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Advances in steam turbines for solar thermal and integrated solar combined cycle power plants

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This paper presents an overview of developments in steam turbines in general with particular application to match concentrating solar thermal (CST) sources.

The design of solar steam generator which has a parabolic output profile is first presented. Technologies for maintaining steady steam inflow into the steam turbine, viz., hybridization of sources like integrated solar combined cycle (ISCC) and thermal energy storage (TES), cogeneration, trigeneration, etc., are explored.

The developments in the steam turbine consist mainly of upgraded materials to operate at higher steam inlet temperature and pressure, ungraded energy efficiency through 3-dimensional computation fluid dynamics (CFD) design, increased fatigue life to withstand large number of cyclic operations, increased dynamic response and automation in manufacture. It is concluded that while 3-d designed steam turbines for elevated temperatures fulfill most requirements of CST power systems, fatigue life improvement and better energy efficiency at part load, needs to be addressed.

Keywords: *steam turbines, isentropic efficiency, 3-d blading, variable reaction blading, stage efficiency, energy efficiency.*

1.0 INTRODUCTION

Concentrating solar thermal (CST) power plants have come of age because the collector efficiencies have been increased by containing the top losses from the systems and steam generation of 400-500 °C is now possible. These have now become a viable competing technology to solar photovoltaic (SPV) (Table 1). It can be seen from Table 1 that while the maximum efficiency of CST is as high as 64 %, SPV maximum efficiency is only 31 %. However, CST have additional requirement of water requirement (~ 3 m^3/MWh) which is totally absent in SPV. In the long run, whenever water is available (a steam based system is already in place) CST power has an edge over SPV power.

High energy efficiency of CST power plants can be ensured only if matching Rankine cycle components like steam turbine, condenser, feed water heaters, etc., are also of high energy efficiency.

In this paper, the design of high efficiency steam turbines and associated sub-systems which are essential for economical operation of CST plants are described. The high temperature creep and rupture vulnerability of the environment inside the turbine has now been overcome by the emergence of newer ferritic and austenitic stainless steel alloys which are capable of operation with steam at 34 MPa and 566-600 ºC.

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TABLE 1				
COMPARISON OF CONCENTRATING SOLAR THERMAL (CST) AND SOLAR PHOTOVOLTAIC (SPV) FOR ELECTRIC POWER GENERATION				
Sl. No.	**Particulars with the present state of the art**	**Units**	**CST**	**SPV**
			Range	**Range**
1	Power potential (steady power and not peak power)	MW/ km^2	30-45	18-25
2	Electrical energy yield	GWh/ km^2/ year	200-300	100-150
3	Maximum possible solar to electric conversion efficiency	%	64	31
4	Solar to electric conversion efficiency	%	17-18	12-14
5	Water requirements for collector cleaning and power process	kl/ MW	2.5-3.0	0.05-0.1
6	Capital cost	Rs. Cr/ MW	8-10	6.0
7	Energy cost	Rs./ kWh	0.50-2.0	6.0

These are now being applied in the CST power generation area for turbines of 250 MW. The technology advances are mainly in the area of steam paths (blading, passages, cross over pipes, exhaust tube, etc.) and to a lesser extent in the areas of balance of turbine (rotor, bearings, etc.), controls and instrumentation, condenser and feed water heaters.

A major reason for new interest in CST power generation through the Rankine route is the attaining of very high internal efficiencies of steam turbine cylinders through the use of 3-d computational fluid dynamic (CFD) design technology. The technology which has been perfected for fossil fired units is quick to penetrate the 250 MW capacity classes of turbines but is still slow in lower sizes of below 200 MW and especially slow in sizes below 30 MW.

2.0 DESIGN OF SOLAR STEAM GENERATORS FOR STEAM TURBINE CYCLES

Parabolic trough concentrators (PTC) are by far the most successful of the CST technologies [1]. A detailed review of parabolic trough collectors for power cycles has brought out that temperatures up to 400°C are achievable with PTC very efficiently [2]. The design efficiency of solar to electrical energy is 22.4 % and the annual average efficiency is 15.3 % [1]. Simulation of integrated steam generating solar tower shows that by separating the external boiler for evaporation and cavity type separator for superheating, steam can be generated at 550ºC and 15 MPa [3]. However, there are other limitations with solar towers. Solar tower type receivers which have separate water cooled receiver for sensible heating, evaporator section and superheating section have been proposed to handle steam dryness in CST with direct steam generation (DSG) systems [4].

PTCs are normally designed to extract thermal energy through thermic fluids like oils. The alternative is to go in for DSG [5]. DSG based parabolic trough CST-PTC collectors give efficiencies as high as PTC collectors give efficiencies as high as 76 % [5]. The study of steam temperature stability in DSG mode indicates that temperature gradients are higher than those in conventional steam generators. Temperature overshoots of up to 45 ºC are common [6]. Steam storage will stabilize the temperatures. The change over from synthetic oil to DSG resulted in an improvement of 6 % in collection efficiency [1, 7]. The first 5 MW DSG recorded steam conditions of 7.0 MPa and 410 ºC [8]. Steam conditions as high as 500 °C and 12.0 MPa are achieved through DSG instead of synthetic oils [9].

A requirement of condensing Rankine cycle steam turbine based power plants is to maintain steady generation to meet steady load. Since solar energy has an inherent parabolic pattern of energy input coupled with stochastic variation due to atmospheric disturbances such as cloud, fog, rain, snow, etc., a steady generation can only be ensured if the system has an alternative source or energy input during the no-solar period. This has led to the concept of integrated solar combined cycle (ISCC) wherein a hybrid of CST and gas turbine in combined cycle (GTCC) are used to maintain steady loading on the steam turbine cycle and associated equipment.

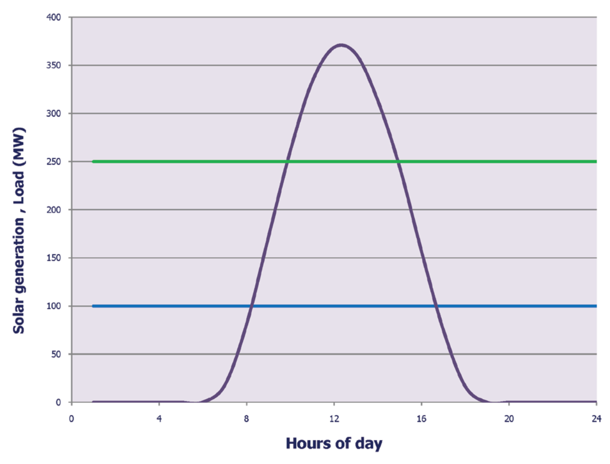

FIG. 1 CHARACTERISTICS OF A 250 MW GENERATING UNIT. IT CAN BE SEEN THAT AVERAGE ROUND THE CLOCK LOAD CATERED TO IS 100 MW

The other option for steady loading of CST based power plants is the thermal energy storage (TES) systems. Figure 1 gives the results of a simulation performance by the author without energy storage. It can be seen that a 250 MW unit can be operated at its near perk period only for a short duration of 4-6 hours and the load is to be backed down during other periods for want of energy input. If around 16 hours of storage is incorporated, a 250 MW unit can be operated steadily at 100 MW. In other words, the unit can be sized down to 100 MW. Hence, storage plays a major role in maintaining steady power and reducing the life cycle cost.

