Proceedings of ASME Turbo Expo 2011
GT2011
June 6-10, 2011, Vancouver, British Columbia, Canada

GT2011-46487

CONCENTRATED SOLAR THERMAL DOWNSTREAM OF THE SOLAR FIELD – DESIGN AND OPTIMIZATION OF THE ASSOCIATED POWER GENERATION CYCLE

Justin Zachary, Natasha Jones, and Aslan Golant
Bechtel Power Corporation
5275 Westview Drive
Frederick, Maryland, USA 21703-8306

ABSTRACT

While major design efforts are dedicated to the development and improvement of solar energy collection technologies, the downstream power generation cycle is often considered a straightforward exercise. The diverse nature of the heat sources and their cyclic behavior make the design of the turbo-machinery and associated balance-of-plant equipment for solar plants quite different from the design for use in conventional fired power plants. The high capital cost of these renewable energy facilities and the limited hours of operation are powerful drivers to increase equipment efficiency and reduce the startup time.

This paper reviews the state of the art regarding hardware selection and design considerations for tower, trough, and Fresnel solar thermal technologies from an engineering, procurement, and construction (EPC) contractor's perspective. It also describes the benefits and limitations of each method and the impact of flow and temperature on cycle efficiency. In particular, it addresses the turbine design challenges for repeated fast startups and plant size optimization. Special emphasis is given to heat sink design in consideration of water scarcity.

In conclusion, the paper provides recommendations for achieving a balance between the economics of generation and cost of equipment and reliability for the downstream power generation system.

1 INTRODUCTION

In the design and development of concentrated solar power (CSP) plants, solar field equipment suppliers devote substantial effort to improvement of the solar energy conversion to steam. Use of sophisticated means for tracking and controls, better optics, and tube coatings are just few of the elements employed by CSP technology developers to achieve their goals. However, optimizing heat input to the system is only half of the effort. The remaining portion, processing the heat into electric power using well-known conventional thermodynamic cycles such as Rankine, Brayton, or Stirling, is the subject of this paper. Special emphasis is given to the main components of a steam cycle: turbine and heat sink.

The nature of the solar heat source and its cyclic behavior make the design of turbo-machinery power generation equipment quite different from that of steam turbines used in conventional power plants. The high capital cost of renewable facilities and the limited hours of operation are powerful drivers to increase turbo-machinery efficiency. Proven technology will be a key advantage in the current project financing situation.

For high-temperature applications such as the power tower or in the medium-temperature solar troughs collector field, the paper will address the unique requirements for performance, integration, and fast startup of the turbines, including the impact of various thermal storage options. Since most of the concentrated thermal solar applications are in arid regions, the paper discusses heat sink selection (air-cooled condenser [ACC], hybrid, Heller tower, etc.) and how it impacts the plant design and performance.

The paper reviews the state of the art of the hardware designs for each application from an engineering, procurement, and construction (EPC) contractor's perspective.

2 EXISTING SOLAR THERMAL TECHNOLOGY CONCEPTS AND IMPACT ON STEAM PRODUCTION

CSP systems require several components to produce electricity: (1) concentrator, (2) receiver, (3) storage or transportation system, and (4) power conversion device. There are different ways of converting the solar energy into electricity. In addition to the Rankine cycle, which is discussed in this paper, there are other conversion systems based on the Stirling cycle and Brayton cycle. For systems based on the Rankine cycle, several types of technologies are available:

- Trough
- Linear Fresnel
- Solar tower

CSP technology type determines the options for interface with a conventional fossil-fired plant. Table 1 summarizes the types of technology and their thermal output.

