Synthesis of Efficient Solar Thermal Power Cycles for Baseload Power Supply

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9 **Abstract**

10 Limited fossil fuel reserves and increasing greenhouse gas emissions from fossil fuels make it
11 essential to develop alternative renewable energy conversion processes to meet energy needs.

12 Advancements in renewable power production are especially important since electric power is
13 the largest consumer of primary energy resources with the highest growth rate among alternate
14 energy use sectors, and is currently responsible for greater than 40% of the global CO₂ emissions.

15 Among the renewable energy sources, solar energy is prominent due to its abundance. Here, we
16 introduce an efficient solar thermal power cycle, solar water power (SWP) cycle, with water as
17 the working fluid, which undergoes reheats between expansions. SWP cycle is developed to
18 maximize exergy efficiency, allow flexible operation and process intensification. Simulation and
19 optimization of the SWP cycles are performed in an integrated Aspen Plus/MATLAB modeling
20 environment. Modeling results predict that SWP cycle with 1 solar reheating stage has a
21 potential to generate electricity with sun-to-electricity efficiencies greater than 30% at solar heat
22 collection temperature as low as 750 K. The cycle also promises sun-to-electricity efficiencies in
23 the unprecedented range of 40 to 46% for the corresponding temperatures above 1400K. Further
24 efficiency increase is estimated by the addition of more reheating stages.

1. Introduction

Decaying fossil fuel reserves [1] and increasing greenhouse gas (GHG) emissions from fossil fuels [2] increase ever-growing interest in developing alternative energy conversion techniques using renewable energy sources to meet energy needs such as solar thermal power generation [3], synergistic biofuel production [4,5], large scale energy storage for uninterrupted renewable power supply [6,7], solar thermal hydrogen production [8]. Renewable electric power production is especially important since electric power is the largest consumer of primary energy resources, has the highest growth rate among alternate energy use sectors, and is currently responsible for >40% of the global CO₂ emissions [9]. Among the renewable energy sources, solar energy is prominent due to its abundance [10]. We envision that producing building blocks foundational to meet all basic human needs of daily existence can create a sustainable economy. Our vision targets the maximization of inherent complementary aspects of energy conversion by creating, synthesizing, and integrating sustainable process solutions to design Solar Electricity Water Food and Chemicals (SEWFAC) processes. The recently proposed Hydricity concept where solar energy is used to efficiently coproduce electricity and hydrogen is an example SEWFAC process to enable a solar centric sustainable future [11]. Here, we present a detailed analysis of an efficient solar thermal power generation process, solar water power (SWP) cycle, which is developed to maximize exergy efficiency, allow flexible operation and optimum process intensification.

Solar power generation methods are generally classified as thermal and photolytic processes. Photovoltaic (PV) cells generate electricity directly from solar light [3,12]. They are capable of absorbing only a portion of the solar spectrum and thus generating electricity from a fraction of the incoming solar rays by a finite number of band gaps [13]. Yet, PV systems are suitable for

both diffuse and direct sunlight applications, which allows them to be installed at a variety of scale and in diverse geographical locations [14]. PV cells are the most widely implemented solar power generation technique. The total global installed capacity at the end of 2014 was 177 GW, 40 GW of which was installed in 2014 [15]. To date, for lab-scale solar cells, the maximum reported STE efficiencies for silicon-crystalline is 27.6%, single-junction gallium arsenide is 29.1%, and concentrator four-junction is 44.7% [16].

Solar thermal power cycles utilize the heat collected from solar energy and thus are able to use a greater fraction of the solar spectrum. This main difference presents an advantage for solar thermal power generation processes to achieve high conversion efficiencies.

From solar ponds to current solar thermal reactors, use of solar thermal energy has always been a part of humanity's life. However, to truly benefit from solar thermal energy, heat must be collected and utilized at high temperatures, which requires the use of solar concentrators to focus the solar rays and increase the solar heat collection temperature. Concentrated solar power technologies are extensively studied in the literature for various concentrators, *i.e.* parabolic dish [17], linear Fresnel reflectors [18,19], central tower [20,21] and parabolic trough [22,23], for various power generation cycles, *i.e.* pressurized air cycles [24], molten salt cycles [25] and direct steam cycles [26,27], and for various operating regimes [28]. The total global installed capacity of CSP at the end of 2014 was 4.3 GW, 0.93 GW of which was installed in 2014 [15].

High solar energy collection and conversion efficiencies are essential to drive down the cost of solar power and to minimize the land area requirements [4,10,29]. Here, we present detailed analysis of recently proposed efficient solar thermal power generation process, solar water power (SWP) cycles. The novelties of SWP cycles, comparison of SWP cycles compared to current power cycle schemes, the effect of solar heat collection temperature, effect of reheating stages

along with the modeling approach using an integrated MATLAB and Aspen Plus environment are discussed in subsequent sections.