The use of thermal salt storage resulted in an improvement of solar to electrical energy efficiency from 15.3 % to 17.8 % [1]. While storage with phase change molten salts for 9-10 hours adds the capital cost of the salt to the overall capital cost, it reduces the operating cost and energy cost due to better capacity utilization and stabilized output [9]. Molten salt storage with both hot and cold storage components to give round the clock operation and storage capacity of 17 hours has been proposed as a solution to the on-demand power [10]. Solar salt pond can be used as TES medium for a regenerative Rankine steam cycle with extraction turbine to cogenerate [11]. The use of steam accumulator technology has been suggested for buffer storage for CST-PTC systems with DSG [12]. A comparison of latent heat versus sensible heat storage (of water) indicates that sensible heat storage has advantages of higher heat transfer rates [13].

High plant load factor {PLF= energy generated/(8760 x plant capacity)}, plant availability factor {PAF=Available number of operating hours/8760} and capacity factor {average load/rated capacity} can be achieved through ISCC and TES options. The focus of CST power plant design must be on the number of full load hours and matching of the load with the source [14]. The performance of CST based power plants is intimately linked to sustainability and energy generation [15].

In an analysis of optimization of CST power plants it is shown that physical optimum conditions must also match with economic optimum. The economic optimum operational conditions lie in the vicinity of the physical optimum [16]. A review of the optimization processes from artificial bee colony optimization to variable neighbor searches indicates that over 35 optimization methods can be applied to renewable power plants which include CST based power generating systems [17]. Supercritical steam regime for Rankine cycle has been established to give stable output from CST systems [18]. Once through flow with recirculation has been shown to have better control over main steam conditions at the entrance to the steam turbine [19]. Header type steam generators with integral economizer, evaporator and superheater in a single shell have been proposed for CST plants.

Alternatives to steam for CST based power generation have been proposed such as CO^2 in ISCC mode [20]. With 10 % solar contribution CO_2 cycle gives an efficiency of 23-25 % in comparison with DSG steam based systems of 22-27 %. Rankine thermodynamic cycle with supercritical CO_2 has been proposed for large scale power production [21]. Binary conversion cycles of alkali-metal (potassium/rubidium) plus steam are showing efficiencies up to 56 % against 43 % for pure steam systems [22]. Calcium looping ($CaO/CaCO_3$) has been proposed to avoid use of water as a heat transfer medium for high temperature systems [23]. A conceptual design of a solar boiler similar to a vertical fossil fuel boiler with heat exchanger coils has been proposed to operate on molten salts, mineral oils, and air or water-steam systems [24]. To improve up to the condensing steam cycles, the Kalina cycle (ammonia-water mixture) in the bottoming mode has been proposed to handle low solar insolation and reduced condenser losses [25]. Improvement of 4-11 % over the normal steam cycle is expected from the Kalina cycle [25].

3.0 TECHNOLOGIES FOR STEADY ENERGY OUTPUT FROM STEAM TURBINES

The design approach for CST-PTC is to go in for the highest working fluid temperature. This approach has reopened interest in the design of CST steam generators to achieve high conversion efficiency. The fossil plant technology of steam turbines has sufficiently advanced in the area of temperature and pressure of operation (34 MPa, 566-600 °C) by designing steels to withstand the creep rupture considerations) which is well within the achievable working fluid temperature range of CST. Moreover, the CST energy sector is restricted to the subcritical steam regime while utility coal fired units are switching over to ultra supercritical operating cycles of operation (34 MPa, 566-600 °C). Also, from the view point of unit size, in the coal fired utility power sector, 300 MW is an average size and individual units of 660 MW and 1200 MW are in operation. Very recently, with breakthrough in CST technology, units of around 300 MW are becoming a reality.

An added feature of CST based steam generation is that the auxiliary steam associated with fossil fuel processing and handling is totally absent in CST systems. Auxiliary steam is only required for internal turbine cycle requirements, air expelling devices and cycle start-up.

In spite of the present steam turbine technology easily meeting the operating temperature, pressure and unit size requirements of CST, there are major differences between CST and conventional fossil fired units which need to be addressed:

- The number of cyclic operations required of CST plants is at least 9,000 per year which is the equivalent of total number of cyclic operations over a 25 year period of a fossil fired plant.
- The ramping rates of parameter variations (% change/min.) during transients of CST plants are much higher than those withstand able by materials than those designed for fossil fired plants.
- Allowable stresses caused operating parameters like inlet steam temperature, etc., exceeding their normative values for larger durations per excursion (> 15 min./excursion), larger total duration (hours/year) and larger physical magnitudes (% over design value).
- High turbine heat rate (reciprocal of turbine energy efficiency in dimensional form) at part loads (40 % to 70 % MCR)

The above problems need to be addressed in steam turbines designed for CST applications. While hybridization through ISCC and TES options [26] are being incorporated to obtain a steady energy output to reduce some of the above effects, the designs must take into consideration the worst scenario of CST variability in its simplex operating mode involving natural and stochastic variation of solar energy.

Solar hybrid gas turbines have been proposed to achieve fuel saving in spite of decrease in overall efficiency and power output [27]. Optimization of GTCCs operation [28] with CST (ISCC) has indicated that exergically optimal conditions can be obtained to give heat to electric efficiencies of over 50 %. Exergy analysis of an ISCC system indicates that the exergy destruction in the turbine, collectors and heat exchangers is 8 %, 9 % and 9 % respectively [29]. These are the three major areas for effecting improvement. Assessment of a working ISCC plant in Iran indicated that energy efficiency improved by 4 %, steam turbine capacity increased by 50 % and life cycle cost reduced by 10 % as compared to conventional combined cycle plant [30]. ISCC of GTCC with CST-PTC systems in the DSG mode coupled are found to the most efficient combination giving solar to electric efficiencies as high as 50 % [31]. Comparative study of ISCC system with CST-PTC in DSG mode in very hot climate on one hand and very hot and very humid climate on the other hand was taken up [32]. The study indicated that generally while hot climates are not good for ISCC plants, very hot and very humid are still worst [32].