TABLE 1. SUMMARY OF CONCENTRATED SOLAR TECHNOLOGIES

Technology Type	Working Fluid	Maximum Temperature (°C)
Tower – direct steam	Steam	550
Tower – molten salt	Salt mixture	575
Trough	Synthetic oil HTF	395
Linear Fresnel	Steam	270 (or higher)

3 CYCLE CONFIGURATION

3.1 Plant Size

Defining the plant size is not only related to the CSP technology but also to availability of appropriate steam turbine and heat sink. Sometimes the size is dictated by the permitting and local legislation. For example, in Spain, the maximum size is set at 50 MW. In the United States, such legal limitations do not exist. The type of technology has also an impact on the size of the plant. There are plans for larger than 200 MW plants for either trough or tower configurations. Intuitively, a larger plant should achieve a lower capital cost. In sizing the plant, one must acknowledge, firstly, the limitations imposed by the solar field. For trough systems, the question is how many loops can be practically connected to a steam generator? For the tower system, these limitations are related to the boiler size, tower height, and distance of the mirrors from the tower. In the final account, it is expected that optimization studies will be conducted to determine the most suitable plant size based on available land for the solar field, standardization to reduce the capital cost, and increased availability. Only a detailed analysis for each specific location can provide a definite answer.

3.2 Number of Feedwater Heaters

An increase in the number of feedwater heaters improves plant efficiency but increases the cost. Figure 1 presents a typical configuration of a plant including an auxiliary boiler. Figure 2 shows how reducing the number of feedwater heaters affects steam cycle efficiency. If the cycle has only four heaters instead of seven, the efficiency is reduced by 0.8%. Depending on the solar multiplier and the economics of the plant, shutting down or throttling heaters could have a positive impact on the plant output. It is imperative to design the steam turbine in a way that allows it to receive the additional steam available when the feedwater heaters are out of service.

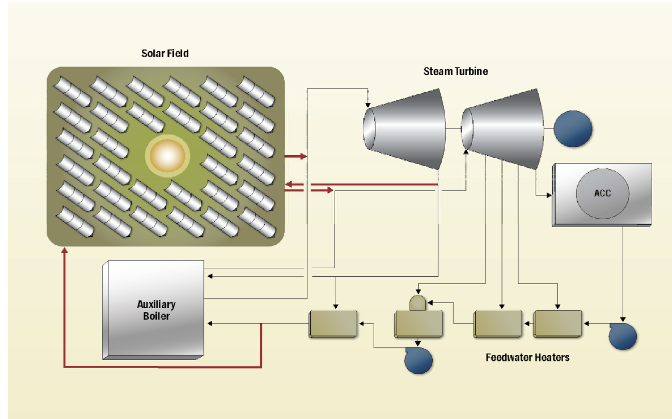

FIGURE 1. CYCLE CONFIGURATION

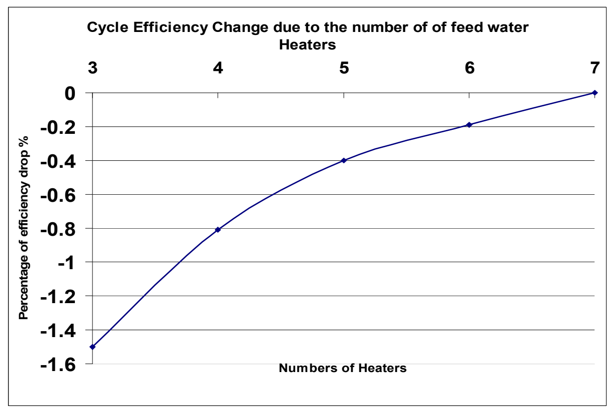

FIGURE 2. NUMBER OF FEEDWATER HEATERS AND RELATIVE CYCLE EFFICIENCY

3.3 Reheat Options

The decision to use a reheat cycle or non-reheat cycle is a function of low pressure (LP) turbine exhaust moisture levels and the desired turbine inlet throttle conditions that provide the optimum ratio of plant efficiency versus capital investment. The renewable technology that is used will provide restrictions on the throttle and reheat temperatures. For example, a power tower plant using molten salt can have main steam temperatures of around 1,000 °F while a parabolic trough plant using heat transfer oil is limited to around 700 °F. With throttle temperatures relatively fixed, the function is reduced to two options: (1) a plant designed with a higher throttle pressure that provides a higher efficiency, but requires a reheat system to lower exhaust moisture levels, or (2) a plant designed with lower throttle pressures that requires less initial capital investment, but suffers from lower efficiency. This paper considers two options using the two throttle temperatures stated above. While such discussion is also pertinent for fossil-fuel-fired plants, the increased efficiency means lower capital cost only for solar applications.

