HIGH PERFORMANCE STEAM DEVELOPMENT

**Final Report
Phase III**

**1500°F Steam Plant for
Industrial Cogeneration
Prototype Development Tests**

MASTER

Solar Turbines

A Caterpillar Company

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HIGH PERFORMANCE STEAM DEVELOPMENT

Final Report Phase III

1500°F Steam Plant for Industrial Cogeneration Prototype Development Tests

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EXECUTIVE SUMMARY

1.1 HIGH PERFORMANCE STEAM SYSTEM SUCCESSFULLY TESTED

As a key part of DOE's and industry's R&D efforts to improve the efficiency, cost, and emissions of power generation, a prototype High Performance Steam System (HPSS) has been designed, built, and demonstrated. The world's highest temperature ASME Section I coded power plant (see Figure 1) successfully completed over 100 hours of development tests at 1500°F and 1500 psig on a 56,000 pound per hour steam generator, control valve and topping turbine at an output power of 5500 hp. This development advances the HPSS to 400°F higher steam temperature than the current best technology being installed around the world. Higher cycle temperatures produce higher conversion efficiencies and since steam is used to produce the large majority of the world's power, the authors expect HPSS developments will have a major impact on electric power production and cogeneration in the twenty-first century.

Coal fueled steam plants now produce the majority of the United States electric power. Cogeneration and reduced costs and availability of natural gas have now made gas turbines using Heat Recovery Steam Generators (HRSG's) and combined cycles for cogeneration and power generation the lowest cost producer of electric power in the United States. These gas fueled combined cycles also have major benefits in reducing emissions while reducing the cost of electricity. Development of HPSS technology can significantly improve the efficiency of cogeneration, steam plants, and combined cycles. Figure 2 is a TS diagram that shows the HPSS has twice the energy available from each pound of steam when expanding from 1500°F and 1500 psia to 165 psia (150 psig, a common cogeneration process steam pressure).

This report describes the prototype component and system design, and results of the 100-hour laboratory tests. The next phase of the development program consists of building up the steam turbine into a generator set, and installing the power plant at an industrial site for extended operation. A host test site with suitable steam and electricity needs is currently being sought.

1.2 BACK PRESSURE STEAM TURBINE PRODUCES TWICE AS MUCH POWER IN COGENERATION

Because of the very high fuel utilization efficiency of back pressure steam turbines in cogeneration, this configuration was chosen for the initial development in a combined cycle arrangement that again doubles the amount of electricity from the steam turbine for each pound of steam to the process. Other cycles such as topping systems for coal burning steam power plants (new or repowered), large combined cycles for power production, Advanced Turbine System (ATS) and integrated coal gasification combined cycles can in general, improve efficiency from 2% to over 15% by application of HPSS technology. Since the U.S. fuel costs are approximately fifty billion dollars per year to raise steam for electric power generation and industrial processes (over 30 quads of energy), it is important to move this technology into field demonstrations. Its importance to the environment is also critical since more energy is consumed in raising steam than any other energy sector. An industrial sized combined cycle for cogeneration was selected by DOE from a number of proposals to advance steam cycle conditions to 1500°F. Figure 3 schematically shows a gas turbine with a supplementary fired HRSG producing steam to be expanded through a back pressure steam turbine exhausting at typical process pressures, up to 200 psig. A combined cycle arrangement supplementary firing the waste exhaust heat to reduce the HRSG stack losses to a minimum produces the maximum possible amount of electricity per pound of steam to the process, at fuel utilization efficiencies in the 90% range.

Figure 1. HPSS in Final Test with OTSG on right connected by expansion loop to control valve and B.P. Steam Turbine connected through a reduction gear to a 6000 BHP water brake dynamometer.

Figure 2. Temperature-entropy of 1500°F steam cycle compared to 900°F steam in an industrial back pressure steam turbine cogeneration system.

Figure 3. HPSS combined cycle with exhaust firing and back pressure steam turbine for cogeneration.

1.3 HPSS INTEGRATED INTO COMBINED CYCLE FOR INITIAL APPLICATION

A small combined cycle (Figure 4) was selected since it had practical development costs and could extend the high efficiency combined cycle technology normally used for large gas fueled cogeneration into the more numerous smaller industrial and commercial sites with steam process requirements. Reference 1 the Phase I Final Report and Reference 2 discuss in detail the cycle optimization and marketing studies that conclude that the application of combined cycles with a backpressure steam turbine exhausting into a process have significant benefits in performance and high market potential.

If all steam cogeneration processes in the United States were to use the HPSS concept, the amount of electricity produced from these processes doubles. The incremental power is produced at 80 to 90% efficiency.

1.4 HPSS BASED ON ONCE-THROUGH STEAM GENERATOR AND GAS TURBINE TECHNOLOGIES

To develop an HPSS, a synthesis of the latest technology in Once-Through Steam Generators (OTSG), superalloy tube process developments and gas turbine technologies was proven to be successful. The most critical phase was to demonstrate an OTSG module with full length tube circuits for 500 hours operation at 1500°F and 1500 psia. Figure 5 shows the 500 hour proof-of-concept module during final instrumentation. The OTSG module had four circuits to minimize fuel costs in a 500 hour test. It was based upon Solar Turbines' patented OTSG technology being marketed by Innovative Steam Technologies (IST) for combined cycles. They are the simplest configuration of a steam generator possible, eliminating many components required by traditional drum HRSG's. At this time 19 OTSG's are in operation in combined cycles at traditional combined cycle steam conditions. Reference 3 describes the OTSG concept, water treatment, controls, and materials that are used in the innovative once-through steam generation technology. All of the OTSG's operating in the field use alloy 800 tubes and carbon steel fins.

Figure 4. HPSS configured as an 8 to 11 MW combined cycle with gas turbine and BPST (or 4 MW with fan only) and 70,000 pph steam to process.

Figure 5. Once-through steam generator module being instrumented prior to 500 hour proof-of-concept test at 1500°F and 1500 psig.

1.5 500 HOUR OTSG TEST PROVES CONCEPT OF 1500°F STEAM GENERATION

OTSG design is ideally matched to very high temperature and pressures well into the supercritical range. For the HPSS steam generator, four rows of screen tubes are used upstream of the final superheater (Figures 6 and 7). Alloy 800 is used for all tubes until metal temperatures reach 1300°F. Inconel 617 (ASME Code rated to 1650°F) is used for the superheater when tube metal temperatures exceed 1300°F. Fin materials change from carbon steel at 950°F to stainless 321, and to stainless steel 310 above 1400°F fin temperatures. All sections of the unit are finned (except for radiation screens made of alloy 800) to minimize the quantity and cost of the superalloy material used. A natural gas combustion system increases the inlet temperature of gases to 2500°F, and a total of 517 hours of operation at rated conditions with a highly instrumented OTSG proved the design concept. Post test inspection of the tubes showed an adherent and protective oxide less than 12 microns thick on both the steam and gas side surfaces. No evidence of spallation or scaling was present. This data correlated well with earlier bench scale testing to 4000 hours of superalloys in steam over 1500°F and 1500 psig.

1.6 COMMERCIAL SCALE OTSG DESIGNED AND FABRICATED FOR 1500°F

Successful results of the 500 hour test initiated the development of a commercial scale HPSS power plant. The HPSS included an OTSG with a full complement of twenty-six circuits (Figure 7), a unique ANSI 9000# class control valve (Figure 8), and a 1500°F steam turbine (Figures 8 and 9) based upon gas turbine technology and materials. Several design modifications based upon the 500 hour OTSG module test were incorporated to improve the commercial OTSG design and reduce costs. The most important was to obtain a production process to replace the final superheater machined from bar stock. A critical cost improvement was the development of a process to cold draw seamless Inconel 617 tubes with a one inch outside diameter and a 0.28 inch wall thickness. Inco International developed this process under a subcontract, and made all of the final superheater tubes as seamless cold drawn tubes in a final form for a production price of 32 dollars per pound (current 1995 price). For a commercial steam generator matched to a 5 to 10 MW gas turbines, this equates to an incremental tube cost for all the 617 tubing of about \$95,000 for the approximately 3000 pounds of superalloy. The incremental power gained is from adding a back pressure steam turbine. The system is in the range of 4 MW valued approximately at 4 million dollars at the installed power plant level. In a similar analysis, the incremental cost of the 200 pounds of superalloys in the steam turbine was shown to incrementally increase the turbines cost by about \$8000 (material cost only). Although many other incremental costs are associated with the ability to go to 1500°F, the highest single cost is the superalloy tubes in the steam generator and they appear to be manageable in a competitive sense at the installed cost gains in electric power production.

Incorporating design and process improvements, the 1500°F OTSG was fabricated from separate OTSG circuits (Figure 10) and instrumented (Figure 11). To match 5 to 7 MW gas turbines 26 once through circuits were welded together in parallel to form a simple OTSG. For larger turbines this circuit consisting of 14 foot long 1.0 inch O.D. tubes can be economically connected in parallel, using up to 75 circuits to form nearly a square frontal area 15 by 14 feet. This size OTSG could recover heat from gas turbines in the size range of 15 to 20 MW providing combined cycles in the range of 30 to 50 MW if used with a fully condensing configuration of steam turbines.

Figure 6. Prototype once-through steam generator flow schematic.

Figure 7. OTSG with 26 circuits producing 56,000 pph at 110 million BTU/hour fuel flow showing screen tubes and final superheater section.

Figure 8. Control valve with OTSG header in foreground and steam turbine in background with by-pass steam flow vertically downward.

Figure 9. Overhung back pressure steam turbine rotor 11.5 inch pitch diameter 30,000 rpm and 5600 BHP.

Figure 10. One of 26 identical OTSG flow circuits showing feedwater in at top right and 1500°F steam discharge at the top four rows from the combustion gas inlet side (left side).

Figure 11. OTSG circuits connected to superheater header and instrumented.

Carbon steel fins are used to provide about 66% of the heat transfer surface, and are 0.5 inch high with 6 fins per inch. At higher steam temperatures the fins are upgraded to stainless steel. The final superheater tubes are 0.28 wall thickness 1.0 inch O.D. tubes of Inconel 617. At lower temperatures in the final superheater the wall thickness is reduced. At less than 1300°F Alloy 800H is used, and Alloy 800 seamless tubing is used for the majority of the unit. Alloy 800 is standard for OTSG's currently in operation in the field at traditional combined cycle steam temperatures less than 1000°F. This alloy is used to allow the gas turbine to operate without water in the OTSG, and to simplify the water treatment system (see Reference 4).

The OTSG was designed to Section I requirements of the ASME Boiler Code. Each circuit and the assembled OTSG was hydrostatically proof tested to the ASME Section I requirements.

A low emissions natural gas combustion system was designed and matched to the OTSG. Figure 12 shows the air supply fan, combustor and exhaust gas recirculation system. A fan was used in the development tests in place of a gas turbine to save money and demonstrate a standalone steam cogeneration system. This configuration could be cost effective for the first field installation system. A Coen lean premixed combustion system designed for fresh air firing was used for tests. It was rated for 110 million BTU's per hour and an exhaust gas recirculation system is used to lower NOx.

1.7 CONTROL VALVE AND STEAM LINES

An ASME/ANSI B16.34 Section 8 9000# Special Class control valve (Figure 8) connects the steam generator header extension directly to the steam turbine. No piping in the ASME code definition is used since this one valve is sufficient as the boiler stop valve and turbine control valve for once-through steam generators.

Figure 12. HPSS viewed from combustor side showing vertical tube OTSG case and exhaust recirculation line into burner wind box, with fan in background.

Pipe bending stresses (other than friction) are eliminated by a unique arrangement of steam line configuration, spring hangers, sliding supports and flexing final superheater tubes. This arrangement has shown to have excellent behavior in over thirty five starts and turbine trips from 1500°F.

1.8 BACK PRESSURE STEAM TURBINE

To make a major advancement in steam turbine temperature technologies, several key design concepts were adapted from industrial gas turbines. It is critical to costs and technical success that the turbine be as small as practical. Small dimensions reduce thermal stresses and distortions during transient and steady state operation. Full admission with complete symmetry also reduce thermal stresses and distortions. Part dimensions must also be within the range of proven gas turbine superalloys and their manufacturing processes. A simple cooling concept was also incorporated to cool the outer case and turbine disc with exhaust steam.

Figure 13 is a schematic section of the turbine flow path. A 30,000 rpm single stage impulse turbine with 11.5 inch pitch diameter rotor (Figure 9) allows a small diameter machine to be used to limit thickness and weight of superalloys. When expanding from 1500°F and 1500 psia the turbine efficiency of 68% results in an exhaust temperature of 990°F. This standard steam temperature can be used for a low pressure off-the-shelf condensing turbine used for maximum electrical power production as a bottoming turbine. The 990°F is also matched to the cooling steam requirements for the outer case and forward face of the rotor. Cooling steam is aspirated through the cavity between the inner case and cast steel outer case, which is made of WC-9 (2-1/4% chrome steel). The differential pressure to cause the flow of cooling steam is generated by turbine blade design configurations and a series of vaned and conical diffusers in the turbine exhaust path. During full power tests this flow was adequate to maintain the outer case temperatures at 1000°F and the forward rotor disc face cooling steam in the 1070°F range.

Figure 13. Back pressure steam turbine (BPST) cross-section showing overhung rotor and 360° symmetry of hot end design (turbine rotor 11.5 inch pitch diameter).

Only three parts, the inner casing, its integral nozzle ring and its floating sleeve attachment to the outer case, operate at 1500°F, and they are made of Inconel 939 castings. Exhaust steam is used to cool the cast low alloy steel outer case. The outer case is operates at exhaust (process steam) pressure, less than 200 psig.

Investment casting development was required to obtain an inner casing design suitable for the casting processes. Final design stresses were a maximum of 8,630 psi at the trailing edge of an internal structural support strut. Allowable stresses per the ASME Boiler and Pressure Vessel Code was 18 ksi. Because statistical data was lacking for hot isostatic pressurized castings, the design allowance was set at 10 ksi, corresponding to 55 percent of the rupture value in 100,000 hours, which includes an 85 percent casting allowance. A total of 24 internal struts structurally reinforce the inner dome to the outer dome of the inner case casting (Figures 14 and 15). Total weight of this critical component is 80 pounds (total weight of all the superalloy parts of the turbine is under 200 pounds). To minimize thermal stresses and costs, the inner casing design uses the thinnest sections consistent with stress requirements. The average thickness of the inner case is 3/8 of an inch. Figure 15 shows the finish machined assembly with the twelve cooling passages machined through struts connecting the inner and outer dome. The 14 convergent divergent supersonic nozzles (Mach 2) machined into the nozzle ring is shown being fitted to the cast casing in Figure 15. The ring is mechanically attached and seal brazed to the inner casing.

1.9 DEVELOPMENT TEST RESULTS

Viability of this technology was confirmed with the successful development tests of the system achieving all test goals. Six major components or systems of the HPSS were tested as an integrated system for 105 hours of development tests that included 35 starts and shutdown transients from full power. The components tested and results are summarized:

Steam Generator:	- Efficiency of 94% LHV- Good stability and control.- Acceptable temperature distribution obtained from circuit to circuit (Figure 16)- Excellent characteristics during full load trips from 1500°F simulating a loss of all electrical power (Figure 17).
Steam Turbine:	- Expansion efficiency of 69%- 2.6 billion fatigue cycles imposed on each blade at first bending undamped natural frequency with maximum forcing function, demonstrating the effectiveness of damping system design- Excellent rotor dynamics
Control Valve:	- Excellent linearity and steam pressure and flow control between turbine and by-pass flow- Good control of steam turbine speed at all conditions- Good performance in 35 turbine trips- Maximum system pressure rise in full power trip limited to 70 psi above operating level- No indication of swell induced temperature reductions during valve transients
Interconnecting Steam Lines:	- Zero bending design concepts worked without binding or indications of excessive friction.

Figure 14. Inner turbine case Inconel 939 investment casting showing 24 structural struts.

Figure 15. Finished inner case from inlet side show 12 cooling passages machined through struts.

Average Temperature = 1508°F

952166-004V

Figure 16. Individual circuit steam temperature outlet at a mixed header temperature of 1508°F.

952166-002V

Figure 17. Maximum steam system transient during total system trip (electrical, air, fuel, water, and turbine shut down simultaneously).

Automatic
Controls:

- In closed loop temperature control by superheater feedback maintained $\pm 5^{\circ}\text{F}$ and a corresponding pressure variation of about ± 10 psi with a cyclic period of about 60 seconds.
- Speed control from a thousand to 30,000 rpm was within a $\pm 0.2\%$ of set point.

Low Emission
Combustion
System:

- Less than 9 ppm NO_x and 7 ppm CO referenced to 15% O₂ (tests were at 4% O₂)

1.10 CONCLUSION

The highly successful prototype development tests warrant continuation of the program, and HPSS development is sufficiently advanced to go the next phase - the field demonstration tests in commercial or industrial cogeneration. Figure 4 shows a plan view of the HPSS in a combined cycle. The maximum economic return would be for a site that required about 70,000 pounds per hour of process steam or 55000 pph of 950°F superheated steam at pressures less than 200 psig. Electrical output would be in a range of from 8 MW to 14 MW depending upon the gas turbine and the pressure level of the process steam. Fuel utilization efficiencies would be in the range of 80 to 90% depending upon the gas turbine and supplementary firing temperatures. The site selected would obtain the equipment discussed in this report from the DOE for consideration of installing the worlds highest temperature steam system for long duration monitoring and demonstration during the first year of operation.

Successful results at 1500°F and 1500 psia also indicate HPSS technology should be considered for large power generation systems such as repowering utility and industrial plants in the 50 to 300 MW range with HPS topping turbines in the 10 to 50 MW range. For large systems steam conditions in the 2400 to 5000 psia range at temperatures of 1300 to 1400°F are feasible. Reference 5 describes HPS topping systems turbine and steam generator conceptual designs for this range of conditions.

2.0

BACKGROUND & CONFIGURATION SELECTION AND APPLICATIONS OF HPSS

2.1 BACKGROUND

Over 35 years ago U.S. industry introduced the world's highest temperature (1200°F at 5000 psig) and most efficient power plant, the Eddystone coal-burning steam plant. The highest alloy material used in the plant was 316 stainless steel. Problems during the first few years of operation caused a reduction in operating temperature to 1100°F which has generally become the highest temperature used in plants around the world. Leadership in high temperature steam has moved to Japan and Europe over the last 30 years.

Because the majority of the U.S. electricity production and much of its industrial heat depends upon steam, the DOE has launched this development to make a step change in technology to 1500°F steam. The previous 100 years of industrial and utility developments have only averaged about 7°F per year gain in main steam temperature. Gas turbine technology, advancements in high temperature superalloys, manufacturing technology advancements, and a new class of once-through steam generators (OTSG) provided the technology base to start this ambitious program.

2.2 COGENERATION CONFIGURATION SELECTION

Thermodynamic analysis show that the overall cycle efficiency of a Rankine cycle increases as temperature and pressure increase. The highest incremental performance gains can be achieved in a 1500°F steam system when using a topping turbine in a back pressure steam turbine configuration (BPST) for cogeneration.