2. Process performance metrics

Solar power production processes are evaluated based on two metrics: the process *heat-to-electricity* (QTE) *efficiency* and *sun-to-electricity* (STE) *efficiency*. While the former metric measures the thermal energy conversion performance of power cycles, the latter evaluates the performance of power cycles from solar energy conversion standpoint.

2.1.Heat-to-electricity (QTE) efficiency

QTE efficiency refers to the fraction of heat (thermal energy) recovered as the electricity output.

In Eq. 1, W_{net} is the net electricity output, Q_{in} is the heat input to the process.

$$\text{QTE efficiency} = \frac{W_{\text{net}}}{Q_{\text{in}}} \times 100 \quad (1)$$

2.2.Sun-to-electricity (STE) efficiency and its theoretical limit

STE efficiency refers to the fraction of incident solar energy recovered as the electricity output.

In Eq. 2, W_{net} is the net electricity output, Q_{in} is the heat requirement of the process, Ω_{opt} is the optical efficiency that accounts for optical losses associated with solar concentration system and is defined as the fraction of total incident solar radiation received by the blackbody absorber, σ is the Stefan-Boltzmann constant, T is the solar heat collection temperature, I is the solar intensity and C is the solar concentration ratio. The expression in the denominator of Eq. 2 refers to the solar heat required for the process. We assume an instantaneous solar intensity of 1000 W/m^2 (AM1.5 solar intensity). Note that Eq. 2 doesn't account for the intermittency and variability

associated with solar energy. As a reference, the theoretical STE efficiency for a given solar concentration and solar intensity is given by Eq. 3 where T_a is the ambient temperature [30].

$$\text{STE efficiency} = \frac{W_{net}}{Q_{in} / W_{opt} \left(1 - \frac{ST^4}{I \times C} \right)} \times 100 \quad (2)$$

$$\text{Theoretical STE efficiency} = \left(1 - \frac{\sigma T^4}{I \cdot C} \right) \left(1 - \frac{T_a}{T} \right) \times 100 \quad (3)$$

3. Solar Thermal Power Generation

Solar thermal systems use concentrators to absorb photons of all wavelengths in the incident spectrum as heat at temperatures that are higher than ambient [31,32]. Due to the use of optical concentrators, solar thermal power systems can only be operated under direct sunlight, which imposes geographical limitations [29]. Further, these systems are anticipated to be cost-effective only as large-scale baseload power plants (100's of MW), owing to the capital costs involved in installing solar concentrators [29,33]. Yet, the highest STE efficiency reported for a solar thermal power plant with molten salt heat transfer fluid is ~22% [3] and using the pilot Stirling engine system is ~31.2% [34].

To benchmark the estimated performance of the proposed power cycles, we simulated a direct solar thermal power (DSTP) cycle as the base case with the maximum and minimum operating pressures of 200 bar and 0.04 bar, respectively. The DSTP cycle, shown in Fig. 1, consists of solar concentrators, solar receiver, high pressure turbine (HPT), medium pressure turbine (MPT), condensing turbine (CT) and a generator. DSTP cycle is a Rankine type [35] power cycle for

which solar energy is the heat source and water is the working fluid. The pumped water is both vaporized and superheated to the solar heat collection temperature, T_{shc} , by direct heat from the solar receivers. Superheated steam is consecutively expanded in HPT, MPT and CT to sub-ambient pressures. For a solar concentration of 8000 and T_{shc} in the range of 650K to 2300K, the calculated STE efficiencies are in the range of 20.1% to 38.5% with the maximum value at 1800K.

4. Solar water power (SWP) cycle

We have conceptualized a solar thermal power production cycle, Solar Water Power (SWP) cycle shown in Fig. 2. The SWP cycle is a Rankine type power cycle with high pressure water as the heat transfer and working fluid. The cycle is developed based on two fundamental considerations; (i) to maximize utilization of high temperature thermal energy for power generation and (ii) to reject heat at the minimal attainable exergetic state. Water is an ideal working fluid because it allows heat rejection close to ambient temperature and it is in liquid state at ambient temperature, the energetic cost of pressurizing is significantly decreased.

Pressurized water (200 bar for the base case) is preheated against a process stream and superheated by solar thermal energy. The superheated steam is then expanded in single or multiple expansion stages with solar reheating in between expansions. The number of reheating stages, n , is indicated in the name of the cycle SWP- n . The discharge of the last medium pressure turbine is cooled down against the pressurized water before being sent to the condensing turbine. The partially condensed low-pressure two phase water stream is fully condensed in a condenser and pumped to the operating pressure of the cycle. The SWP-1 cycle with one reheat between high pressure expansion turbines shown as dotted lines is presented in Fig. 2. In the base

SWP cycle, the exhaust of the high pressure turbine (HPT) is directly fed via line A to the medium pressure (MPT) turbine.