Solar powered fuel assisted Rankine cycle configurations have been established based on pre-generation of steam through solar and superheating in fossil fired super heaters to give steady conditions of steam entering into the steam turbine. With a 20 % fossil support, the system can achieve a basic power cycle efficiency of 18.3 % [33]. Exergy analysis of CST and natural gas hybrid systems indicates with introduction of solar energy for feed heating an improvement of 10 % in the overall energy efficiency and power output can be achieved [34]. CST powered steam injected (at 200 °C) gas turbine power plants have shown solar to electric efficiency of 15-24 %, and combined cycle efficiency of 22-37 % has improved to 40-55 % for solar fraction up to 50 % [35]. Seven possible methods of augmenting a gas turbine plant have been proposed. These include heating of feed water, evaporation (latent heat addition), superheating, and combined evaporation and superheating, air preheating, reheating with air preheating, air preheating and feed water heating [36]. While feed water heating is most economical it is low in exergy efficiency [36].

Some of the developments aimed at stabilization of the steam output from CST systems are described herein. A control scheme is designed to forced recirculate the water to obtain constant outlet steam pressure and temperature [37]. By dividing the heat absorption in the CST system into sensible heating, latent heating and superheating portions, the steam output can be stabilized [3]. Various types of routes such as trigeneration {generation of thermal energy (heating and cooling) plus electrical energy}, organic Rankine cycles, pure electrical power, cooling energy-co-generation, heating energy-cogeneration are analyzed for maintaining a steady output [38]. It is seen that the main sources of exergy destruction are solar collectors and evaporators which must be focus areas [38]. Another study of trigeneration plants indicated that maximum energy efficiency of 60.33 % was achieved [39]. From overall considerations of energy efficiency, environmental and economic considerations, CST based cogeneration technologies are more superior to pure power generation [40]. The design of CST based system must focus on cogeneration for getting stable output. An exergy analysis of CST systems with DSG for power generation indicates that three or more feed water heaters result in increase of energy efficiency of the turbine cycle [41].

On the power systems side also, the response of other generating systems to sudden changes in CST power output is a problem which can lead to frequency, voltage and reactive power variations or instabilities. This problem is presently covered in passing and not addressed fully in this paper which pertains only to steam turbines.

2.0 Overall steam turbine design

In the context of this paper steam turbine refers to the composite train of individual steam turbine modules mounted on a common axis shaft and connected to the generator which is connected to the power grid.

There has been continuous improvement in the size, design and performance of steam turbines since the development of the impulse steam turbine by De Laval (isentropic efficiency: 44 %) and the reaction turbine by Charles A. Parsons (isentropic efficiency: 75 %). Unlike the turbines operating on incompressible fluids like water turbines, steam turbines cannot convert all the energy in the steam into rotational kinetic energy through one set of blades but require a large number of sets. The set of blades arranged serially are called as stages. The set of serial stages in a turbine cylinder are called as a stack or group. Each stage consists of a pair of fixed and rotating blades also called as disk-diaphragm, nozzle-bucket, and guide-working blades. As the steam moves through each stage of blades it expands and it would be very unwieldy to have a single long turbine with blades heights ranging from 25 mm to almost 1 m. This is handled by arranged groups of stages into modules. Each turbine module (also called as a cylinder) and is composed of a number of stages (adjacent stationary and moving blades).

It is now generally accepted that turbines of 50 MW to 250 MW are designed in 3 tandem compounding modules composed of single flow high pressure turbine (HPT), double flow intermediate pressure turbine (IPT) (1 modules) and double low pressure turbine (LPT). There also designs with 4 modules composed of 2 IPT modules. For units larger than 500 MW it is one HPT module, 2 IPT modules and 2 LPT modules. The volume of individual turbines increases with decrease in steam pressure from 1.5 m3 for the HPT to 2 m^3 for IPT and to 6 m^3 for the LPT because of the expansion of the steam which necessitates larger handling volumes. The staging of turbine power has been optimized and constantly refined by both thermodynamic and thermo-economic criteria [42-44].

The degree of reaction or stage reaction is defined as the enthalpy drop across the moving blades as a percentage of the total enthalpy drop across the stage (moving plus stationary set). The purely impulse (all enthalpy drop in fixed blades) and purely reaction stages (all enthalpy drop across moving stages) in a turbine have given way to impulse-reaction stages typically the 50 % reaction type where the enthalpy drop of the stage is shared equally between the stationary and moving sets of blades [45]. The expansion of steam takes place in two steps hence the flow velocity can be controlled more precisely and flow losses minimized. Though various manufacturers have their unique design philosophy and strategy, the technology has been moving in a common direction due to globalization and rapid cross-ventilation of ideas. Power loss factors have been characterized [46].

Figure 1 gives the typical variation of turbine heat rate over the full load range. Fossil fuel plants are rarely operated at part loads below 90 % MCR and are continuously maintained at near 100 % MCR loading. In the case of CST power plants, part load operation is essential. Hence, the design needs to be modified to obtain a flatter heat rate performance over the 40-70 % MCR range.

The focus of all turbine manufacturers' designs is to:

- Maximize isentropic efficiency (actual enthalpy drop/isentropic enthalpy drop) without compromising on the structural design. This is through optimal blade and steam path design.
- Maximize the number of safe permissible cyclic operations during the life time of the turbine (i.e., improvement in fatigue life).
- Choose materials which give maximum temperature and pressure of steam (i.e., maximize creep-rupture life).
- Improve the heat rate in the range of 40-70 % MCR.

Figure 2 gives the typical variation of turbine heat rate over the load range of 0-100 %. Fossil fired plants are rarely, operated at part loads and always continuously maintained at near 100 % MCR. In case of CST powered plants, part load operation is essential. Hence the design needs to be modified to flatten the energy efficiency curve (represented by heat rate) over the 40 % to 70 % range.

