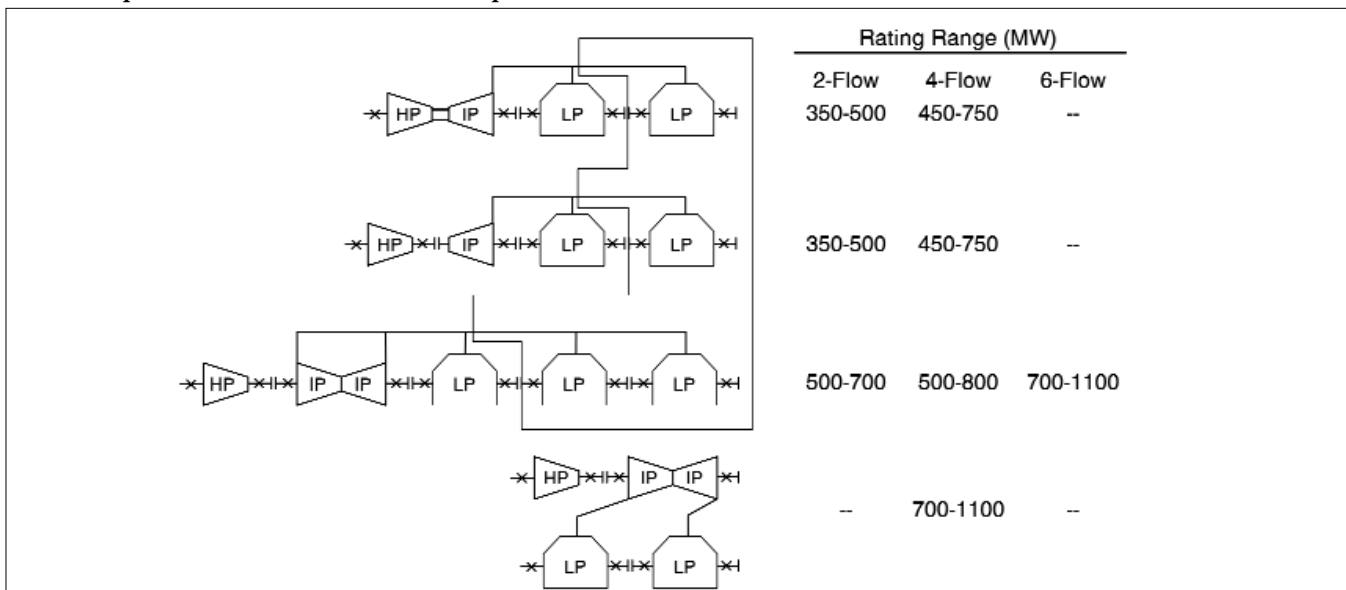
STEAM TURBINE DESIGN & MATERIAL SELECTION

Steam Turbine Configurations

The appropriate steam turbine configuration for a given ultrasupercritical application is largely a function of the number of reheats selected, the unit rating, the site backpressure characteristics and any special requirements such as district heating extractions.

Single Reheat Power Generation Applications

The available configurations for single-reheat applications are shown in Figure 9. For most applications, an opposed flow HP/IP section in a single casing can be utilized. This section would be combined with either one or two double-flow LP sections depending on the actual rating and design exhaust pressure. The use of the combined HP/IP section makes possible a smaller overall power island with its resultant savings in turbine building, foundation and maintenance costs. Supercritical units with this type of construction have operated successfully at ratings above 600 MW for many years. To meet the requirements of specialized applications and customer preferences, single-flow HP and IP sections in separate casings are also available. The HP and IP turbine cross-sections of these two configurations are shown in Figures 10 and 11 respectively.



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Figure 9. Single-Reheat Ultrasupercritical Product Line