The proposed SWP cycle consists of a central tower and direct steam cycle type CSP plant. An attractive feature of the SWP cycle is the introduction of intermediate heat exchange between the discharge of final medium pressure turbine stage and the pressurized cold water stream. The proposed new heat exchange scheme allows to preserve high temperature heat in the system and to reject heat from the system only at ambient temperature after expansion in the condensing turbine with a minimum exergy loss. This allows the cycle to operate efficiently at various solar heat collection temperatures. Moreover, the introduced intermediate heat exchange enables the cycle to derive maximum benefit from reheating between expansion stages regardless of the temperature level of the solar collector. Without the proposed intermediate heat exchange, the reheating stages may not contribute to the same benefits especially under higher operating temperatures.

4.1.Basic SWP configuration

The stream flows for the basic SWP configuration are shown as solid lines in Fig. 2. SWP cycle is designed for high temperature operation, preferably greater than 1000 K. The temperature of the outlet of the last turbine stage is decreased to a temperature that warrants the allowable condensation to occur at the outlet of the condensing turbine. Furthermore, the preheating of the pressurized water reduces the exergy losses in the solar receiver/heat exchanger due to decrease in temperature difference between the maximum heat collection temperature and the incoming pressurized water stream. The described approach allows the rejection of heat from the system only in condensation step, which is the lowest temperature in the system. The proposed heat

exchange scheme allows the power generation to occur at higher temperatures and reaching the condensation pressures at ambient temperature despite the high starting temperature.

4.2. SWP with n reheating stages

Further STE efficiency improvements are anticipated by keeping the temperature during steam expansion in the turbines as high as possible. Hence, we implemented reheating stage(s) following the HPT, resulting in a modified cycle called the SWP- n cycle, where n represents the number of reheating stages (SWP-1 cycle is shown in Fig. 2 with a combination of solid and dashed lines). In SWP- n cycle, after the expansion in HPT, the stream is sent to the solar receiver/heat exchanger and reheated to a high temperature. The high temperature can be similar to the maximum temperature. The reheated stream is then fed to the next turbine and further expanded. The effect of number (n) of reheating stages is further discussed in subsequent sections. The flowsheet of SWP cycle with multiple reheating is given in Fig. S4.

4.3. Water Dissociation and Catalytic Combustion of Unreacted Hydrogen

At high operating temperatures water dissociation to its elements (i.e. H^+ , H_2 , O^- , O_2 and OH) might be observed. Hence, this phenomenon is included in our simulations by adding reactor models that considers equilibrium composition of the elements at each temperature levels as shown in Fig. S1. The equilibrium water dissociation at the highest temperature considered in our calculations of 2300 K is less than 0.4%. In our simulations, we allowed for equilibrium water dissociation assuming the following species to be present: H_2O , OH , H_2 , H , O_2 , and O [36].

In all cycles presented in this study, to prevent the accumulation of potentially unconverted H_2 , OH and O_2 molecules, which can cause a safety hazard, a catalytic reactor should be located

prior to the condensing turbine inlet (reactor R4 in Fig. S1). By addition of the catalytic combustion step (at temperature 385.9 K for the base case), complete recombination of H₂, OH and O₂ molecules are ensured. Hence, the discharge stream of condensing turbine does not contain any noncondensable molecules and is composed of 100% mix phase H₂O (partially condensed water vapor). The catalytic combustion is modeled as an equilibrium reactor in our simulations.

5. Process modeling, simulation and optimization

Detailed calculations and process simulations have been performed in an integrated Aspen Plus and MATLAB modeling environment [37]. The integrated simulation flow diagram is shown in Fig. 3A and the content of base and iterative model for the power cycle is shown in Fig. 3B. The Core Communication Script (CCS) developed in MATLAB allows the communication among Aspen Plus simulation files and MATLAB scripts. Transferring data between different models enables integration of various models for design and optimization of complex flowsheets [38]. The dotted box in Fig. 3A is used to optimize process parameters and not enabled for the regular runs and sensitivity analyses.

5.1.Process simulation models

Various Aspen Plus and MATLAB models are developed for the simulations, i.e. Preprocessing and Initiator model, Base model, Iterative model, and Heat Integration model. The content of each model is shown in Fig. 4 and briefly described below. The material flows for power generation processes are calculated via Base model, which contains all the process units such as turbines, reactors, heat exchangers, other than reheating and expansion prior to condensing turbine. For multiple reheating stages the Iterative model is developed and integrated with the Base model through CCS. The minimum heating and cooling requirement for the process is

calculated from the overall process heat balance using heat cascade principle of the pinch analysis method [39], which is implemented in the Heat Integration model. The minimum solar thermal energy requirement is then calculated in the Efficiency Calculation model, which takes into account the solar heat collection efficiency by calculating radiative losses and losses due to optical efficiency of the concentrators as described in Eq. 2.

