Thermo-economic evaluation of an innovative direct steam generation solar power system using screw expanders in a tandem configuration

Pengcheng Li^{a,b}, Jing Li^{c,*}, Ronghui Tan^d, Yandong Wang^e, Gang Pei^f, Bin Jiang^a, Jingchun Tang^a

^a*School of Automobile and Traffic Engineering, Hefei University of Technology, 193 Tunxi Road, Hefei, China*

^b*Y&C ENGINE CO.,LTD.NO.2 Eqiao Road, Sanshan, Wuhu, China*

^c*Department of Architecture and Built Environment, University of Nottingham, University Park, Nottingham NG7 2RD, UK*

^d*Jiangxi HuaDian Electrical Power CO. Ltd, 398 Saiwei Road, Xinyu, China*

^e*Hefei General Machinery Research Institute, 888 Changjiang Road ,Hefei, China*

^f*Department of Thermal Science and Energy Engineering, University of Science and Technology of China, 96 Jinzhai Road ,Hefei, China*

Abstract: Tandem screw expander (SE) technology is a promising solution for large volume ratio situations. Tandem SE driven direct steam generation (DSG) system is first proposed for distributed solar thermal power generation. Steam accumulator is adopted for storage due to its moderate heat source temperature. Compared with the cascade steam-organic Rankine cycle (SORC) system, the novel tandem system has simpler structure, easier control strategy and lower technical requirements, less operating and maintenance fees, more stable power output and higher security. Thermodynamic analysis and economic evaluation of the novel system are conducted

based on some parameters of a recently constructed tandem SE project. Steam Rankine cycle (SRC) efficiency of 18.49% is achieved by employing built-in volume ratio ($r_{v,b}$) of 3 for the high-pressure SE (SE1) and $r_{v,b}$ of 7 for the low-pressure SE (SE2). The cost-effectiveness of the system is improved as the power capacity and heat storage time increase. Levelized electricity cost (LEC) of 0.118 \$/kWh and payback period (PP) of 10.48 years are obtained for 1 MW tandem plant with 6.5 h heat storage. The cost of parabolic trough collectors accounts for nearly half of the total investment while the accumulators occupy less than 7%. The cost of SE2 is approximately seven times that of SE1 due to the larger design outlet volume flow rate and rotor diameter.

Keywords: screw expander; tandem system; levelized electricity cost; payback period

*Corresponding author. Tel./Fax: +86 551 63607517. E-mail: lijing83@ustc.edu.cn

1. Introduction

Screw expander (SE) has favorable match with parabolic trough collector (PTC) for direct steam generation (DSG) solar thermal power applications. The advantages are embodied in the following aspects. First, superheater in conventional island is eliminated due to the high tolerance for multi-phase expansion of SE (allowance for steam, vapor, liquid and their mixtures). Second, excellent part-load behavior of SE enables system to run steadily in off-design conditions. Third, moderate operating temperature ($<250^{\circ}\text{C}$) and pressure ($<4\text{MPa}$) facilitate lower technical requirements in heat collection and storage. Fourth, mass-produced small scale prototypes (range from several kW to megawatt) make distributed solar power generation easy to implement.

However, low built-in volume ratio ($r_{v,b}$) and limited inlet/outlet pressure difference are the drawbacks SE-driven system has to handle. It means the available energy of high temperature fluid cannot be fully extracted through a single SE. One solution is to couple with a turbine-related organic Rankine cycle (ORC) in the bottom [1]. The ORC fluid of low boiling point shows good reaction to the fluctuation in ambient temperature. The detailed fundamentals and superiorities of the cascade steam-organic Rankine cycle (SORC) over PTC-ORC and conventional PTC-steam Rankine cycle (SRC) have been illuminated [1]. Modeling and optimization of the cascade system with respect to the characteristics of SE are carried out [2]. Water is appropriate storage medium on account of the moderate operating temperature. Mathematical model of the steam accumulator is built and the cost of steel is less than one-third of the oil's [3].

Besides cascade arrangement, SEs in a tandem configuration (SE1 connected with SE2) as depicted in Fig. 1 is another alternative solution for the cases characterized by large, but not extreme volume ratios. The binary phase steam at the outlet of SE1 has relatively high temperature and pressure. The exhaust steam then goes into SE2 and continues to expand and export power. Similarly with multistage expansion in turbine, the exergy existing in the exhaust steam is further harnessed.

At present literature on tandem positive displacement (particularly screw) expanders is rare. Georges et. al designed two scroll expanders in series for solar ORC power plant. Thermal efficiency of 11.36% was reached for evaporation and condensation temperatures of 140 °C and 35 °C, respectively. The small high pressure expander cost was only 33% of the low pressure expander [4]. Quoilin et. al simulated

double-stage scroll expander in PTC-ORC system by connecting validated component submodels. An overall electrical efficiency between 7% and 8% was obtained and the most efficient fluid was Solkatherm. A dynamic model was needed and the behavior of the storage tank should be modeled to perform a one-year simulation [5]. Astolfi investigated three plant layouts of PTC-ORC: single screw, tandem screw and flash trilateral cycle. By comparison with the single stage expansion, power output increased by more than 19% with the tandem configuration since the condensing temperature was reduced by 5 °C and a higher enthalpy drop was available [6]. But the in-depth thermodynamic optimization process and thermo-economic evaluation were not conducted. Read et. al simulated mass flow rate, intermediate pressure, discharge pressure ratio, isentropic efficiency, overall adiabatic efficiency, total shaft power, cycle efficiency and net power output for a flash trilateral cycle using a two-stage expander. The overall conversion efficiency was predicted to be similar to those of the Smith cycle (consists of a high-pressure screw and a low-pressure turbine) and conventional saturated vapor ORC [7].

Notably, tandem SE technology has some technical challenges that need to be addressed. On the one hand, axial thrust is higher and vibration is greater than those of single expander case, which calls for higher processing and assembly accuracy. The coincidence of axes is thus of prime importance. On the other hand, large rotor diameter is required for the low pressure expander (i.e. SE2 as illustrated in Fig. 1) due to the large volume flow rate. The operating pressure of SE2 is lower than that of SE1, and the corresponding working fluid's density inside SE2 is smaller. Consequently, volume

flow rate through SE2 is greater than that of SE1 on account of constant mass flow rate. Greater volume flow rate results in larger rotor diameter and thus SE2 is larger than SE1.

In the past few years, progress on the tandem technology has been made with solutions to tackle the above challenges. Some demonstration projects in the field of waste heat recovery have been built and the parameters are listed in Table 1 [8]. Fig. 2 is a photograph of the third engineering project in Table 1. It is located in India and constructed by Jiangxi Huadian Electric Power Co., Ltd. The high and low pressure expanders are connected by an intermediate coupling. Successful applications of the tandem technology in industrial waste heat utilization lay a solid foundation for solar thermal electric generation.