Condensing steam turbines commonly operate with saturated steam exhaust conditions. However, if there is too much moisture, the turbine blades will suffer from erosion, causing decreased efficiency and eventually leading to an earlier-than-normal overhaul. Based on information from several original equipment manufacturers, it is common to see reheat turbines designed to safely operate with 8% exhaust moisture content, while non-reheat turbines are allowed to go to 11% moisture. This analysis uses these values as design constraints.

For the range of steam turbines of interest to designers of renewable resource power plants, isentropic efficiencies can vary from 80% to 90%. Temperature versus entropy can be shown diagrammatically to illustrate the performance impacts via available heat energy for the use of reheat versus non-reheat cycles. It is at this end that the authors arbitrarily chose 85% isentropic efficiency to form a basis for our comparisons. Furthermore, the LP exhaust pressure will be kept constant to aid in comparison.

Figure 3 shows a typical reheat cycle versus a non-reheat cycle designed at the same main steam temperature and pressure combination. The reheat option has a moisture content of 8% while the non-reheat section's moisture content is 14.6%. The higher moisture content in the LP section should be avoided. Figure 4 shows a comparison between a reheat cycle and a non-reheat cycle designed with a throttle temperature of 1,000 ˚F and with a lower throttle pressure in the non-reheat case to maintain an acceptable moisture level.