The Predictive Redlich-Kwong-Soave equation of state and NBS/NRC steam tables are used to retrieve the relevant thermodynamic properties for all the calculations such as chemical equilibrium, heat of reaction and heat capacity. Process simulations are prepared in Aspen Plus V8.6 [40]. General assumptions for the calculations are given in Table 1.

5.1.1. Water equilibrium

Water dissociation can take place especially at high operating temperatures, hence, all potential chemical species, *i.e.* H₂O, OH, H₂, H, O₂ and O, are defined in the Aspen simulation and the equilibrium composition is determined by a Gibbs free energy minimization approach. A phase and chemical equilibrium calculation is carried out for every temperature change in the flowsheet as shown through units R1 through R6 in Fig. S1 of the Supplementary Information. Chemical equilibrium is calculated for zero pressure drop and zero net heat duty in the reactor. This approach also takes into account possible temperature variations due to the water dissociation or association of H, H₂, O and O₂ species. In cases of water dissociation, the temperature of the outlet of the stream decreased to the equilibrium temperature. Similarly, in case of association of components of water, the temperature of the stream is increased to the equilibrium temperature.

5.2.Process parameters and simulation initialization

Key processing parameters such as solar heat collection temperature, pressure ratio across turbines, and number of reheating stages are identified and implemented as variables that can be

controlled via CCS. A simulation is initiated by specifying variables such as solar heat collection temperature, number of reheating stages and allowable wetness of the CT. Preprocessing and Initiator model generates the initial set of specifications. For example, once the solar heat collection temperature is set the same temperature is used for all superheating and reheating heat exchangers. Similarly, the number of reheating stages determines the pressure ratio across each turbine for the general case. The Pressure Ratio (PR) of the turbines are assumed to be equal for multi reheating SWP cycles and calculated by Eq. 4 where $P_{\max}$ is the maximum pressure in the system, $P_{CT,in}$ is the condensing turbine inlet pressure and n is the number of reheating stages.

$$PR = \left(\frac{P_{\max}}{P_{CT,in}} \right)^{\frac{1}{n+2}} \quad (4)$$

The equal pressure ratio does not necessarily results in the optimum performance of the power cycles, however, it creates a fast and easily implementable basis for sensitivity analysis over number of reheating stages and temperature levels. The initial set of specifications is then applied to the corresponding models via CCS and the simulation starts.

5.3. Sensitivity analysis and optimization

The operating conditions of various units are determined using sensitivity analysis and optimization. The objective function is to maximize the STE and QTE efficiency for SWP cycle. The effects of solar heat collection temperature ($650 \text{ K} \leq T \leq 2300 \text{ K}$), operating pressure ($100 \text{ bar} \leq P \leq 500 \text{ bar}$), solar concentration ($2000 \leq C \leq 10,000$), etc. have been investigated by sensitivity analysis. An outer loop is introduced for the manipulated variable(s) and the simulation is performed as described in Section 3.2 for multiple initial sets. The applied simulation strategy allows various studies, which are unpractical using solely a process simulation software, for example, adding multiple reheating stages or running sensitivity analysis

with a number simultaneous variable changes. Process parameters such as pressure ratio of individual turbines, have been optimized using genetic algorithm to determine the process design with maximum STE efficiency for a given solar heat collection temperature and maximum operating pressure. Pressure ratio of turbines and CT inlet temperature are some of the optimized parameters. For example the optimized CT inlet temperatures are 523 K and 385 K for 1% and 10% CT allowable wetness, respectively. The simulations for power cycles are performed in Aspen Plus and the results are incorporated with MATLAB results as described in detail in previous sections.

6. Results and Discussion

STE and QTE efficiencies of DSTP, SWP and SWP-1 cycles for the base case operating conditions and variable solar heat collection temperature are shown in Figs. 5A and 5B, respectively. The solar heat collection temperature range can be divided into three zones: i) low temperature (LT) for $T_{shc} < 1000K$, ii) high temperature (HT) for $1000K \leq T_{shc} \leq 1600K$, and iii) super high temperature (SHT) for $T_{shc} > 1600K$. In LT zone, as seen in Fig. 5, the performance of DSTP and SWP cycles are exactly the same. In LT zone, the STE efficiency of the SWP cycle and the DSTP are indistinguishable due to the inability to reach low sub-ambient pressures (e.g. 0.04 bar) at the exhaust of the CT if the cycle is operated at the base case operating conditions (200 bar operating pressure). In other words, in LT zone, available thermal energy in the system limits the performance of the cycle. Hence, for this zone, operating pressures of DSTP and SWP cycles need to be modified. We have optimized the operating conditions of SWP and DSTP cycles at LT zone to maximize the power cycle efficiency. The optimum operating conditions are given in Table 2. Alternatively, the operating pressure of 200 bar can be used, in this case depending on the solar heat collection temperature, QTE and STE efficiencies are 1% to 35% lower than these cycles' optimized conditions. This poor performance is due to the fact that the