In this work, tandem configuration is applied for DSG power generation as shown in Fig. 1. To the best knowledge of the authors, it's the first time that SEs in series have been combined with a solar DSG system. Water is heat transfer, storage and working fluid. Recirculation pump consumption and extra investment associated with secondary heat transfer fluid such as thermal oil are eliminated. The collector tubes benefit from approximately 1 MPa pressure (the specific parameters are provided in Table 1), which is much lower than the commercial turbine-related PTC plants (6.5-10 MPa) [9-11]. Higher reliability and longer lifetime are expected for the tubes.

Tandem system has following advantages in comparison with the cascade SORC system. First, simpler structure. Single cycle saves a set of speed governing device, a set of control equipment, a grid-connected system and a heat exchanger (because of the

absence of steam/organic fluid heat transfer). A gearbox in turbine-related cycle is left out and the two SEs share a generator. Second, easier control strategy and lower technical requirements. SE is able to start up and shut down more quickly than the turbine in SORC. Drawbacks of cavitation phenomenon, strict sealing, high cost and low global pumping efficiency associated with organic fluid pump are inexistent due to the omission of the bottom ORC. Third, no special warm-up, less faults from over-speeding and turning, lower operating and maintenance fees than turbine on account of SE's solid and simple rotor structure. Fourth, annual power generation is more stable because the condensation temperature of steam fluctuates slightly in different seasons. Fifth, problems of toxicity, flammability, ozone depletion potential and global warming potential accompanied with organic fluid in cascade SORC are eliminated.

The structure of the work is presented in Fig. 3. Thermodynamic analysis and economic investigation of the solar DSG tandem SE system are carried out. Its performance is identified with a variable heat storage capacity as well as system scale. Furthermore, investment proportion of each unit is clarified.

2. Mathematical models

2.1. Thermodynamic models

2.1.1. Solar energy collection efficiency

A type of PTC installed in the USA with up to 2700 m² of aperture area is referenced [12]. The performance formula of a single PTC is [13]:

$$\eta_{PTC}(T) = 0.762\kappa - 0.2125 \times \frac{T - T_a}{G_b} - 0.001672 \times \frac{(T - T_a)^2}{G_b} \quad (1)$$

where T is collector inlet temperature; κ is incidence angle modifier and is

expressed by

$$\kappa = \cos \varphi + 0.0003178\varphi - 0.00003985\varphi^2 \quad (2)$$

φ is incidence angle and its calculation process is shown in Section 2.1.2.

Thousands of collectors are usually adopted and the temperature difference between neighboring collectors is supposed to be small. To calculate the overall efficiency of PTC, it is reasonable to assume that the average operating temperature of the collector changes continuously from one module to another.

For liquid water, in order to reach an outlet temperature T_{out} with an inlet temperature T_{in} , the required collector area is obtained by

$$A_l = \int_{T_{in}}^{T_{out}} \frac{m \cdot C_p(T)}{\eta_{PTC}(T) \cdot G_b} dT \quad (3)$$

where m is mass flow rate of water.

Heat capacity of water can be expressed by a first order approximation:

$$C_p(T) = C_{p,0} + \alpha(T - T_0) \quad (4)$$

Where $C_{p,0}$ is heat capacity corresponding to a reference temperature T_0 .

With $c_1 = 0.2125 / G_b$, $c_2 = 0.001672 / G_b$, the collector area according to Eqs. (1), (3) and (4) is calculated by

$$A_l = \frac{m}{c_2 G_b (\theta_2 - \theta_1)} \left[(C_{p,a} + \alpha \theta_1) \ln \frac{T_{out} - T_a - \theta_1}{T_{in} - T_a - \theta_1} + (C_{p,a} + \alpha \theta_2) \ln \frac{\theta_2 - T_{in} + T_a}{\theta_2 - T_{out} + T_a} \right] \quad (5)$$

Eq. (5) is the analytic solution to the formula of integration of Eq. (3). c_1 and c_2 are two defined intermediate parameters. θ_1 and θ_2 are the arithmetical solutions of Eq. (6) ($\theta_1 < 0$, $\theta_2 > 0$).

$$0.762 - c_1 \theta - c_2 \theta^2 = 0 \quad (6)$$

$$C_{p,a} = C_{p,0} + \alpha(T_a - T_0) \quad (7)$$

θ_1 and θ_2 can be determined once G_b is known since Eq. (6) is a common quadratic function.

Collector efficiency in liquid phase region is

$$\eta_{PTC,l} = \frac{m\Delta h_l}{G_b A_l} \quad (8)$$

The solar field contains steam-liquid mixture. Overall efficiency is

$$\eta_{PTC} = \frac{Q}{G_b(A_l + A_b)} = \frac{\Delta h_l + \Delta h_b}{\frac{\Delta h_l}{\eta_{PTC,l}} + \frac{\Delta h_b}{\eta_{PTC,b}}} \quad (9)$$

where Δh_l and Δh_b are the enthalpy increments of water in liquid and binary phase regions. $\eta_{PTC,b}$ can be easily calculated since the temperature is constant.

2.1.2. Calculation of the incidence angle

When the PTC is north-south oriented and with east-west tracking, the incidence angle is calculated by [13]

$$\cos \varphi = \sqrt{\sin^2 \alpha_s + \cos^2 \delta \sin^2 \omega} \quad (10)$$

where α_s is the solar altitude angle ($^\circ$); δ is the solar declination ($^\circ$),

; ω is the solar hour angle ($^\circ$).

For horizontal PTCs, α_s is determined by [14]

$$\sin \alpha_s = \sin \phi \sin \delta + \cos \phi \cos \delta \cos \omega \quad (11)$$

where ϕ is the geographic latitude ($^\circ$), $-90^\circ \leq \phi \leq 90^\circ$;

is expressed by

$$\delta = 23.45 \sin \left(360 \frac{284 + n}{365} \right) \quad (12)$$

where n represents the n^{th} day in a year, .

ω is calculated by

$$\omega = 0.25(AST - 720) \quad (13)$$

where AST is the apparent solar time (min).