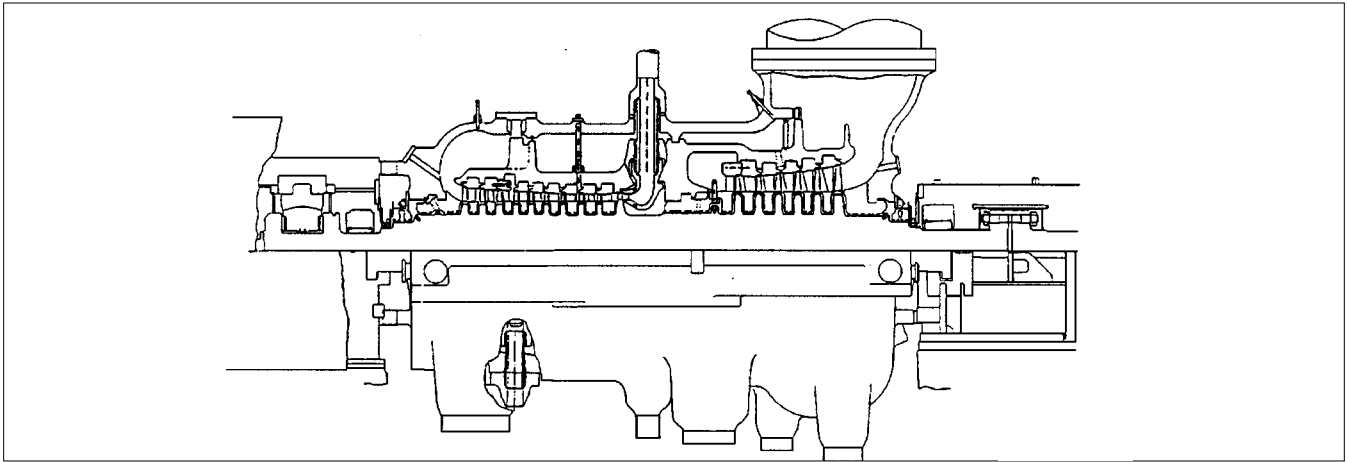


Figure 10. Combined HP/IP Section of Ultrasupercritical Turbine

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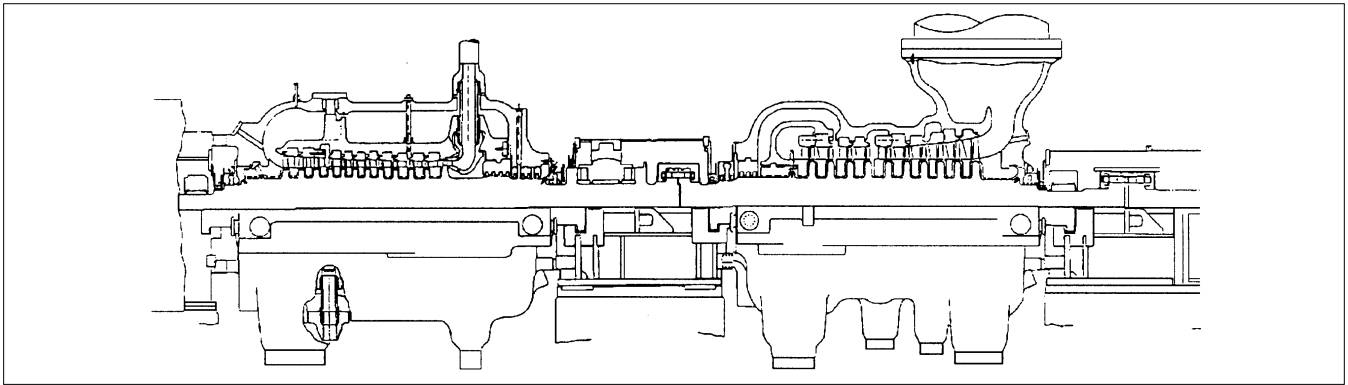


Figure 11. Separate HP and IP Sections of Ultrasupercritical Turbine

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As unit rating increases, stability requirements and last IP bucket length make a configuration utilizing a single flow HP section and double flow IP section in separate casings the appropriate selection. These two high temperature sections can be combined with one, two or three double-flow LP sections depending on the design exhaust pressure. Tandem compound

configurations of this type with three LP sections are capable of the highest unit ratings currently contemplated for ultrasupercritical power plants. The HP and RH cross-section of such a unit is shown in Figure 12.

For the highest unit ratings and those instances where the customer prefers it, cross-compound units are also available. These units