discharge pressure of CT should be high (>1 bar) not to violate allowable operating conditions, *i.e.* outlet temperature and/or outlet wetness. Hence, to prevent these violations, CT discharge pressure has been adjusted, which resulted in lower performance of power cycles. However, this limitation can be overcome by adding reheating stages. As seen in Fig. 5, addition of one solar reheating stage (SWP-1) cycle provides 10% to 20% increase in efficiency. The proposed reheat schemes of the SWP-1 cycle increase the portion of solar heat that is converted to electricity by allowing generation of electricity at higher temperatures and lowering the condensing turbine outlet pressures for temperatures lower than 1000 K. In the LT zone, SWP-1 has about 3% better efficiency than the other two power cycles.

As the solar heat collection temperature is increased the effect of proposed heat exchange scheme shows its benefit. For HT zone, the efficiency of SWP cycle is 1.3% to 4.6% higher than the efficiency of DSTP cycle at the same solar heat collection temperature. HT zone is mainly pressure driven, meaning that the thermal energy at high temperature in the system is in excess and the heat can be rejected at a higher than ambient temperature in case of DSTP cycle, or with the proposed heat exchange scheme it can be rejected at ambient temperature as in the case of SWP and SWP-1 cycles. The STE and QTE efficiencies of SWP-1 cycle is 7.2% - 9.9% higher than the efficiencies of SWP cycle at the same solar heat collection temperature. The cooling curves and Carnot cooling curves of DSTP, SWP and SWP-1 cycles at three different temperatures in HT zone are shown in Fig. 6.

Figs. 5 A-C depict the variation of heat exchange between hot and cold streams in HT zone. At 1000 K as shown in Fig. 6A, DSTP and SWP cycles behave almost the same and requires similar solar heat to generate 100 MW of net electricity. As temperature increases, as seen from Fig. 6 B and C, the behavior of DSTP and SWP cycle begins to differentiate. The extra thermal energy

available in the system is used to preheat the pressurized water in SWP and SWP-1 cycles, which result in reduced solar heat input to generate the same amount of electricity. Carnot cooling curves shown in Figs. 5 D-F, demonstrate the utilization of exergy for each cycle at different temperatures [41]. Carnot factor in Y-axis is the Carnot efficiency at the solar collector temperature. The area between hot and cold curves represents the exergy losses from the power cycle. As seen from Figs. 6 D-F, as temperature increases the exergy losses decrease. SWP cycle improve exergy efficiency especially at temperatures higher than 1000K. SWP-1 cycle's exergy losses are lower than that of SWP cycle, primarily due to its ability to recover high temperature thermal energy as electric power.

The maximum STE efficiency of all three power cycles is achieved in SHT zone at 1800K, 1900K and 1850K for DSTP, SWP and SWP-1 cycles, respectively. To understand why the efficiency is maximum at an intermediate temperature, observe that there are two competing effects: a higher temperature imparts a higher value of exergy to a unit quantity of heat, while a higher temperature blackbody absorber (*i.e. solar receiver in Figs. 2 and 3*) irradiates a greater fraction of the incident solar radiation. The efficiency gap between DSTP and SWP cycle is significantly increased in SHT zone reaching 9.7%. However, as observed from Fig. 5, the QTE efficiency of SWP and SWP-1 cycles come closer and the calculated improvement is 4.5% to 7%.

As described in previous sections, the pressure ratio across turbines is assumed to be equal but this assumption might not give the optimum efficiency for the power cycles. To demonstrate the effect of this assumption, we have optimized the SWP-1 cycle to determine the pressure ratio across each turbine that will yield the maximum QTE efficiency. As expected, the maximum QTE efficiency is reached at different pressure ratios for different solar heat collection

temperatures. For equal pressure ratio outlet of HPT turbine is supposed to be 15.5 bar however, as shown in Table 3, depending on the temperature the optimum pressure varies. The efficiency of optimized cycles is shown in Fig. 5 with crossed circles ($\otimes$) and it is almost the same as the base case results. Hence, the equal pressure ratio assumption is suitable for the purpose of this study.

Further efficiency improvements are anticipated by keeping the temperature during steam expansion in the turbines as high as possible. Hence, the implementation of reheating stages maintains the temperature level higher for each expansion stage. In addition to the benefit of ensuring the attainment of the specified low condensing turbine outlet pressures at low solar heat collection temperatures, keeping the temperature at higher levels increase the power production efficiency.

Fig. 7 summarizes the estimated STE efficiency and QTE efficiency for the SWP cycle involving various reheating steps ($n = 1-5, 10$) relative to the theoretical efficiency values for the particular solar heat collection temperature. It should be noted that a significant fraction of the solar irradiation is lost in optical concentration and blackbody re-radiation. This can be observed from the difference between STE and QTE efficiencies and Fig. 8 summarizes the fraction of solar energy lost as a function of solar heat collection temperature. Therefore, any improvement in solar heat collection technology will further improve the STE efficiency of the solar power cycles.