AST is expressed by

$$AST = LST + ET - 4(SL - LL) \quad (14)$$

where LST is the local standard time (min); ET is the equation of time (min); SL

is the standard meridian for the local time zone ($^{\circ}$); LL is the local longitude ($^{\circ}$),

$$-180^{\circ} \leq LL \leq 180^{\circ}.$$

is calculated by

$$ET = 9.87 \sin 2B - 7.53 \cos B - 1.5 \sin B \quad (15)$$

2.1.3. Thermodynamic efficiency

Thermal power efficiency (η_T) is

where

$$\eta_{SRC} = \frac{(h_1 - h_{2s})\epsilon_{SE1}\epsilon_g + (h_2 - h_{3s})\epsilon_{SE2}\epsilon_g - (h_{5s} - h_4)/\epsilon_p}{h_1 - h_5} \quad (18)$$

2.1.4. Part-load behavior of SE

$r_{v,b}$ is defined as

$$r_{v,b} = \frac{v_{out,b}}{v_{in,b}} \quad (19)$$

The operating pressure ratio (r_p) is

(20)

There are four types of losses in SE operation: (i) loss due to mismatch of the r_p , (ii) fluid leakage loss, (iii) loss due to thermodynamic irreversibilities and (iv) mechanical friction loss from the rotating shaft. For each stage of the work loss, an efficiency term can be defined to account for it. They are theoretical, leakage, thermodynamic and mechanical efficiencies for (i)-(iv), respectively [15]. The actual overall isentropic efficiency can thus be defined as:

$$\varepsilon_{os} = \varepsilon_{Th} \varepsilon_L \varepsilon_{TM} \varepsilon_M = \varepsilon_{Th} \varepsilon_D \varepsilon_M \quad (21)$$

where

(22)

ε_{Th} is determined by the characteristic of SE, and it has a constant value for a specific SE. By definition, the theoretical efficiency is given as:

$$\varepsilon_{Th} = \frac{W_{TD}}{W_{TI}} = \frac{(1 - r_{v,b}^{1-\gamma}) + (\gamma - 1)(1 - r_{v,b} / r_p)}{\gamma(1 - r_p^{1-\gamma/\gamma})} \quad (23)$$

where γ is the isentropic index. It varies according to the working fluid and its state.

γ is 1.13 for dry saturated steam.

The thermodynamic work output W_{TM} , can be estimated by summing the shaft work output and the frictional work, i.e.

$$W_{TM} = W_S + W_F \quad (24)$$

where W_F is a direct function of the shaft speed N . It is generally assumed to be a constant if the rotation speed is unchanged. The diagram efficiency is given by the ratio

of the thermodynamic work to the theoretical diagram work, i.e.

$$\varepsilon_D = \frac{W_{TM}}{W_{TD}} = \frac{W_s + W_F}{m_t P_{su} V_s \left[\frac{(1 - r_{v,b}^{1-\gamma})}{\gamma - 1} + (1 - \frac{r_{v,b}}{r_p}) \right]} \quad (25)$$

The definition of ε_D includes the effect of leakage and thermodynamic irreversibility. The mass flow rate related to leakage in an expander can be estimated using the ideal-gas choked flow model with a given leakage flow area [16]. Therefore, it is expected that the expression of diagram efficiency is similar to that in the conventional turbomachines. Increment in $r_{v,b}$ has a negative effect on the diagram efficiency, but the efficiency variation is quite slight at high r_p [17]. Peak isentropic efficiency ($\varepsilon_{os,p}$) can be achieved when $\varepsilon_{Th,p}$ is 1.

$$\varepsilon_{os,p} = \varepsilon_{Th,p} \varepsilon_D \varepsilon_M \quad (26)$$

Combine Eqs. (21), (23) and (26), ε_{os} can be estimated as:

$$\varepsilon_{os} = \varepsilon_{os,p} \varepsilon_{Th} = \varepsilon_{os,p} \cdot \frac{\varepsilon_D}{\varepsilon_{D,p}} \frac{(1 - r_{v,b}^{1-\gamma}) + (\gamma - 1)(1 - r_{v,b} / r_p)}{\gamma(1 - r_p^{1-\gamma/\gamma})} \quad (27)$$

Thus the actual SE efficiency is affected by r_p , $\varepsilon_{os,p}$, , working fluid and state.

2.1.5. Condenser area

The cost of large heat exchangers is mainly contributed by the exchanger area and hence total amount of materials in use [18,19]. HTRI software is the industry's most advanced thermal process design and simulation software [20], and it is used to estimate the heat transfer area.

The condenser is shell and tube type with single shell and double tube pass configuration. Single segmental baffle type is assumed. Hot fluid is located in shell side while cold fluid is in tube side. Shell and tube heat exchanger shows good flexibility in

terms of great availability of construction materials, high value of both heat power transferred /weight and volume ratio and finally low costs [21, 22].

2.2. Economic models

A detailed economic analysis is much complicated. An alternative option is to calculate only the approximate purchase prices of the major components.

2.2.1. Cost of accumulators

Large pressure vessels have the cost in approximate proportion to its weight [24].

The material cost of accumulator is

$$C_{steel} = P_{steel} M_{steel} = P_{steel} \rho_{steel} V_{steel} \times 10^{-9} \quad (28)$$

C_{steel} is the steel cost. P_{steel} is the cost per kilogram. Cylinder vessel is commonly adopted. The total volume of steel (V_{steel}) is a function of the diameter (D_i), thickness and height (H) of the vessel. Design of the cylinder is correlated with the design pressure [25]

$$\delta_{cy} = \frac{p D_i}{2[\sigma]^t \phi - p} \quad (29)$$

where the units of p and D_i are MPa and mm; $[\sigma]^t$ is permissible stress [26].

The material mass used for the cylinder is

$$M_{steel,cy} = \rho_{steel} V_{steel,cy} \times 10^{-9} = \rho_{steel} \pi [(0.5 D_i + \delta_{cy})^2 - 0.25 D_i^2] H_{cy} \times 10^{-9} \quad (30)$$

$V_{steel,cy}$ is the volume of cylinder.

$$V_{steel,cy} = V_w = \frac{3600 t_H \cdot W_{net}}{\rho_w \cdot \eta_{SORC} \cdot C_{p,w} \cdot \Delta T} \quad (31)$$

V_w is water volume. t_H is storage time in hour. is the heat capacity of water.

ΔT is the temperature drop in discharge process.

Cylinder vessel generally has two elliptical heads at the top and the bottom. The standard ratio of the half long axis (a) and the half short axis (b) of an ellipse is 2:1. The design thickness is expressed by

(32)

The mass (kg) of an elliptical head is calculated by [27]

$$M_{steel,head} = \rho_{steel} \pi \left\{ \left[\frac{2}{3} (a + \delta_{head})^2 (b + \delta_{head}) + (a + \delta_{head})^2 h_{head} \right] - \left(\frac{2}{3} a^2 b + a^2 h_{head} \right) \right\} \times 10^{-9} \quad (33)$$

For a standard head as shown in Fig. 4, $a=0.5D_i$, $b=0.5a$.

$$M_{steel,head} = \rho_{steel} \pi \delta_{head} \left[\frac{D_i^2}{3} + \frac{5}{6} D_i \delta_{head} + \frac{2}{3} \delta_{head}^2 + (D_i + \delta_{head}) h_{head} \right] \times 10^{-9} \quad (34)$$

where h_{head} is the edge height of head (mm) and is regulated by the standard [28].