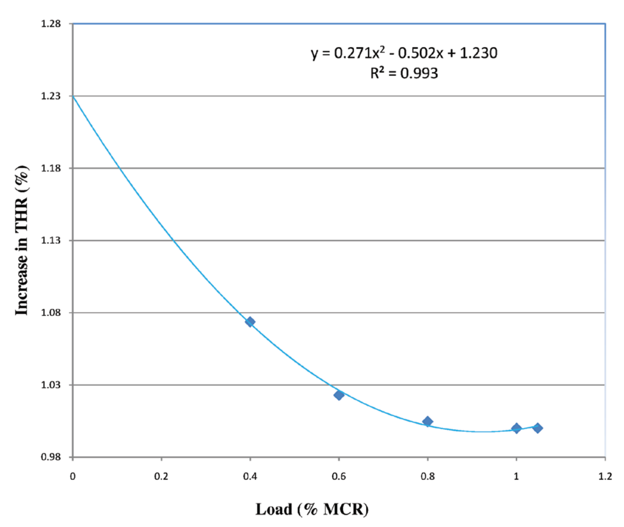

FIG. 2 VARIATION OF TURBINE HEAT RATE WITH LOADING ON THE UNIT

3.0 DESIGN TOOLS AND MANUFACTURING PROCESSES

The developments in computer capabilities, information technology, electronics and their combinations (embedded systems) have created unparalleled computing power which has enabled visualization of phenomenon inside the turbine and 3-d design of energy converting machine parts and flow paths. Flow solvers are basically matrix inverters with ability to invert large matrices. The exponential grown of matrix inversion capability of computers is the key to design optimization. Digitization of many analog functions, automation of manual operations and new digital based capabilities has enabled fine tuning of machine operation. Digitization of design drawings has enabled automated manufacturing without any human intervention from the design desk to the final product.

While the new developments in steam turbines have been implemented in large sized turbines of 660 MW and above, as one moves down in capacity to 250 MW and to 100 MW and further down to 30 MW, many of the new developments have not penetrated in the design philosophy and the older technology continues to be use. Steam turbines for CST applications must be of the state-of-the-art with the highest possible energy efficiency.

The development of steam turbines for maximum isentropic efficiency involves 3-d design optimization of stage specific blading (variable reaction along every successive stage) with twist, taper and bowing and also variable reaction radially across a blade through biasing of mass flow rate from the tip (40 %) to the root (60 %).

For computing the mechanical energy losses due to friction, eddies and leakage, 3-dimensional Navier Stokes equations are used through flow solvers. Viscous solvers with realistic boundary conditions are now being used in place of invisid solvers. The 3-d analysis is conducted using commercial general purpose software CFX™, TASCflow™, NASTRAN™, etc... The isentropic drops of a stage are fixed and losses are computed by 1-d models, then 2-d and then 3-d models for acceleration of results. Parallel computers with mesh fineness of almost 0.25 million grid points for a blade passage are in use.

For implementation of 3-d profiles the manufacturing processes have also to be fully automated. The trends are to make designs modular to decouple manufacturing processes from specific customer requirements. The 3-d CAD (computer aided design) and CAE (computer aided engineering) processes generate drawings, process instructions, specifications, electronic product manuals and manufacturing instructions without manual intervention. Blading plans, groove design, drawings of unmatched parts and final drawings are all generated automatically as a computer sequence.

The introduction of electronics and software into machine tool technology (numerical controlled machine tools) has enabled fabrication of blades using 3-d drawings generated from commercial patented/trademarked software utilities (PRO ENGINEER™, IDEAS™ & CATIA™). Very sophisticated blade profiles can be fabricated through numerical controlled 5-axis milling machines.

The use of mobile machine tools for on-site work on castings, rotors, couplings and valves has reduced installation and commissioning time.

Software based programs like PROTEC™ reduce the down time by auto positioning of coupling bores, etc., with ease.

4.0 BLADES AND STEAM PATH STRUCTURES

The major thrust of the advances in steam turbines is the design of blading (rotary and stationary). The conversion of energy in steam into rotational kinetic energy is achieved through a large number of stages. Typically high pressure (HP) blading is small in size and as the steam expands along the steam path the length of the blades increase from 25 mm (at 20.0 MPa) to 770 mm (at 20 kPa) [47]. Around two decades, back blade materials were not available to fabricate blades of 400 mm height and so around 50 % of the steam was diverted to the condenser through the penultimate stage without going to the last stage. This flow arrangement was called as a Baumann exhaust. Presently, large blade sizes being possible, the Baumann exhausts are of vintage design. For typical 250 MW units, the power generated increases from 18-20 kW/blade in the HP section to 30-40 kW/blade in the LP section [48].

4.1 Classification of losses in a steam turbine and their reduction

The focus of steam path optimization has been on the stationary and moving blades. The losses in each stage of the blade can be classified [48] as:

a. Flow friction over the blades. This is also known as the profile loss and occurs on account of the viscous nature of steam and roughness of the blade surface.

b. Secondary re-circulation flow losses. Steam through the turbine blade passages does not all flow in the axially but swirls radially causing local eddies.

c. Tip leakage. The flow through clearances between the blade and the casing is nearly ten times that of the flow through the blades.

The total of the above three losses gives the blading loss.

The other stage related losses [48] are as follows:

a. Moisture losses: The wetness of the bulk of fluid is determined from the average of local experimental wetness. Alternatively, the moisture in exit steam is computed from the energy balance of the composite turbine. The exit enthalpy is the unknown which is determined from which the dryness fraction is calculated. The moisture loss is given by the product of the mass flow rate of moisture in steam (bulk of fluid) and the enthalpy change across the stage.

b. Exhaust outflow and extraction losses: This is based on the averaged values of the fluid parameters at the LPT blade section outlet. This loss is the exit kinetic energy of bulk of fluid.

c. Losses from clearance seals of blading and shafts: This is calculated based on turbine designers guidelines:

d. Cross over pipes: This is calculated based on the frictional pressure drop across the flow length of the piping.

The reductions in losses are achieved primarily by controlling the profile losses in blades. The most significant losses are those due to blade friction-profile losses. For standard high efficiency blading offered by major manufacturers, the profile section is optimally designed to achieve to the flow conditions in that particular stage. Increase of pitch to chord ratio decreases the profile losses. The blade airfoils of the stator and rotor rows comprise of a family of profiles with different cambers and thickness. The twists of the airfoils implies a corresponding variation in the stacking angle of the different profile sections, the chord length of which are governed by the conditions of a constant axial width. The high efficiency blade is ensured to provide insensitivity to incident angle and thin trailing edge.

The second most important loss in the blading is the secondary losses. Secondary flow occurs because of flow deflection in the blade channel. This is minimized by bowing the blade circumferentially. Minimization of cross flow velocity and uniformity in axial flow velocity decreases secondary flow losses. By incorporation of local leaning at the blade outlet and through change in blade outlet angle to account for disturbance caused by the immediately next blade row, secondary flows can be controlled by increase in aspect ratio (blade height to chord ratio). Secondary losses can be minimized by biasing the flow - more towards the root and less towards the tip of the blade.