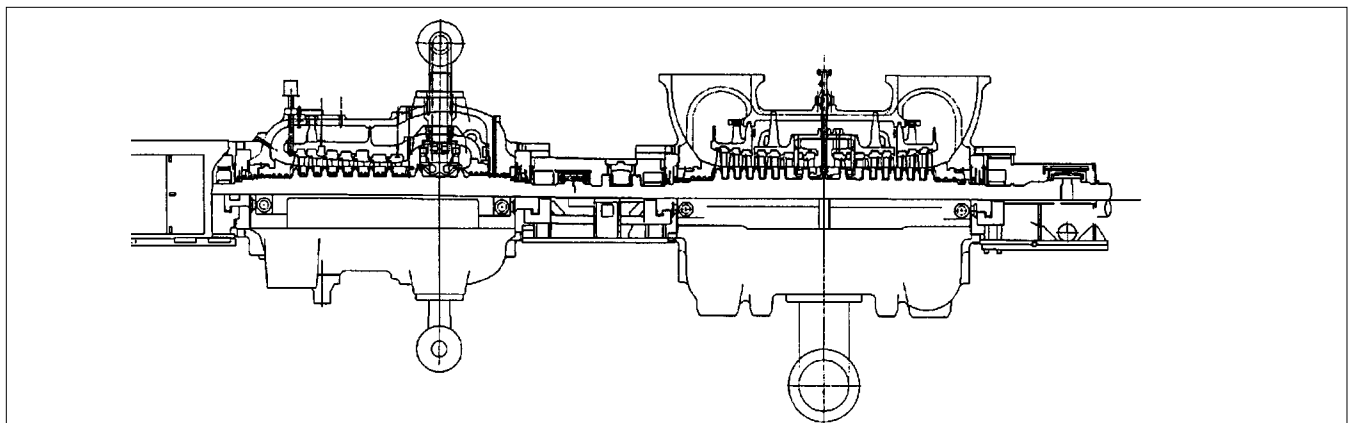
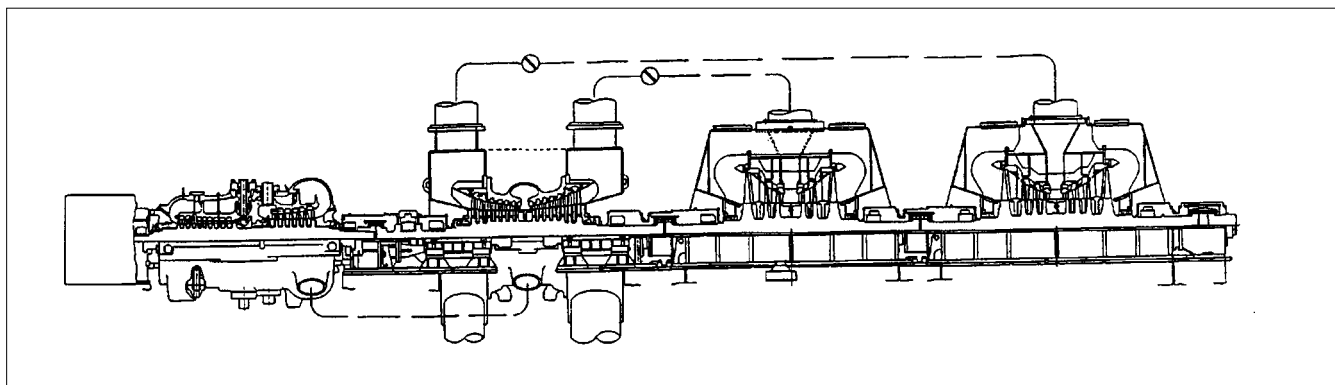


Figure 12. Separate HP and Double-Flow IP Sections of Ultrasupercritical Turbine

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Figure 13. Ultrasupercritical Steam Turbine Designed for 2-Stage District Heating Application

include a full speed shaft line having a single-flow HP section and a double-flow IP section, as described, above driving a two pole generator. A second half-speed shaft line consisting of two double-flow LP sections driving a four pole generator is also included. Steam exhausting from the IP section of the full-speed shaft-line is fed to the inlet of the LP sections in the half-speed shaft line via two crossovers.

Single Reheat District Heating Applications

A number of single-reheat ultrasupercritical projects have been used for district heating applications and this requirement can significantly affect both the steam cycle parameters and turbine configuration. The optimal turbine configuration that meets the functional requirements of district heating operation as well as the high performance and economical turbine island arrangement, will depend primarily on the need for controllability of district heat over the load range. A study done recently on a 440 MW ultrasupercritical district heating application concluded that if part load district heat controllability is not a requirement, a compact three-casing configuration using an opposed flow HP/IP section, such as that shown in Figure 10, was the best choice from a systems cost perspective. With this configuration, the district heaters would be fed from uncontrolled extractions in the LP sections and control would be achieved on the water side of the district heating system [4].

In district heating applications where part load district heat controllability is a requirement, a four-casing configuration such as that shown in Figure 13 is more appropriate. This configuration, which was developed for another 400 MW ultrasupercritical application features a first casing containing the HP section and the single flow portion of the IP section in an opposed-flow arrangement. Exhaust from the

single-flow IP section is directed into a separate double-flow asymmetrical IP section in a separate casing. The two district heating extractions are taken from the exhausts of this casing and the district heating pressure is controlled by way of butterfly valves in the crossovers to the LP sections. In comparison to an alternative construction with totally separate HP and IP sections, the use of single-flow IP staging for the first part of the reheat expansion enables longer buckets with associated better stage performance. Additional benefits include confining all the high temperature steam to the center of the first section, better rotor cooling steam utilization and overall reduced machine length.

Double Reheat Applications

The available configurations for double-reheat applications are shown in Figure 14. For many applications, a single-flow HP section in its own casing can be combined with a second casing having the two reheat sections in an opposed flow arrangement. The high pressure and reheat sections are directly coupled to one, two or three double-flow LP sections depending on the application rating and design exhaust pressure.

For units of higher rating, a configuration with a single-flow HP section and single-flow first reheat section, located in a common casing and coupled to a double-flow second reheat section in a separate casing, is utilized. As with the configuration described above, the high temperature sections are directly coupled to one, two or three double-flow LP sections based on the rating and exhaust pressure. Figure 15 shows a cross-section of the HP and RH sections of such a design.