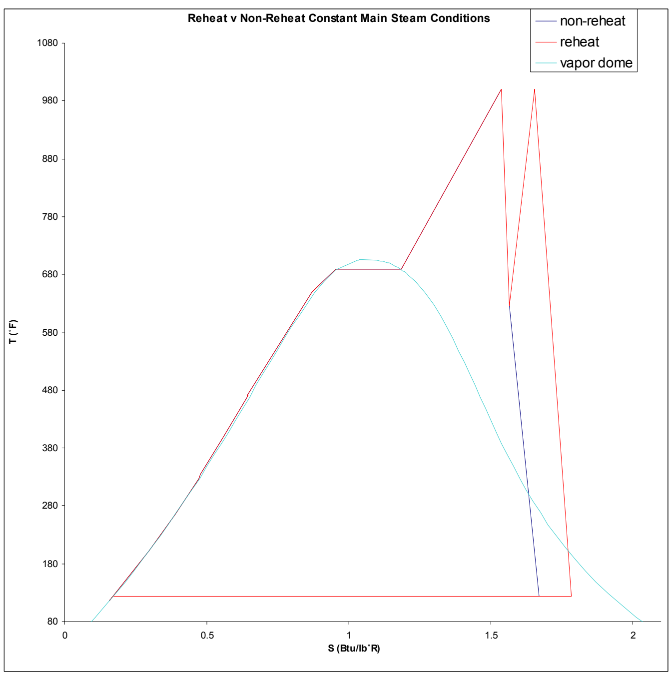

FIGURE 3. REHEAT VS. NON-REHEAT CONSTANT MAIN STEAM CONDITIONS

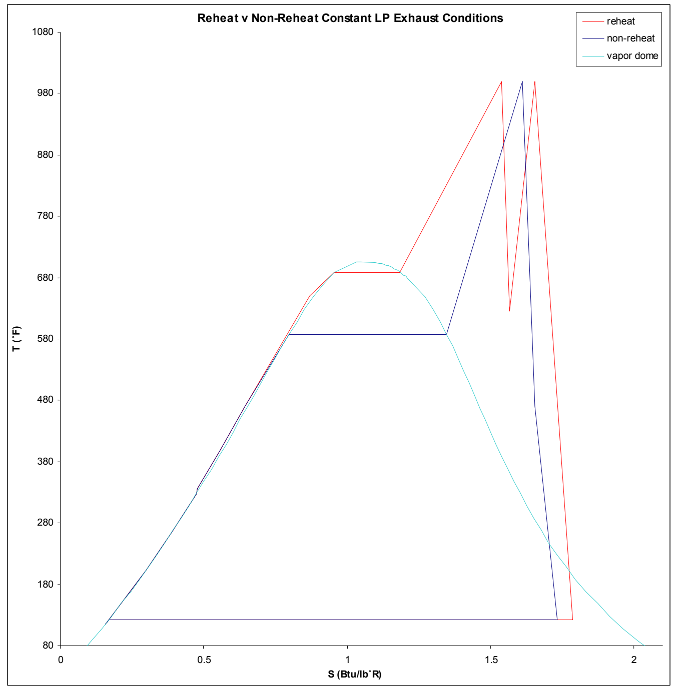

FIGURE 4. REHEAT VS. NON-REHEAT FOR CONSTANT EXHAUST CONDITIONS

Reduction in the turbine throttle pressure has protected the LP turbine blades from erosion due to high moisture levels. However, the amount of recoverable energy has been reduced as well. This is represented by the area encompassed by the blue, non-reheat, line compared to the area encompassed by the red, reheat, line. For the cases used in this study, the amount of heat available for conversion to power is 18% lower in the non-reheat case. The magnitude of the reduction in recoverable energy will vary with the temperature and pressure constraints imposed by the renewable resource.

Figure 5 shows a comparison between a reheat cycle and a non-reheat cycle designed with a throttle temperature of 700 ˚F and with a lower throttle pressure to maintain an acceptable moisture level. In this case, a moisture reduction device is used, similar to the practice in nuclear plants, which allows an improved expansion line.

The ratio of recoverable energy has shifted in favor of using the non-reheat cycle by using a lower throttle temperature by a magnitude of 9%. Therefore, the performance advantage of the reheat cycle in the 1,000 °F case has been reduced when using a throttle temperature of 700 °F. A summary of the performance appears in Table 2. This trend indicates that there is a point where the performance gains of using a reheat

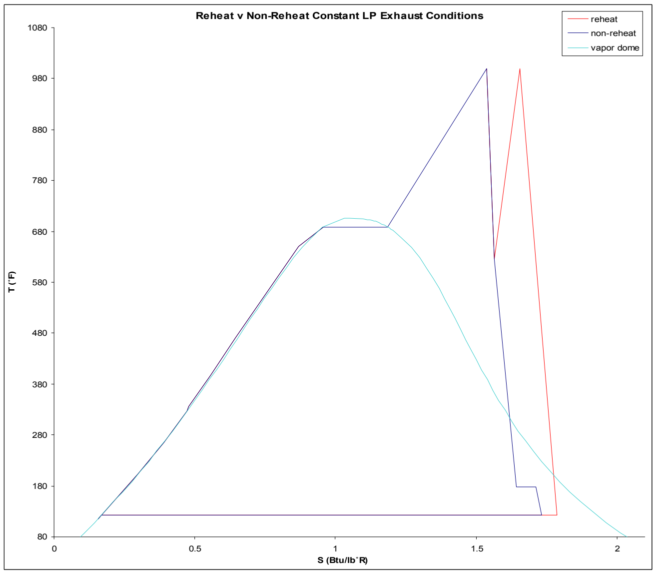

FIGURE 5. REHEAT VS. NON-REHEAT AT CONSTANT EXHAUST CONDITIONS WITH MOISTURE REDUCTION

cycle will be outweighed by the additional capital investment required.