Fig. 9 presents the cascading energy losses involved in the solar power generation for 1000 K, 1250 K and 1600 K temperature levels. For a 100 units of incident solar radiation, 79.43, 78.61 and 76.28 units of thermal energy can be recovered for temperatures 1000 K, 1250 K and 1600 K, respectively. Approximately 25 units of energy is lost due to inefficiencies in optical

concentration and blackbody re-radiation. From the approximately 75 units of thermal energy, reversible power can be generated with energy efficiency of 70.18%, 76.15% and 81.25% for temperatures 1000 K, 1250 K, and 1600 K, respectively. Finally, when the equipment efficiencies are taken into account (turbine efficiency of 80% is assumed in this study), the STE efficiency of the various power generation cycles are lower than 45%, because of the inefficiencies associated with the finite temperature differences during heat exchange between the process streams.

For solar heat collection temperatures higher than 700 K, the STE efficiency of the SWP-1 cycle is estimated to be greater than 30%. This in itself is remarkable as it allows the possibility of achieving unprecedented STE efficiencies of greater than 30% at temperatures that are within the limits of operation of commercially available steam turbines. Additionally, if the materials of construction, particularly for the steam turbines, are modified to enable operation at higher temperatures above 1250 K the SWP cycles can achieve STE efficiencies that are greater than 40%. As solar reheating number is increased, the STE efficiency improves, albeit with diminishing STE efficiency improvements. The STE efficiency of 46.2% is achieved at 1800 K for the SWP-10 cycle.

6.1.Improvements from Advancements in Equipment Technology

Some technological advancement can allow the implementation of the presented power cycles and further improve the achievable STE and QTE efficiencies. As discussed in the previous sections, temperature is the most critical parameter to reach high efficiencies, hence the availability of materials and equipment to handle high temperatures will be a key step for SWP cycles. Similarly, turbine technologies that can handle higher pressures and allow high wetness

will improve the performance of SWP cycle. Solar concentration plays an important role in achieving high temperatures and it directly affects STE efficiency of the solar power cycles as described in Eq.2.

6.1.1. Availability of materials for high temperature applications

The implementation of the processes at the preferred temperature level depends on the availability of materials for equipment construction. Commercially available One Atmosphere Boiler [42] can generate steam up to 1573 K. Current advanced gas turbine engines are operated at turbine inlet temperature of 1473 K – 1773 K and with recently developed cooling techniques the allowable inlet temperature of turbines are expected to reach above ~2000 K [43]. Hence, the system that operates at solar heat collection temperature in LT and HT zones seems to be promising for efficient solar thermal electricity production solution.

6.1.2. Turbine technology improvement

Improvement in turbine technologies would further improve the STE efficiency of SWP cycles. Water is an ideal working fluid for the proposed power cycles, as it is a liquid at ambient conditions and facilitates operation of the system at high pressures while consuming minimal energy consumption for pumping. The STE efficiency results in Figs. 5 and 7 are based on pumping the feed water to a pressure of 200 bar. With increasing feed water pressure, the STE efficiency for each power cycle could be further improved. The sensitivity analysis on the operating pressure is shown in Fig. 10. Up to 3% QTE efficiency improvement is calculated for the pressure range included in the study. The CT outlet conditions significantly affect the performance of the cycle as it determines the heat rejection to the cooling water. The variation in STE efficiency of SWP-1 cycle for allowed CT outlet wetness of 1% and 10% is shown in Fig.

11. Current advanced CTs can withstand up to 14% wetness, however this value is lower for the most of the CTs in operation [44]. The drop in allowed wetness from 10% to 1% decreases the STE efficiency of the cycle about 2% irrespective of the solar heat collection temperature. Similarly, the effect of CT outlet pressure is depicted in Fig. 12 for 0.04 bar - 0.1 bar. Maximum 2% QTE efficiency change is estimated between 0.04 bar and 0.1 bar CT outlet pressure cases. Current steam cycles are generally operated with maximum steam temperatures of 650-900 K and pressures in the range of 30-140 bar [45]. In addition to these well-known Rankine cycles, high temperature water cycles with CO₂ recovery have been discussed in the literature using fossil fuel-fired derived heat to operate at high temperatures of ~1625 K (Graz-cycle, Matiant-cycle, water cycle, etc.) [46–50]. Turbine technologies designed for high pressure, high temperature operations and condensing turbines with high allowable wetness will ensure the efficient conversion of thermal energy into electric power as demonstrated in this study.

6.1.3. Solar Concentration

The STE efficiency changes for different values of solar concentration ratio according to Eq. 2. As an example, Fig. 13 illustrates the STE efficiency of the SWP-1 cycle with 200 bar maximum pressure for various solar concentration ratios between 2,000 and 10,000.