For units of the highest rating, a cross-compound configuration can be used. This configuration would utilize a full-speed shaft line having sections basically the same as the HP and RH

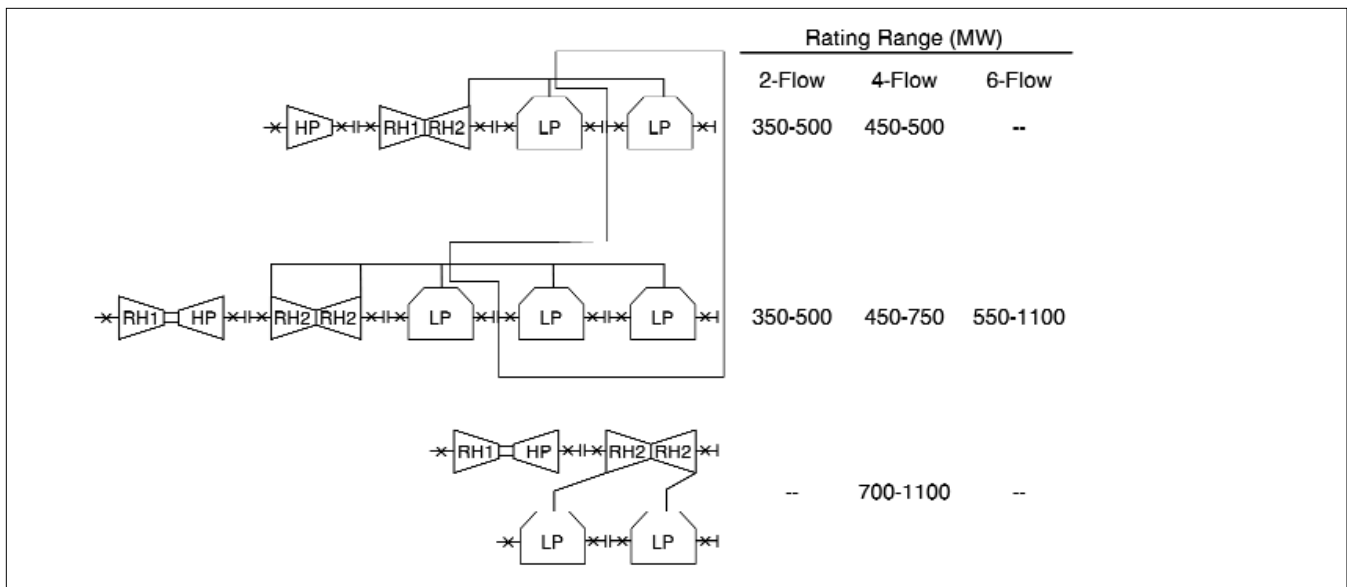


Figure 14. Double-Reheat Ultrasupercritical Product Line

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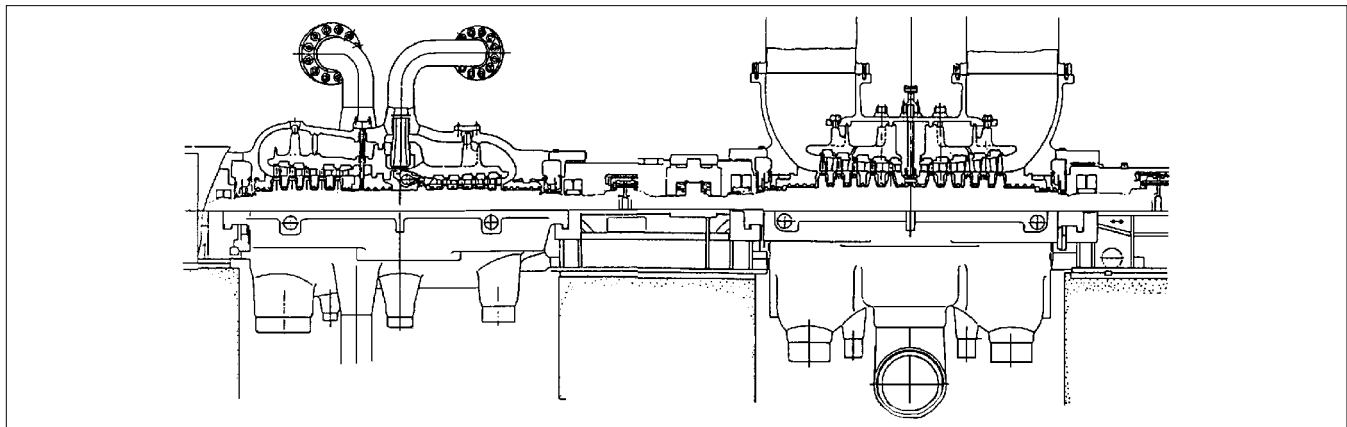


Figure 15. HP and Reheat Sections of a Double-Reheat Ultrasupercritical Turbine

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sections just described. Rather than being coupled to full-speed LP sections, these sections would be directly coupled to a 3600 or 3000 RPM generator. A separate half-speed LP shaft train similar to that used in single-reheat applications would be utilized in conjunction with the full-speed HP/IP shaft train.