Development costs and potential payback to industry of the program also favored the small size for the cogeneration system. On natural gas, small combustion turbines with process boilers have been the most cost competitive. In large systems combined cycles are standard. In small systems combined cycles are not as competitive and need innovative technologies to bring the performance and economic advantages of combined cycles to smaller industrial and commercial plants. Developments of the OTSG has greatly simplified the steam generation and dramatically reduces systems complexity for combined cycles. Supplementary firing in combined cycles with BPST arrangements are normally the best economic solution, because they can provide fuel utilization efficiencies of about 90 percent in cogeneration.

Figure 4 schematically shows this simple arrangement for a BPST combined cycle. In this arrangement, the maximum amount of electricity is produced per unit of steam to the process. An incremental fuel efficiency, to produce electricity of 80-90% results from high rates of supplementary firing in the gas turbine exhaust. Other combined-cycle systems can be configured by adding a condensing low pressure turbine to produce much more electricity, but the overall fuel utilization efficiency decreases as the heat of vaporization is lost in the condenser.

A high production family of gas turbines (Solar Turbines, Inc. 5 to 10 MW machines), was selected to size the steam generator system. Although some combined cycles using these gas turbines have been installed, the large majority of cogeneration systems in this size range use only a process steam generator, typically with supplementary firing, but do not incorporate a steam turbine. This decision is based upon current economic conditions. With traditional technology, the ability to "put-

the-steam-to-work" on its way to process heating, as is common practice with large systems, is only marginal due to low power recovery and consequential costs for small cogeneration systems.

2.3 OTHER APPLICATIONS

In cogeneration with biomass-produced gas, industrial waste gases and low-Btu landfill fuels, the HPSS can be configured as a 4 MWe steam plant for cogeneration or a 10 MWe condensing steam power plant without the gas turbine. In some cases fuel gas contamination clean up costs or compression costs tend to favor atmospheric pressure combustion processes.

In coal plants the HPSS technology has been studied in a separate program (co-funded by EPRI) for repowering old coal steam plants and for new advanced steam power plants. By operating at higher pressures (to 5000 psig) and lower temperatures of about 1300°F, heat rate reductions of 12 percent can be achieved using the same tubes developed by the DOE cogeneration program and many HPSS technology developments. Reference 5 details the materials, manufacturing, design and corrosion issues that have directly supported scale-up to utility systems using HPSS technology for repowering 100 to 300 MWe plants with topping turbines of 25 to 50 MWe. Turbine design concepts developed for these larger HPSS plants are also small diameter (12.5 inches) but incorporate two stages with lower expansion ratios and rotating speeds of 16,000 rpm. These turbines have much longer blades than the HPSS turbine (about 1.0 inch) and design efficiencies of about 80%. However, they use the same materials and design concepts as the HPSS. The EPRI study also discusses the application to topping the steam plant of large combined cycles and steam injected combined cycles using topping steam turbines. Table 1 summarizes the positive impact the HPSS technology can have on utility ATS combined cycles, utility repowering, new steam plants and industrial cogeneration.

Table 1. The Effect of HPSS on Cycle Performance

	Combined Cycle (LHV)		Repowering (HHV)		New Steam (HHV)		Cogeneration (LHV)	
Fuel (Normalized)	1.00	1.14	1.00	1.10	1.00	1.11	1.00	2.5
	Stand.	HPSS	Stand.	HPSS	Stand.	HPSS	Stand.	HPSS
Gas Turbine	0.45	0.45					0.31	0.31
HPSS Turbine		0.12		0.09		0.08		0.29
Base Turbines	0.12	0.12	0.35	0.35	0.38	0.38		
Efficiency	57%	61%	35%	40%	38%	42%	78%	90%
Output	0.57	0.69	0.35	0.44	0.38	0.46	0.31	0.60
Heat Rate, Btu/kwh	5,988	5,595	9,714	8,674	8,981	8,126	4,376	3,792

3.0

OBJECTIVES

This R&D program had many objectives for both DOE and Solar, as listed below:

- Increase industrial and electric utility efficiency through the development of high temperature steam technology.
- Increase steam temperature to 1500°F at 1500 psig for applications for small combined cycles for industrial cogeneration using supplementary firing in the gas turbine exhaust.
- Design, fabricate and test for 500 hours a full tube length steam generator test module at 1500°F and 1500 psig.
- Develop production cold drawn final superheater tubes to meet all of the ASME Section I Fired Boiler Code requirements and the economic goals of high performance industrial steam generators.
- Design and fabricate an industrial sized 1500°F heat recovery steam generator matched to a supplementary fired 5 MWe industrial gas turbine.
- Design and fabricate a back pressure steam turbine (BPST) to operate at 1500°F and 1500 psia and produce 4 MWe.
- Develop a high temperature valve to control the system as a combination throttle, bypass, and stop valve.
- Design a 1500°F piping system to minimize thermal stress and costs in a high temperature system.
- Incorporate a low emissions burner.
- Test the High Performance Steam System (HPSS) in a factory proof development test for 100 hours prior to placing in a site cogeneration system.
- Install HPSS at a permanent site in cogeneration and monitor it for the first 1000 hours.

4.0

PROTOTYPE SYSTEM DESIGN, ANALYSIS, FABRICATION AND DESCRIPTION

4.1 THERMODYNAMIC CYCLE ANALYSIS

4.1.1 Cycle Efficiency

The high temperature steam technology developed in this program can be utilized by several different thermodynamic cycles, but the cycle selected for design and demonstration in this program is the combined (Brayton/Rankine) cycle. More specifically, the proposed cycle for this program is a supplementary-fired combined cycle, in which the exhaust from the gas turbine is heated prior to entering the heat recovery steam generator (HRSG). This cycle is illustrated in the schematic diagram of Figure 18. In the steam portion of the cycle, a high-pressure steam turbine exhausts to a process header, typically at 150 psig for industrial applications. A variation of the cycle can incorporate a low pressure (condensing) steam turbine to expand a portion, or all, of the steam from 150 psig to a vacuum. In this configuration it will produce more electrical power, and less thermal energy (steam) for the process.

One of the primary objectives of the High Performance Steam Program was to increase the output from the topping portion of the steam cycle, or the amount of electricity that can be generated from the steam before it is sent to a thermal process. The key to accomplishing this is to generate the steam at a very high pressure and temperature such that there is a large amount of energy available in the superheated steam. This energy can then be recovered as the steam is expanded through the topping (back pressure) steam turbine. The thermal energy above 1100°F is made available for conversion to work by the HPSS developments.

Figure 2 (Page 2) illustrates this concept in the form of a temperature - entropy diagram. Two curves are shown, the solid line representing steam produced at 1500 psia and 1500°F, and the broken line representing steam produced at 900 psia and 900°F. The lower curve is considered to be the maximum conditions currently used by industry (a more typical industrial steam plant might operate at 600 psig and 750°F). The amount of turbine work done per pound of steam is proportional to the vertical drop from the initial steam conditions to the 150 psig line. For the 56,000 pound per hour steam flow design point, the power available from the 1500 psi, 1500°F point is four megawatts, compared to two megawatts from the 900 psi, 900°F point. Thus the potential power from the topping portion of the steam cycle can be doubled compared today's best industrial practice.

Figure 19 illustrates the turbine power available from currently available back pressure steam turbines compared to that available from the high temperature turbine developed during this program. Electrical power output is plotted versus steam flowrate, since it is often process steam flowrate which determines the most economical size of an industrial cogeneration system. This figure includes the effects of the thermodynamic cycle as well as the machine efficiency. There are two bands for the standard industrial practice machines, the lower one for simple single stage machines, and the higher one for more complex, and expensive, multi-stage turbines. The high temperature turbine is a simple single-stage machine, but it has an expansion efficiency approaching that of the complex multi-stage turbines.

Figure 18. Advanced Cogeneration System, Combined Cycle with Back-Pressure Steam Turbine

Notes: 1. Includes effect of steam temperature and pressure
 2. 150 psig back pressure
 3. SIP = Standard industrial practice

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Figure 19. Electrical Output From Steam Turbo-Generators

Efficiency of the overall cogeneration thermodynamic cycle is also enhanced by the temperature to which the supplemental burner is fired. In this system, the gas turbine exhaust (typically at 950°F) is supplementary fired to as high as 2500°F before it enters the steam generator. As firing temperature increases from the turbine exhaust temperature, cogeneration efficiency increases because all of the heat from the incremental fuel is either captured as additional steam production for the thermal process, or as additional superheat, which is converted to electricity in the steam

turbine/generator set. This is illustrated by Table 2. A comparison is made between a simple cycle gas turbine and process steam boiler and the advanced combined cycle with supplemental firing.

Table 2. Efficiency of HPSS vs. Simple Cycle Cogen

	Simple Cycle (1)	HPSS	Incremental Delta
Firing Temp (°F)	945	2330	
kW Net	4048	7630	3582
Process Steam (pph)	25,800	70,780	44,980
Fuel Heat (MMB/hr)	50.0	117.4	67.4
Fuel Utilization Efficiency (%)	81.6	85.4	99.4
Notes: (1) Based on Solar Centaur 50 and 150 psig steam (2) Based on drain temperature of 180°F			

Fuel utilization efficiency is the sum of the electrical output and the useful thermal output, divided by the fuel input. The incremental fuel burned in the HPSS is used at an efficiency of 99%, between electrical generation and steam production.

For the case where a condensing steam turbine is added to the system, the advantage of adding the extra superheat in the high pressure steam carries over to the low pressure expansion, due to the higher enthalpy and the reduced moisture in the last stage(s) of the steam turbine. In terms of thermodynamic cycle efficiency for condensing turbines the efficiency varies with initial pressure and temperature as shown in Figure 20. These curves are based on a condenser pressure of 1 psia and condensate at 90°F, and include no losses in the steam turbine. Steam cycle efficiency for the high performance steam conditions is 46% compared to 36 % for standard industrial practice steam conditions.

When the system is a full combined cycle (no steam extracted for thermal use) the overall efficiency (electric output divided by the fuel input) is as shown in Figure 21. For comparison, the efficiency of a simple cycle gas turbine and for an unfired combined cycle are shown as well. The peak numbers shown in this figure are less than those in the previous figure due to the inclusion of losses in turbine bearings, reduction gears, generators, and parasitic power. Although the peak electrical efficiency of the high performance combined cycle is slightly less than that of the unfired combined cycle, a big advantage of the supplementary-fired high temperature combined cycle is that it offers a high efficiency over a large power turndown range. Efficiency remains nearly constant between 50% and 100% power. For simple cycle and unfired combined cycles, the part load efficiency falls off substantially.

4.1.2 Flow Schematics and State Point Diagrams

As stated above, the primary system configuration evaluated during the design phase was the combined cycle with a back pressure steam turbine (BPST) only. The simplified flow diagram was shown in Figure 18. Figure 22 is an energy balance diagram of this system, as designed, with state point locations indicated by the circled numbers, and the energy quantities in millions of Btu's per hour following the designation "E=". Energy losses are indicated by the zig-zag arrow exiting the various components.

Figure 20. Ideal Rankine Cycle Efficiency Comparison of HPSS and Traditional Industrial Steam Conditions

Figure 21. High-Performance Steam Combined cycle With Condensing Turbine Efficiency

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Firing temperature varies from a minimum of about 1000°F at low power to 2500°F at full power, corresponding to heat release rates of 20 million to 105 million Btu per hour (lower heating value).

In the test rig, the steam generator and the steam turbine function as they would in the combined cycle. Steam is generated in a floating pressure and temperature mode, in which steam pressure varies nearly directly with the flowrate, due to the fixed area of the steam turbine nozzle. Flow is choked across the nozzle at all but the very lowest flowrates. Downstream of the turbine, a back pressure control valve is installed to allow regulation of the turbine exhaust pressure. This valve is manually controlled by the test operator. The steam turbine exhausts to atmosphere via a steam silencer. In an industrial situation, the steam turbine would exhaust to a process steam header, typically at a constant pressure of 150 psig.

Table 3. HPSS Combined Cycle State Point Values

State Point No.	Fluid	Pressure (psia)	Temperature (°F)	Enthalpy (Btu/lb)	Mass Flow Rate (lb/hr)	Energy Flow (MMBtu/lb)
1	Steam	1500	1500	1784	56,943	101.59
2	Steam	166	936	1496	56,943	85.19
3	Steam	166	366	1196	73,093	87.42
4	Water	14.7	180	148	58,474	8.65
5	Water	14.7	156	124	73,093	9.06
6	Water	50	157	125	73,093	9.14
7	Water	1800	170	138	73,093	10.09
8	Water	1750	170	138	56,943	7.86
9	Water	200	170	138	16,150	2.23
10	Water	15	60	28	14,619	0.41
11	Air	14.7	59	124	139,774	17.33
12	Hot Gas	15.2	949	356.3	142,080	50.62
13	Hot Gas	15.1	2500	834.7	142,517	121.46
14	Hot Gas	15.1	2493	832.5	145,517	121.14
15	Fuel Gas	14.7	293	188.4	145,517	27.41
21	Nat. Gas	180	60	20610	2,306	47.53
22	Nat. Gas	3	60	20610	3,437	70.84

Both the industrial system, be it combined cycle or stand alone, and the test rig incorporate a turbine bypass system to allow steam to bypass the turbine during start-up and in the event of a sudden load shed, or turbine overspeed. This system consists of a turbine bypass valve and desuperheating system. The valve is a three-way valve which either ports the incoming steam to the turbine, or to the bypass, or to a combination of both. The valve cannot close off the steam pipe from the steam generator, so it can never cause excessive pressure at the steam generator header. This makes the system inherently safe, and functionally eliminates the need for a boiler safety relief valve. However, in the actual industrial site, a safety valve may be needed to meet regulations and will be added as required.

The turbine bypass system incorporates a desuperheating section in the bypass outlet from the three-way steam valve. Water flow rate is regulated by a valve which is controlled automatically to maintain a set temperature in the bypass piping. Typically this temperature is set equal to the turbine exhaust temperature, so the system does not experience a temperature upset when the turbine power is suddenly reduced, or the turbine is tripped off line.

For the laboratory testing the water/steam side was run open loop to avoid the large capital cost of a steam condensing system. City water flowed through a deionization system and mixed bed polishing tank to provide specified water to the system. Deaeration is not required and was not used during tests. Water pressure is raised to about 2500 psig by the feedwater pump, and sent through a flow measurement orifice plate and a flow control valve prior to entering the steam generator.

For the test rig, both the incoming air and water are at approximately ambient temperature. In the industrial combined cycle system, the incoming air is replaced by turbine exhaust products at approximately 950°F, and the feedwater will be supplied from process drains, likely to be at 160°F or thereabouts. The stack temperature for either system will be approximately 100°F warmer than the incoming water. For the test rig the stack temperature is about 190°F.

4.2 PROTOTYPE HPSS TEST ARRANGEMENT

Because of the major step in steam temperature taken in the development of each component, the system component arrangement had to be designed with similar attention to materials, cost and innovation to achieve the challenging goals. Several traditional arrangements of steam generator and turbines were considered. They generally required long steam line runs with considerable thermal expansion and pipe bending stress allowances. In the previous high temperature system, the Eddystone 1200°F plant, four separate steam leads of about 150 feet in length were made of forged and gun drilled stainless steel lines. These were expensive and made it difficult to accommodate thermal expansion. The development of Inconel 617 hot extruded seamless piping of 5.25 inch O.D. and a wall thickness of 1.10 inch allowed the HPSS system to obtain a cost effective solution to system "piping". The arrangement shown in Figures 4 and 24 was selected to solve many issues associated with high temperature steam line systems.

The arrangement shown where the steam turbine, control valve and steam generator header are all in the same plane, eliminates vertical flexing loads. With the valve arranged on the vertical and horizontal centerline on the end of an expansion section of piping, all bending loads are eliminated. The centerline of the turbine trunnion mount is on the horizontal centerline of the header within the OTSG (a slight offset for the turbine case temperature being at 1000°F rather than the 1500°F for the rest of the system is provided in the design details). A centerline key on the turbine and four 2 inch slide keys on the valve keep the turbine centerline fixed in space, and the trunnion centerline fixes the axial position of the turbine and reacts axial frictional loads. Slides on the valve allow it to expand away from the turbine. Valve weight is carried by spring hangers on two superalloy rods (Figure 25). Slide pads under the OTSG steam generator header allows the header to move into the OTSG to accommodate expansion away from the turbine centerline and to a limited extent in the turbine axial direction to allow for warm up and cooldown transients between the various components and lines.

Figure 24 schematically shows the low pressure piping used for testing. The turbine exhaust was arranged to exhaust down through low pressure piping at 1000°F. This piping is connected to a vent silencer on the right that provided about 40 psig back pressure on the turbine. A restrictor and valve were arranged to increase the back pressure on the turbine to as high as 200 psig. When in turbine by-pass mode, the steam discharged down through an integral spray desuperheater in the left hand leg of the low pressure piping. Several bellows allowed for pipe expansion in the low pressure and temperature piping.

Figure 24. Elevation view of HPSS Prototype Test System Arrangement

Figure 25. Control Valve Weight Carried by Super Alloy Rods to Spring Hanger

It was planned that the prototype HPSS be configured as it would be in the demonstration site with respect to the high temperature component and line arrangements. Low pressure piping would be different since the demonstration site would use the low pressure steam for a process or for a low pressure condensing turbine. In the prototype test for 100 hours it was less expensive to exhaust the steam to atmosphere rather than condensing.

4.3 STEAM GENERATOR

4.3.1 Steam Generation Concept - OTSG Selected

The steam generator is the most critical component in the development of high performance steam plants since it is the necessary precursor in providing the source of steam. In recognition of this, the development of the complete power plant was constrained until a proof-of-concept 500 hour test module was successfully tested. Figure 5 shows the full length tube, four-circuit test module. The 500 hour test module was based upon a commercial OTSG, a new concept in steam generation developed specifically for the next generation of combined cycles. At traditional steam temperatures (less than 1000°F) about twenty of these OTSG's are in industrial power generation and cogeneration (end of 1995). Three units have also been fabricated for steam injected peaking gas turbines in the 80 MW range, but are not yet operating. Unique and patented features of these commercial OTSG's make them ideal for future HPSS applications and the OTSG design concept was used as the basis of the steam generator developed.

A OTSG is the simplest configuration possible to generate steam and is inherently ideally suitable for the advancement of steam conditions to obtain higher performance. For smaller power plants (less than 20 MW) pressure levels in the range of 1200 to 1800 psig are optimum for high performance at high temperatures. The next size range of high performance steam systems would need to use higher steam pressures with somewhat lower temperatures between 1300 and 1400°F for optimum performance and costs. As pressures advance above 2400 psig to supercritical ranges, the OTSG is essentially a requirement with drum configurations ineffective and expensive.

4.3.2 Steam Generation Flow Path and Materials

Simplicity is the key feature of an OTSG. Figures 26 and 27 schematically illustrate typical flow paths and components required for steam generation in combined cycles. They show dual pressure drum HRSG unit compared with a dual pressure OTSG. Dual pressure is typical with unfired combined cycles and generally not required at high supplementary firing temperatures.