The total mass of material used for the vessel is

$$M_{steel} = M_{steel,cy} + 2M_{steel,head} \quad (35)$$

Design pressure p is the sum of the saturation pressure (p_s) of water at the design temperature and the static pressure (p_g) caused by gravity. The accumulator can be laid in vertical or horizontal way. But the horizontal layout in Fig. 5 appears to save more material of steel [3]. For the horizontal disposition, the design pressure is

$$p = p_s + p_g = p_s + \rho_w g D_i \times 10^{-9} \quad (36)$$

Q345R is exemplified and the price is 3600 RMB/ton [3]. Its permissible stress is posted in Table 2. The exchange rate of RMB against US dollar is 0.1537.

2.2.2. Cost of other units

Cost of condenser and pump can be calculated by the equations of purchased and bare module cost as provided in Table 3. Parameters and coefficients of the equations are indexed in Table 4 [30, 31]. The effect of volume ratio on SE cost is not taken into

account because usually it does not change the actual size of the device. The cost correlation of SE is function of the volumetric flow rate at the outlet in m^3 / s [34, 35]. The actual cost in 2014 need to be converted from the cost in 2001 by introducing the CEPCI (Chemical Engineering Plant Cost Index) [36].

The required aperture area of PTC not only meet the simultaneous heat collection and power conversion duration (t_{sim}), but also fulfill the capacity of heat storage:

$$A_{PTC} \sum_{i=1}^{8760} G_{b,i} \eta_{PTC,i} \eta_{SORC,P} = (t_{sim} + t_H \cdot 365) W_{net} \quad (37)$$

Cost of PTC is assumed to be 170 \$/m², the same as potential installed cost assessed by SkyTrough [37]. The specific land requirement is $3.5A_{PTC}$ [38].

To take the effective operation time, the capital recovery factor (CRF) and the cost of operation and maintenance (COM) into account, levelized electricity cost (LEC) is considered as the evaluation criteria. Its formula is given by [38, 39, 41]

$$LEC = (C_{tot} \cdot CRF + COM) / [(t_{sim} + t_H \cdot 365) W_{net}] \quad (38)$$

$$CRF = \frac{i(1+i)^{LT}}{(1+i)^{LT} - 1} \quad (39)$$

COM is divided into annual solar field component replacement cost (2.5% of PTC cost) and annual operation and maintenance cost (4% of the investment cost) [38, 41].

$$COM = 0.025C_{BM,PTC} + 0.04(C_{tot} - C_{BM,PTC}) \quad (40)$$

Payback period (PP) can be calculated by [30]:

$$PP = \ln \left(\frac{W_{net} C_{elec} - COM}{W_{net} C_{elec} - COM - i \cdot C_{tot}} \right) / \ln(1+i) \quad (41)$$

Electricity price (C_{elec}) is 0.18 \$/kWh [31, 42].

3. Results and discussion

In this section, the heat-power conversion in design condition and off-design behavior of heat discharge process are analyzed. Economic evaluation on 1 MW output with 6.5 h heat storage is carried out, followed by estimation on different storage capacities and generating scales. Besides, investment proportion of each unit is clarified.

Thermodynamic performance optimization is the prerequisite for economic investigation. However, optimum design parameters are not at disposal because tandem technology remains to be improved. To guarantee the reliability of the simulation results, inlet and outlet pressures of the tandem SEs are taken from the practical Nepal project case in Table 1. Phoenix is selected as territory in the simulation. Only over-expansion process is considered for the SEs and saturated vapor is injected into SE1 inlet. An over design area of approximately 10% is ensured for the condenser. All the other fixed parameters are displayed in Table 5.

3.1. Thermodynamic performance analysis

3.1.1. Heat-power conversion in design condition

Fig. 6 exhibits variations of SRC efficiency with the pressure ratio in SE1 for different $r_{v,b}$ combinations of SEs. The case with $r_{v,b} = 5$ for both SEs shows the optimal property while the one with $r_{v,b} = 2$ for SE1 and $r_{v,b} = 8$ for SE2 shows the worst. Performance degrades as the gap between the built-in volume ratios of two SEs enlarges. All the curves first climb and then go down. There exists an optimal pressure ratio in SE1 (which corresponds to the optimal intermediate pressure of tandem SEs) to maximize SRC efficiency for different $r_{v,b}$ combinations. The maximum range of

SRC efficiency is between 17.94% and 18.77%.

Here, the case with $r_{v,b} = 3$ for SE1 and $r_{v,b} = 7$ for SE2 is exemplified, since the cases with other combinations are similar. As depicted in Fig. 6, the maximum SRC efficiency of 18.49% is reached when the pressure ratio for SE1 is 6. Further observations of tandem SEs' power output and isentropic efficiencies with the pressure ratio in SE1 are graphed in Fig. 7. The overall capacity is 1 MW. It is shown that SE1 efficiency decreases while SE2 efficiency increases as the pressure ratio in SE1 elevates. The variations of two SEs' power output demonstrate the opposite trend. The optimal SRC efficiency is a compromise between two SEs' efficiencies.

Design parameters of the condenser and operating parameters of 1 MW system are presented in Tables 6 and 7, respectively. The operating pressure ratios of two SEs are the values when the maximum SRC efficiency of 18.49% is achieved. The built-in pressure ratios and design outlet volume flow rate are the values at which SEs' peak efficiencies of 0.75 are reached. SE usually works in over-expansion condition for optimum heat-power conversion performance by sacrificing its isentropic efficiency.

3.1.2. Off-design behavior of heat discharge process

In heat release process, V1, V2, V5 are open and P1 runs. The system works in this mode when solar radiation is unavailable and power is demanded.

It is a crucial issue to allow temperature drop in the storage unit because it influences the area of PTCs, the accumulator size and annual output. A larger design temperature drop of water during exothermic process will lead to a smaller accumulator. However, as the temperature descends, the supply pressure for SEs will be diminished, thus

affecting the power conversion. The designed T_1 is 180 °C as indexed in Table 7, and assuming that T_1 drops from 200 °C to 180 °C during heat release process. G_b is 700 W/m² and the ambient temperature is 20 °C. The variations of η_{SRC} , η_T and efficiency of SE2 with the pressure ratio in SE1 at given T_1 are exhibited in Fig. 8. Efficiency of SE1 can be kept constant by adjusting its pressure ratio. Efficiency curve of SE2 ascends in discharge process owing to the decline of its pressure ratio. Given T_1 , there exists an optimum pressure ratio in SE1 which makes and η_T maximum.