Steam Turbine Component/System Design

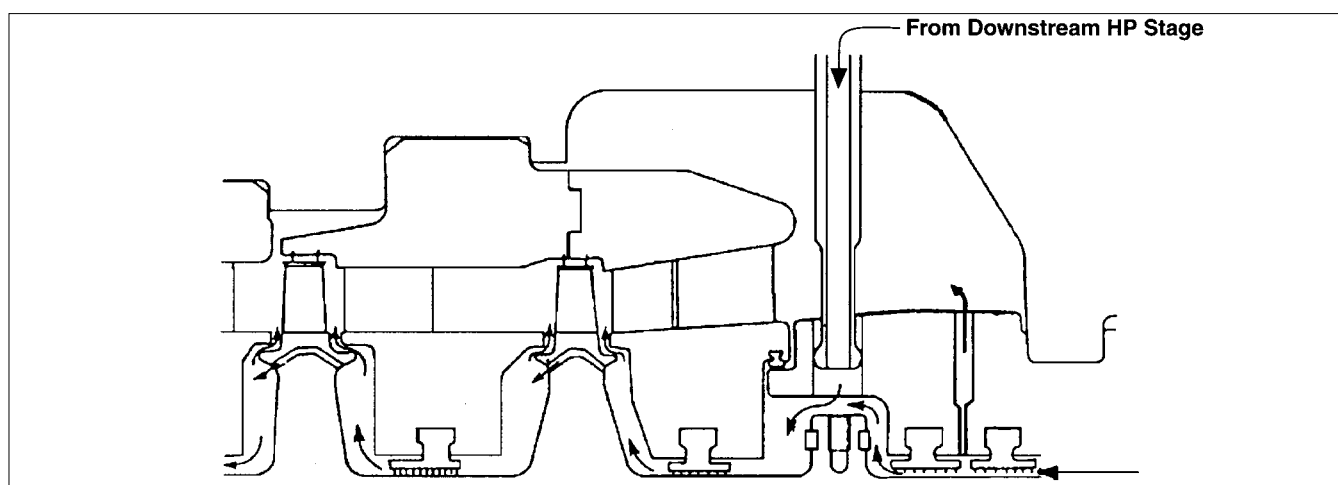
The design of high temperature steam turbines has evolved and is strongly influenced by the development of improved materials and by the use of more effective cooling steam arrangements. Both factors are discussed for the various critical components which are affected by advanced steam conditions.

Rotor Material

GE has extensive experience with two rotor alloy steels in high-pressure rotor applications: CrMoV and 12CrMoVCbN. The 12Cr steel is generally used when a higher rupture strength is required at elevated temperatures, or when a higher than normal operating temperature (1050 F/566 C) is required.

The first 12Cr rotor was placed into service in 1959. This material was developed and patented by the authors' company in anticipation of a market need for steam turbines capable of operating at ultrasupercritical steam temperatures. Since 1959, a total of 63 rotors have been built with 12Cr forgings. These rotors have successfully operated in some of the most challenging applications in units rated between 500 and 1000 MW.

The result of these extensive service experi-



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Figure 16. Reheat Stage Cooling Configuration for Opposed Flow HP/IP Sections

ences and long-term material tests has confirmed that the 12Cr rotor alloy has a rupture strength at 1100 F/593 C that is equivalent to the corresponding value for CrMoV material at 1050 F/566 C. Therefore, no compromise is required for the design of a high temperature rotor operating at 1100 F/593 C with the 12Cr material [5].

Weld Inlay of Rotor Bearing Journals

The 12Cr rotor material has very poor journal running characteristics due to its high chrome content. Under abnormal running conditions, the rotor journal surface can gall and parts of the surface can be chafed off, resulting in bearing failure. Traditionally, this problem was solved by employing shrunk-on low alloy journal sleeves. However, the use of shrunk-on sleeves also requires the use of shrunk-on couplings and, depending on the unit configuration, the use of shrunk-on thrust runners. Although these designs have been shown to operate reliably, current designs employ a low alloy weld inlay to the journal and thrust runner surfaces, which addresses the galling issue without resorting to the use of shrunk-on components. This approach provides the additional benefit of allowing the turbine designer to locate the thrust bearing in a position such that optimum clearance control in the HP section is achieved.

Rotor Cooling

At the elevated temperatures associated with ultrasupercritical applications, the first and second stage of the reheat sections generally require external cooling of the wheel and bucket dovetail region. This design approach has been successfully employed on many previously

built turbines utilizing conventional materials and operating at traditional temperatures.