The last and least of the losses is the clearance or leakage loss. In reaction blading, the traditional radial variation of flow is 40 % at the hub section to 60 % at the tip section. The clearance loss is minimized by reversing the radial flow profile to 60 % at the hub section and 40 % at the tip section. This will bring down the tip leakage by approximately 0.1 % points in the second last stage but it is not very significant in the other stages. Tip leakage is further minimized by inserting of spring backed seals in the tip block of both fixed and rotating blades.

When changing over from Baumann exhausts (divided flow) to normal exhausts (free standing blades), an increased hub reaction (flow expansion and wetness) of the last stage blades minimizes the risk of flow separation in the hub region at part loading which minimizes blading losses. Optimized radial exit angle of the last stage minimizes blading plus non-blading losses. Also, channel shock (due to transonic flow) is minimized.

4.2 Design of high efficiency blading

One dimensional model is well established. Earlier, for blade design, the losses in the steam path were estimated based on a number of semi-empirical correlations and relations for blade losses, stage loss, tip leakage and seal leakage. The turbine blade designs have steadily upgraded from 2-d to 3-d design.

Presently, the blades are designed with twist, taper and bowing (leaning) using 3-dimensional computational fluid dynamics (CFD) for optimization of the losses due to friction, leakage and secondary eddies [49]. Twisting of blades minimizes the incident effect of entering steam, taper minimizes the profile friction losses and permits aerodynamic loading while bowing enables radial biasing of flow across the blade. Rotor blades are designed with variable airfoil sections with reverse twist and no bowing. Stationary blades are designed with variable airfoil sections, twisted and bowed. The blade designs have successfully addressed problems of non-uniform flow distribution across the blade, transonic flows; wetness in steam, vortex formation, swirling and exit eddies. The stage efficiencies for fixed and rotating blades in terms of stage loading, stage reaction, flow coefficients, etc., are described by Simon et al. [50].

Vortex transport and blade interactions have been investigated in HPT [51]. Flows into blade shroud clearances [52], entrance zones [53], exit zones [54]; and aeroelastic behavior of turbine blades in transonic flow [55] have been optimized. Low solidity blades are preferred at the entrance nozzle section [56].

Several turbine manufacturers such as Siemens [57], ABB [58], GE [59], LMZ [60], Alstom [61], Parsons [62], BHEL, Skoda, Hitachi, Mitsubishi, etc., have contributed to the design of blades with higher stage isentropic efficiencies.

The 3-d optimization of individual stage of blades has given way to that of entire turbine cylinder with each row of blades being of stage specific design (group or stack optimization) [63, 64]. Stack optimization has been enabled because of three transitions in the design philosophy:

i. From fixed impulse and reaction stages to 50 % reaction stages where the enthalpy drop across the stage is shared equally between the stationary and moving blades [65]. The number of stages in a 50 % reaction turbine cylinder is almost twice that of a purely impulse cylinder for the same energy change across it.

ii. From 50 % reaction stages to variable reaction stages with the degree of reaction different for each stage in a cylinder and varying between 10 % and 60 %. Hence the design of each set of blades is stage specific and different from the other blades in the stack in its stage reaction. However identical profiles for the fixed and moving blades results in simplified design and economy in manufacture.

iii. The total enthalpy drop across a given turbine cylinder which was earlier set approximately equal for each stage is now biased axially with stages and radially across the length of each row of blades.

Having optimized the enthalpy drop across a stage and the degree of reaction of the stage, the stage losses are reduced by optimization of 3-d design of the individual stages. Further the optimization of the entire stack in the cylinder is taken up.

The Siemens been successively re-designing blades from the original T2™ to T4™, TX™, 3DS™, 3DV™ (variable reaction) blading [66]. The optimized 50 % reaction stages have been developed for the entire HPT and IPT cylinders and initial sections of the LPT [49].

The ABB has replaced Series 1000 blades with Series 8000 blades ™ and with HPB (high performance blading) Series blades [67]. These are cylindrical prismatic blades which are insensitive to inlet angle variations with an optimized controlled diffusion zone and thin trailing edge [45]. These have integral blade root and shroud.

Parsons have come out with the R series blading ™ as a replacement for the '600' series blading ™ [62].

The added feature of the 3-d designed turbines are dimensionally solidified blading, design of blading to ensure good creep rupture strength and stiffness; and insensitivity to resonant frequency, deviation in incident angle and thin trailing edge. Traditionally integral blade, root and shroud are milled out of solid bar stock of 12 % Cr austenitic steel. The 3-d directionally solidified blading for the HPT stages are grown out of single crystals.

In the HPT stage there is one piece admission segment. Full arc admission tilted stationary blades are proposed for the first stage. Double T-root blade designs are proposed for the 1st and 2nd stages of the HPT. A tilted first stationary blade arrangement with a low reaction blade profile is used for the IPT stage. A low IPT inlet pressure yields a higher efficiency in the IPT and LPT because of the effect of aspect ratio and lower LPT temperatures.

The number of blades in each stage also affects the stage efficiency. The larger the number of blades, the lower the efficiency. The number of blades can be reduced by maintaining a higher aspect ratio. To overcome this drawback the concept of 'dense pack' blading is introduced. This involves a newer design analogous to gas turbine blading with increase in no of stages of the HP/IP cylinders without increase in overall length of the machine.

The last stages of the LPT are free standing with optimal pressure ratios. The last stage blading is made of alloy steel for blade lengths up to 950 mm (axial exhaust areas up to 15.0 m^2) and from titanium of length up to 1200 mm for axial exhaust areas of 23.5 m^2 [59,49]. Baumann stages are now replaced with free standing blades capable of handling the total LP flow. The last stage blades have special design features- these are slotted hollow blades or hollow blades with heating using steam from earlier extraction to avoid moisture condensation on its surface and thereby reducing corrosion-erosion damage.

Heating system for the surface of the last stationary stage of hollow blades (which have to handle moist steam of as high as 10 % local wetness and 6 % total wetness) have been developed based on injection of steam from an earlier extraction. A number of technologies for cooling of initial sets of blades have emerged [68-70]. Optimization of wake-blade interaction is also taken up for the last stage where the turbulence is high [71].

Figures 3-5 give views of high efficiency 3-d blading in a steam turbine with twist taper and bowing. The bowing and twist are exaggerated to show the design feature.