In an OTSG, no drums are used and all of the steam is generated in a continuous flow path from inlet water to outlet superheated steam. Feedwater enters at the combustion gas outlet exhaust gas stack. It is heated as it flows in a cross counter flow arrangement of fin tubes. In Figure 10 (Page 8) the water enters at the top right hand side and flows towards the left as the combustion gases flow to the right. A continuous heating occurs as circulation is forced by the feedpump pressure differential pressure from inlet to outlet of the circuit. Water is evaporated and converted to superheated steam in the small diameter continuously connected tubes. By welding many circuits in parallel, the OTSG's output can match different exhaust gas flows. In order for a OTSG to function without scale formation, the feedwater has been limited by condensate polishing to total dissolved solids averaging less than 50 ppb for steady state operation. Startup and transients to solid levels of as high as 500 ppb do not appear to cause significant problems with deposits after many years of operational experience with OTSG combined cycles (Reference 3).

Figure 26. Dual Pressure Drum-Type HRSG Flow Diagram

Figure 27. Dual Pressure OTSG Flow Diagram

A OTSG provides an order of magnitude of simplification with one tenth the number of valves and components, including no drums. To achieve this high degree of simplification corrosion resistant high nickel stainless steel, alloy 800 is used in the current production units. This advanced alloy has proven to be cost competitive in OTSG's when competing with traditional carbon steel tubes used in drum HRSG's during several recent competitions. Although the tube material is several times more expensive the dramatic simplifications and elimination of components such as drums allows a high alloy tube to be competitively used by OTSG's. The high nickel stainless tube provides features such as the ability to operate the gas turbine with the steam generator dry and to use passive water treatment, a patented "non-water-treatment system" in which no chemicals are added to feedwater or drum water. In passive water treatment oxygen and pH are not controlled and no blowdown is incorporated to remove solids from the steam generator. This replaces the traditional approach to steam generation of removing solids in the HRSG by blowdown. In the OTSG approach, solids are removed prior to introduction into the steam generator with a full flow water DI polishing system (References 3 and 4). In some cogeneration installations, control of oxygen and pH may be required to protect downstream carbon steel piping.

Alloy 800 tubes are allowed to be used to 1500°F average tube temperature per ASME Section I Fired Boiler Code. Thus by simply increasing the tube wall thickness to match the allowable stress levels it is practical to obtain 1250 to 1300°F steam temperature with the standard materials used in production OTSG's. Figure 28 is a photo of a typical production OTSG fabricated with Alloy 800 tubes using carbon steel fins. This is a OTSG with horizontal tubes of Alloy 800 with a vertical gas path for use with a 10 MW gas turbine supplementary fired to 1200°F for industrial cogeneration. At fin temperatures over 900°F, the carbon steel fins are replaced with ferritic stainless 409 fins and over 1100°F fin temperatures austenitic stainless steel is used. A unique method of metallurgically bonding fins to different tube materials is resistant to temperatures over 1800°F.

Figure 28. Production OTSG for Industrial Cogeneration

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To obtain 1500°F steam at high pressure alloy 800 would not be viable in the final superheater. Inconel 617 was chosen from the available ASME Section I Boiler Code approved alloys to operate to average tube wall temperatures of 1650°F. Other materials in the ASME Code were considered, and are acceptable, but Inconel 617's capability was clearly proven during the Phase I and Phase II testing. Data on use in coal ash environments and the designer's experience indicated it was an excellent choice for the first 1500°F steam generator using ASME Code allowable design stress levels.

4.3.3 Control Concept

The OTSG uses one control the feed water rate to control output steam temperature for a given steady state operating point. Other controls such as fuel and mass flow of gases are possible but are not necessary in general. The development of digital computer controls have been important in the practical development of OTSG units. For good control, a predictive or feed forward control has proven to be important. The thermal energy entering the OTSG on the gas side is calculated from key measurements. The correct steady state amount of feedwater is calculated for a given outlet temperature and flow needed for optimum operation. Proportional, integral and differential gains can be modified to account for transient characteristics of the system to minimize pressure and temperature transients in the outlet steam. The high degree of flexibility obtained with digital computer controls make the OTSG control relatively simple and practical. Figure 29 is a printout of the HPSS control schematic screen used as part of the development test control system.

4.3.4 Design Improvements Based Upon 500 Hour Module Test

During the construction and testing of the 500 hour Test Module (Figure 5) a number of important design improvements were evolved. The initial design had a total of eight radiation screen tubes in the initial evaporator section. During testing it was determined that this was too conservative. Not only was it not necessary to have so many radiation screen tubes to limit the final superheater tube temperature, it also caused the final dry out zone to move into the screen and cause excessive screen tube temperatures under some operating conditions. As a consequence, four of the radiation screen tubes were removed during preliminary testing. In the HPSS 26 circuit system design, this was reduced to a total of four rows which proved to be satisfactory.

Figure 29. HPSS Control Screen Print Out of Steam Turbine Control Screen

Other major improvements developed from data and experience on the 500 hour test module include:

- Elimination of intermediate headers in the superheater.
- Improve tube bundle support system to reduce cost and improve sealing for production configurations.
- Change of final superheat tubing to a production configuration (see Section 4.3.6).

4.3.5 Tube Selection and Development

To reduce cost and program development time, the 500 hour test module was designed to the stresses allowed by the ASME Code but not actually coded. Since the plan is to place the development HPSS system into a commercial or industrial site, it was necessary to design and fabricate the OTSG to all of the ASME Code requirements that normally would anticipate 20 to 30 years of revenue operation without problems. Some of the candidate materials are listed in Table 4 together with the ASME allowable design stresses. Reference 5 has additional data and results of 4000 hour oxidation and steam corrosion rates of superalloys in 1500°F and 1500 psig steam environments. Table 4 shows Code required allowable stresses and corresponding wall thicknesses for 1.0 inch outside diameter tubing. From it is seen that Alloy 800 is optimum up to about 1100°F, Alloy 800H to 1250°F, and Inconel 617 above 1250 or 1300°F. The final selection is based upon internal tube pressure loss, steam generator fabrication costs and tube manufacturing process capabilities.

4.3.6 Tubing Development

In the 500 hour proof-of-concept test module the final superheater tubes were machined from bar stock 15 feet long and gun drilled to form tubes 1.25 inch OD with 0.38 inch thick wall. Although this manufacturing technique is too expensive for production tubes, it was cost and time effective for a proof-of-concept test, since only five final superheater tubes were required for the 500 hour test module. Successful results of the 500 hour test initiated the HPSS development. A critical initial phase of the HPSS development was the establishment of a viable production process for making heavy walled Inconel Alloy 617 tubing with an OD/wall ratio of about 3.0 to the ASME Code Case 1956-2 requirements. The Case allows use of solution annealed Ni-Cr-Co-Mo alloy (UNS N06617) in seamless tube in the construction of fired boilers. Previous attempts to make heavy wall tubing in this alloy were only marginally successful.

Inco Alloy International of Huntington West Virginia was selected, as it was the inventor of the alloy and owned the patents on the alloy. A special heat was melted where chemistry was adjusted to enhance cold workability. From this heat, 0.5 inch wall 5.25 OD shells were extruded which was the starting material for all five wall sizes fabricated (Table 5). A series of tube reducing operations followed, along with appropriate cleaning, annealing and pickling steps. Previous work also indicated that cold working could be improved with a fine grain anneal at 2050°F and limiting reductions to 50%. At strategic points in the process, room temperature tensile tests were conducted as a check on the effectiveness of the process parameters. Ultrasonic testing was also employed periodically to monitor the soundness and integrity of the ID. The standard used was a 5% longitudinal ID notch. Table 6 was the process outline used in producing tube items 2 through 6. Longitudinal cracks developed with item 1 and the material was scrapped. Items 3, 4, 5 and 6 were final sized with plug mandrel drawing in conjunction with a final sink pass.

Table 4. Required Tube Wall Thickness

Metal Temperature (°F)	Alloy 800H		Alloy 800		Alloy 617	
	Maximum Allowable Stress (ksi) ²	Estimated Wall Thickness (in.) ¹	Maximum Allowable Stress (ksi) ²	Estimated Wall Thickness (in.) ¹	Maximum Allowable Stress (ksi) ²	Estimated Wall Thickness (in.) ¹
500	12.9	0.059	16.7	0.048	17.3	0.047
600	12.2	0.063	16.3	0.049	16.7	0.048
700	11.7	0.065	15.9	0.050	16.2	0.049
800	11.1	0.068	15.5	0.051	15.9	0.050
900	10.7	0.070	15.1	0.052	15.7	0.051
1000	10.3	0.073	14.7	0.054	15.5	0.051
1100	10.0	0.075	13.0	0.060	15.4	0.051
1150	9.8	0.076	9.8	0.076	15.4	0.051
1200	8.4	0.087	6.6	0.107	15.3	0.052
1250	6.9	0.103	4.7	0.143	13.0	0.060
1300	5.4	0.127	2.0	0.311(3)	10.0	0.075
1350	4.5	0.148	1.6	0.410(3)	7.7	0.094
1400	3.6	0.177	1.1	(4)	6.0	0.116
1450	3.0	0.205	1.0	(4)	4.6	0.145
1500	2.5	0.236	0.8	(4)	3.6	0.177
(1) $t = PD/(25 + P) + 0.005D$		D = 1.0 inch P = 1500 psia		(2) ASME BPVC, Section I, 1983 ed. (Pipe & Tube)		
(3) Thick Wall $t > 1/4D$				(4) Wall Thickness $> D/2$		
				$t = \left(\frac{\sqrt{Z}-1}{\sqrt{Z}} \right) R_o$ $z = (SE + P)/(SE - P)$		

Table 5. Production tube Process Development Size Items

Item	Proposed Size	Finish Size	Item	Proposed Size	Finish Size
1	.240" MW x 1.000" Dia.	--	4	.280" MW x 1.000" Dia.	.300" MW x 1.000" Dia.
2	.150" MW x 1.000" Dia.	.150" MW x 1.000" Dia.	5	.150" MW x 1.000" Dia.	.180" MW x 1.000" Dia.
3	.240" MW x 1.000" Dia.	.270" MW x 1.000" Dia.	6	.180" MW x 1.000" Dia.	.185" MW x 1.000" Dia.

Table 6. Process Outline (T00409-2)

1	11.5" Dia. Rods Hot Rolled from ESR Ingots	12	Degrease
2	Machine 10.7" Dia. Extrusion Billets	13	Anneal and Pickle
3	Extrude .500" Wall x 5.250" Dia. Shell	14	Tube Reduce .150" x 1.050" Dia.
4	Anneal and Pickle	15	Degrease
5	Tube Reduce .385" Wall x 3.500" Dia.	16	Anneal and Pickle
6	Degrease	17	Sink Pass to .150" x 1.050" Dia.
7	Anneal and Pickle	18	Degrease
8	Tube Reduce .275" Wall x 2.500" Dia.	19	Final Anneal and Pickle
9	Degrease	20	Straightening
10	Anneal and Pickle	21	UT and Hydrostatic Test
11	Tube Reduce .175" Wall x 1.900" Dia.	22	Tests Cut (RTT, Hardness, Grain Size, etc.)

A lab annealing study was conducted on samples of the final tubes in the as-cold-drawn condition to determine the optimum final anneal treatment on the tubes (see Table 7). It was known from previous work on alloy 617 that the material should be annealed in a manner that would result in an ASTM #4-5 grain size. Products of this alloy with grain structures in this size range have the best combination of tensile and stress rupture properties.

Based on the results of the lab study, the final anneal for the tubes was conducted at 2150°F for 15-20 minutes, except for items 3 and 4, which were annealed for only 5 minutes due to a calibration error in the anneal furnace. The effect of the difference in anneal treatments can be seen in Table 8, which summarizes test results after the final anneal. It should be noted that the stress rupture life for items 3 and 4 would be expected to improve after a re-anneal treatment.

Yield for this order is defined as the percent weight of finished tubes obtained from the starting tube shells, .500" wall x 5.250" OD. Yield losses include various saw cuts, OD grinding, ID reaming, and the in-process rejection for surface quality, internal soundness, and tolerances (see Table 9).

92% of rejections at UT were for longitudinal ID cracks. These ID defects would have been much higher if the tubes had not been reamed at 3.500" OD. Approximately .040" was removed from the ID. Generally speaking, the overall quality level of these tubes was very good.

It appears that a process where most reductions are accomplished on the Wean tube reducer, where sizes 1.050" OD are possible, would be the process of choice. Tubes with 1.050" OD are easily sunk to 1.000" OD with good results and no outworkers have to be used.

Actual methods of production (MOP) were written for each item and submitted to IAI Accounting and Pricing for analysis. The proposed selling price for the various sizes listed in Table 10 are based on metal prices in effect on November 9, 1990.

Table 7. Effect of Annealing on Grain Size

Time	Temperature		
	2125°F	2150°F	2175°F
5 Min, * WQ	Avg. #6 ALA #3	Avg. #5-1/2 ALA #3-1/2	Avg. #4-1/2 ALA #2
15 Min, WQ	Avg. #5-1/2 ALA #3	Avg. #4-1/2 ALA #3	Avg. #4 ALA #2
* In general, 5 minute anneals resulted in a less uniform grain size.			

Table 8. Test Results After Final Anneal

Item	Nominal Wall Thickness	Grain Size	Room Temperature Tensile			Stress Rupture*		
			Y.S. (ksi)	T.S. (ksi)	EI (%)	Life (hr)	EI (%)	RA (%)
2	.150"	#5	56	119	58	..**	-	-
3	.240"	#6	58	122	57	22	45	40
4	.280"	#5-1/2	57	120	57	34	36	52
5	.150"	#5	59	125	58	..**	-	-
6	.180"	#5	57	122	62	..**	-	-
* - 1600°F/13 ksi			..** - Unable to test because of small dimensions of the tube.					

Table 9. Yield Results

Item	Start Weight (lb)	Finish Weight (lb)	Start Pieces	Finish Pieces	Pieces Rejected (UT)	Reject Rate (%)	% Yield
1	1300	--	4	8	8	100	0
2	1255	994	3	40	14	35	79.2
3	1755	1176	5	18	1	5.5	67.0
4	1770	1324	5	19	0	0	74.8
5	972	732	3	15	1	6.6	75.3
6	1284	919	4	17	1	5.8	71.6

Table 10. Tube Selling Price for Inconel 617

Item	Finished Size	Proposed Selling Price (\$/Ft)	Item	Finished Size	Proposed Selling Price (\$/Ft)
1	--	--	4	.300" MW x 1.000" Dia.	86.60
2	.150" MW x 1.000" Dia.	58.09	5	.180" MW x 1.000" Dia.	81.87
3	.270" MW x 1.000" Dia.	93.01	6	.180" MW x 1.000" Dia.	77.01

The objective of this project was met; i.e. develop a repeatable process for the production of heavy walled Inconel alloy 617 tubing. Tubes with heavier than expected walls were produced having OD/wall ratios of 3.33 to 6.7. Also, all mechanical testing requirements as spelled out in the initial proposal were met. The surface, size, and appearance quality met commercial standards and tolerances. It appears that any future orders for heavy walled Inconel alloy 617 tubing, when produced within the guidelines described herein, could be successfully completed and shipped within current production lead times.

Using the costs in Table 10, the total cost of tubing for the superalloy Inconel 617 superheater is about \$125,000 (dated November 9, 1990). More recent quotes from Inco International (November 1994) place the cost of the final superheater at about \$95,000. This is the heart of a HPSS that produces over 70,000 pph of process steam, 4 MW of topping power and 6 MW of power for a condensing steam plant. At competitive installed costs of about \$1000 per kW, the plant would install at about 5 to 10 million dollars. Since the final superheater tube costs are a small fraction of the approximate installed cost of a cogeneration plant, the incremental cost of the OTSG tubing to go to high temperatures is not a major constraint on the concept acceptance. Additional costs associated with welding, superalloy header and piping are important also but would not limit its commercialization. The header and steam line connections to the tubing are also expensive at about \$1300 per foot. However, the HPSS design has optimized the header and piping costs by minimizing the total length of 5.25 inch OD by 1.12 inch wall header and piping to a total length of less than 12 feet. With this length, the total pipe material cost of high temperature lines is less than \$16,000 for the HPSS.

4.3.7 Heat Transfer Matrix Description

The following design criteria was important in finalization of the heat exchanger matrix configuration:

- Limit average tube wall temperature to less than 1585°F at an outlet temperature of 1535°F and a pressure of 1550 psia to allow for system losses to the steam turbine and control variations in a 1500°F and 1500 psig system.
- Configure tubes to minimize the cost of superalloys.
- Configure matrix spacing to allow high temperature support structure to be installed.
- Use hot wall, rather than water walls to minimize emissions and reduce cost.
- Maximize cycle and steam generator efficiency consistent with low emissions.
- Maintain final superheater fin tips temperatures less than 2000°F and attachments less than 1700°F.
- Minimize gas side and water side pressure losses.
- Design for dry operation at an exhaust gas temperature from gas turbine of 1000°F.
- Provide for field maintainability and access to the final superheater.
- Provide low leakage seals around the entire heat transfer matrix.

A number of candidate matrix configurations were considered and the final arrangement is shown in Figure 30. All 38 rows of tubing in each parallel circuit (Figure 10) except for the four screen tubes are arranged in an in-line mechanical arrangement. This arrangement allows individual circuits (Figure 10) to be mechanically supported and sealed at the u-bends with a box beam upper support when using vertical tubes that are supported from the top. A vertical tube matrix with a horizontal gas flow allows the tubes to be self-supporting. Fin heights are 0.5 inch except in the four screen tubes that are unfinned and the final superheater that has a 0.1875 inch high fins to control final superheater tube wall and fin maximum temperatures. Section 4.3.7 discusses materials, wall thickness and fin configuration and materials throughout the matrix.

Figure 31 shows the gas side and water side calculated temperatures within the matrix. Table 11 lists the final superheater row configurations and heat transfer data. It starts at a steam temperature of 837.3°F and at a gas temperature of 1600°F within the matrix and documents the next five rows of tubing that increase the steam temperature to 1505°F at an inlet gas temperature of 2122°F. These are the five rows fabricated from Inconel 617 and 310 stainless steel fins. Table 12 is a summary of the total heat exchanger dimensions showing water side and gas side pressure drops when producing 57,170 pph of steam at 1505°F and 1496 psia steam and a gas flow of 142,000 pph at 2500°F.

4.3.8 Mechanical Design of OTSG

Important criteria used to design the highest temperature steam generator include:

- Conformance with the ASME Section I Boiler Code for a 1500°F and 1500 psig HPSS.
- Dry operation (no water in tubes) at gas turbine exhaust temperatures of 1000°F.
- Hot start capability at 1000°F and normal dry start as gas turbine starts and or as fresh air combustion system lights off. Unit is normally dried out by shutting feedwater off and running the gas turbine at no load for a few minutes to boilout the water to prevent freezing.