For opposed flow HP/IP sections, the cooling steam is extracted from the third or fourth HP stage and re-admitted into the mid-span packing. To improve the cooling effectiveness, a portion of the mid-span packing leakage flow can be bled off prior to mixing. The HP/IP cooling scheme is shown in Figure 16.

For the first stage of a double-flow second reheat section, the cooling steam is extracted from the first reheat extraction stage and is piped into the upstream first stage wheel space below the double flow tub. By judicious use of bucket dovetail steam balance holes and root radial spill strips on both sides of the dovetail, it is possible to direct the cooler steam to the second stage upstream wheel space.

In all cases, the cooling steam effectiveness must be evaluated at full load and at the load point where the reheat temperature normally starts to drop off, typically at 40-50% load. This effect is shown in Figures 17 and 18.

High Temperature Bucket / Diaphragm Designs and Materials

Buckets for the early HP and reheat stages of steam turbines must have good high-temperature strength and low thermal expansion to minimize thermal stresses. For ultrasupercritical applications, a 10CrMoVCbN bucket alloy similar to the rotor forging alloy was developed. This alloy possesses a rupture strength nearly 50% higher at 1050 F/566 C than the AISI 422 alloy traditionally used in applications of up to 1050 F/566 C. Together with use of axial entry type bucket dovetails, judicious application of rotor cooling schemes, reheat pressure optimization

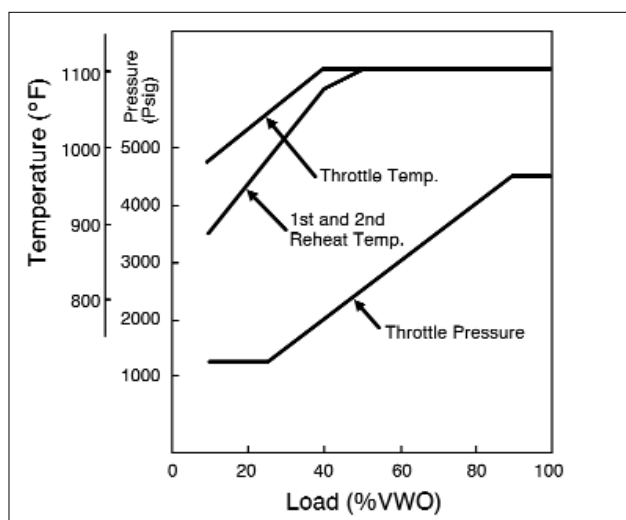


Figure 17. Typical Boiler Characteristic for USC Unit (Hybrid Pressure 310 bar, 395 C/593 C/593 C Cycle)

and the use of double-flow configurations for HP control stages at higher ratings, acceptable high temperature bucket designs can be achieved to cover the rating range of 350 MW to 1100 MW.

In all turbine sections employing 12CrMoVCbN rotors, diaphragms and packing casings are constructed of 12Cr material to match the thermal expansion characteristics of the 12Cr rotor material.

Shells and Nozzle Boxes

Low alloy CrMoV materials generally suitable for stationary components in turbines designed for conventional steam conditions are not suitable for the higher temperature regions of ultrasupercritical steam turbines. High strength martensitic stainless steel casting alloys (10CrMoVCb) were developed by the authors' company in the late 1950s for valve bodies and nozzle boxes in applications with 1050 F/566 C and 1100 F/593 C inlet temperatures. Last year, four large turbine shells were made from this material and work has been completed with a vendor to improve its producibility for large castings.

HP sections of ultrasupercritical steam turbines generally utilize triple-shell construction to minimize the thermal and operating stresses the various pressure containment parts are subjected to. The highest pressures and temperatures are borne by a nozzle box constructed of forged 12CrMoVCbN steel. The inner shells are constructed of cast 10CrMoVCb or CrMoV material depending on the specific temperatures associated with the ultrasupercritical appli-

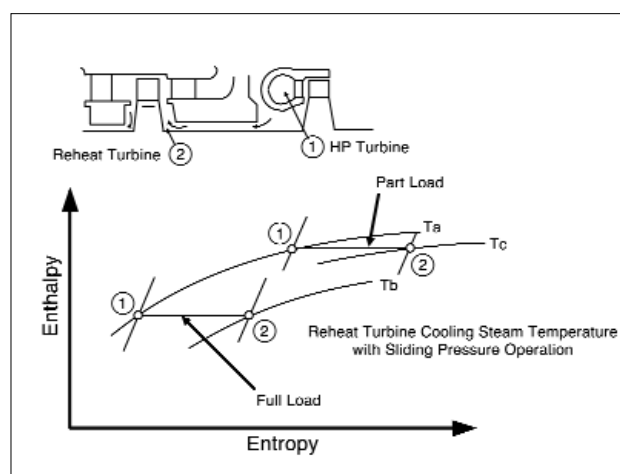


Figure 18. Effect of Part Load Operation on Cooling Effectiveness

cation. With this type of construction, the outer shell is not subjected to elevated temperatures and can thus be constructed of traditional CrMoV material.