One further option is to borrow from nuclear's idea, where moisture removal is added near the final stage of the LP turbine. Figure 4 provides a comparison of the 1,000 °F throttle temperature where the reheat option is compared with a non-reheat cycle that has a moisture removal stage. This shows a single-stage moisture removal section with a moisture removal effectiveness of 40%. The comparison is qualitative in nature due to the theoretical basis of using this technology derived from much larger applications and viewing it solely from a thermodynamic standpoint. A quantitative analysis would require further investigation with steam turbine manufacturers.

In Figure 5, it can be seen that in the case where throttle temperature is 1,000 °F, throttle pressure can be maintained at the reheat level while moisture is kept to a safe level. There is a 12% reduction in available heat energy in the moisture removal case. Therefore, the performance losses are less than the case where throttle pressure was lowered. A summary of the performance is provided in Table 2.

TABLE 2. SUMMARY OF THERMAL ANALYSIS RESULTS

		1,000 F Reheat	1,000 °F Reheat Constant Pressure	1,000 °F Non-Reheat Reduced Pressure	1,000 °F Non-Reheat Moisture Removal	700 °F Reheat	700 °F Non-Reheat Reduced Pressure
Heat Input	Btu/lb	1,706.2	1,544.9	1,542.3	1,581.9	1,471.4	1392.7
Heat Rejected	Btu/lb	942.0	874.1	911.3	911.2	942.0	911.3
Work Output	Btu/lb	764.2	670.8	631.1	670.7	529.4	481.4
Eff. LP Turbine	%	44.8	43.4	40.9	42.4	36.0	34.6
Exhaust Moisture	%	8.0	14.6	11.0	11.0	8.0	11.0

The choice of whether or not to add reheat to the cycle has significant impact on performance and the initial capital investment required for construction of the plant. LP turbine exhaust conditions must be maintained at sufficiently low moisture levels to ensure long, reliable operation. The cases above have shown the performance reduction on a basis of available energy, but it is important to note that the performance increase in the reheat option has come at the price of increased heat transfer surface area. It is to this end that plant efficiency must be evaluated as well as the plant output.

Either additional initial capital must be invested to keep turbine throttle pressure up and increase performance with a reheat cycle, or throttle pressure can be allowed to fall with a lower initial capital investment and lower performance. The use of moisture removal stages offers a compromise between reheat and non-reheat throttle pressures, but requires further quantitative analysis from turbine manufacturers. In summary, it is imperative in the renewable technologies market that a comprehensive engineering analysis of the various options in available turbo-machinery be conducted to ensure that capital investment is optimized for the renewable resource being used.

4. HEAT SINK CONSIDERATION

In this section, various options for the heat sink are described. Since heat sink selection is dictated not only by cycle design, but also by water availability, a detailed discussion is needed.

4.1 Air-Cooled Condenser

In an ACC (see Fig. 6), heat is transferred from the steam to the air using fin tube bundles. The ACC tube bundles have a relatively large tube side cross section and are usually arranged in an A-frame configuration, resulting in a high ratio of heat exchange surface area versus plot area. The tubes are kept cool by the heat being conducted across the tube thickness to the finned outer surface. Air is continuously circulated over the (dry) outside surface of the tubes. Heat transfer from this outside surface of the tubes to the air takes place by forced convection heat transfer (heating of the air). No evaporation of water is involved. Thus, for ACCs, the condenser performance with regard to turbine exhaust pressure is directly related to the ambient (dry bulb) air temperature, as well as to the condenser design and operating conditions. This results in a higher turbine back pressure for given ambient atmospheric conditions, with a resultant decrease in turbine generator output when compared to wet cooling technologies. An ACC eliminates the entire circulating water system, circulating water pumps and surface condenser.