Figure 30. OTSG Heat Transfer Matrix Geometry

Figure 31. OTSG Temperatures, Gas Side and Water Side

Table 11. OTSG Inconel 617 Final Superheater Heat Transfer Design Data

WATER/STEAM FLOW	5.717E+04	LBm/HR	GAS FLOWRATE	1.420E+05	LBm/HR
HEAT TRANSFER AREA	2.910E+03	SQ.FT.	PRESS. DROP LENGTH	4.800E+01	FT.
TUBE O.D.	1.000E+00	INCH	TUBE WALL THICKNESS	1.200E-01	INCH
NUMBER OF CIRCUITS	2.600E+01		FREE FLOW AREA	3.551E+01	SQ.FT.
WATER/STEAM SIDE DP	3.245E+01	PSI	GAS SIDE DEL P	-6.529E-01	IN H2O
TOTAL H.T. LENGTH	1.092E+03	FT.	H.T. LENGTH/TUBE	1.400E+01	FT.
NUMBER OF FINS	6.000E+00	FINS/INCH	TRANS. TUBE PITCH	4.700E+00	INCH
FIN THICKNESS	3.000E-02	INCH	LONG. TUBE PITCH	1.200E+00	INCH
FIN HEIGHT	5.000E-01	INCH	BUNDLE HEIGHT	6.000E-01	FT.
FREE FIN SPACING	1.367E-01	INCH	BUNDLE WIDTH	5.287E+00	FT.
FIN EFFICIENCY	9.484E+01	PCT.	BUNDLE MASS	2.993E+03	LBm
AVE. GAS VISCOSITY	1.130E-01	LBm/FT/HR	CROSS SECTION AREA	7.403E+01	SQ.FT.
AVE. GAS CONDUCTIV.	4.603E-02	BTU/FT/HR/F	BUNDLE VOLUME	4.442E+01	CU.FT.

GROUP	STM ENTH BTU/LBm	STM TEMP Deg F	STM PRESS PSIA	GAS TEMP Deg F	AVE TUBE DEL Deg F	TUBE TEMP Deg F
1	1.371E+03	8.373E+02	1.861E+03	1.600E+03		0.000E+00
2	1.468E+03	9.832E+02	1.852E+03	1.724E+03	1.056E+03	5.708E+01
3	1.564E+03	1.139E+03	1.841E+03	1.845E+03	1.207E+03	5.131E+01
4	1.655E+03	1.292E+03	1.829E+03	1.960E+03	1.356E+03	4.588E+01

GROUP	GAS REY	GAS REY	EFF GAS HTC Btu/Hr/SqFt/F	TUBE HTC Btu/Hr/SqFt/F	TOTAL HTC	HEAT FLUX Btu/Hr SqFt
1	0.000E+00	0.000E+00	0.000E+00	0.000E+00	7.306E+00	0.000E+00
2	2.073E+03	6.751E+03	8.707E+00	1.000E+04	7.510E+00	7.673E+04
3	1.994E+03	6.071E+03	8.746E+00	1.000E+04	7.578E+00	7.592E+04
4	1.925E+03	5.467E+03	8.799E+00	1.000E+04	7.654E+00	7.302E+04

WATER/STEAM FLOW	5.717E+04	LBm/HR	GAS FLOWRATE	1.420E+05	LBm/HR
HEAT TRANSFER AREA	9.701E+02	SQ.FT.	PRESS. DROP LENGTH	1.600E+01	FT.
TUBE O.D.	1.000E+00	INCH	TUBE WALL THICKNESS	2.000E-01	INCH
NUMBER OF CIRCUITS	2.600E+01		FREE FLOW AREA	3.551E+01	SQ.FT.
WATER/STEAM SIDE DP	4.685E+01	PSI	GAS SIDE DEL P	-2.510E-01	IN H2O
TOTAL H.T. LENGTH	3.640E+02	FT.	H.T. LENGTH/TUBE	1.400E+01	FT.
NUMBER OF FINS	6.000E+00	FINS/INCH	TRANS. TUBE PITCH	4.700E+00	INCH
FIN THICKNESS	3.000E-02	INCH	LONG. TUBE PITCH	1.200E+00	INCH
FIN HEIGHT	5.000E-01	INCH	BUNDLE HEIGHT	2.000E-01	FT.
FREE FIN SPACING	1.367E-01	INCH	BUNDLE WIDTH	5.287E+00	FT.
FIN EFFICIENCY	5.447E+01	PCT.	BUNDLE MASS	1.243E+03	LBm
AVE. GAS VISCOSITY	1.204E-01	LBm/FT/HR	CROSS SECTION AREA	7.403E+01	SQ.FT.
AVE. GAS CONDUCTIV.	4.980E-02	BTU/FT/HR/F	BUNDLE VOLUME	1.481E+01	CU.FT.

GROUP	STM ENTH BTU/LBm	STM TEMP Deg F	STM PRESS PSIA	GAS TEMP Deg F	AVE TUBE DEL Deg F	TUBE TEMP Deg F
1	1.655E+03	1.292E+03	1.829E+03	1.960E+03		0.000E+00
2	1.737E+03	1.428E+03	1.782E+03	2.061E+03	1.509E+03	7.003E+01

GROUP	GAS REY	GAS REY	EFF GAS HTC Btu/Hr/SqFt/F	TUBE HTC Btu/Hr/SqFt/F	TOTAL HTC	HEAT FLUX Btu/Hr SqFt
1	0.000E+00	0.000E+00	0.000E+00	0.000E+00	7.654E+00	0.000E+00
2	1.865E+03	6.298E+03	8.838E+00	1.000E+04	7.212E+00	8.174E+04

WATER/STEAM FLOW	5.717E+04	LBm/HR	GAS FLOWRATE	1.420E+05	LBm/HR
HEAT TRANSFER AREA	3.563E+02	SQ.FT.	PRESS. DROP LENGTH	1.600E+01	FT.
TUBE O.D.	1.000E+00	INCH	TUBE WALL THICKNESS	2.800E-01	INCH
NUMBER OF CIRCUITS	2.600E+01		FREE FLOW AREA	3.892E+01	SQ.FT.
WATER/STEAM SIDE DP	2.856E+02	PSI	GAS SIDE DEL P	-6.658E-02	IN H2O
TOTAL H.T. LENGTH	3.640E+02	FT.	H.T. LENGTH/TUBE	1.400E+01	FT.
NUMBER OF FINS	6.000E+00	FINS/INCH	TRANS. TUBE PITCH	2.350E+00	INCH
FIN THICKNESS	3.000E-02	INCH	LONG. TUBE PITCH	2.000E+00	INCH
FIN HEIGHT	1.875E-01	INCH	BUNDLE HEIGHT	1.667E-01	FT.
FREE FIN SPACING	1.367E-01	INCH	BUNDLE WIDTH	5.092E+00	FT.
FIN EFFICIENCY	8.398E+01	PCT.	BUNDLE MASS	1.062E+03	LBm
AVE. GAS VISCOSITY	1.230E-01	LBm/FT/HR	CROSS SECTION AREA	7.128E+01	SQ.FT.
AVE. GAS CONDUCTIV.	5.116E-02	BTU/FT/HR/F	BUNDLE VOLUME	1.188E+01	CU.FT.

GROUP	STM ENTH BTU/LBm	STM TEMP Deg F	STM PRESS PSIA	GAS TEMP Deg F	AVE TUBE DEL Deg F	TUBE TEMP Deg F
1	1.737E+03	1.428E+03	1.782E+03	2.061E+03		0.000E+00
2	1.787E+03	1.505E+03	1.496E+03	2.122E+03	1.577E+03	6.482E+01

GROUP	GAS REY	GAS REY	EFF GAS HTC Btu/Hr/SqFt/F	TUBE HTC Btu/Hr/SqFt/F	TOTAL HTC	HEAT FLUX Btu/Hr SqFt
1	0.000E+00	0.000E+00	0.000E+00	0.000E+00	7.212E+00	0.000E+00
2	2.491E+03	7.949E+03	1.513E+01	1.000E+04	1.257E+01	6.767E+04

Next to
Final
Superheat
Row

Final
Superheat
Row

Table 12. OTSG Heat Exchange Module Size Summary

Gas Side Del P	-3.9 In H₂O	Total Bundle Volume	548 Cu. Ft.
Total H.T. Length	13,832 Ft.	Total Bundle Mass	36,330 Lbm
Total H.T. Area	32,280 Sq. Ft.	Total Bundle Depth	7.4 Ft.

- Transportation loads of 2 g's in all directions. Normal shipping would be with OTSG complete with insulation laying on side opposite superheater outlet. Combustion section to be shipped separately.
- Maintainability. Each circuit individually supported and removable by cutting inlet and outlet tubes to allow the circuit to be slid out of the matrix into the combustion duct or exhaust for repair or maintenance as needed.
- Allowance for thermal expansion between each individual tube and each circuit.
- The steam generator to supply turbine piping system thermal expansion with minimum frictional loads.
- Positive gas path seals around the perimeter of the matrix for performance and prevention of high temperature gas blow-by over heating carbon steel exhaust components.
- Maximize simplicity for high reliability and reduced costs.

The basic concept is an innovation on the well proven pendent superheater designs used around the world in thousands of steam generators. In a high temperature gas stream the tubes cooled by steam can easily support their own weight. By supporting the vertical tubes with a cooled support structure at the top of the casing out of the gas flow path a highly reliable mechanical design can be configured for 2500°F inlet gas temperatures.

Figure 32 identifies ASME code calculation tube locations that correspond to steam conditions, tube dimensions and materials used to establish code compliance (Table 13). It should be noted that OTSG is code certified for 1536 psig and 1535°F to allow for system pressure losses and control temperature tolerance and temperature losses for a 1500°F and 1500 psig HPSS. Actual test temperature losses were lower than expected being only about 5°F and control system tolerance was lower than expected being about 5°F. However, pressure loss to the turbine was higher being a total of 65 psi. The turbine was designed at a match steam condition of 1500 psia or 1485 psig and was operated with the steam generator at 1535 psig and the turbine at 1495 psia. Figure 33 is a copy of the page from the approved Code calculations for location 11 at the final superheater outlet.

Methods of construction of the OTSG are shown in Figures 34 through 36. Each circuit (Figure 10) is supported as shown by a box beam fabricated from 316 SS and Hastelloy X in the hot section up to a gas temperature of about 2000°F. In the hotter zones cooled support pipes support the tubes. The header is not required to support the hot end of the bundle. Figure 35 shows the final superheater support with a cooled support pipe to the right of the outlet header. This structural pipe is insulated and outside of gas flow path. Both ends of the circuit box beam are supported by links to two eight inch square carbon steel beams on the outer casing structure. At the cold end (Figure 36) each circuit is supported by individual links that allow circuit to circuit temperature differential expansion during hot starts and normal shutdown dry out procedures.

Figure 32. ASME Code Calculation Tube Locations

Table 13. Summary of Conditions at Each Location

Location	Alloy	S (psi)	Temp (F)	Pressure (psia)	Pressure (psig)	Tube OD (In)	Wall Thick (In)
1	800	18700	170	2100	2086	1.000	0.083
2	800	16100	650	2100	2086	1.000	0.083
3	800	16100	650	2100	2086	1.000	0.083
4	800H	11900	650	2060	2046	1.000	0.120
5	800H	9300	1150	1950	1936	1.000	0.120
6	617	15400	1150	1950	1936	1.000	0.120
7	617	10000	1300	1925	1911	1.000	0.120
8	617	10000	1300	1925	1911	1.000	0.200
9	617	5020	1435	1875	1861	1.000	0.200
10	617	5020	1435	1875	1861	1.000	0.280
11	617	2380	1585	1550	1536	1.000	0.280
12	617	3040	1535	1550	1536	1.000	0.020
13	800	16100	650	2100	2086	1.000	0.120
14	800H	9300	1150	1950	1936	1.000	0.176
15	617	15400	1150	1950	1936	1.000	0.120
16	617	11500	1275	1925	1911	1.000	0.120
17	617	5580	1415	1875	1861	1.000	0.200
18	617	5580	1415	1875	1861	1.000	0.280
19	617	3040	1535	1550	1536	1.000	0.280

Notes: (1) Wall Thickness is nominal supplied; (2) Locations 1-12 are for straight sections;
(3) Locations 13-19 are for u-bends.

ASME BOILER AND PRESSURE VESSEL CODE CALCULATIONS
SECTION I 1989 EDITION
DECEMBER 31, 1989 ADDENDA

REFERENCE: Location 11 - Superheater Outlet (Within Heat Transfer Section)
Straight Section
Minimum Tube Thickness
PG-27 Cylindrical Components Under Internal Pressure
PG-27.2 Required Tube Thickness

DESIGN PARAMETERS

Material: Alloy 617 (UNS N06617)
per ASME SB444
Code Case 1956-2

Outside Diameter: $D := 1.0 \text{ in}$
Nominal Shell Thickness: $t_n := .28 \text{ in}$
Thickness Factor: $e := 0 \text{ in}$
Pressure: $P := 1536 \text{ psi}$
Temperature: $T := 1585 \text{ F(max)}$
Allowable Stress: $S = 2380 \text{ psi}$

Nominal Thickness $t_n = 0.28 \text{ in}$
Less Tolerance - 10% $\text{tol} := t_n \cdot .1$ $\text{tol} = 0.028 \text{ in}$
Less Corrosion $C := 0 \text{ in}$

Minimum Shell Thickness $t := t_n - \text{tol} - C$ $t = 0.252 \text{ in}$

CYLINDRICAL SHELLS

Required Tube Thickness PG 27.2.1

$$t_r := \frac{P \cdot D}{2 \cdot S + P} + 0.005 \cdot D + e$$

$$t_r = 0.249 \text{ in}$$

Difference between minimum and required thickness
 $dt := t - t_r$ $dt = 0.003 \text{ in}$

Maximum Allowable Pressure PG 27.2.1

$$P_m := S \cdot \frac{2 \cdot t - .01 \cdot D - 2 \cdot e}{D - (t - 0.005 \cdot D - e)}$$

$$P_m = 1561 \text{ psi}$$

Difference between actual and allowable pressure
 $dP := P_m - P$ $dP = 25 \text{ psi}$

Figure 33. Page 10 of Code Calculations for Superheater Outlet Tube

Figure 34. OTSG Heat Transfer Matrix Cross-Section

Figure 35. Superheater Support Using Cooled Insulated Structure

The complete OTSG used 26 identical circuits stacked on a horizontal assembly fixture (Figure 37). This fixture is also used for transportation and installing the OTSG in the vertical tube arrangement of the HPSS. After stacking the 26 circuits using permanent tube spacers and temporary spacers the header was welded on to complete the OTSG matrix (Figure 11).

Proprietary fin fabrication allowed three different fin materials to be bonded to the three different tube materials with crevice free joints at the fin to tube bond capable of operation to 1800°F without oxidation damage. Tubes were welded to form circuits by automatic orbital welding using a TIG process to join tubes by butt welding without a filler by melting the two sides of each butt joint. For tubes over 0.083 inch wall thickness, the tubes were machined to 0.083 inch thick and the automatic orbital weld of butt joint was made for the root pass without a filler. After inspection, the root pass diameter was filled by TIG manual welding using a filler rod to the OD of the two tubes. By welding u-bends or crossover tubes to the 39 tubes that form a continuous flow path, a complete OTSG circuit (Figure 10) was completed. It was hydrostatically tested to ASME requirements at this stage to allow individual inspection of each weld joint.

4.3.9 Heat Transfer Matrix Fabrication

Simplicity is emphasized in the steam outlet header design that is made combustion gas tight by an insulated bellows designed for 20 inch water column differential pressure (Figure 36). The header is also completely free in the horizontal plane being mounted on slide pads with low coefficients of friction (graphite impregnated menenilite) and since it is at the center line of the valve (keyed to the horizontal and vertical centerline of the turbine) the header only has sliding friction loads and no pipe bending loads. All of the thermal expansion flexing is taken over the 14 foot length of the superheater tubes. Reaction forces at 14 foot length to flex the tubes the required 0.75 inch distance are approximately two pounds force.

Figure 36. Cold End Circuit Support Links

An 8 inch thick insulation system is used to support the weight of the steam generator to allow horizontal circuit assembly and transportation. Structural casing is the first layer with an egg crate arrangement of stainless steel scalloped plates welded as shown in Figure 39. A similar structure of scalloped plates is jointed at 90 degrees to those in the photograph. These plates are 0.1 inch thick and designed to minimize heat loss while supporting twice the matrix weight for transportation loads. Eight inches of compressed ceramic insulation fills all of the space between the stainless steel liner plates that protect the insulation and seal the gas flow path within the matrix. These liner plates also 0.1 inch thick, are made of many small plates to allow thermal expansion. They are held in position by pins welded to the scalloped plates and retained by washers welded to the pins. Figure 34 shows a typical pattern of plates, pins, and washers used at the gas exit end of the steam generator. At a gas temperature of 1600 to 1800°F Hastelloy X plates and washers are used. Above 1800°F twelve inch cube ceramic fiber blocks are used for insulation. They are rated to 2600°F and are fastened with a stainless steel wire frame welded to the outer case and buried 6 inches in the ceramic fiber block. High density 12 pounds per cubic foot of almost pure alumina

4.3.10 Insulation

Each circuit is connected at its outlet to the header in parallel and instrumented in final assembly with over one hundred thermocouples (T.C.) used to evaluate operation. Each outlet tube has two T.C.'s, a T.C. is used at the inlet and outlet of the four circuit screen tubes. Circuit 12, a middle circuit of the bundle, instrumentation is shown in Figure 38. The outlets are 1201 and 1202 with 1203 and 1204 the screen instrumentation. Circuit 12 and an end circuit 1 had additional instrumentation 1205, 1206, 1207, 1208, and 1209 for further evaluation. Key structural areas were also instrumented.

Figure 37. Horizontal Assembly Fixture for OTSG.

Figure 38. Circuit 12 Instrumentation Locations

Figure 39. Scalloped Plate Egg Crate Insulation Support

block has proven to be a good insulation at the 2500°F operating temperatures of the combustion burn-out duct and steam generator high temperature end. Other systems such as hardboard ceramic liners, vacuum formed blocks and rammed plastic refractory have been tested or used for sections of the high temperature end but are more costly and do not now appear to be necessary, but offer higher temperature and velocity capabilities.

4.3.11 Combustion System

During the HPSS concept development phase, the issue of emissions was a critical factor. Low emissions of NO_x was achieved by reducing flame temperature and premixing the fuel gas with the combustion air. CO was minimized by long residence times and hot walls to prevent quenching of reaction zone products. In the combined cycle configuration of HPSS combustion air comes from the gas turbine vitiated exhaust gas containing large amounts of CO₂ and H₂O that reduce flame temperatures and NO_x, but can also increase CO formation. To obtain highest fuel utilization efficiency, the final stack losses must be minimized by reducing the oxygen level as low as possible consistent with NO_x and CO emissions. A theoretical flame temperature of 2500°F was selected for the HPSS as a compromise temperature biased towards obtaining ultra low NO_x. At 2500°F ultra-low CO was obtained by high residence time to complete the reactions. The 2500°F combustion temperature with hot walls combined with a 16 foot long combustor and a plug velocity calculated residence time of over 500 msec reduces CO to very low values.