Variations of peak SRC and equivalent thermal efficiencies in heat release process are graphed in Fig. 9. Each point stands for the peak value at a given T_1 . When heat is released due to an accumulator temperature decrement from $(T + \Delta T)$ to T , equal energy should be harvested from the solar field in the periodical operation. So the equivalent η_T is deemed as the product of η_{SRC} and at T . suffers from smaller temperature difference driving heat-power conversion during exothermic process and it decreases from 19.51% to 18.49%. The deviation from the design is only 1.02%. But lower operating temperature results in a rise of η_{PTC} (from 64.92% to 66.55%, which can be deduced from Section 2.1.1). The conflicting effects lead to a reduction in equivalent η_T from 12.66% to 12.30%. The degradation is only 0.36%, which is slight. Thus it is reasonable to select the design temperature of 200 °C for the accumulator while the annual output is simulated under the condition that T_1 is 180 °C. Thermo-economic performance is not overestimated based on these considerations.

3.2. Economic evaluation

Economic investigation is on the basis of the design parameters in Table 7.

Simultaneous heat collection and power conversion is assumed, i.e., V1, V2, V3 are open and P1 works. The case with $r_{v,b} = 3$ for SE1 and $r_{v,b} = 7$ for SE2 is illustrated, which produces lower SRC efficiency in comparison with that using $r_{v,b} = 5$ for both SEs. From this viewpoint, economic performance of the proposed system is conservatively assessed.

Fig. 10 shows the monthly beam solar irradiance, irradiation time and t_{sim} in Phoenix. Hourly weather data in a typical year is used [43]. The beam solar irradiance is most abundant with the longest irradiation time in May, while it is the scarcest in January. Notably, t_{sim} is shorter than irradiation time in each month. The reason is that when G_b is weak, there is no positive analytic solution to Eq. (3). The corresponding liquid state collector area A_l is nonexistent or negative. of 3806 h can be calculated in a typical year.

3.2.1. Economic evaluation on 1 MW scale with 6.5 h storage duration

Storage has a significant impact as it increases the capital cost but allows the plant to operate for longer. Horizontal multi-vessels that function in parallel can effectively reduce the device length and satisfy heat storage requirement. Variations of the total mass and diameter with number of vessels are exhibited in Fig. 11. The storage capacity is 6.5 h. Though multi-vessels offer low steel material cost, the disadvantages are obvious. As the number increases, the processing cost is supposed to rise and the heat loss from the vessels could be more remarkable [3]. Comprehensively considering the pros and cons, ideal number of 6 seems to be acceptable.

The investment proportion of each unit is shown in Fig. 12. The cost of PTCs

occupies nearly half of C_{tot} , followed by SE2. The cost of SE2 is approximately seven times that of SE1 due to the larger design outlet volume flow rate and rotor diameter. Water storage only accounts for 6.59%. LEC of 0.118 \$/kWh and PP of 10.48 years can be obtained using the economic models.

3.2.2. Economic evaluation on different storage capacities and generating scales

Variations of number of vessels and the cost of steel with the capacity of heat storage are illustrated in Fig. 13. Storage time period of 1-10 h is calculated because the longest in one day is 13 h according to the hourly weather data in a typical year. As the capacity of heat storage enlarges, the cost of steel increases approximately linearly from 71.31 to 716.70 thousand dollars. When $1 \leq t_H \leq 5$, the number of vessels equals to t_H ; when $6 \leq t_H \leq 10$, the number is $(t_H - 1)$.

Fig. 14 presents the variations of LEC, C_{tot} and PP with the capacity of heat storage. Though C_{tot} rises with increasing vessel number, LEC and PP decrease from 0.136 to 0.112 \$/kWh, 13.23 to 9.64 years, respectively. It indicates that longer storage time is more profitable as the cost for more steam accumulators can be compensated by electricity earnings.

The proposed system is especially suitable for distributed cogeneration applications. However, it is worth noting that the maximum rated power of single SE is lower than 3 MW according to the present technical state of art. The reasons are as follows: First, larger SE requires larger helical rotors. Generally, the maximum diameter of the rotor is 500 mm due to the restriction of manufacturing technology level. Second, the market demand for distributed energy is generally below 3 MW. Third, steam turbines above

3 MW are common and mature. It's not necessary to develop large SE.

The parameters of the condensers corresponding to 0.2-0.8 MW are listed in Table 8. The investment proportion of each unit and economic indicators are indexed in Table 9. LEC and PP increase as the scale turns smaller. Similar to the conventional turbine-based solar power stations, the tandem system is more profitable at larger scales. This is because some fixed costs for a larger plant are almost the same as those for a smaller plant, while the devices (e.g., expander, pump and generator) are more efficient at higher power.

It is remarkable that the tandem SE technology is still at a promoting stage. Further improvement of the economic indexes could be anticipated if more efficient combinations of SEs are employed, higher SE1 inlet pressure/temperature are admitted or larger operating pressure ratio is adopted.

4. Conclusion

Tandem technology is an effective compensation solution for low $r_{v,b}$ of SE. It is innovative to combine tandem SE and DSG for solar electricity generation. In this study, the technical superiorities of the tandem system over the cascade SORC system are illuminated. Thermodynamic analyses on the design and off-design behaviors of the tandem system are carried out. And the thermo-economic investigation is conducted on the basis of the operating parameters of an engineering project. Following conclusions can be drawn:

(1) There exists an optimal pressure ratio in SE1 (which corresponds to the optimal intermediate pressure of tandem SEs) to maximize SRC efficiency for different $r_{v,b}$

combinations of SEs. The design with $r_{v,b} = 5$ for both SEs exhibits the optimum thermodynamic property. SRC efficiency of 18.49% is achieved using $r_{v,b} = 3$ for SE1 and $r_{v,b} = 7$ for SE2.

(2) LEC of 0.118 \$/kWh and PP of 10.48 years are obtained for 1 MW plant with 6.5 h heat storage. The cost of PTCs accounts for nearly half of C_{tot} while steam accumulators occupy less than 7%. The cost of SE2 is approximately seven times that of SE1 due to the larger design outlet volume flow rate.

(3) The tandem system is more profitable with widened scale range and extended storage capacity. It is expected that lower LEC and PP can be achieved by thermodynamic optimization and technological improvement of the tandem configuration.

Acknowledgment

This study was sponsored by EU Marie Curie International Incoming Fellowships Program (703746) and Special Funding For Doctoral Staffs (JZ2016HGBZ1016) by Hefei University of Technology.

REFERENCES

[1] J. Li, P. Li, G. Pei, J.Z. Alvi, J. Ji, Analysis of a novel solar electricity generation system using cascade Rankine cycle and steam screw expander, Appl. Energy 165 (2016) 627-638.