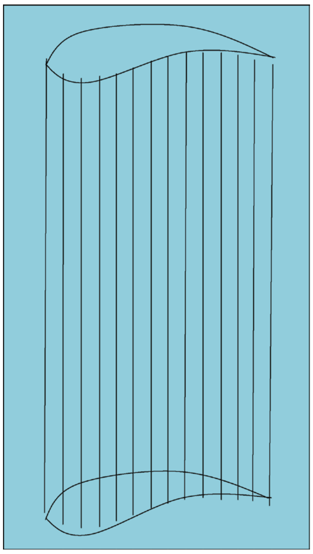

FIG. 3 VIEW OF A 3-D TAPERED BLADE

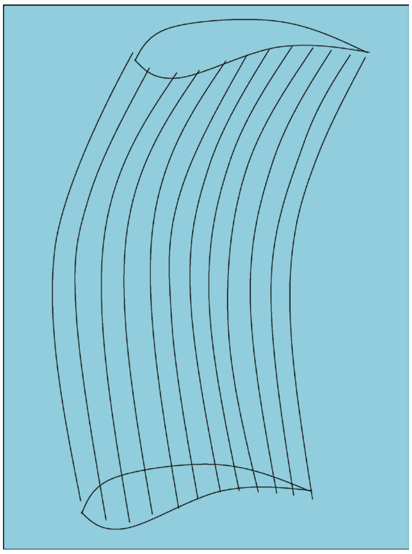

FIG. 4 VIEW OF A 3-D TAPERED BLADE WITH BOWING

The improvement in change over from 2-d designed machines to 3-d designed stack optimized design of machines with variable reaction is as around 6-7 % for units up to 250-300 MW, 5 % for units of 500-660 MW [59] and 3 % for units of 800 MW and above. The reason for reduction in improvement margin in larger machines is that there is already a good degree of optimization in the original 2-d machines.

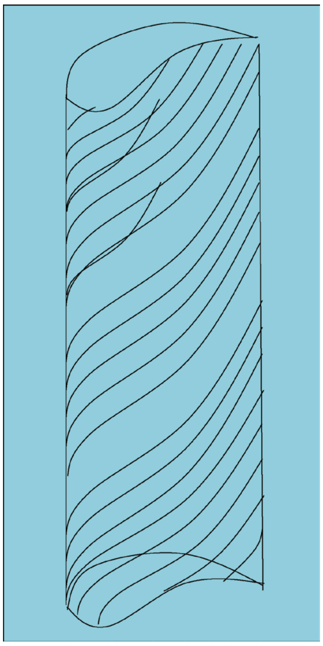

FIG. 5 VIEW OF A 3-D BLADE WITH TAPER AND TWIST

In the small range of turbine of 1 MW to 30 MW in CST plants, the possible improvement is nearly 11-12 % because the designs are antiquated and frozen rather than matched to the user's requirements. Though there is a large market for efficient 3-d designed blading in this capacity range, the improvements have not penetrated into the field. This is partly because of high technology costs involved in 3-d optimization and partly because of conservative approach of this small capacity sector (who are presently catering only to the cogeneration sector) (30 MW and below).

The efficiency of steam turbines in the 250 MW range and above is constantly and continuously increasing over the years. However, mature technologies as this one are expected to saturate and tend towards asymptotic values of maximum efficiency. With the incorporation of ultra super critical steam cycles, the thermodynamic efficiency can be boosted to 64 %. The maximum isentropic efficiency is can reach ~96 %. The overall turbine efficiency of the composite turbine can reach ~56.7 %. The presently available flow solvers do not consider the effects of compressibility, unsteadiness, realistic turbulent flow representation, realistic two phase flow models. These effects if incorporated could further refine the optimization process and isentropic efficiency.

5.0 NON BLADED STEAM PATHS-CASING PIPES CROSS OVER PIPES, EXHAUST HOOD AND DIFFUSERS

The considerations for optimization of casings are attenuation of thermal losses, balancing of axial thrust and ease of maintenance (dismantling and boxing up).

The HPT are of barrel/double barrel design which are compact and of rotationally symmetric casing geometry, without the need for flanges along the sides of the outer casing and eliminates non-uniform changes in radial clearances [72]. It provides a staggered envelope avoiding the need for horizontal joints with large bolts and axi-symmetric geometry resulting in good thermal flexibility.

Two shell horizontally split cylinders are used for IP cylinders with rotor thermal shield with a swirl cooling device and all extraction points being located on the bottom half of the casing [72]. The rotor heat shield in the inlet region prevents the rotor from excessive heating. Heat shield rings with vortex formation provides rotor cooling in the middle sections.

For the LP cylinder, the inner casing is supported directly on the slide bearings. A push rod connection from the outer casing of the IP enables inner casing of the LPT to follow shaft expansion by shifting towards the generator. The LP casing is integral with the condenser. The inner casings are of nodular cast iron design. Modification have been effected in location of exhaust hood spray such that the excessive ventilation heating (during low load operation and load rejection) is not experienced in the LPT on the generator side.

In the non-blade steam paths, IP-LP cross over pipes, 30 % HP/LP turbine bypass (to save the unit from tripping in case of load rejection on the electrical side) and LPT exhaust hood are 3-d designed with a conical profile to minimize the mechanical losses [57].

The exhaust outflow and extraction losses are reduced through improvements in diffuser geometry. Presently, steam from the free standing blades is passing into the diffuser axially so that it has to turn almost 90º. The loss minimized diffuser accommodates high velocity flow, retards the steam and recovers considerable kinetic energy thereby minimized the energy losses. Also, minimized radial exit angles of the last stage helps to minimize the exit losses Thus, incorporation of 3-d designed diffusers reduces the losses by almost 0.9 % points. The velocity of 2 phase stream reduces (by ~20 %) to 180 m/s from 220 m/s after incorporation of the standard last stage blading (i.e., increasing the blade height from 770.0 mm to 850.0 mm).

6.0 SEALS, VALVES, ROTOR

CST based systems do not have any auxiliary steam requirement for fuel preparation or processing as in fossil fueled plants. Auxiliary steam is only for the turbine cycle but the auxiliary steam in internal leakage circuits external to the turbine cycle is not recoverable into the turbine-condenser-feed water heater cycle and must be minimized. In 250 MW class turbines, out of the total auxiliary steam of 8 kg/s, 4 kg/s is required in the turbine cycle as sealing steam, leakage into LPT, spindle leakage and leakages in HPH, LPH and LPT. The auxiliary steam flow rate (loss through seals, spindle leak off, etc.) for the HPT, IPT and LPT are 1.7 kg/s, 0.8 kg/s and 0.7 kg/s respectively [48].

Current developments such as tip to tip see-through seals enable both radial clearance/gaps and axial widths to be reduced by 0.1 mm with reduced risks of rubbing. The losses due to clearance seals are reduced (for the HPT, IPT & LPT) to 0.92 %, 0.50 % & 0.53 % giving an overall reduction in steam flow for the complete turbine by 1 kg/s [48].