A production COEN lean premixed register swirl burner (Figure 40) was selected to be integrated with the OTSG's hot wall combustion reaction zone. In Figure 40 the eight fuel injection nozzles are shown in the plane of the swirl flame holder. The burner is 33 inches in diameter and is rated at a heat release of 110 million Btu/Hour at a pressure drop 10 inches of water column. This configuration although not completely premixing is a production proven arrangement with hundreds in the field in low emission service. Although a greater amount of premixing is possible, this configuration meets the most stringent United States NO_x and CO emissions now required. At the gas turbine O₂ reference level, the NO_x and CO are both less than 10 ppm (less than 30 ppm at boiler O₂ reference level of 3% O₂).

Extensive work on increasing the level of premixing by several companies in the industry have demonstrated much lower emissions with equipment now in operation in the field but not yet in production.

In the HPSS 100 hour development test, exhaust gas recirculation of about 22% from the OTSG exhaust stack is drawn into fan suction (downstream of the air flow control valve, Figure 41). This reduces the O₂ content to about 15% in the windbox to simulate vitiated gas turbine exhaust gases in the development tests. For a standalone (without gas turbine) steam plant exhaust gas recirculation would be used (as it was tested) to reduce emissions and to maintain higher than 90% boiler efficiency with the 2500°F gas inlet design temperature to the OTSG.

4.4 STEAM TURBINE

4.4.1 Design Approach

To a large extent the design approach for the steam turbine was driven by the high temperature (1500°F) steam selected for the cycle. For the turbine to operate at elevated temperature and pressure, gas turbine developed super-alloy materials were required. To keep the cost of the turbine in check, it was decided to design as simple and as compact a machine as possible. This approach led to a single stage design, with a high pressure ratio across the stage. Solar applied design concepts and materials developed for gas turbines to the steam turbine.

Figure 40. Lean Premixed Coen Low Emissions Burner (33 inch diameter, 110 million Btu/hr).

Figure 41. Air Supply Fan with Exhaust Gas Recirculation Connection

To minimize thermal distortion in the high temperature portion of the turbine inlet, and to avoid losses due to partial arc of admission, a full 360 degree arc of admission was used for the nozzles. By taking the full pressure reduction in the single stage nozzle, and using impulse (low pressure drop) rotor blading, the static pressure in the rotor flow path is minimized, which allows the use of the highest, and most efficient, blading. In addition temperature in the rotor was minimized.

To fully expand the steam from 1500 psia to 170 psia in a single stage nozzle requires a converging-diverging nozzle, which provides a supersonic spouting velocity (4000 ft/sec). To effectively utilize this velocity, and keep the rotor as small as practical, a turbine rotational speed of 29900 rpm was selected, with a rotor pitch diameter of eleven inches, and a tip speed of about 1540 ft/sec.

An over-hung rotor approach was selected to avoid the need of a shaft penetration at the high temperature (inlet) end of the turbine. Figure 13 (Page 10) is a cross-sectional drawing showing the single-stage steam turbine designed for this program. At the design flowrate of 56,000 pounds/hour of steam at 1500 psia and 1500°F, and exhausting to a pressure of 150 psig, the turbine produces 5600 horsepower. The turbine package itself is about three feet in diameter, five feet long, and weighs about 2500 pounds.

To cool the rotor disk, and the outer casing, a cooling concept was designed which extracts some relatively cool steam (1000°F) from the turbine exhaust, and circulates it back between the inner and outer casings, through the inlet casing, and inserts it at the centerline of the disk. This cool steam flows radially outward along the front of the disk to help cool it and reenters the flow stream between the nozzle and the rotor blade. The back side of the disk is cooled by exhaust steam leaking past the labyrinth seal on the shaft. The cooling issue is discussed in more detail in Section 4.7.3.

4.4.2 Mechanical Arrangement

Steam enters from left (Figure 13), on the turbine centerline, where it enters the inner casing which distributes the steam to a full admission nozzle ring. The steam passage is formed by two concentric hemispherical shapes connected with twenty-four radial ribs. This results in a light weight, high strength casting to contain the high temperature, high pressure steam. Figures 14 and 15 (Page 12) are photos of the inner casing, which is made of Inconel 939, a high temperature, high strength alloy. The inlet end of the turbine is actually in two parts, with the high pressure internal casing surrounded by an outer casing. By surrounding the inner casing within the steam at exhaust conditions, the critical inner casing is subjected to reduced differential pressure and temperature, thus reducing the loads and stresses on the casting. Differential expansion between the hot inner casing and the relatively cool outer casing is accommodated by a separate floating sleeve which contains the hot steam, and incorporates 20 sliding solid ring type seals. The hot inner ring has attached heat shields, one to shield the relatively cool outer casing and the other to shield the rotor disk. Both of these pieces are designed to operate at about 1100 to 1200°F, compared to 1500°F for the inlet steam ring.

A vertical flange, normal to the axis of the turbine, joins the inlet and exhaust sections of the outer casing. Concentricity is held with a rabbet fit and sealing is metal to metal, with triple-boiled linseed oil as a non-filled sealer against exhaust pressure and temperature. The inlet section is connected to the steam pipe via a convoluted section made from Inconel 617 ring forgings that provides for the gradual temperature gradient from the 1500°F steam pipe to the 1000°F outer casing. A critical electron beam weld was developed to connect the nickel alloy inlet steam line to the cast steel casing. The structure of the exhaust casing is used to carry the internal bearing housing, and to connect the turbine to the base plate, via a trunnion mount.

The bearing housing carries the two journal bearings and the thrust bearing, and includes porting for the utilities; lubrication oil, buffer air, and instrumentation lines. A convoluted section is designed to minimize the thermal conduction from the exhaust casing, at 1000°F, to the bearings and oil, which run at about 200°F.

The rotor is a single stage impulse type, which is over-hung from the two tilting-pad journal bearings. A photo of the rotor is shown in Figure 9 (Page 7). There is no shaft penetration into the hot end casing. On the first build, the blading was integral with the disk, having been electro-chemically milled into the periphery of the disk (Figure 42). There were 91 impulse blades with a span of 0.58 inches and a chord of 0.8 inches. Following a fatigue failure of the integral blades, which is discussed in some detail in Section 4.4.7, the blades and disk were redesigned for inserted blades on fir tree attachments.

Figure 42. Turbine Disk with 91 Integral Blades and No Shroud

Sixty two shrouded blades (Figure 43) were used in this final configuration. Leading and trailing edge radii were relatively large at .012 inches, to reduce the stress concentrations at the blade root. Generous fillets were incorporated between the airfoil and the platform and between the airfoil and the shroud.

The disk (Figure 44) is solid, without a central hole, and is located and connected to the shaft with a rabbet fit and bolted flange. Labyrinth seals on the shaft, with intermediate buffer air gallery, prevent exhaust steam from entering the lubrication oil. A tilting-pad thrust bearing (Figure 45) is located at the drive end of the turbine shaft, to carry the axial load on the rotor. A dry, flexible disk pack coupling is used to connect the 29,900 rpm turbine shaft to the reduction gearbox.

For the Phase III testing, a 2.02/1 reduction gearbox was used to lower shaft speed to 14,800 rpm, which matches an available water brake dynamometer. When the generator set is assembled for the Phase IV field tests, a different gear box will be purchased, which reduces the shaft speed to 1800 rpm needed to drive a 60 Hz generator.

Figure 43. Shrouded Inserted Blade Configuration

Figure 44. Steam Turbine Disk

Figure 45. Tilt Pad Thrust Bearings

4.4.3 Materials (High Temperature Solutions)

A key material issue was selecting the best material for the turbine inlet inner casing, and nozzle ring, which operate continuously at 1500°F with an internal pressure of 1500 psia. To minimize thermal stresses during start transients, and to keep cost down, it was desirable to keep the section thickness as thin as possible. This required a material with high strength at elevated temperature, and a design that minimized stresses. After considerable materials research, the selected material is Inconel 939, a vacuum-melted and investment cast cobalt-chromium-nickel super-alloy. It has a rupture stress (100,000 hour) of 18 ksi at 1500°F, and an allowable design stress of 10 ksi was used.

The transition piece between the inlet steam line and the turbine outer casing was made of Inconel 617, the same material used in the steam piping and the superheater section of the steam generator. This is a forged nickel-chromium-molybdenum super alloy. Inconel 617 is one of the few materials approved by the ASME Boiler and Pressure Vessel Code for operation at 1500°F. Allowable design stress per code is 3600 psi at 1500°F.

An electron beam weld was made between the Inconel 617 transition piece and the outer casing, which is made of cast WC-9 (2-1/4% chromium, 1% molybdenum alloy steel). The chrome-moly portion of the casing runs at 1100°F maximum, at which temperature the WC-9 has an allowable design stress of 4400 psi, according to the ASME Boiler and Pressure Vessel Code.

A high strength material capable of operation at 1050°F was required for the rotor disk and blades. Inconel 718 was selected for this application. Alloy 718 has excellent creep-rupture properties, and is widely used in gas turbine applications. At 1050°F it has an average yield strength of 150 ksi, and a 100,000-hour creep rupture strength of 84 ksi.

4.4.4 Flow Path Design

An aerothermodynamically matched steam flow path was designed, consisting of the supersonic nozzles, rotating buckets (blades), exit guide vanes, diffuser, and exhaust hood collector section. A sectional view of the flow path is shown in Figure 46. Fourteen full admission supersonic, converging-diverging, nozzles expand the steam to over Mach 1.9. The diverging section of the nozzle incorporates a bell shape, like a rocket nozzle, to minimize losses. Due to the supersonic flow at the nozzle exit, and the finite leading edge thickness (.025 in.) of the rotor blades, a bow wave crosses the entrance to the rotor flow path, and a normal shock wave occurs. Downstream of the shock the flow is subsonic as it passes between blade surfaces generated by circular arcs that control the turning to produce favorable pressure distribution along the concave and convex surfaces. The rotor blading is impulse type, which has very low pressure drop.

The rotor exit steam has a large swirl velocity, because of the low velocity ratio (0.33) of bucket peripheral velocity to available energy velocity released by the nozzles. The swirl energy is recovered by the tandem exit guide vanes that turn the flow to the axial direction. Substantial static pressure is recovered in the diffuser section aft of the guide vanes. The steam then dumps into the exhaust hood collector slightly above the exhaust header pressure, and exits the machine at a modest velocity of less than 150 feet per second. Consequently, the static pressure between nozzle exit and bucket inlet is lower than the exhaust back pressure (by 8 to 10 psi), which increases the power output due to increased nozzle exit velocity. Also, this makes possible the cooling steam flow that is explained in Section 4.4.1.

4.4.5 Rotor Dynamics and Bearing Analysis

An overhung rotor configuration was selected to avoid the need for a shaft penetration at the hot end of the turbine. This results in a 28 pound disk overhanging the forward bearing by 6.6 inches, a 12 inch bearing span, and the center of gravity of the thrust runner and coupling combined is 6.6 inches aft of the aft journal bearing. Resulting bearing loads are 55 pounds on the forward bearing and 22 pounds on the aft bearing. The bearings selected are 2.5 inch diameter, four-pad tilting pad journal bearings, with the gravity load acting between pads. Bearing length/diameter is 0.4.

Figure 46. Turbine Flow Path Arrangement

Rotor dynamic analyses included undamped critical speed analysis, journal bearing definition, damped critical speed analysis, and forced response calculations. Calculations showed the rotor would operate between the second and third critical speeds at design operating speed of 29,900 rpm. The first two criticals are rigid body modes, with the third critical speed being the first bending mode of the shaft. Figure 47 is the damped critical speed map as predicted for the final rotor configuration as tested during the HPSS development tests.

Figure 47. Damped Critical Speed Map

4.4.6 Critical Component Analysis

Nozzle Ring: A critical area of the steam turbine design was the nozzle ring and inlet casting which had to operate at a temperature of 1500°F, with an internal pressure of 1500 psia. It was important to keep the section thickness as thin as possible to minimize thermal gradients and stresses during warm-up. Selection of the Inco 939 material was discussed in Section 4.4.3. Detailed design of the casting was iterated many times with the engineers at the selected foundry (Precision Castparts Corp., PCP) until the design was suitable for casting. PCP developed a proprietary version of the Inco 939 for casting. The design included internal webbing to carry the internal pressure load, and ports through the webs to pass the cooling steam. Several attempts at casting the part were made before a satisfactory part was completed. At the time, this casting was the largest investment casting of this material ever produced. Figure 14 is a photograph of the finished casting.

Rotor: The turbine disk is a highly stressed part, which operates at a temperature 1000 to 1100°F, and is a critical component due to the high of kinetic energy it contains when rotating at 29,900 rpm. Finite element models were analyzed to establish the stress distribution and fundamental vibration modes of the disk. Disk profile was modified to equalize the stress in the disk, and keep the total mass to a minimum. Material selected for the disk is INCO 718, commonly used for turbine disks in gas turbine engines. Average yield strength at 1050 F is 150 ksi, 100,000 hour average rupture strength is 84 ksi. Steady design stress is about 74 ksi at the disk center, and 80 ksi peak (40 ksi average) in the neck of the blades. Bearing stresses on the blade attachments (fir-trees) are allowed

to reach 98 ksi. Disk and blade life is calculated to exceed 100,000 hours of operation and 10,000 start/stop cycles.

Vibration analysis of the turbine disk assembly with the blades produced the natural frequency vs. speed curve shown in Figure 48. In all cases, the natural frequencies are well above the impulse line, which means there are no fundamental resonances within the operating disk. The only interferences are at higher orders, at frequencies above 15 kHz, compared to an operating speed of 500 Hz, and a nozzle passing frequency of 7 kHz. It was concluded that the very high frequencies would not have sufficient energy to cause excessive vibratory stresses in the turbine disk, or the blading.

Figure 48. Disk Assembly Natural Frequency at Operating Speed

4.4.7 Steam Turbine With Inserted Blades

During the initial testing of the steam turbine, the integral turbine blades experienced failures on the leading and trailing edges at the blade root. Failure analysis confirmed that the cracks were due to high cycle fatigue. On the day of the failure, the turbine was running at reduced speed to evaluate a cooling flow question. It was concluded that the most likely cause of the failure was the lack of damping due to the integral blades, and that the blade had a fundamental frequency equal to the fourth harmonic of nozzle passing frequency, which is 7000 Hz at 30,000 rpm.

Following the failure, it was decided to redesign the turbine such that it had shrouded blades and a means of vibration damping. This design included 62 inserted blades, compared to the 91 integral blades on the first build, with fir-tree roots scaled from a proven production gas turbine. Figure 44 is a photograph of the turbine disk with fir-tree slots for the blades. The individual blades have integral tip shrouds as shown in Figure 43. Figure 49 is a photo of the assembled disk with the 62 inserted blades and Figure 50 shows the complete rotor assembly.

Figure 50. Rotor Assembly

Figure 49. Assembled Disk and Blades

With the blade tips connected together, the individual blades are not free to vibrate at their fundamental bending frequency. Since each blade is loaded by the steam jet from each nozzle at a slightly later time than its neighbor, the forcing functions on adjacent blades are out of phase with each other. Because the blade tips are constrained to move together, the blade displacement cannot coincide with the forcing function, and the blade can not vibrate in resonance. A damping device is included which dampens any blade vibration that may occur.

4.5 SYSTEM CONTROL VALVE

A single system valve, designed and built by Con-Tek Valves Division of Fisher Controls International, (Figure 8, Page 7) is the only valve in the entire high temperature side of the steam system. It also has the only component (the valve plug) that is designed to have motion at 1500°F. It was uniquely designed to have only one moving part, the plug and integral hollow ported actuator stem that allows the valve to provide multiple functions. Figure 1 (Page 2) illustrates the simple header connection to valve in an expansion loop arrangement. Steam from the header enters the valve on the OTSG side and makes a 90° turn through the valves internal 3.0 inch port and is directed into the turbine (Figure 8). The vertical line connected downward to the 150 psig low pressure system in Figure 8 is the steam by-pass to an industrial process or a relief vent depending upon the site arrangement. A spray desuperheater is integrated internal to the by-pass leg of the valve to reduce the discharge temperature to less than 1000°F.

The valve although simple, provides many functions such as:

- Turbine stop valve
- Turbine by-pass
- By-pass desuperheater
- Turbine throttle valve
- System drain valve
- System vent valve
- Boiler stop valve

A crossection (Figure 51) shows the valve has only one moving part a valve plug combined with a hollow stem cylindrical multiple orifice by-pass.

A 2000 psig hydraulic actuator system is designed to operate the valve. Multiple fail safe features are included in the actuation system to close the valve to the turbine in case of overspeed with or without the hydraulic pump functioning. In the normal rated operating condition the turbine throttle and stop plug will be in the back seated condition with the valve fully open to the turbine. A number of small holes drilled in a thick cylinder around the entrance to the valve plug chamber protects the turbine from larger particles that may be left in the steam system.

Since the HPSS is designed to operate in the floating pressure and temperature mode, the valve would only move from the back seated position if there was a turbine trip or sudden reduction of load. Planned changes of load would generally be slow and controlled by reducing pressure and temperature produced by the OTSG. The turbine when used as a generator drive would be locked onto the grid frequency and the throttle would be wide open and back seated.

In the case of a sudden turbine trip or loss of load, the valve is designed to move full stroke in 150 msec to the full by-pass or no load position, to prevent turbine overspeed. Traces in Figure 52 show cold test results where the valve closing was measured to be 100 msec.

Figure 51. HPSS Control Valve Cross-Section

Figure 52. Valve Trip Closing Time Cold Tests

A key functional requirement of the valve is to maintain system pressure constant during rapid changes in steam flow to the turbine. Either high or low pressure transients can be potentially damaging to components in the HPSS. Over 35 hot system trips were made with the complete system at full rated conditions, and excellent results with response to turbine speed reduction and system pressure fluctuations were recorded. Pressure transients of about 70 psi above the operating level pressure occurred at initiation of the trip. If operating at 1540 psig, the pressure would go to 1610 for a second or two during valve trips at full steam flow. This is in the desired direction to prevent an OTSG swell transient that could cause a rapid reduction in system temperatures and resulting thermal stresses. No valve transients resulted in negative (swell) temperature transients.

In the start cycle, the valve is used to vent the system and warm-up and drain any condensate from the cold tubing and piping. It would be in the by-pass mode with only a small flow through the turbine. Since the OTSG is started dry, superheated steam starts flowing in a few minutes at a temperature near at the gas temperature entering the OTSG's final superheater. Thus only a small amount of condensed steam flows through the valve before it is superheated. When the valve is electrically heated to about 800°F to speed up the warm up cycle and minimize thermal stresses, 800 or 900°F superheated steam at 50 psig is used to roll the turbine during warm up. Speed control through the steps of turbine warm up is excellent, and good speed control with $\pm 0.1\%$ could be maintained at full speed and full power on a water brake dynamometer.

4.6 SYSTEM CONTROLS

Control of the prototype system was accomplished using a modified version of the control package used for production Solar gas turbine engine/generator systems. A Turbotronics control console (shown in Figure 53) incorporating an Allen-Bradley programmable logic controller (PLC), and a Bently-Nevada vibration monitor is used. The control system provides automated start-up and warm up of the system, as well as steady state control of steam pressure and temperature and turbine speed. It also provides numerous alarms for parameters out of normal range, and safe shutdowns in the event that speed, pressure, or temperature exceed allowable values. In addition to the primary control system, there are two supplementary systems which control the COEN Low NOx burner and the hydraulic system for the steam turbine control valve. These systems were supplied by the major component suppliers.