[2] P. Li, J. Li, G. Gao, G. Pei, Y. Su, J. Ji, B. Ye, Modeling and optimization of solar-

454 powered cascade Rankine cycle system with respect to the characteristics of steam
 455 screw expander, *Renew. Energy* 112 (2017) 398-412.

456 [3] J. Li, P. Li, G. Gao, G. Pei, Y. Su, J. Ji, Thermodynamic and economic investigation
 457 of a screw expander-based direct steam generation solar cascade Rankine cycle system
 458 using water as thermal storage fluid, *Appl. Energy* 195 (2017) 137-151.

459 [4] E. Georges, S. Declaye, O. Dumont, S. Quoilin, V. Lemort, Design of a mall-scale
 460 organic Rankine cycle engine used in a solar power plant, *Int. J. Low Carbon Tech.* 8
 461 (2013) i34-i41.

462 [5] S. Quoilin, M. Orosz, H. Hemond, V. Lemort, Performance and design optimization
 463 of a low-cost solar organic Rankine cycle for remote power generation, *Sol. Energy* 85
 464 (2011) 955-966.

465 [6] M. Astolfi, Techno-economic optimization of low temperature CSP systems based
 466 on ORC with screw expanders, *Energy Procedia* 69 (2015) 1100-1112.

467 [7] MG Read, IK Smith, N Stosic, Optimisation of power generation cycles using
 468 saturated liquid expansion to maximise heat recovery, *Proceedings of the Institution of*
 469 *Mechanical Engineers, Part E: J. Process. Mech. Eng.* 231 (2017) 57-69.

470 [8] <http://www.jxhdep.com/project.asp> (accessed 21.06.2018).

471 [9] Ochoa LR, Engineering Aspects of a Parabolic Trough Collector Field with Direct
 472 Steam Generation and an Organic Rankine Cycle, Master of Science in Energy Systems,
 473 Cologne, Germany, October 2014.

474 [10] Pitz-Paal R, Dersch J, Milow B, Ferriere A, Romero M, Tellez F, et al, ECOSTAR
 475 roadmap document, DLR, Germany, 2005.

- 476 [11] Zarza E, Rojas ME, González L, Caballero JM, Rueda F, INDITEP: The first pre-
477 commercial DSG solar power plant, *Sol. Energy* 80 (2006) 1270-1276.
- 478 [12] D. Krueger, A. Heller, K. Hennecke, K. Duer, Parabolic trough collectors for
479 district heating systems at high latitudes: a case study, *Proceedings of Eurosun 2000*.
- 480 [13] S.A. Kalogirou, Solar thermal collectors and applications, *Prog. Energy Combust.*
481 *Sci*, 30 (2004) 231-295.
- 482 [14] J.A. Duffie, W.A. Beckman, *Solar Engineering of Thermal Processes*, 4th ed. John
483 Wiley & Sons; 2013.
- 484 [15] K. C. NG, T. Y. Bong, T. B. Lim, A thermodynamic model for the analysis of screw
485 expander performance, *Heat Recovery Systems and CHP* (10) 1990 119-133.
- 486 [16] G. Rogers, Y. R. Mayhew, *Engineering Thermodynamics-Heat and Work Transfer*,
487 ELBS Book 1964.
- 488 [17] B. K. Ku, Screw type steam engine, Internal report of Mayekawa Mfg Co Ltd. 2-
489 3-1 1984 Tokyo 135, Japan.
- 490 [18] H.D.M. Hettiarachchi, M. Golubovic, W.M. Worek, Y. Ikegami, Optimum design
491 criteria for an Organic Rankine cycle using low-temperature geothermal heat sources,
492 *Energy* 32 (2007) 1698-1706.
- 493 [19] F. Cataldo, R. Mastrullo, A.W. Mauro, G.P. Vanoli, Fluid selection of organic
494 Rankine cycle for low-temperature waste heat recovery based on thermal optimization,
495 *Energy* 72 (2014) 159-167.
- 496 [20] HTRI Software, <http://www.htri.net>; 2018 (accessed 11.06.2018).
- 497 [21] R.K. Shah, D.P. Sekulic, *Fundamentals of Heat Exchanger Design*, Wiley 2003.

- 498 [22] F. Calise, C. Capuozzo, A. Carotenuto, L. Vanoli, Thermoeconomic analysis and
499 off-design performance of an organic Rankine cycle powered by medium-temperature
500 heat sources, *Sol. Energy* 103 (2014) 595-609.
- 501 [23] Y. Li, M. Du, C. Wu, S. Wu, C. Liu, J. Xu, Economical evaluation and optimization
502 of subcritical organic Rankine cycle based on temperature matching analysis, *Energy*
503 68 (2014) 238-247.
- 504 [24] Vessel Cost Estimate. <<http://matche.com/equipcost/Vessel.html>> (accessed
505 26.07.2016).
- 506 [25] Steel pressure vessels, National Standard of the People's Republic of China. GB
507 150-1998, pp:28.
- 508 [26] Steel plates for boilers and pressure vessels, National Standard of the People's
509 Republic of China, GB 713-2008.
- 510 [27] Heads for pressure vessels, National Standard of the People's Republic of China,
511 GB/T 25198-2010, pp:13.
- 512 [28] Elliptical heads. Standards of the Machinery Department of the People's Republic
513 of China, JB1154-73.
- 514 [29] R. Turton, R.C. Bailie, W.B. Whiting, J.A. Shaeiwit, Analysis, synthesis, and
515 design of chemical processes (4th ed.), Prentice Hall PTR, New Jersey 2013.
- 516 [30] C. Zhang, C. Liu, S. Wang, X. Xu, Q. Li, Thermo-economic comparison of
517 subcritical organic Rankine cycle based on different heat exchanger configurations,
518 *Energy* 123 (2017) 728-741.
- 519 [31] M.H. Yang, R.H. Yeh, Thermodynamic and economic performances optimization

520 of an organic Rankine cycle system utilizing exhaust gas of a large marine diesel engine,
 521 Appl. Energy 149 (2015) 1-12.

522 [32] V.L. Le, A. Kheiri, M. Feidt, S.P. Prayer, Thermodynamic and economic
 523 optimizations of a waste heat to power plant driven by a subcritical ORC (Organic
 524 Rankine Cycle) using pure or zeotropic working fluid, Energy 78 (2014) 622-638.

525 [33] L. Pierobon, T.V. Nguyen, U. Larsen, F. Haglind, B. Elmegaard, Multi-objective
 526 optimization of organic Rankine cycles for waste heat recovery: Application in an
 527 offshore platform, Energy 58 (2013) 538-549.

528 [34] M. Astolfi, Techno-economic optimization of low temperature CSP systems based
 529 on ORC with screw expanders, Energy Procedia 69 (2015) 1100-1112.