The transition between the main steam leads and the outer shell has traditionally been designed as a flanged connection with thermal sleeves. Today's ultrasupercritical designs employ a welded connection. The welded connection is cooled by the cold reheat steam on the inner wall to a temperature level of 1025 F - 1050 F/550-565 C. To assure sufficient heat transfer near the weld, a small amount of steam is blown down to the next extraction point. Figure 19 illustrates the ultrasupercritical multi-shell HP section construction described above.

IP sections of ultrasupercritical turbines utilize double shell construction with the high temperature inner shell being constructed of cast 10CrMoVCb material and the outer shell and

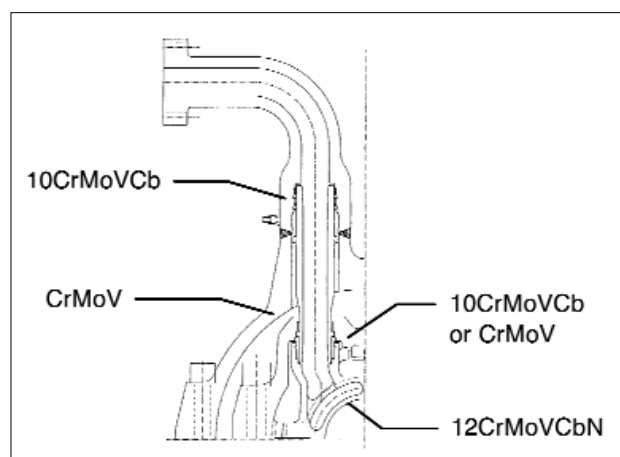


Figure 19. Main Steam Inlet Construction

low temperature inner shell constructed of traditional CrMoV material.

Advancements in finite element (FE) calculation capabilities enable designers to assess the local stress field in these high temperature components and, thus, selectively add material only where needed for strength purposes. This results in a shell design that satisfies all stress limitations and is thermally flexible to meet the shorter start-up times required by today's customers. Figure 20 shows an example of a FE mesh for an ultrasupercritical HP/IP inner shell. Figure 21 shows a typical stress plot for full load steady state conditions.

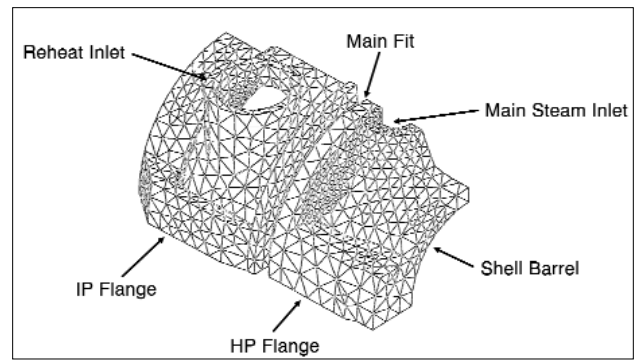


Figure 20. Finite Element Model of USC HP/IP Inner Shell GT25601

Bolting

For shell bolting applications at temperatures up to 1050 F/566 C, 12Cr alloys and low alloy steels have been used. However, the more demanding ultrasupercritical steam conditions exceed the capabilities of these materials, thus dictating the requirement for nickel-based alloys in high-temperature regions.

A comparison of candidate bolting materials possessing higher temperature strength was recently made and Inconel 718 was selected as the material possessing the best combination of all the bolting requirements. The use of Inconel bolts results in smaller bolt diameters and, therefore, narrower flanges. This, in turn, leads to lower transient thermal stresses during turbine start-ups. This material has been successfully used by the authors' company in gas turbine, aircraft engine and conventional steam turbine applications for many years.

LP Section Design

The primary LP section design issue associated with ultrasupercritical turbines is the elevated

crossover temperature that is frequently encountered with these power cycles. It has been found that conventional NiCrMoV rotor materials have a tendency to embrittle at LP bowl temperatures above 660 - 710 F/350 - 375 C. In order to avoid this phenomenon, past high temperature designs have used an internal cooling scheme that circulates the exhaust steam of the first LP stage into the upstream wheel space by virtue of special wheel hole scoops and a slightly negative root reaction. This design approach, however, results in a performance loss.