Figure 54 is a piping and instrument diagram, which illustrates the fluid system, the various control valves and pumps, and the instrumentation used for control of the system as tested in the test cell. All active control elements are shown as are the instrumentation points used as inputs to the control system. All control inputs and outputs are designated with circles inside of squares. Letters in the top half of the circle designate Analog or Discrete, and Input or Output. An analog input is designated AI etc. Inputs to the control system include some 23 analog inputs, consisting of measured pressures and temperatures, and 9 discrete inputs, consisting of pressure and level switches. Outputs from the control include six analog outputs to control modulating valves, and five discrete outputs to control pumps and solenoid valves.

Hexagons represent parameters which are sent to the data acquisition system for performance analysis purposes, but are not used in control functions. Each portion of the system is briefly described below.

4.6.1 Water/Steam System

Deionized water is supplied to the run tank through a modulating valve which is controlled by an analog level sensor. At start-up, the water boost pump is signaled on with the start signal. When the feedwater pump inlet pressure switch indicates the boost pump is functioning properly, the main feedwater pump is turned on. Since the water valve to the steam generator is closed at start, and since the feedwater and boost pumps need water flow to maintain cooling, a water bypass system is needed. This consists of a modulating valve which maintains the appropriate pressure set point on pump discharge pressure. The control system includes an algorithm to maintain the necessary pump discharge pressure as a function of power level, as illustrated in Figure 55. This saves pump horsepower at reduced system power levels, and reduces the pressure drop across the feedwater valve, which reduces the potential of cavitation damage to the valve trim at low power conditions.

Figure 53. Control Console

Feedwater is supplied to the steam generator via the feedwater control valve, and the control maintains the required flow set point sensing feedback from the flow measurement. The flow measurement is made by sensing the pressure differential across a precision orifice plate. During warm-up, the flow setpoint is ramped up with the power control to the burner fuel valve, as shown in Figure 56. This open loop water flow setpoint provides rapid adjustment to changes in power level. In addition, and for the steady state control, there is a feedback loop based on a steam temperature error signal. When the measured steam temperature exceeds the temperature setpoint, the water flow set point is increased, which adds more feedwater to cool off the steam. The steam temperature setpoint is increased gradually during warm-up, from an initial value of 800°F to 1500°F at full power, Figure 57.

For part load operation, the steam temperature must be reduced to preclude overtemperature of the turbine exhaust and casing. In a field application where the turbine is exhausting to a constant pressure steam process header, as the turbine power is reduced, the steam inlet pressure decreases. This necessarily reduces the pressure ratio across the turbine and the amount of work taken out of the steam, which also decreases the temperature reduction across the turbine.

Figure 54. High Performance Steam System - Test P and ID.

Figure 55. Feedwater Pressure Setpoint

Figure 56. Feedwater Flow Setpoint

Figure 57. Steam Temperature Setpoint

4.6.2 Steam Turbine & Control Valve

Steam turbine speed is controlled by the main steam valve, which serves both as a turbine throttle valve and a steam bypass valve. To make the system inherently safe, a three-way valve was especially designed which provides a constant resistance to the steam flowing from the steam generator, whether the steam is flowing to the turbine or the bypass, or partially to each. With this system, the steam generator can not be over pressurized when the turbine throttle is closed. The main steam valve primarily responds to steam turbine speed, to maintain the measured speed at the setpoint called for by the controller. During warm-up, the turbine speed is ramped up over a period of time, until it reaches full operating speed. Due the small inertia of the steam turbine, this main steam valve must close very rapidly in the event of a turbine overspeed condition.

A dedicated high pressure hydraulic control system with accumulator and quick release solenoid accomplishes this overspeed trip function. The valve modulation for speed control is also done by the hydraulic system, which has its own controls incorporating several safety shutdowns in the event of low hydraulic pressure or level, or excessive oil temperature or pressure. A hydraulic accumulator provides sufficient fluid to close the valve quickly in the event of an electrical or system failure, and a large mechanical spring will close the valve in the event of a failure in the hydraulic line from the accumulator.

Built into the main steam valve is a desuperheater nozzle which cools the steam sent to the bypass (low pressure) leg of the system. Bypass steam is maintained at a temperature approximately equal to the turbine exhaust temperature to prevent rapid changes of temperature on the piping system. Water flow to the desuperheater is regulated by the desuperheater valve, and is controlled to maintain the proper temperature at the downstream thermocouple.

Within the steam turbine, the two journal bearing temperatures and the thrust bearing temperature are measured and sent to the control monitor. Excessive temperatures will cause alarms and eventual turbine shutdowns. A magnetic pickup reads the teeth on a special gear to provide accurate turbine speed indication. Proximity probes are located adjacent to each journal bearing, which sense the proximity of the rotating shaft. This information is sent to the vibration monitor, which again will initiate an alarm and a shutdown if shaft vibrations become excessive. It was this system which sensed the blade failure during the initial testing in 1994.

For the prototype system in the test cell, an operator controlled turbine back-pressure valve was included in the system. This allowed the turbine to be tested at back pressures to be used in the eventual cogeneration system, while venting the steam to atmosphere.

Figure 58. Air & Fuel Flow and A/Fuel Ratio

4.6.3 Burner/Steam Generator

Power to the system is regulated by the amount of fuel sent to the burner. An open-loop signal is sent from the system controller to the burner controller. During warm-up the power control is ramped up from idle (about 20% fuel flow) to full fuel flow. Fuel flowrate is controlled by the modulating valve. The low NO_x burner system actually incorporates three valves, one in the air supply at the blower inlet, one in the flue gas recirculation (FGR) system, and the fuel valve. A separate control system supplied by the burner manufacturer, COEN, takes the power control signal from the main controller, and creates three parallel signals for the three valves. A separate PLC includes 10-segment characterizers which allow curve fits to be created to establish the proper relationships between the three valve positions, to minimize emissions at all power levels. Figure 58 illustrates the air flow, fuel flow, and air/fuel ratio as a function of the power command signal.

The burner control system also includes start-up and shut down logic, and several safety controls. During start, there is a purge cycle in which the air valve is full open for a period of time prior to ignition to purge any residual fuel from the system. Then the air valve returns to idle and the fuel pilot valve opens and the spark igniter is energized. When flame is detected, the main fuel valve is opened. After a brief warm-up period, the igniter and pilot valves are turned off. Several conditions such as loss of flame, loss of combustion air or instrument air, or high or low fuel pressure will cause the burner to shut down immediately. In addition, each of the three modulating valves are fitted with position feedback devices. If the feedback does not agree with the control signal, which means the air/fuel ratio could be out of control, the burner is shut down. On shutdown the air valve goes full open again to purge any excess fuel from the burner/steam generator volume.

4.6.4 Lubrication

Lubrication of the steam turbine, the reduction gear, and the water brake dynamometer are provided by a skid packaged system which is common to all Solar gas turbine/generator sets. Control of the lubrication system is built into the Turbotronics system. Oil pressure is maintained by a mechanical pressure regulating valve and temperature is maintained with a thermostatic valve which ports oil to an external cooler, when needed. Tank level sensors, and oil pressure and temperature sensors feed inputs to the electronic controls, which in turn sound alarms or initiate system shutdowns in the event of various parameters being out of range. On the prototype system the main lube pump is AC motor driven, and there is a small back-up DC motor driven pump to provide oil to the bearings during spin down in the event of an AC power failure.

4.6.5 Generator Set

Several control system changes will be required when the generator set is assembled during Phase IV. The turbine speed ramp during warm-up will be accelerated to allow the generator to get to synchronous speed prior to adding any load to the turbine. Synchronization will be automated within the Turbotronics control console, as will the controls for the generator. Once the circuit breakers are closed and the turbine is loaded, the main steam valve will be signaled to wide open to the turbine to close off the bypass leg. Turbine speed will then be controlled by the grid frequency exciting the generator.

For the industrial site, the turbine back pressure valve will be eliminated, since the turbine will exhaust to the process steam header, which is normally held at constant pressure.

If the plant is to be operated based on a steam need following mode, the burner power loop will likely be closed and tied to steam header pressure. When steam need increases, header pressure will fall, causing fuel flow to the burner to increase.

If the plant is installed as part of a combined cycle plant, there will be further changes. The fresh air blower and air and FGR valves may be eliminated. The burner will be replaced with a supplemental burner to boost the gas turbine exhaust temperature, instead of heating ambient air. A smaller fuel control system could be used since about 40% of the heat will now be supplied by the gas turbine.

4.6.6 Instrumentation

Instrumentation used for inputs to the control computer are typically 4 to 20 milliamp transmitters for both pressure and temperature measurements. Temperature elements can be either thermocouples or resistance temperature devices (RTD's). The only RTD's used in the system are the turbine bearing temperature measurements. The transmitters were all supplied by Rosemount. Proximity probes and proximiters used to monitor shaft vibration (position) were supplied by Bently-Nevada.

For development purposes numerous temperatures, pressures, and flows were recorded on a data acquisition system which was totally independent of the control system. For these measurements thermocouples, pressure transducers, turbine flowmeters, and strain gage load cells were used. Data from these sensors were sent through a Datronics data collection system and the data was stored on the Vax computer system at Solar. Critical operating parameters could be displayed on a CRT on the operator's panel.

In accordance with Solar's policy on taking performance test data, all transducers are recalibrated every six months, against standards traceable to the National Bureau of Standards. The entire instrumentation system went through an end-to-end calibration verification check before testing commenced. Typically the flow transducers are accurate to 1% and pressure transducers are accurate to 0.5% of full scale. Thermocouples are typically accurate to 4 degrees F, or 0.75%, which ever is greater.

4.6.7 Start Sequence

At start, the lubrication pump is started and the buffer air is turned on. This allows for prelube of the steam turbine bearings, and the buffer air keeps the oil out of the steam seals (and later keeps steam out of the oil). The combustion air blower, and steam generator cooling air are also signaled on at start, as is the feedwater boost pump. When minimum supply pressure is sensed at the feedwater pressure pump, it is started.

When water and combustion air pressures have risen to acceptable values, the burner start is initiated. This includes a 60 second purge with the air damper wide open, to assure all fuel gas is purged from the furnace and steam generator volumes. Following the purge, the torch igniter is started, followed by the main burner after torch ignition is detected. The main burner remains at low fire condition (1000°F to 1200°F) during the first 30 minutes of the warm-up period.

Feedwater flow to the steam generator is initiated when the burner is started. The flow command is gradually increased over about a 15 minute period of the initial warm-up. Feedwater flow command then reverts to the control algorithm which specifies a flowrate based on the firing command signal. A closed loop portion of the water flow control corrects for steam temperature. During warm-up the setpoint for steam temperature is 800°F.

Steam temperature and pressure gradually increase as the OTSG warms up and the water flowrate increases. The feedwater bypass valve remains open, limiting the pump pressure to about 1500 psi. When steam temperature at the OTSG exit indicates 200 degrees of superheat for two minutes (approximately 12 to 15 minutes after burner ignition), the steam turbine speed control setpoint begins to increase from zero to the 20% speed plateau for warm-up. This signal causes the steam turbine inlet valve to open to start the turbine rotating. The valve modulates as required to follow the speed setpoint ramp. After 15 minutes at 20% speed, the turbine speed setpoint is ramped up to 35%. After another 10 minute warm-up period, the turbine is ready for acceleration to operation speed. Here there is a difference between the component test procedure, and that which will be used for the generator set.

For the dynamometer test, the power level must be increased with the turbine speed to preclude a low-load rumble phenomenon within the dyno. In the generator set, the speed will be ramped to synchronous speed with virtually no load. Once the circuit breaker is closed, load will be added. To limit the rate of change of steam temperature to about 600 degrees per hour, the burner command signal is slowly ramped up. This temperature rise rate was selected to limit the thermal gradient within the turbine casing to conservative industry standards, for the first 1500°F turbine. With planned detailed analysis and addition of flange heaters this rate is expected to be increased by over 70%. The steam temperature setpoint and steam turbine speed setpoint are also ramped up on a predetermined schedule. It takes 70 minutes to ramp the steam temperature set point from 800°F to 1500°F. The total start-up time from cold to full power is about 100 minutes.

Once at operating condition, the steam turbine valve is wide open to the turbine. If less than full power, or full steam to process, is desired, the burner command is reduced, which reduces the amount of steam generated. The feedwater flow control valve is modulated to regulate the steam temperature equal to the setpoint.

4.7 PROTOTYPE SYSTEM TEST RESULTS

Initial testing of the prototype system occurred during the summer of 1994. The steam generator was tested up to full power to measure both thermal and emissions performance. Steam generator performance was close to that predicted, and exhaust emissions were better than the guaranteed values. The steam turbine was tested at speeds to full operating speed, but it did not reach design power. A mechanical fatigue failure was experienced in the blading prior to testing at full steam inlet pressure and temperature. Following the failure, the steam turbine blading was redesigned to eliminate the fatigue problem. The failure, and redesign effort is discussed in Section 4.4.7. During the 1994 tests the total accumulated test time on the steam generator was 60.1 hours and on the steam turbine was 19.3 hours.

With the redesigned steam turbine blading, the system was retested quite successfully in summer of 1995. After initial part load testing to establish start and warm up procedures, and to adjust controls and instrumentation, the system was tested for over 100 hours at design turbine inlet steam conditions of 1500 psia and 1500°F, and design turbine speed of 29,900 rpm. Performance of both the steam generator and steam turbine were quite close to the predicted values, and will be discussed in some detail below. Total test time on the steam generator is now 234 hours and on the current steam turbine is 160 hours. Of these approximately 105 hours have been accumulated at the full design turbine conditions.

Over 2.5 billion fatigue cycles were accumulated on each of the 62 turbine blades at their undamped first flap natural frequency of 7000 Hz. The high number of cycles per blade were due to blades passing the 14 nozzles at 30,000 rpm resulting in 25.2 million cycles per hour. The undamped blade resonant frequency coincides with the 7000 forcing functions per second. The damping system

worked well since no evidence of any cracking or distress was found on any of the 62 blades, proving fatigue stresses were well below the endurance limit. The original blade design had failed in a classic high cycle fatigue mode with millions of cycles accumulated in a few minutes of operation at a resonant frequency of 25000 Hz.

4.7.1 Overall System Performance

Overall system performance as tested was quite close to that predicted by the analytical design calculations. Minor differences were evident at various components, and they will each be discussed below. Figure 23 is a flow diagram of the system showing various state points, and Table 14 lists the properties of the flow at each of the state points, for both the design case and the test data values. The HRSG design model was rerun at the test cell feedwater temperature and combustion air conditions.

The largest discrepancy is the steam flowrate through the turbine nozzles. Design steam flowrate is 56,000 pounds per hour (pph) at rated conditions, and yet the feedwater flowmeter measured only about 54,500 pph. This discrepancy is discussed in some detail in the following sections, on the steam generator and steam turbine performance.

4.7.2 Steam Generator Performance

Design calculations on the steam generator and burner combination predicted that the steam generator would generate 55,940 pph of steam 1550 psig and 1504°F, when the burner was fired with 100.8 million Btu/hr (lower heating value [LHV]) and with an air flow of 28 pounds/second and flue gas recirculation flow of 25% of air flow. Test data at these conditions show a slight disparity between the water/steam side measurements and those on the exhaust gas side. Based on the measured feedwater flowrate, the steam production is about 54,500 pph, or 2.6% lower than predicted. However, based on the measured stack temperatures, and accounting for heat losses to the environment, the steam generator should be producing 55,540 pph, or only 0.7% lower than predicted. The actual measured stack temperature, with feedwater at 87°F, is 192°F, compared to a predicted value of 177°F. This slight shortfall in performance could either be due to optimism in the analytical model, some hot gas leakage around the tube bundle, degradation of the carbon steel fins in the cold end of the unit, or to ineffective surface area caused by condensate filling the gaps between fins. Some fin degradation has occurred due to rust from condensation taking place on the cold tubes in the first two rows of the steam generator. Based on visible evidence of condensate filling the fin gaps, this is the most likely explanation of the steam generator performance discrepancy. In fact, if the first row in the analytical model is made ineffective, a stack temperature of 192°F is calculated, which agrees with the measured value. Significant leakage at the seal at the bottom of the bundle (see Section 4.7.6) was also a contributor to reduced OTSG performance.

Table 15 lists the performance parameters of the steam generator, based on the design model calculations, and on the measured test values. Following the tests, the model was rerun to make the calculations for the measured feedwater temperature, fuel flowrate and combustion air flowrate. Percentage of flue gas recirculation was calculated based on measured temperatures of the fresh air, the flue gas reentering the system, and the resulting windbox temperature. Temperature rise across the blower was accounted for based on the horsepower input to the blower.

Measured steam generator efficiency, based on the lower heating value of the fuel (LHV) was 93.4% compared to 95.7% calculated by the model.

Table 14. State Point Properties for HPSS As-Tested

State Point	Description	Property	Design	Test Data
1	Water tank discharge	Pressure, psia Temperature, F Enthalpy, B/# Flowrate, pph	14.7 70 38 94000	
2	Boost pump discharge	Pressure, psia Temperature, F Enthalpy, B/# Flowrate, pph	240 72 40 94000	
3	Feedwater pump discharge	Pressure, psia Temperature, F Enthalpy, B/# Flowrate, pph	2500 89 62 94000	2488 89 62 94000
4	Steam generator inlet	Pressure, psia Temperature, F Enthalpy, B/# Flowrate, pph	2100 87 (2) 60 56000	2104 87 60 54500
5	Superheater outlet from steam generator	Pressure, psia Temperature, F Enthalpy, B/# Flowrate, pph	1565 1504 1786 56000	1565 1504 1786 54500
6	Steam turbine inlet	Pressure, psia Temperature, F Enthalpy, B/# Flowrate, pph	1500 1500 1784 56000	1500 1500 1784 54500
7	Steam turbine exhaust	Pressure, psia Temperature, F Enthalpy, B/# Flowrate, pph	165 991 1525 55600	165 991 1525 54100
11	Combustion air inlet	Pressure, psia Temperature, F Enthalpy, B/# Flowrate, pph	14.5 72 127 104400	14.5 72 127 100800
12	Flue gas recirculation	Pressure, psia Temperature, F Enthalpy, B/# Flowrate, pph	14.5 180 (2) 164 29600	14.5 180 164 23400 (1)
13	Windbox	Pressure, psia Temperature, F Enthalpy, B/# Flowrate, pph	15 109 145 134000	15 106 145 124200 (1)
14	Fuel	LHV, B/# Flowrate, pph Heat, MMB/h	20475 4930 100.9	20475 4924 100.8
15	Combustion products	Pressure, psia Temperature, F Enthalpy, B/# Flowrate, pph	14.8 2500 885 138900	14.8 2640 (1) 885 129100
16	Stack	Pressure, psia Temperature, F Enthalpy, B/# Flowrate, pph	14.6 182 164 109330	14.6 192 167 105724

(1) Calculated Values

(2) Adjusted to Measured Values

Table 15. OTSG Performance, Calculated Model Versus Test Data

Parameter	Model	Test Data
Air flowrate (lb/sec)	28.0	28.0
Fuel flowrate (MMB/h)	100.8	100.8
Flue gas recirc (%of air)	22.0	22.0
Feedwater temp (F)	87.0	87.0
Steam pressure (psig)	1550	1550
Steam temperature (F)	1504	1504
Steam production (pph)	55940	54500
Stack temperature (F)	177	192
Efficiency, Q_{stm}/Q_f (%)	95.7	93.4

Heat Balance/Heat Losses:

Heat balance calculations for the burner and steam generator do not correlate perfectly with the efficiency measurements. Stack losses are calculated at about 3.5%, based on the measured stack temperature, air mass flowrate, and fuel flowrate. Structure cooling losses are calculated at 0.3%, based on cooling air flowrate and temperature rise. Thermal loss from the flue gas recirculation ducting is calculated at 0.2% based on the measured temperature loss and mass flowrate. Losses through the casing to the environment calculate to about 0.3%, based on approximate surface temperatures and estimated external heat transfer coefficients. These heat losses total to 4.3% of the fuel input, which would correspond to an efficiency of 95.7%, which agrees well with the design model, but is 2.3% higher than the value calculated based on measured steam production. Redundant feedwater and fuel flowmeters were added to check the primary measurements, but this heat balance discrepancy still exists.