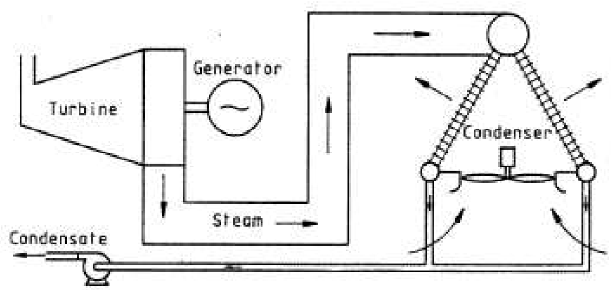

FIGURE 6. AIR-COOLED CONDENSER

4.2 Parallel Condensing System

Exhaust steam from the steam turbine is separated into two streams. One stream flows into a water-cooled surface condenser while the other is directed to an ACC. Condensate from the surface condenser and the ACC can be collected in a common hotwell. Water consumption is controlled by the distribution of the heat load between the two condensers.

The PAC System ™ (see Fig. 7) should not be confused with a “hybrid” cooling tower, which is used primarily to reduce visible plume from a wet cooling tower. A hybrid cooling tower has practical limits to the amount of heat that can be rejected in the dry section, since the latter is sized for plume abatement only. With the PAC System, there is complete flexibility in the amount of heat rejected in the dry section.

The dry section of the PAC System employs direct condensation in contrast to most hybrid systems, which are indirect condensing systems (i.e., water is cooled through both the wet and dry sections and is then pumped through a common condenser). As a result, the dry section of the PAC System can efficiently reject a substantial amount of heat even on hot days, thereby reducing peak water usage. During cooler periods, the amount of heat rejected in the dry section can be increased up to 100% if so designed, thus further reducing the plant's water consumption. An additional benefit of the PAC System is the reduction of plume. Plume can be reduced or eliminated entirely when danger of icing exists, simply by shutting off the wet section.

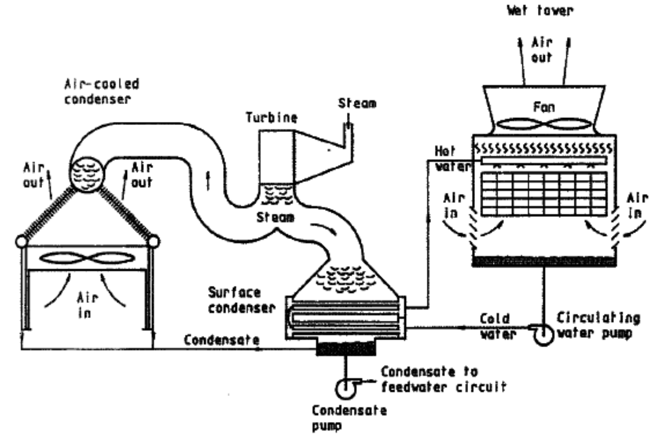

FIGURE 7. PARALLEL COOLING SYSTEM

4.3 Heller System

The Heller system (see Fig. 8) is an indirect dry cooling technology that requires a separate condenser and circulating water pump. The heat is initially exchanged in a condenser to a closed water circuit where the heat is rejected to ambient air utilizing a dry tower with water-to-air heat exchangers, typically in a natural draft configuration. However, mechanical draft is also available. The tower may be equipped with an additional system that sprays water on part of the heat exchanger bundles during hot ambient conditions for peak condenser load reduction purposes. A direct contact jet condenser is typically used, since it is characterized by low terminal temperature difference (TTD) values, but surface condensers have been used as well. Because Heller systems are indirect, there is no need for a large-diameter steam duct between the steam turbine and condenser.

There is no general solution for determining the heat sink. As mentioned earlier, many factors should be taken into account before a final decision is made. Capital cost, scarcity of water, and plant location are some of the many determining factors. Experienced plant design could assist in the selection process.

The advantages and disadvantages of the systems are summarized in Table 3.