The increase in turbine cycle efficiency due to the reduction in steam consumption is around ~0.5 % [48]. Approximately when the clearance exceeds 0.6 mm, the heat loss due to steam leakage (W/μm) the heat loss due to steak leakage is: Blade tips: HPT & IPT: 200, LPT: 400; inter-stage seals: HPT & IPT: 100, LPT: 200; end glands: HPT & IPT: 60, LPT: 80 [48].

Tip-to-tip double strip seal design allowing unlimited relative axial expansion of shaft system [72] help in reducing the internal steam leaks in the system. The loss due to seal leaks in conventional designs can be as high as 2-4 %. The new designs not only reduce the leakage down to 1.2-1.4 % but also maintain consistency over the period between two capital overhauls by attenuation of flow coefficients from 0.6 to 0.3. Seal widths have gradually reduced from 2 mm to 0.6 mm. The radial sealing gaps are reduced; double strip seals are used and caulked in.

Hydraulically operating digitally controlled valves are used. In the HP section pilot valves are used prior to the main valves. Control valve design is now based on entrance loss minimization [73]. 3-d designed entrance, cross over and exit piping are used. Since the mode of operation is boiler follow turbine co-ordinated control, the response of the control valves also have a role to play in the energy efficiency.

Steels with good toughness characteristics and critical speeds below their rated speeds have been used to form integral forged steel shaft rotors without axial through-bore [72] or welded rotors [74]. Minimization of temperature fluctuations in CST powered turbines are proposed through three methods- maintaining higher internal steam temperature during shut down periods, use of heat blankets to maintain high rotor temperatures and increase in gland steam temperature [75].

Each shaft is fitted with one bearing only. Double wedge journal bearings are used for HP shaft. Plain sleeve bearings are used for IP and LP shafts. All bearing pedestals are designed as fixed anchor points.

7.0 MATERIAL UPGRADATION TO WITHSTAND HIGHER INLET STEAM CONDITIONS, MORE SEVERE RAMPING RATES, AND HIGHER FATIGUE RESISTANCE.

The common parameters of the main steam entering the steam turbine are 13-15 MPa and 535-550 °C [47], the upper limits being the creep-rupture properties of steels. Ultra supercritical plants with operating steam cycle parameters of 31.0-34.5 MPa and 566/566/566 °C (main steam/1st reheat/2nd reheat) are the favored base load and peak load solutions the world over because they are capable of delivering gross overall efficiencies (coal to power) as high as 43-44 % and better load response. In the sub-critical range the parameters are improved to 18.0 MPa and 566 ºC [76].

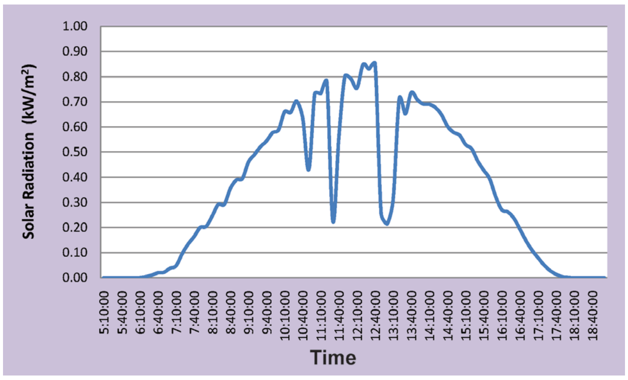

FIG. 6 VARIATIONS IN SOLAR RADIATION RESULTING IN RAMPING OF THE STEAM TURBINE

The fossil fired turbines are designed for around 9,000 cycles over a 25 year period but for CST powered systems the minimum number of cyclic operations are over 9,000/year because of the solar energy pattern alone (Figure 6). This is increased by another 20 % due to stochastic variations in solar radiation. The materials have to be improved to withstand the cyclic operations in fatigue. To improve fatigue resistance improvement in ductility, impact energy and fracture toughness is required. The normal fracture toughness is 50-60 MPa.m½ which needs to be increased to 60-70 MPa.m½. Forging heat treatment of rotors to obtain fine grained structure of rotors is called for. In the event of existence of tensile stresses, failure well below 40 % of the yield strength is possible even at temperatures far lower than the limits under a fatigue environment. High compressive state of stresses at the surface of components and especially at locations prone to failure such as blade roots, dove tail grooves, etc., must be ensured. Shot peening of components is one of the ways of creating a compressive environment.

The ramping rates for fossil fired turbines are 15 %/min. for load and turbine speed, 5 %/min for temperature and pressure. In the use of CST power plants, the natural variation of solar radiation (0-1000 W/m^2) in 6 hours is 2.8 W/m^2/min. or 0.28 %/min. This is well within the limits of the steam turbine design. Besides the natural variations, the stochastic variation of drop of 300 W/m^2 due to cloud cover (range of drop: 50 W/m^2 to 600 W/m^2) can occur over a 1 second interval.

The variation on this count is about 30 %/s to 60 %/s. The ramping rates are damped because of the inertial mass of the water-steam or oil based working fluid in the collector to around 5 %/min. Hence, these are well within the limits of the turbine design.

Increase in operating pressure can be handled by increase in the thickness of the pipe or tube to some extent (20 %) but major increase in pressure requires materials with upgraded properties. New materials with upgraded properties are inescapable for increase in operating temperature levels.

Problems arising in the previous decades in the areas of rotor cooling, turbine bypass and creep-fatigue resistance necessitating the use of materials for rotor forgings, blades and diaphragms, casings, valves, etc. which have high resistance to creep rupture and erosion-corrosion. The search for materials for high stressed components is centered on material optimization of alloying elements and preventing detrimental impurities. There is convergence of view among turbine manufacturers on use of forged steels with 9-12 % Cr, 0.1-0.2 % C and with molybdenum, niobium, vanadium & boron for steam turbine components [77]. Simultaneously improvements in steel making, forging, welding, heat treatment have resulted in new products such as G-X21CrMoV121 steel, 9-10 % Cr steels (X12 CrMoWNiVNbN 1011), T91/P91 (10 CrMoVNb 91) steels, etc., which are capable of long service life at 560-580 °C. Nickel based super alloys have now replaced ferrite steels [77]. T91 is suitable for tube /pipe materials. Inner casings, valve liners, middle rotor liners, etc. are of 10 % Cr steels. The terminal stages are coated with 13 % Cr steel spray coatings while the entering stages are coated with 80 % chromium carbide spray.