Based on the steam turbine performance, discussed below, the steam flowrate calculated from turbine performance is about 54,970 pph, or in between the two values measured by the water-side and the gas-side measurements on the OTSG.

If the steam generator model is modified by reducing fin effectiveness on the cold end rows until the stack temperature agrees with the measured value, then the calculated steam production is 55,540 pph. Hence, by using the steam turbine to calculate steam flow, and by degrading the model performance to obtain the measured stack temperature, the steam flows agree within about 1%. Since a small amount of steam (estimated at 300 pph) leaks past the piston rings in the steam valve, the flowrate through the turbine would be about 55,240 pph, which agrees quite well with the 54,970 calculated based on turbine horsepower and enthalpy drop.

Exhaust Emissions:

A low NO_x burner was installed in the system for stand-alone (non combined-cycle) operation. The manufacturer guaranteed NO_x emissions of less than 30 parts per million, volumetric dry (ppmvd) and 200 ppm CO, when corrected to oxygen content of 3%. Emissions tests were conducted by a third party testing company under the direction of two on-site engineers of the San Diego Air Quality Management District. Tests were conducted at 50% and 100% power in August of 1994. Table 16 summarizes the test results:

Table 16. OTSG Emissions Test Results

	50% Power	100% Power
NOx ppmvd @ 3% O ₂	18.8	25.7
CO ppmvd @ 3% O ₂	23.6	19.2

Circuit to Circuit Temperature Variation:

One of the concerns about the operation of the high temperature once through steam generator was the potential variation in superheater temperature due to different water flowrates and / or different heat additions to the various circuits. The steam generator was designed with variable orifices (hand valves) at the feedwater inlet to each of the 26 parallel steam generating circuits. During the test program, these valves were adjusted as required to even out the superheater temperatures as well as possible. The goal was to maintain the variation to less than $\pm 50^{\circ}\text{F}$ from the nominal temperature. Figure 16 is a plot of the superheater temperatures measured on the tube walls of the 26 circuits. Redundant thermocouples were installed, which is why there are two values plotted for most of the circuits. Measured temperatures were between 1484°F and 1558°F for an average temperature of 1508°F , except for circuit # 1, which had a temperature of 1270°F . It was decided not to reduce the feedwater to this circuit, because in contrast to the low superheater temperature, the temperature of the water entering the evaporator at the furnace end of the steam generator was abnormally high. If the feedwater flow were reduced, it would be possible that the circuit would dry out in the hot end, and lead to potential burnout of that circuit. Circuit 1 is at the edge of the steam generator, and it is hypothesized that partial imbedding of the tubes into the ceramic fiber insulation is limiting the heat addition to the superheater of this edge circuit.

Correlation to the Design Model:

Overall performance of the steam generator was compared to the prediction of the analytical model above. In addition, a comparison of the temperature rise of the water/steam as it progresses through the unit was made by plotting the measured temperature of specific tube rows against the temperatures predicted by the model. Figure 59 is an illustration of the cross section of the steam generator circuit, showing where the measurements were made, and Figure 60 shows these temperatures for circuit 12, which is near the center of the gas flow path. The measured temperatures are at rows numbered 16, 19, 24, 28, 32, 33, 34, 35, and 39. All of the measured temperatures are within a few degrees of those predicted by the model, which indicates the model is quite accurate in the calculations for the preheat, evaporator, and superheater sections of the steam generator.

4.7.3 Steam Turbine Performance

Over 105 hours were accumulated on the steam turbine at or near the design inlet conditions of 1500 psia and 1500°F steam, and an operating speed of 29,900 rpm. Table 17 lists the performance of the steam turbine compared to the predictions from the analog model developed to project turbine performance at various operating conditions.

Brake horsepower predicted by the performance model is 5634 horsepower at the design back pressure of 150 psig. The turbine model calculates a steam flowrate of 56,000 pph passing through the nozzle at these conditions, an aerodynamic efficiency of 69.0%, and a brake (shaft) efficiency of 68.2%.

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Figure 59. Steam Generator Circuit Thermocouple Locations

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Figure 60. HRSG Temperatures

Table 17. Steam Turbine Performance Analytical Model Versus Test Data

Parameter	Design Model	Data Point #1	Data Point #2
Turbine inlet pressure (psia)	1500	1500	1497
Turbine inlet temperature (F)	1500	1501	1500
Turbine exhaust pressure (psia)	165	165	164
Steam flowrate, measured (pph *)	56000	54600	54300
Steam flowrate, calculated (pph **)		54970	54970
Steam exhaust temperature (F)	991	991	992
Turbine speed (%)	100	100.6	99.6
Brake horsepower	5634	5522	5498
Aerodynamic efficiency (%)	69.0	68.9	68.6

* Measured by feedwater flowmeter

** Calculated from brake horsepower and exhaust temp (delta enthalpy)

Test data at near the design inlet and exhaust conditions showed an average brake horsepower of 5510 and an aerodynamic efficiency of 68.7%. Horsepower being low by 2.1% while efficiency is only low by 0.4% indicates that steam flow through the turbine is low. When turbine flowrate is calculated by the brake horsepower and the measured enthalpy (temperature) drop across the turbine, the steam flowrate is calculated to be 54,970 pph. This is 1.8% lower than the 56,000 pph calculated based on the turbine nozzle area and inlet conditions, with an estimated nozzle flow coefficient. Measured feedwater flowrate of approximately 54,500 pph supports the value calculated by the measured enthalpy drop across the turbine. Apparently the flow coefficient calculation in the analytical model is slightly optimistic.

Cooling System Performance:

As described in the design section, the steam turbine incorporates a flow passage for relatively cool exhaust steam to flow back and cool the turbine casing and the forward side of the rotor. This system is driven by the pressure recovery in the exit guide vanes and diffuser causing the turbine exhaust pressure to be higher than the pressure at the turbine rotor inlet. It was crucial for the proper cooling of the high speed rotor that this system function and supply the cool steam correctly. A cross section of the turbine, showing the cooling flow path is shown in Figure 61. Thermocouples were located on the heat shield and in the flow path where the steam exits the ports in the inlet casting. The goal was that the cooling steam pick up no more than 100°F as is flowed from the turbine exhaust back along the casing and in though the inlet casting. This would limit the maximum disk temperature to less than the 1050°F design point. Since exhaust steam flows inward along the back side of the disk, it is estimated that the disk is at a temperature midway between the cooling steam temperature and the exhaust steam temperature.

Table 18 lists the predicted and measured values for the cooling steam. At the design test point (data point #1) the cooling steam temperature was only 79 degrees higher than the turbine exhaust temperature. The cooling system worked very well at the conditions for which it was tested, back pressures of 165 psia and 145 psia. Due to the characteristics of the turbine blading and exit guide vanes, as the turbine back pressure is reduced, the pressure loss in the blading and guide vanes increases, and the delta pressure available to drive the cooling flow decreases. Reduction in cooling flow is expected to control the minimum turbine exhaust pressure to about 135 psia, before the turbine inlet temperature will have to be reduced from the 1500°F design point. Otherwise the turbine rotor may overheat.

Figure 61. Steam Turbine Cooling Flow

Table 18. Turbine Cooling System Performance

Parameter	Design Value	Data Pt. #1	Data Pt. #2
Turbine back pressure (psia)	165	165	145
Turbine exhaust temp. (F)	991	991	973
Heat shield temp. (F)	1090	1100	1123
Cooling steam temp. (F)	1090	1070	1087
Estimated disk temp. (F)	1040	1030	1030

Start-up and Shutdown Transients:

Steam turbine rotation was usually initiated as soon as superheated steam was available at the steam valve for two minutes. This was done by gradually opening the steam turbine inlet valve until rotation started. Turbine speed setpoint was ramped quickly to 20% speed for the initial 15 minute warm-up period. Then the turbine speed is ramped to 35% for the next 10 minutes, followed by a very slow ramp up in speed to 65%. Speed is held at 65% until the boiler power is sufficient to drive the turbine to 90% speed at 3000 plus horsepower. The speed setpoint must rapidly be ramped from 65% to 90% to pass through a critical speed of the pinion shaft of the reduction gear box. This problem is unique to the test rig, and will not be required on the generator set. On the generator set, turbine speed can be ramped to 100% synchronous speed, and then the circuit breaker would be closed and load could be added.

To minimize thermal stresses due to differential temperatures in the steam turbine casing, the warm-up rate of the steam was set at 600°F per hour. Initial steam temperature set point is 800°F, and it takes another 70 minutes to heat up to the design point of 1500°F. This steam temperature ramp begins after the initial 15 minute warm-up period on the turbine. Figure 62 shows steam turbine metal temperatures during a typical warm-up. The time scale is from the time the burner is ignited, and the turbine rotation is initiated 15 minutes later. The upper two curve is the steam temperature leaving the boiler, and entering the turbine. It reaches 1500°F at 90 minutes after start. The lowest curve is the heavy vertical split flange on the turbine which is still approaching its steady state operating temperature of 990°F after 200 minutes. Intermediate curves show the hotter parts of the turbine casing, which reach operating temperature of 1000°F after about 100 minutes. Location of the various thermocouples on the steam turbine are shown in Figure 63.

On shutdown, the minimum thermal stress is achieved by shutting the turbine down from full operating temperature and letting the machine slowly cool, rather than cooling off the steam inlet before shutdown. One difficulty with gradually decreasing power to the turbine before shutdown is that as turbine inlet pressure is reduced, the exhaust temperature rises. This is because the turbine exhaust pressure would normally be to a constant pressure process header, and as inlet pressure is reduced, pressure ratio and energy removed from the steam decrease, which increases exhaust temperature. Since the normal operating temperature of the turbine exhaust and turbine casing are near the maximum allowable values, there is little room for increased temperature. Hence the rapid closing of the turbine valve to shutdown the turbine from full power is actually the most gentle method.

Figure 64 illustrate the temperature transients in the turbine which result from this shutdown from full power. A typical system trip cool-off rate for most of the hardware is 100°F per hour, which is very gentle on the machine. Actually in this instance there was still a small amount of steam flowing through the steam turbine due to the setup on the steam valve which prevented it from closing fully. Consequently, in the final system, the cool-down rate may be even more gradual.

Cooling steam temperature and the temperature at the convolution actually increase for the first half hour after shutdown. These parts are close to the inlet casting, which is cooling from 1500°F, so they absorb heat initially during the shutdown.

Figure 62. Steam Turbine Warm-Up

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Figure 63. Turbine Thermocouple Locations

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Figure 64. Turbine Temperature Decay on Trip From 1500°F

Part Load Operation:

Operation of the steam turbine at part load differs substantially between the test cell, where a water brake dynamometer and fixed resistance turbine exhaust are used, and the generator set in cogen service, where the load is constant speed, and the turbine exhaust pressure is constant. In the test cell, load and speed are varied together due to the nature of the water brake dynamometer, which has a relatively narrow power band which increases with speed. It was not feasible to perform part load, full speed tests to simulate the generator set operation. Also because the turbine exhaust system vented to atmosphere through a fixed resistance, it was not feasible to run reduced power tests at the design back pressure. Consequently, no effort was spent to obtain part load data in the test cell.

In the generator set application, where the turbine operates at one speed and exhausts to a constant pressure header, the steam inlet temperature will need to be reduced as the steam inlet pressure is reduced to decrease output power. This will be necessary because the heat removed from the is a function of the expansion (pressure) ratio across the turbine. For a fixed exhaust pressure, as inlet pressure is reduced less energy is taken out of the steam, and the temperature drop in the steam is reduced. At full power, the exhaust temperature of the steam and the turbine casing are operating at near their maximum design values of 1000°F (for a steam inlet temperature of 1500°F). To prevent the turbine exhaust and casing from going over temperature at part load, the inlet temperature will be reduced along with the pressure. This relationship based on the performance model calculations is shown in Figure 65. Testing of this relationship will be done on the generator set at the host site during Phase IV.

Since generator load can be shed rapidly, which would cause the speed control to close the turbine valve, reducing inlet pressure, there will be a short term temperature overshoot in the turbine exhaust temperature until the inlet temperature can be reduced appropriately. It is uncertain at this time how rapidly the temperature can be safely reduced, without excessive thermal transients and stresses in the steam generator header, piping, and steam valve. Consequently there may well be a limit to the magnitude of load reduction step which can be absorbed safely without taking the turbine completely off line. Further analysis and testing of this load shed issue will be done during Phase IV.

Rotor Dynamics:

Projected rotor dynamics performance of the high speed steam turbine was calculated. The first two critical speeds are rigid body modes, the first being translational and the second a conical mode. A third critical speed is the first bending mode of the shaft. The steam turbine was to operate between the second and third critical speeds as shown in Figure 47. In all cases the shaft criticals were calculated to be well damped, and would not limit turbine operating speeds in any way. During turbine assembly, the diametral clearance of the journal bearings was adjusted to 6 mils (0.006 inches), and the rotor was balanced to less than .007 oz inches of unbalance.

Proximity probes were installed at each of the two journal bearings to monitor shaft position (vibration) for use in the rotor dynamics analysis. A run-down test was conducted to assess shaft vibration at all speeds, with minimal load on the shaft. Results of this test are shown in Figures 66 through 70. In each figure, the upper frame shows the vibration amplitude (lower curve) and phase angle (upper curve) versus rotational speed, or frequency. The lower frame is a polar plot of the same information. In order, the figures show the forward (inboard) bearing vertical probe, the forward bearing horizontal probe, the aft bearing vertical, the aft bearing horizontal, and the pinion bearing on the gear box input shaft.

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Figure 65. Part-Load Steam Temperature and Turbine Power Versus Steam Supply Pressure

The forward bearing carries most of the rotor load, 55 of the 77 pounds, and shows critical speeds at approximately 7500 rpm and 18,600 to 20,400 rpm, corresponding to the two amplitude peaks at 125 HZ and 310 to 340 HZ on the figures. These are somewhat different than those shown in the design calculations as illustrated in Figure 47. The difference in the frequency of the second critical speed is likely the assumptions made for the stiffness of the bearings for the calculations. During the tests, the oil temperature was substantially cooler than that assumed, 120°F vs. 145°F. In addition, the bearing support in the actual machine may well be less stiff than that assumed for the calculations. In any event the critical speeds are well damped and present no operational issues. The amplification factor calculated for the first critical is 1.1, indicating it is extremely well damped. Amplification factor on the second critical is about 4.5, which is also more than adequate damping. Vibration amplitudes are about 0.5 and 1.0 to 1.2 mils at these critical speeds, which are only slightly greater than the full load running amplitude of 0.3 to 0.8 mils.

The aft bearing is very lightly loaded, and does not even respond to the first critical speed. It does show an amplitude of about 0.7 mils at and above the second critical speed. It is very likely that the activity seen in both bearings at speeds above the second critical speed is vibration fed back through the shaft from the reduction gear box pinion shaft. Figure 70 shows the vibration at the pinion bearing with a critical speed at about 21,700 rpm and an amplitude of 1.8 mils. This critical speed vibration on the pinion is quite pronounced, and amplitudes can exceed 4 mils on a gradual speed run-up. Note: the reduction gear used for the component testing will not be used with the future generator set.

Under full load operating conditions, shaft vibrations at both journal bearings is the order of 0.5 to 0.7 mils peak-to-peak. This magnitude of vibration is very acceptable, and meets the conservative allowable criteria of the API (American Petroleum Institute) Standards for a 30,000 rpm turbine.

Bearing Clearance and Oil Temp Rise:

The turbine rotor is supported by two tilting-pad journal bearings. Each bearing has four pads and the load is centered between pads. Diametral clearance measured on assembly was .006 inches, with a shaft diameter of 2.5 inches. The bearing supplier estimated a power consumption of 9.5 horsepower per bearing for oil supplied at 135°F, and a turbine speed of 29,900 rpm.

Figure 66. Forward Bearing Vertical Dynamics

Figure 67. Forward Bearing Horizontal Dynamics

Figure 68. Aft Bearing Vertical Dynamics

Figure 69. Aft Bearing Horizontal Dynamics

Figure 70. Gearbox Pinion Dynamics

The thrust bearing is a 12-pad tilting-pad bearing designed to carry up to 3000 pounds of axial thrust. Power consumption was estimated at 49 horsepower at rated conditions. Including the journal bearings, the total estimated power consumption of the bearings was 68 horsepower.

During the actual testing, oil temperature was somewhat lower than the design condition, 120°F instead of 135°F. This was done to stiffen the journal bearings slightly, and decrease shaft vibrations. At full power and full speed operation, the thermal horsepower absorbed by the oil passing through the turbine calculated to about 85 horsepower. This is very reasonable, since the power absorbed by the bearings would likely increase about 5% due to the cooler oil temperature, and some heat is transferred from the hot turbine casing to the cooler bearing housing, and then removed by the oil.

Labyrinth Seal Performance:

A labyrinth seal on the turbine shaft limits the steam leakage from the 150 psig exhaust cavity of the turbine. Analytical calculations predicted steam leakage of about 380 pounds per hour, or 0.7% of the steam passing through the turbine.

Based on the steam/air temperature measured at the casing vent, and the calculated quantity of buffer air mixing with the steam leakage, the calculated value of steam leakage is between 200 and 300 pph. Thus the labyrinth seal performs better than originally estimated analytically.

4.7.4 Control System Evaluation

The control system for the Phase III prototype tests evolved from an earlier steam turbine cogeneration system control developed at Solar. A control console, programmable logic controller (PLC), and input/output devices identical to those used for standard gas turbine generator sets was used. Complete description of the system was given in Section 4.6.

During the testing the control system was adjusted as required to provide the desired start-up sequence, warm-up ramps, and steady state set points for the various controlled parameters. Five analog variables are controlled during steady state operation: steam turbine speed, steam temperature (feedwater flowrate), feedwater supply pressure, desuperheater (bypass system) temperature, and water tank level. Turbine speed, and steam temperature are the two primary control loops of concern. The other three are quite straightforward and of secondary importance.