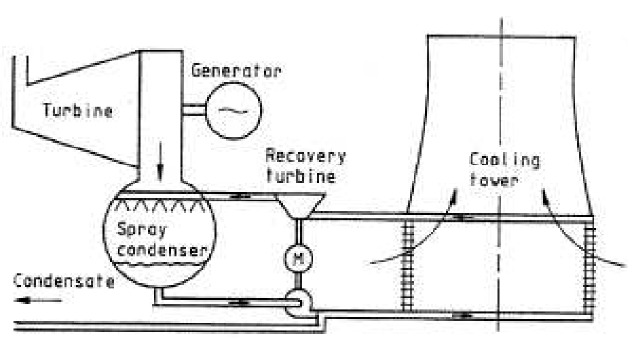

FIGURE 8. HELLER SYSTEM

TABLE 3. ADVANTAGES AND DISADVANTAGES OF VARIOUS HEAT SINK OPTIONS

Pros	Cons
ACC	
▪ Lower capital cost ▪ Does not require full design fan power year round ▪ Shorter and more accurate construction and schedule considerations ▪ Lower visual impact ▪ Smaller plot area requirement (ACC vs. tower)	▪ Potentially higher auxiliary power requirement ▪ Susceptible to recirculation ▪ Performance more negatively impacted by wind ▪ More sensitive to plot plan layout ▪ Higher noise ▪ Nearly 100% of cooling system under vacuum ▪ Higher anticipated time spent performing operation and maintenance activities
INDIRECT DRY	
▪ Potentially lower auxiliary power requirement at design ▪ Use of direct contact condenser – lower turbine backpressures at certain temperatures ▪ Relatively insensitive to distance from turbine hall ▪ More options available for supplemental wet cooling ▪ Heat exchangers do not require gas-tight welds in the field ▪ Decreased maintenance with use of direct contact condenser	▪ Higher capital cost ▪ No installations in the US ▪ Natural draft tower may not be an option for the site ▪ Effects of shadowing mirrors ▪ Large underground tanks required for protection against freezing and turbine flooding ▪ Overnight condensate pump considerations

4.4 Selection Criteria

It is clear that the natural draft Heller system can be justified purely on an economic basis against evaporative cooling even at a medium makeup water cost—thus providing environmental advantages as an extra benefit. In the absence of any moving parts, the dry natural draft Heller system reduces the auxiliary loads of the plant when compared with an ACC.

Since the size of CSP plants continues to increase, the Heller solution becomes more attractive economically. In terms of performance, a Heller tower will reduce the auxiliary load by 44% and increase average electrical production by 1.8% for a nominal 250 MW power plant.

5. STEAM TURBINE

5.1 Requirements

The steam turbine requirements are quite different from those for conventional steam turbines for fossil applications. Important features that equipment suppliers must provide include:

- Modular design
- Capability to accommodate variable high-pressure (HP) flows and high LP flows
- Fast and easy assembly
- Robust design for daily startup (low mass rotors, casings, and reduced seal leakages, etc.)
- Fast-responding controls
- Capability to operate at high back pressure due to extensive use of ACCs for solar applications
- Use of high-quality materials for cycling operation
- Fast startup

It should be emphasized that steam turbine startup and warming must be done as quickly as possible. Designs should consider use of a conventional natural gas firing system to ensure that warm-up of the steam lines and turbine casing takes place before sunrise. In the case of systems with thermal storage, this task can be accomplished by using any available heat stored in the tanks.

In terms of thermal performance, the turbo-machinery should be designed for the following requirements

- High efficiency to reduce solar field
- Low minimum load capability
- Convenient steam extractions locations
- Flexibility to cope with thermal transients
- Proven technology

The main goal of a solar power plant is to produce as many MWh per year as possible. At the beginning and end of the day, the solar radiation is substantially lower. Therefore, to maximize the power production, the turbine should be capable of operation at extreme low loads. While conventional steam turbine minimum load is about 12–15% of the base load, turbo-machinery designers for these special applications should find innovative solutions to continuously operate at 3–5% of the base load. This is not a trivial task, due to the effects of low flow on a fixed exhaust geometry and high ventilation losses.