6.2. Energy storage

Solar energy is available only during a small fraction of a twenty-four-hour day. Hence, to operate a baseload solar power plant, an integrated energy storage solution is indispensable. The energy storage technologies can be divided in two general categories: i) Energy storage from electricity, ii) Direct energy storage. Energy storage techniques such as pumped hydroelectric, batteries, compressed air energy storage lie in the first category where the energy input to the storage facility is electricity [51–55]. The options that underlie in the second category such as

carbon storage cycle, thermal storage and chemical storage can take a non-electricity form of energy usually thermal energy for energy storage [51,54,56,57]. The second category reduces the number of conversion steps. The proposed power cycle especially operated at high temperatures brings an advantage: Solar thermal energy recovered at high temperatures could also be utilized to drive thermochemical processes like water-splitting reaction [30,32,58–60]. SWP cycles can be synergistically integrated with solar thermal hydrogen production processes or any other chemical facilities not necessarily for energy storage.

6.3. Power cycle with other energy sources

Although SWP cycle is primarily designed as a concentrated solar power plant, the power cycle can be operated by any heat source such as nuclear energy, natural gas or hydrogen. For example, an interesting scenario is the production and storage of hydrogen as an energy storage medium when solar energy is available and then power generation using the stored hydrogen and oxygen in the same power cycle by replacing the heat source from solar energy with hydrogen oxy-combustion [11]. The attractive feature of this integration is that it continues to use water as the working fluid. Such a cycle allows for the uninterrupted operation of rotating machines, heat exchangers, etc. that were deployed in the SWP cycle. This approach can overcome startup and shutdown impacts on solar systems [61]. Similarly, any fuel can be used in a modified power cycle scheme that runs at high temperature.

7. Conclusion

We presented detailed analysis of a new family of solar thermal power cycles, solar water power cycle (SWP), which can reach high *sun-to-electricity* and *heat-to-electricity* efficiencies. These cycles are quite efficient with SWP-1 having the potential to provide STE efficiencies greater than 30% at heat collection temperatures above 700 K and greater than 40% for solar heat

428 collection temperatures above 1400K. Although STE efficiency of SWP cycles are higher at high
429 temperatures, the implementation at the optimal conditions might require technological
430 advancements. However, the cycle also allows the construction of a power production process at
431 low collection temperatures using current technology that has high STE efficiency.

432 The novelty of the process is the addition of an intermediate heat exchange step between the
433 discharge of the last turbine stage and the pumped cold water. This proposed heat exchange
434 scheme enables the use of the same power cycle for a range of operating temperatures especially
435 at higher temperatures. Moreover, it makes the reheating stages between expansion stages more
436 effective. The reheating of expanded streams have been studied in the literature, however, the
437 proposed heat exchange truly allows the exploitation of the benefit due to reheat by enabling the
438 power generation at the highest possible temperature, which increase the efficiency of the power
439 cycle.

440 Another advantage of SWP cycle is the efficient use of high temperature heat for power
441 production, which creates numerous opportunities to integrate the power cycle with chemical
442 plants such as solar thermal-driven hydrogen production. One potential application is the storage
443 and utilization of hydrogen for energy storage to supply electricity round-the-clock.

444 Solar energy is the most important component of a sustainable future due to its abundance and
445 the availability of a variety of harnessing opportunities. Solar thermal technologies are
446 particularly promising since thermal energy is a well-known form of energy with many
447 applications in place and it was being created using various energy sources such as fossil
448 resources for centuries. In this realm, with more than 20% increase in the global installed
449 capacity in 2014, solar thermal power production plants are rapidly getting implemented and will

450 be implemented. Solar processes designed for the optimal utilization of resources and flexible
451 operation; such as the presented SWP cycle have a critical role to play in the envisioned Solar
452 Electricity Water Food and Chemicals (SEWFAC) concept, which aims to produce building
453 blocks foundational to meeting all basic human needs of daily existence.

454 **Acknowledgments:**

455 The authors would like to acknowledge Prof. Mohit Tawarmalani for valuable help and
456 numerous discussions. Research supported as part of the Center for Direct Catalytic Conversion
457 of Biomass to Biofuels, an Energy Frontier Research Center funded by U.S. Department of
458 Energy, Office of Science, Basic Energy Sciences (BES), under Award #, DE-SC SC0000997;
459 by the National Science Foundation Solar Economy IGERT (0903670-DGE).

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Figure Legends

Fig. 1 Direct Solar Thermal Power (DSTP) cycle. HPT: High pressure turbine, MPT: Medium pressure turbine, CT: Condensing turbine, G: Generator.

Fig. 2 Solar Water Power (SWP) and Solar Water Power with 1-solar reheating stage (SWP-1) Cycles. HPT: High pressure turbine, MPT: Medium pressure turbine, CT: Condensing turbine, G: Generator.