530 [35] BERNARDO PERIS PÉREZ, Thermo-economic assessment of small-scale
 531 organic Rankine cycle for low-grade industrial waste heat recovery based on an
 532 experimental application, Doctoral Thesis, Universitat Jaume, July 2017, Chapter IV
 533 pp: 110.

534 [36] R. Turton, R.C. Bailie, W.B. Whiting, J.A. Shaeiwit, Analysis, synthesis, and
 535 design of chemical processes, Pearson Education Inc 2009.

536 [37] P. Kurup, C.S. Turchi, Parabolic trough collector cost update for the system
 537 advisor model (SAM), NREL/TP-6A20-65228 National Renewable Energy Laboratory
 538 2015 pp: V.

539 [38] N.B. Desai, S. Bandyopadhyay, Thermo-economic comparisons between solar
 540 steam Rankine and organic Rankine cycles, Appl. Therm. Eng. 105 (2016) 862-875.

541 [39] N.B. Desai, S. Bandyopadhyay, Thermo-economic analysis and selection of

working fluid for solar organic Rankine cycle, Appl. Therm. Eng. 95 (2016) 471-481.

[40] I.I.T. Bombay, Solar thermal simulator version 2.0, 2014.

[41] N.B. Desai, S. Bandyopadhyay, Optimization of concentrating solar thermal power plant based on parabolic trough collector, J. Cleaner Prod, 89 (2015) 262-271.

[42] Price of electricity, diesel fuel, and CO₂ emission of electricity, 2012. <<http://www.eia.gov/>>. (accessed 01.08.2018).

[43] Weather Data. <<https://energyplus.net/weather>>Energyplus (accessed 12.05.2018).

Figure Legend

Fig. 1. Solar DSG tandem SE system

Fig. 2. A picture of the SE tandem engineering project in India

Fig. 3. Overview of the work

Fig. 4. Cross-section of elliptical head

Fig. 5. Horizontal placement of cylinder accumulator

Fig. 6. Variations of SRC efficiency with the pressure ratio in SE1 for different $r_{v,b}$ combinations of SEs

Fig. 7. Variations of tandem SEs' power output and isentropic efficiencies with the pressure ratio in SE1 for 1MW tandem system

Fig. 8. Variations of η_{SRC} , η_T and efficiency of SE2 with the pressure ratio in SE1 at given T_1

Fig. 9. Variations of peak SRC efficiency, peak thermal efficiency when the discharge temperature drop is 20 °C

564 Fig. 10. Monthly beam solar irradiance, irradiation time and t_{sim} in Phoenix.

565 Fig. 11. Variations of the total mass and the diameter with number of vessels in

566 horizontal

567 Fig. 12. The investment proportion of each unit for 1MW scale

568 Fig. 13. Variations of number of vessels in horizontal and the cost of steel with the

569 capacity of heat storage

570 Fig. 14. Variations of LEC, C_{tot} and PP with the capacity of heat storage for 1MW

571 scale

572 **Table legend**

573 Table 1. Design parameters of some tandem project cases

574 Table 2. Permissible stress for Q345R

575 Table 3. The cost estimating equations

576 Table 4. Values of constants for different equipments

577 Table 5. Fixed parameters

578 Table 6. Parameters of the condenser in 1MW system

579 Table 7. Design and operating parameters of 1MW system

580 Table 8. Parameters of condensers in different scales of system

581 Table 9. The investment proportion of each unit and economic indicators for 0.2-0.8

582 MW systems

588

587

Fig. 3. Overview of the work

Fig. 4. Cross-section of elliptical head

Fig. 5. Horizontal placement of cylinder accumulator

Fig. 6. Variations of SRC efficiency with the pressure ratio in SE1 for different combinations of SEs

Fig. 7. Variations of tandem SEs' power output and isentropic efficiencies with the pressure ratio in SE1 for 1MW tandem system

Fig. 8. Variations of η_{SRC} , η_T and efficiency of SE2 with the pressure ratio in SE1 at given T_1

Fig. 9. Variations of peak SRC efficiency, peak thermal efficiency when the discharge temperature drop is 20 °C

Fig. 10. Monthly beam solar irradiance, irradiation time and t_{sim} in Phoenix.

Fig. 11. Variations of the total mass and the diameter with number of vessels in

horizontal

Fig. 12. The investment proportion of each unit for 1MW scale

Fig. 13. Variations of number of vessels in horizontal and the cost of steel with the capacity of heat storage

Fig. 14. Variations of LEC, C_{tot} and PP with the capacity of heat storage for 1MW

scale

651

Table 1. Design parameters of some tandem project cases [8]

Parameters	Project case	Tianjing Xinli	Nepal ACPL	India keerthi
		steel plant	cement plant	cement plant
Inlet pressure (MPa)		0.5	1.0	0.95
Inlet temperature (°C)		152	300	314
Inlet mass flow rate (t/h)		11.5	18.5	17.5
Outlet pressure (MPa)		0.01	0.01	0.02
Outlet temperature (°C)		46	45	60
Rated power (kW)		950	2400	2400
Installed power (kW)		1000	2600	2600
Delivery time		2014	Not available	2016

652

653

Table 2. Permissible stress for Q345R [27], unit: MPa

Standard	Thickness	Temperature /°C					
		200	250	300	350	400	450
	/mm						
	3-16	183	167	153	143	125	66
	>16-36	170	157	143	133	125	66
GB 713	>36-60	160	147	133	123	117	66
	>60-100	150	137	123	117	110	66
	>100-150	147	133	120	113	107	66
	>150-200	143	130	117	110	103	66

660

661

662

663

664

Table 3. The cost estimating equations

Item	Equation
Purchased cost of condenser [23, 29, 30]	$\log_{10} C_p = K_1 + K_2 \log_{10} A + K_3 (\log_{10} A)^2$
Purchased cost of pump [23, 29, 30]	$\log_{10} C_p = K_1 + K_2 \log_{10} W + K_3 (\log_{10} W)^2$
Bare module cost of condenser and pump [23, 29, 30]
Pressure factor [23, 29, 30]	$\log_{10} F_p = C_1 + C_2 \log_{10} (10p - 1) + C_3 [\log_{10} (10p - 1)]^2$
Cost of generator [32, 33]	$C_{p,g} = 60(W_g)^{0.95}$
Cost correlation of SE [34, 35]	$C_{p,SE} = 3143.7 + 217423V_{out}$
Cost in the year 2014 [36]
Land and site development cost [39, 40]
Miscellaneous cost [39, 40]	$C_{mis} = 183W_{net}$
Total cost	$C_{tot} = C_{BM,condenser} + \sum_{i=1}^2 C_{BM,SEi} + C_{BM,g} + C_{BM,PTC} + C_{BM,steel}$ $+ \sum_{i=1}^2 C_{BM,pi} + C_{land} + C_{mis}$

665

666

667

668

669

Table 4. Values of constants for different equipments [30, 31].