The turbo-machinery for solar applications should meet the following requirements:

- Achieve the environmental emissions requirements
- Offer design simplicity
- Achieve high availability and reliability
- Provide low operation and maintenance cost

5.2 Turbine Back Pressure

In selecting the steam turbine, it is important to consider the exhaust back pressure and last-stage blade (LSB) design. As can be seen in Fig. 9, a larger exhaust blade will not necessarily provide an optimum solution for the system. A shorter blade (26 inches) yields a lower power loss than a larger blade (33.5 inches) as the exhaust pressure becomes higher.

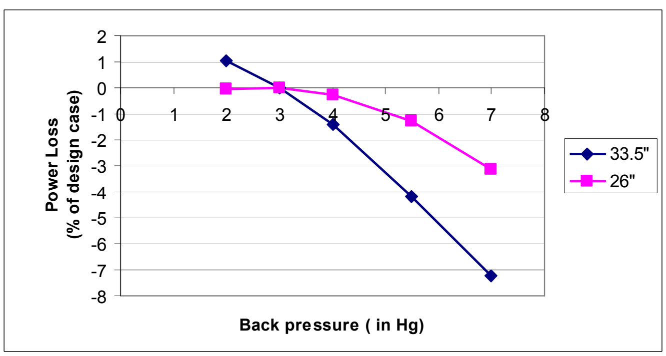

FIGURE 9. POWER LOSS VERSUS STEAM TURBINE BACK PRESSURE

5.3 Startup

An additional important consideration in selecting the most appropriate turbine is the startup time. In the absence of natural gas or other heat source to facilitate startup by warming up the lines, valves, and turbine casing, the turbine's ability to accept steam at lower temperature becomes a significant consideration. Figure 10 depicts such a start. As the figure shows, despite the fact that the heat input from the solar field achieves substantial heat generation more quickly, the turbine startup requires almost 30 minutes to generate any power. Finally, close to 60 minutes is needed to reach full power. This behavior has a direct effect on the number of kWh produced annually. Efforts should be dedicated to improve startup time either by use of conventional heat sources or use of thermal storage.

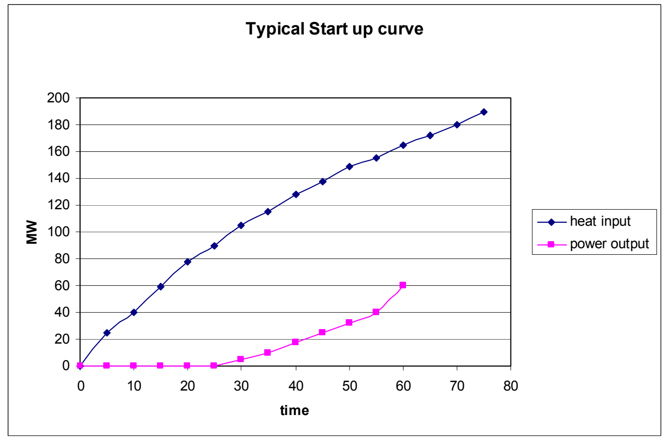

FIGURE 10. TURBINE STARTUP CURVES

6. SUMMARY AND CONCLUSIONS

While significant effort has been dedicated to solar field improvements, a comprehensive understanding of the interaction between the solar field and the heat energy conversion system is vital to development of a successful project. Selection of the two major components, the turbo-machinery and the heat sink, must be coordinated and integrated to meet the specific requirements of the site.

The unique requirements for solar power plants have given rise to specific types of turbo-machinery. The continuous demand for renewable energy will lead to development of more efficient and reliable equipment.

The optimum equipment selection requires detailed analysis of site-specific climate conditions, commercial drivers, and equipment capability to respond to the intermittent heat source behavior. Selection of experienced power plant design and construction firms could certainly facilitate the process.

More than 4,000 MW of CSP power are currently in the planning, design, or construction stages. Given this large volume, it is obvious that dedicated equipment for these applications has entered the mainstream of standard products offered by major manufacturers.