The successful materials for turbines on supercritical and ultra-supercritical steam cycles [77] which can be applied to CST energized steam turbines are:

- HPT rotor: 12CrMoVCbN (steel)
- IPT rotor: NiCrMoV (steel)
- Blades & diaphragms: 10CrMoVCbN (steel)
- Nozzles, Shells, boxes, valve bodies: 10CrMoVCb (Martensitic stainless steel castings)
- Bolts: Inconel 718

Advances in steel making such as de-oxidation and vacuum treatment for enhancing homogeneity and reduction of segregation have yielded better creep-rupture resistant steels for turbine components [72]. Improvements in welding, heat treatment, forging processes have resulted in more homogenous materials with lower probability of fatigue failure [72].

8.0 ELECTRONICS, INSTRUMENTATION& CONTROLS

Turbine cycle instrumentation is generally turbine-integral (supplied along with the turbine). Add on instrumentation and controls are also mostly turbine integral. Smart, intelligent and software controlled instrumentation has resulted in turbine supervisory instrumentation which includes the following:

- Steam parameters, turbine & bearing metal and drain oil temperatures

- Automatic turbine run up system
- Turbine speed, positions of emergency stop valve and control valve; and load limiter
- Shaft eccentricity, relative shaft vibration, absolute vibration
- Axial shift, differential expansion of rotor and cylinders, overall expansion

Endoscopes, taking cue from gas turbine instrumentation, are now available for examination of blading without opening of the casing. Expert systems are available for signature analysis, fault diagnosis, heat rate calculation and efficiency monitoring for online/offline decision support.

Hydro mechanical speed governors are now replaced by electronic speed governors whose time constant is reduced from 2-6 s to 20 ms and sensitivity of 0.01 % against 0.04 %. The governing valve jacks with mechanical movements need to be changed to pneumatic control. The overall time of electronic governors is now the operating time of the governor valves. The electronic speed governor operation is independent of the control power supply due to inbuilt power sources. Electronic governors facilitate automatic control features such as automatic synchronization of the unit with the grid. The grid code necessitates operation of the generating units composing the grid in the free governor mode of operation (FGMO) making provision for increase of load by 5-8 % in the event of drop in power frequency of the grid due to sudden drop in the power output of CST power plants. For CST energized power plants higher margins and faster responses are required to handle the load drops and gains.

The turbine load governing is gradually moving from nozzle governing (sequential/serial valve opening) to throttle governing (simultaneous/parallel opening) mode because of the better dynamic response. Majority of the turbines now operate in the variable pressure operation or sliding pressure operation modes where steam generator and the turbine stop valves are kept fully opened and steam pressure is controlled through the feed pump. The CST system operates in the turbine follow boiler coordinated mode to accommodate sudden changes in boiler output.

9.0 BALANCE OF TURBINE CYCLE-CONDENSER

Condensers are now CFD designed with wide steam passages and narrow bundles [72]. Horizontally positioned bundles provide counter current heating of condensate. Totally enclosed inlet gas systems prevents formation of gas particles in tubing. The steam in 'Church-window' type condenser modules [78] below the lower tube bundle has a lower velocity than in the upper bundle as a result of which the condensate gets heated up by 2-3 °C higher than its saturation temperature. This is negative sub-cooling (heating) which is beneficial for energy efficiency because the condensate ultimately has to be heated in the feed water heaters [78].

10.0 BALANCE OF TURBINE CYCLE-FEED WATER HEATERS

Feed water heaters in the turbine cycle have a direct effect on the capacity of total CST collector surface area. Feed water just below the saturation temperature of its operating pressure is required to be accomplished through a set of low pressure (LP) heaters (extraction steam pressure below 1.0 MPa and feed water pressure below 6.0 MPa), deaerator and HP (high pressure) heaters. The configuration of LP and HP heaters used in fossil fired plants are applicable to CST based power plants with the following differences:

- During part load operation when there is severe decay in the deaerator pressure, auxiliary pegging steam is required to be provided to stabilize the deaerator pressure.
- Heaters must be designed for high fatigue life because of cyclic and part load operation.
- T22 (2¼Cr1Mo low alloy carbon steel) is the recommended heat exchanger material from considerations of both heat transfer and fatigue life.

- Final feed water temperature must be optimized with reference to the main steam temperature.

Header type feed water heaters have an edge over the U tube in shell (tube sheet) heaters from the angle of operating flexibility, fatigue life and layout arrangement. Double tube bundle heater in horizontal configurations is also suitable for CST based cyclic environment. Issues of flow mal distribution and optimal sizing of de-superheating, condensing and sub-cooling zones have been sorted out through 3-d CFD solutions.

To handle sustained operation at partial load, intelligent system for regulation of optimal percentage of main steam into heaters would be required. Presently, the fossil fuel plants have uncontrolled extractions for all feed water heaters. Drain level control and feed water level control is also quite important in CST based systems.

11.0 CONCLUSIONS

The main conclusions of the study are as follows:

1. Concentrating solar thermal (CST) based power generation is a competing option to solar photovoltaic (SPV) from the from the stand point of area requirement and energy efficiency. Water requirement in CST which is totally absent in SPV is managed by integrating with a process already with a steam cycle like in a sugar mill power process.
2. The present steam turbine technology is capable of operation at steam conditions of 34.0 MPa and 566-600 ºC which is well within the CST system limits. The creep rupture strength requirements are easily satisfied for CST based systems.
3. The 3-d CFD designed steam turbines are capable of high isentropic efficiencies and energy efficiencies which meet the requirements of CST systems to a large extent for units of 250 MW. As one moves down in size the availability of the technology is not fully certain. In units of 100 MW and below the technology is not available as it is presently catering only to the cogeneration sector.
4. The developments in controls are adequate to meet the transient environment prevalent in the CST systems.
5. The technology for obtaining a steady steam output from CST energized plants such as hybridization with gas based systems (ISCC), thermal energy storage (TES), cogeneration and trigeneration are feasible and proven in a few cases.
6. Fatigue life of turbine needs to be improved by creating a compressive stress state and improving fatigue toughness.
7. Part load characteristics of CST powered turbine needs to be improved by designing for almost constant efficiency over the load range of 40-70 % MCR.

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NOMENCLATURE

CFD	=	Computational fluid dynamics
CST	=	Concentrating solar thermal
DSG	=	Direct solar generation
FGMO	=	Free governor mode of operation
HPT	=	High pressure turbine
IPT	=	Intermediate pressure turbine
LPT	=	Low pressure turbine
MCR	=	Maximum continuous rating (MW)
MS	=	Main steam
PLF	=	Plant load factor
PTC	=	Parabolic trough concentrators
SSC	=	Specific steam consumption (kg/kWh)
TES	=	Thermal energy storage
THR	=	Turbine heat rate (kJ/kWh)
3-d	=	3 dimensionally designed