Turbine Speed Control:

Turbine speed is controlled by the main steam valve, which ports steam either to the turbine, or to the turbine bypass piping. When speed is higher than the speed set point, the valve moves to decrease steam flow to the turbine. Speed is sensed by a magnetic pick-up reading the teeth on a 24-tooth gear rotating at shaft speed. 11,960 Hz represents a 100% speed signal. The valve is controlled by a dedicated 2000 psi hydraulic system driving a large cylinder to move the valve. Due to the potential damage from a turbine overspeed condition, the hydraulic system is sized to move the valve quite rapidly. In addition to the servo providing the analog valve position control loop, there is an overspeed trip function for which a solenoid valve dumps hydraulic pressure, closing the valve very rapidly.

During the speed ramp-up the turbine is on speed control, following a linear speed set point vs. time ramp. Speed regulation during this period was always excellent, with the actual turbine speed

staying within about 0.3% of the set point. The valve responded quickly when the dynamometer load was gradually increased.

For the full power performance testing it was necessary to hold the steam valve wide open (to the turbine) to minimize the steam leakage to the turbine bypass leg. To accomplish this, the speed set point was typically set at near 101%, and the dynamometer was loaded to hold turbine speed between 99 and 100% speed. Due to fluctuations in the dynamometer water pressure (flow) turbine speed would vary a percent or two under these conditions.

The overspeed trip function was tested several times, and it always brought the turbine speed down very rapidly. During the initial check-out of the steam valve it was verified that the valve would be effectively closed within 150 milliseconds of the electrical signal (drop out) for the overspeed trip (Figure 52).

With the exception of the steam valve initially not having sufficient mechanical clearance to tolerate the thermal gradient experienced during operation, the steam turbine control valve performed very well during the turbine testing.

Steam Temperature Regulation:

Steam temperature exiting the steam generator is controlled by modulating the feedwater flowrate to the unit. When the measured steam temperature does not equal the set point value, the error signal is used to bias the set point on the feedwater flowrate. A water flow measurement taken across an orifice plate is compared to the set point value, and the flow error signal is used in turn to vary the position of the feedwater flow control valve, until the error signal returns to zero. Feedwater flow setpoint is actually a sum of two signals, one being an open loop signal which proportions the basic flow requirement to the heat being added in the combustor. A second term is based on the feedback of the temperature error signal. The purpose of the open loop portion is to anticipate the need for a feedwater flow change when the burner power is changed rapidly, and the purpose of the feedback loop is to provide accurate temperature control during steady state operation.

During the steam generator warm up, the steam temperature setpoint is gradually ramped up as a function of the power level, from 800°F at idle to 1504°F at full power. Figure 71 shows the relationship between the steam temperature setpoint and the measured steam temperature on a typical test. In general the actual steam temperature lags the set point on the way up, primarily due to the thermal inertia of the mass of tubing and fins in the steam generator. Once the steam temperature set point is increased to the full power point of 1504°F, the steam temperature variation was limited to about $\pm 4^\circ\text{F}$. This is surprisingly close temperature control considering the long flow path between the water inlet and the steam outlet.

Water Pump Pressure Control:

To economize for the laboratory test, a positive displacement high pressure feedwater pump was purchased. Since this pump was also a constant speed motor-driven pump, the water flowrate through the pump is constant. Consequently a water bypass system was required to discharge back to the run tank the water not needed by the steam generator. A modulating valve with a simple pressure control loop was installed, to maintain the pump discharge pressure at the desired value.

Figure 71. OTSG Temperature Setpoint versus Superheater Outlet Temperature

Since both steam pressure and steam generator pressure drop are direct functions of steam production rate, or power level, the feedwater supply pressure required varies greatly with power level. To save the pump horsepower at low operating power, and to reduce the pressure drop across the feedwater control valve (to prevent cavitation damage) the set point pressure for pump discharge pressure was varied with system power level. It varied from 1200 psig at idle, to 2500 psig at full power, Figure 55 (Page 60).

Transient Operation:

To minimize thermal stresses due to temperature gradients in the heavy walled sections of the steam generator, steam valve, and steam turbine, it is desirable to make load transients gradually. For initial warm-up, a steam temperature rise rate of 600°F/hour was conservatively selected based on traditional steam turbine practice to provide satisfactory differential temperature in the steam turbine outer case cast steel vertical flange, which is the thickest section of the turbine. The high temperature superalloy inner case can be cycled much faster and a heater can be added to the flange to decrease warm up times if required for a given site. This limitation controls the system warm-up time, which takes approximately 100 minutes from burner ignition. Figure 71 shows the steam temperature and pressure as a function of time for a typical start. Figure 72 shows temperatures at various tube rows within the OTSG during a typical start with only minor excursions of the screen outlet temperature (Scm out) at low pressures and low firing temperatures.

Shutdown transients can be quite gentle by shutting down the feedwater flow at the same time the burner is shut off. The hot end of the steam generator has enough sensible heat to vaporize all the water remaining in the tubing. Steam temperature and pressure decay rates are shown in Figure 17 (page 13) for a typical shutdown from full power, and Figure 73 shows the temperature transient for various sections of the OTSG.

By closing the steam valve to the turbine, the turbine cools off very slowly, over many hours. Figure 74 shows the decay rates of the various turbine temperatures at shutdown.

Figure 72. OTSG Internal Temperatures Compared with Superheater Outlet (T_{stm}) during Start Transient (Scrn = screen temp, Sup1 = First Superheater Out Temp.)

Figure 73. OTSG Temp. Transient on trip from 1500°F and 1500 psig.

Figure 74. Typical Turbine Temp. Transient Decay from 1500°F and 1500 psig.

Under certain partial load shed events on a generator set, it may be desirable to keep the steam turbine operating and generating electricity. The potential problem is that if the turbine inlet pressure is decreases rapidly without reducing the temperature accordingly, the exhaust end of the turbine can overheat due to insufficient energy being removed from the incoming steam. To prevent this occurrence, the steam temperature leaving the steam generator must be reduced quickly. A test was run to investigate the steam temperature decay rate when the fuel is rapidly reduced, while the feedwater flowrate is maintained. Figure 75 shows the decay rate in the superheater, turbine inlet, turbine exhaust, and turbine cooling steam temperatures with time. Fuel flow had been reduced from 100% to 88% at time zero. In this case the critical cooling steam temperature only rose a few degrees, and then began to decay with the exhaust temperature. This test could not precisely simulate a transient in an industrial situation because the turbine exhaust pressure decayed as the temperature decayed since the exhaust resistance in the test rig is constant. After 15 minutes, the

pressure had decayed approximately 10%, or 15 psig. Consequently, in a site situation, the cooling steam temperature will probably not decay as much due to the exhaust pressure being held constant by the process header, and the consequent reduction in pressure ratio and energy extraction. None the less, the test was encouraging in that some off-load transients can be tolerated without excessive cooling steam temperatures. In a site application, a rapid off-load transient on the generator set will require the turbine bypass valve to open (partially closing the turbine throttle valve) to lower turbine mass flow to that required by the new load level. The response time of the OTSG is the order of 15 or 20 minutes to settle at a new operating level.

Figure 75. Partial Load Shed Turbine Cooling Steam Temperature Transient

4.7.5 Noise Measurements of Test System

Noise measurements were made of the HPSS test loop (Figure 37) and were as expected, dominated by the steam vent discharge into the ambient air. Figure 76 shows noise measured at different locations. The steam turbine exhaust vent stack at location 6, 16 feet away from the vent was 108 dBA. At another location further from the vent discharge, such as location 4 about 3 feet from the steam turbine, were more representative of the specific component. At 103.5 dBA at the turbine and 102 dBA at the steam generator locations the noise level of these components can be related to unenclosed industrial gas turbines. Unenclosed 1 MW gas turbines are typically measured at about 99 dBA 3 feet from the turbine. A 5 MW industrial gas turbine would be typically at 101 dBA and a 10 MW machine at 3 feet would be 106 MW.

By estimating the contribution of the steam vent, it is estimated the unenclosed HPSS system would at a site fall between the 99 and 101 dBA levels. This estimate places the noise emissions at a level between a 1 MW and a 5 MW industrial gas turbine.

4.7.6 Post Test Steam Generator Inspection

All accessible areas and components of the combustion system and steam generator were inspected after more than 120 hours of operation at rated conditions. Access was gained by means of bolted plates into the firebox and exhaust stack.

Figure 76. Sound Pressure Levels (dBA) at Locations "1" through "6" around the HPSS.

OTSG Inspection: The screen tubes and the final superheater appeared to be in excellent condition with no significant evidence of distortion, buckling, tube overtemperature, fin overtemperature (on final superheater) or insulation caused fouling. Figure 77 shows a general overall view of the fireside screen tubes after the test. It can be seen all tubes except circuit eight, highlighted in Figure 78, were in their original position. Circuit 8 was moved about one inch from the original position with a slight bow in its fifteen foot length as shown in Figure 79. Since it is only one inch in diameter and 15 feet long and thus very flexible, a small amount of binding in the crossover tube at the bottom can cause this out of position condition. This binding can be realigned during preparation for shipping of the OTSG to the site and is not considered a significant problem.

Examination of the insulation system in the area where flame impingement could be anticipated showed no insulation erosion due to velocity or temperature. However, about 20% of the sprayed on thin hard coating shed from the surface of the ceramic fiber insulation. It is assumed that the approximately 45 on and off cycles caused the coating to shed.

The coating was an added protection against high velocity erosion. However at the very low average flow velocities used (less than 100 ft/sec) the basic Thermal Ceramics Incorporated uncoated fiber showed no apparent erosion either on the coated or uncoated sections. No evidence of fiber fouling on the tubes or fins was seen in either the high temperature or cold wet end of the steam generator.

Combustor burner register swirlers were in excellent condition with no evidence of overheating or distortion (Figure 80). The burner head wall used a rammed plastic refractory that was in good condition except for the normal shrinkage cracks seen in Figure 80.

Figure 77. Post Test Condition of OTSG Fire Side

Figure 78. Location of Circuit Number 8 Highlighted by Flashlight Beam

The cold end of the OTSG showed no evidence of distortion or fouling. All circuits were in their as built location and alignment (even though they are designed to take most of the relative horizontal motion of the tube bundle. The unit was designed to operate above the dew point in cogeneration service (feedwater inlet temperature of 160°F). During the test the unit always operated below the dew point in the last two rows with condensate flooding the last row at 70°F feedwater inlet temperatures. As a result considerable rust developed on the last two rows of carbon steel fins and significant rust deposits were observed. These were soft and flocculent and would blow off easily and are not a problem if the unit is operated above the dew point for the field installation with 160°F feedwater. Fins removed from the unit showed a pitting of about 0.004 inch. At this level of pitting no significant loss of unit performance would be observed in the site installation.

Figure 79. Circuit Number 8 About 1.0 Inch Out of Position

Figure 80. Combustor Swirler After Test

Examination of the OTSG's perimeter sealing systems showed no problems except at the bottom of the bundle. This area of the unit is the most difficult to seal. Since the bundle is top supported the total vertical expansion of the tubes of up to 1.8 inches must be allowed by the bottom sealing system. Three flexible metal seals shown in Figure 34 (Page 41) were used to control leakage. Only two of the seals could be viewed for inspection. Both seals viewed showed significant distortion and allowed hot gases to by pass the heat transfer surfaces at the bottom of the unit. A thermocouple installed at this location showed 1000°F streaks at the bottom when the stack gas temperature averaged 190°F. Although the mass flow rate of leakage was relatively low and had little effect on the overall performance (see Section 4.7.2), this part of the sealing system will be modified and fixed before field installation. Use of a superalloy of about half the thickness with greater high temperature strength is considered a better choice than the thick 316 series stainless steel seals that appear to have gotten hotter than estimated. Gases leaking in this area are cooled by the crossover tubes and were estimated to be reduced from 2500°F to less than 1000°F. Evidence indicates parts of the sheetmetal vee shaped seals were above 1500°F.

Inspection of the fiber insulation system showed no erosion damage to the insulation but some gaps opening up as is generally anticipated in the first 50 hours. The fiber system shrinks 5 to 7% in the first 50 hours. A 10% compression is used to install the fiber blocks. If a uniform compression is not obtained gaps between blocks can open up as shown in Figure 81. About 20% of the joints had gaps. Normally these gaps do not cause hot spots since the gap is only near the surface. If a gap opens sufficiently to cause a hot spot it is usually fixed by inserting and packing it with 1.0 inch thick blanket fiber insulation. No hot spots on the outer carbon steel casing of the approximately 1500 square feet of casing were observed except at the top transition between the OTSG case and combustor. Since the 500 hour test module combustor case was used to save money a special transition piece was used to match the two units. In place of the standard production 12 inch by 12 inch by 12 inch blocks special shapes were used. These special blocks had 18 and 24 inch corner shapes with standard supports. The extra weight (and possible rain damage during erection and installation) caused these larger insulation shaped blocks to sag more and caused local hot spots at the top of the transition. These will be reinforced with additional support and gaps packed with insulation prior to field operations.

Figure 81. Ceramic Insulation Showing Gap Between Blocks in Post Test Inspection

4.7.7 Post Test Turbine Inspection

Following the turbine testing during which 160 total rotating hours were accumulated, with 105 hours at full operating temperature of 1500°F and full speed of 29,900 rpm, the steam turbine was disassembled for inspection.

Thrust bearing surfaces looked excellent, with only one small particle imbedded into one of the loaded side pads. The thrust runner had a very small shiny circle at the radius of the imbedded particle. All parts are usable as is, except that the failed temperature sensor in the thrust bearing is being replaced.

A very slight radial rub occurred between the blade shrouds and the honeycomb tip seal. An approximately .010 inch deep groove was created in the honeycomb. This is considered to be a nearly perfect fit, although the design intent was for no contact. Further dimensional checks will be made to assess whether the rotor increased in diameter more than anticipated due to the combination of operating temperature and rotational speed.

A more significant axial rub occurred between the blade retention hooks on the aft side of the blades and disk, and the inside shroud on the exit guide vane assembly. In this case, there was slight initial contact with the high spot on the shroud and it began machining the blades and disk. Material built up on the shroud to a thickness of .010 to .015 inches, increasing the interference with the rotating parts and machining a groove. Groove depth ended up at about .025 inches. Unfortunately the groove location is such that it substantially reduced the section thickness of the blade retention hooks, as shown in Figure 82. The oxide layer on the groove indicates that the rub occurred early, and that 100+ hours of full temperature, full speed testing was done with the damaged hooks. Analysis indicates the stresses are not excessive even in the thin section of the worn hooks. Nevertheless, it was decided to redesign the blade retention system so the hooks can be machined off. It is not good practice to have such a thin section operating at the high velocity and acceleration of the blade platform. The new retention system will likely use axial rivets in the attachments to preclude axial motion.

The turbine blades and disk were fluorescent particle inspected for any indications of cracks in the critical areas of the blade attachment fir trees. No indications of any cracks were found.

Figure 82. Blade Retention Hooks

Inspection of the blades and shrouds verified the the shroud and damping system designed for the inserted blades worked well. There is some indications on the fir tree contact surfaces that routine fretting occurred, as is normal on the gas turbine blade attachments. There are also "hammering" contact indications on the faces of the blade shrouds and platforms indicating the blades were contacting adjacent blades periodically. It was concluded that the relative blade motion occurred during the slow accelerations, not at rated speed. There is no concern of excessive wear issues in this area.

Visual inspection of the coloring of the turbine disk, made of Inconel 718, indicates that the hub of the disk operated at a temperature of 900 to 1000°F, as expected. Out toward the blade attachment area at the rim of the disk, however, the temperature appears to have been as high as 1150°F. The design intent was that the cooling steam would keep the attachment below 1100°F.

Initially it was calculated that the steam flow through the rotor blading would be supersonic, resulting in low exhaust temperatures, and low relative total temperatures in the blading. Following the failure of the rotor with 91 integral blades, the rotor was redesigned with 62 inserted blades to resolve the mechanical fatigue issue. At this time, the leading edge thickness of the blades was increased from 0.012 inches to 0.025 inches to decrease the stress concentration locally. The decrease in calculated turbine performance resulted in an increase in the design turbine exhaust temperature from 950°F to 991°F, which decreased the effectiveness of the cooling steam and increased the relative total temperature in the blading. As a result, the blade attachment temperature has increased from an estimated 1075°F to approximately 1100 to 1140°F.

Attachment stress values, and actual disk material properties (creep strength) were reviewed to assess the effects of the higher attachmt temperature. The life versus temperature curves in Figure 83 illustrate that in this range, a 25° temperature change can have a large influence on service (creep) life. Since there is still some uncertainty in attachment temperature (as well as material properties) it may be prudent to reduce the operating temperature for extended service life of the turbine. As the testing demonstrated, the turbine can operate satisfactorily at these elevated temperatures, but the service life could be shortened based on creep strength reduction, if the attachment temperature reaches 1140°F. At an attachment tempeprature of 1140°F, the service life could be about 40,000 hours. If the inlet steam tempeprature is reduced by 25°, the service life would increase to over 100,000 hours. This issue will be evaluated following the 1000-hour field demonstration test at 1500°F inlet steam temperature.

Figure 83. Life vs. Attach Temperature, Estimated Disk Material Properties

5.0

RECOMMENDATIONS & CONCLUSIONS

5.1 RECOMMENDATIONS

The highly successful prototype development tests warrant further operational evaluation in a long duration industrial or commercial site for over 100,000 hours of operation. Figure 4 shows a plan view of the HPSS with a combined cycle arrangement. The maximum economic return would be at a site that can use about 70,000 pounds per hour of saturated process steam or 55000 pounds per hour of 950°F superheated steam at pressures less than 200 psig for a process or a condensing steam turbine. Electrical output would be in a range of 8 MW to 14 MW depending upon the gas turbine and pressure level of the process steam. Fuel utilization efficiencies would be in the range of 80% to 90% when using large amounts of process steam. The site selected would obtain the equipment developed and described in this report: OTSG, control valve and steam turbine from the DOE to assist the host site in reducing the costs of the installation. In addition to the site design, construction and installation costs the host site would have to pay for the standard electric generator and other equipment to modify the prototype HPSS (Figure 1) into a commercial electric generator set. The site would also have to permit the DOE and contractors to monitor the first 1000 hours of operation and demonstrate to US industry.

5.2 CONCLUSIONS

The work performed demonstrates that higher steam conditions are technically feasible at temperatures to 1500°F at pressures of 1540 psig. The developments, materials and processes appear scalable to 50 MW (Reference 5) with a two stage, 12 inch diameter 16,000 rpm turbine operating at lower temperature but much higher pressures. Depending upon size of the part, pressures of up to 6000 psig with temperatures reduced to 1300°F are feasible with the tubes developed in this program and should be considered. Repowering with 3500 to 4000 psig systems at temperatures in the range of 1300 to 1400°F is well within the technical envelope for a number of old steam power plants and old combined cycles.

The economic tradeoff for a wide range of plant sizes and applications of higher steam conditions has not been evaluated in detail. Successful demonstration of the HPSS should be a starting point to optimize the performance versus economic payback of higher steam conditions based on materials, concepts, and processes of the HPSS development. Future application studies to Advanced Turbine Systems (ATS) and new power plants is recommended.

6.0

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