Studies performed by EPRI and others over the past several years have demonstrated that NiCrMoV material can be made virtually immune to embrittlement by reducing the levels of P, Sn, Mn and Si. Utilization of this "super-clean" chemistry combined with other enhancements such as raising the nickel content and gashing between the wheels prior to quenching, result in rotor forgings with superior embrittlement, fracture toughness and tensile ductility properties in comparison to previously available

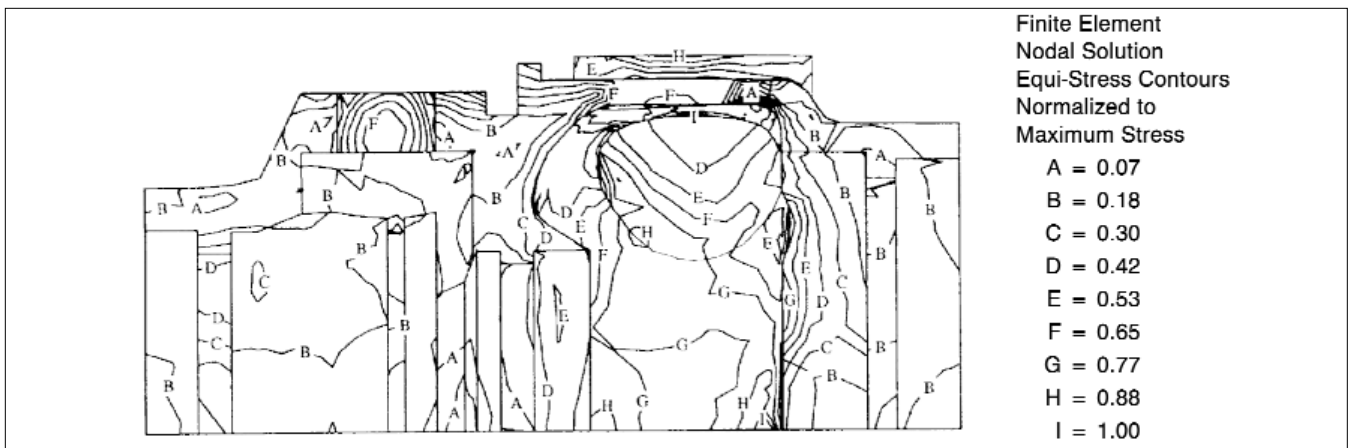
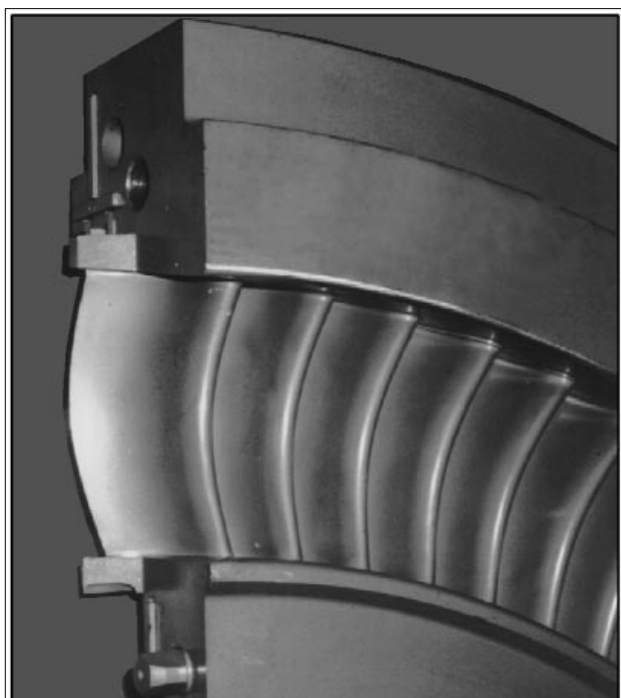


Figure 21. Predicted HP/RHT Inner Shell Stress Distribution at Peak Load (Normalized to Maximum Stress) HT25602



GT25603

Figure 22. Diaphragm Segment with Advanced Nozzle Partitions

NiCrMoV materials. This improvement provides additional freedom to optimize the cycle parameters, in particular the crossover temperature for double reheat units, to achieve higher efficiency levels without performance losses associated with previously used cooling schemes.

Advanced Steam Path Design

Recent years have seen the rapid advancement of computational fluid dynamics (CFD). Based on this new capability, turbine components can be better optimized for reduced flow losses [6]. The performance of steampath components such as nozzles, buckets and seals have been significantly enhanced as a result of applying this new technology and the resultant performance gains have been verified both in test turbines and operating units. A segment of an IP section diaphragm utilizing advanced nozzle partition designs is shown in Figure 22.

In addition to the performance improvements attributable to CFD in the steampath, performance gains can also be achieved by optimizing stationary components such as valves, inlets and exhausts using the same tools. All ultrasupercritical designs in the future will incorporate these CFD-based design enhancements.

CONCLUSION

Increased fuel costs, improved technology and an a heightened focus on reducing power plant emissions have combined to revitalize power industry interest in coal-fired power plants utilizing ultrasupercritical steam conditions. To achieve an economically optimized plant, the cycle conditions under which these plants operate need to be carefully evaluated, taking into account such parameters as the number of reheats employed, inlet steam conditions and feedwater heater arrangement. A variety of steam turbine configurations for ultrasupercritical applications are available. Each of these configurations utilizes materials and design features appropriate to ensure long turbine life with reliability levels comparable to conventional designs.

Note: This paper was originally presented at Power Gen Europe '96.

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