Fig. 3 A: Simplified flow diagram of integrated MATLAB and Aspen Plus modeling environment for simulation and optimization. B: Schematic of the content of Base (black lines) and Iterative (red split lines) Aspen Plus models.

Fig. 4 Detailed contents of each model in the integrated MATLAB and Aspen Plus modeling environment.

Fig. 5 STE and QTE efficiencies of DSTP, SWP and SWP-1 cycle involving for different solar heat collection temperatures at concentration factor of 8000. The maximum STE efficiencies of DSTP, SWP and SWP-1 cycles are shown with void black square, full blue square and red full circle, respectively. STE and QTE efficiencies of SWP-1 for optimized HPT pressure ratio are shown with gray void crossed circles.

Fig. 6 Cooling curves of DSTP, SWP and SWP-1 cycles at different solar heat collection temperatures in HT zone. A: Cooling curve at 1000K, B: Cooling curve at 1250K, C: Cooling curve at 1600K. D-F: Carnot cooling curves where Carnot factor on y-axis is the Carnot efficiency at the solar heat collection temperature. D: Carnot cooling curve at 1000K, E: Carnot curve at 1250K, F: Carnot curve at 1600K (Solar concentration= 8000).

Fig. 7 STE and QTE efficiencies of SWP cycle involving various reheating stages for different solar heat collection temperatures.

627 **Fig. 8** Energy losses due to solar energy collection as a function of solar heat collection
628 temperature.

629 **Fig. 9** Energy losses involved in each step of solar heat collection and power generation.

630 **Fig. 10** QTE efficiency of the SWP-1 cycle for different solar heat collection temperatures and
631 different operating pressures

632 **Fig. 11** STE efficiency of the SWP-1 cycle for different solar heat collection temperatures and
633 allowable condensed liquid ratio at the outlet of the condensing turbine

634 **Fig. 12** QTE efficiency of the SWP-1 cycle for different solar heat collection temperatures and
635 different condensing turbine outlet pressures

636 **Fig. 13** STE efficiency of the SWP-1 cycle for different solar concentration ratios.

Table 1 Parameters for the STE efficiency calculations and modeling assumptions

	Units	Value
Solar energy availability (base case)	hr	4.8
Day	hr	24
Solar intensity, I	W/m ²	1000
Solar concentration ratio, C		2000-10,000
<i>for the base case</i>		8000
Optical efficiency, Ω_{opt}	%	80
Modeling assumptions		
Minimum temperature of approach during heat exchange for temperatures		
<i>ambient condenser</i>	ΔT_{min}	5K
<i>below 600 K</i>	ΔT_{min}	10 K
<i>above or at 600 K</i>	ΔT_{min}	28 K
Isentropic turbine efficiency		
High Pressure Turbine (HPT)	%	80
Medium Pressure Turbine (MPT)	%	80
Condensing Turbine (CT)	%	80
Isentropic compressor efficiency	%	80
Pump Efficiencies	%	95
Pressure ratio across H ₂ /O ₂ permeable membrane		5
Cooling water pressure	atm	1
Cooling water temperature	K	293
Pressure drop	bar	0
Operating pressure (base case)	bar	200
Solar heat collection temperature range		
<i>low temperature</i>	K	650-1000
<i>high temperature</i>	K	1000-1600
<i>super high temperature</i>	K	1600-2300
Condensing turbine outlet liquid water fraction	%	<14%
<i>for the base case</i>	%	10%
Condensing Turbine outlet pressure	bar	0.04

Table 2 Optimum operating pressures of DSTP and SWP cycles at 650 K – 950 K solar heat collection temperatures

T _{shc} [K]	Operating Pressure [bar]	Condensing turbine outlet pressure [bar]
650	22	0.05
700	34	0.05
750	50	0.05
800	70	0.05
850	103	0.05
900	144	0.05
950	199	0.05

Table 3 Optimum pressure ratios of HPT and corresponding QTE efficiencies of SWP-1 cycle at 800K - 1600 K solar heat collection temperatures.

Temperature [K]	Outlet pressure of HPT [bar]	QTE efficiency [%]
800	16.9	39.59
900	17.4	42.42
1000	16.8	44.91
1100	16.3	47.15
1200	16.1	49.19
1300	16.0	51.08
1400	15.9	52.82
1500	15.8	54.44
1600	15.7	55.94

Fig. 1

Fig. 2

Fig. 3A

Fig. 3B

Fig. 4

Fig. 5A

Fig. 5B

Fig. 6A

Fig. 6B

Fig. 6C

Fig. 6D

Fig. 6E

Fig. 6F

Fig. 7A

Fig. 7B

Fig. 8

Fig. 9

Fig. 10

Fig. 11

Fig. 12

Fig. 13