Equipment	Condenser	Pump
K_1	4.3247	3.3892
K_2	-0.3030	0.0536
K_3	0.1634	0.1538
C_1	0.0388	-0.3935
-0.11272	0.3957
C_3	0.08183	-0.00226
B_1	1.63	1.89
B_2	1.66	1.35
F_M	1.40	1.60

670

671

672

673

674

675

676

677

678

679

680

681

Table 5. Fixed parameters

Term	Value
Pinch-point temperature difference,	5 °C
Pump efficiency,	0.8
Peak isentropic efficiency of SE, $\varepsilon_{os,p}$	0.75
Generator efficiency, ε_g	0.95
Welding coefficient, ϕ	0.8
Height of cylinder, H_{cy}	25m
$CEPCI_{2001}$ /	397/586.77
Interest rate, i	5%
Plant life time,	25 years

682

683

684

685

686

687

688

689

690

691

692

693

Table 6. Parameters of the condenser in 1MW system

Process data	Hot shell side	Cold tube side
Mass flow rate (kg/s)	2.094	49.40
Heat transfer coefficient (kW/m ² ·K)	12.83	2.70
Velocity (m/s)	24.45	0.47
Shell ID/ Tube OD (mm)	1100	19
Inlet/Outlet quality	0.867/0	0/0
Fouling resistance (m ² ·K/W)	0.000086	0.000174
Pressure drop (kPa)	0.93	4.99
Baffle spacing (mm)	1700	
Tube length (m)	5.0	
Tube count	1222	
Overall heat transfer coefficient (kW/m ² ·K)	1.123	
Duty (kW)	4343.3	
Area (m ²)	357.404	
Over design (%)	17.95	

694

695

696

697

698

699

Table 7. Design and operating parameters of 1MW system

Parameter	Value
$T_1(^{\circ}\text{C}) / P_1(\text{MPa})$	180/1.00
$T_2(^{\circ}\text{C}) / P_2(\text{MPa})$	114.63/0.17
$T_3(^{\circ}\text{C}) / P_3(\text{MPa})$	51/0.01
$T_4(^{\circ}\text{C}) / P_4(\text{MPa})$	46/0.01
$T_5(^{\circ}\text{C}) / P_5(\text{MPa})$	46.09/1.00
SRC efficiency (%)	18.49
Operating pressure ratio of SE1 / SE2	6/16.67
Built-in pressure ratio of SE1 / SE2	3.46/9.00
Design outlet volume flow rate of SE1 / SE2 (m^3/s)	1.24/9.06
Power output of SE1 / SE2 (kW)	439.34/563.27

700

701

702

703

704

705

706

707

708

709

710

Table 8. Parameters of the condensers in different scales of system

Scale	0.2	0.4	0.6	0.8
Process data				
Shell side heat transfer coefficient (kW/m ² ·K)	17.60	15.01	13.75	13.04
Shell ID (mm)	700	700	800	900
Baffle spacing (mm)	1000	1400	1400	1700
Tube side heat transfer coefficient (kW/m ² ·K)	1.24	2.77	3.17	3.19
Tube length (m)	4	5	5	5
Tube count	475	475	625	822
Overall heat transfer coefficient (kW/m ² ·K)	0.702	0.971	1.226	1.223
Area (m ²)	111.61	139.96	183.82	241.31
Over design (%)	15.07	18.76	10.35	8.14

711

712

713

714

715

716

717

718

Table 9. The investment proportion of each unit and economic indicators for 0.2-0.8

MW systems				
Scale	0.2	0.4	0.6	0.8
Units/ Indicators				
PTC (%)	44.68	47.20	48.03	48.41
SE1 (%)	3.73	3.83	3.86	3.87
SE2 (%)	25.92	27.27	27.71	27.92
Generator (%)	0.60	0.61	0.61	0.61
Condenser (%)	9.36	5.39	4.10	3.51
Water storage (%)	5.94	6.35	6.50	6.57
Land and site development cost (%)	5.52	5.83	5.93	5.98
Pumps (%)	1.86	0.99	0.70	0.55
Miscellaneous cost (%)	2.39	2.52	2.57	2.59
C_{tot} (million dollars)	1.532	2.901	4.276	5.656
LEC (\$/kWh)	0.129	0.122	0.120	0.119
PP (years)	12.139	10.995	10.666	10.519

Nomenclature

A	aperture area, m^2 / heat transfer area, m^2	SRC	steam Rankine cycle
a	half long axis, mm	V	valve
b	half short axis, mm		
C	cost, \$	Subscripts	
C_p	heat capacity, $kJ / (kg \cdot ^\circ C)$ / equipment cost, \$	0	reference state
D	diameter, mm	1-5	state points
G	solar radiation, W / m^2	a	ambient
g	gravity, N/kg	b	binary phase/ beam/ built-in
H	height, mm	SORC	steam-organic Rankine cycle
h	enthalpy, kJ / kg / edge height of head, mm	BM	bare module
i	interest rate	ch	characteristic
M	mass, kg		cylinder
m	mass flow rate, kg / s	D	diagram
P	cost per kilogram, \$/kg	$elec$	electricity
pressure, MPa		friction
r	ratio	g	generator/ static
T	temperature, $^\circ C$		hot side/ hour
volume, mm^3 / volumetric flow rate, m^3 / s	$head$	elliptical head
v	specific volume, cm^3 / kg		inner
W	power output,	in	inlet
α_s	solar altitude angle, $^\circ$	L	leakage

δ	solar declination, °/ thickness, <i>mm</i>	<i>l</i>	liquid phase
ω	solar hour angle, °	<i>land</i>	land
κ	incidence angle modifier	<i>M</i>	mechanical
ε	device efficiency	<i>mean</i>	mean
ϕ	welding coefficient/ geographic latitude, °	<i>mis</i>	miscellaneous
γ	isentropic index	<i>net</i>	net
η	efficiency	<i>os</i>	overall isentropic
$[\sigma]$	permissible stress, <i>MPa</i>	<i>out</i>	outlet
ρ	density, <i>kg / m³</i>	<i>P</i>	pressure/ pump/ peak
φ	incidence angle	<i>s</i>	shaft/ isentropic/ saturation
		<i>sim</i>	simultaneous heat collection
			and power conversion
Abbreviation			
CEPCI	Chemical Engineering Plant Cost Index	<i>steel</i>	steel
COM	cost of operation and maintenance		supply
CRF	capital recovery factor	<i>T</i>	thermal
DSG	direct steam generation	<i>Th</i>	theoretical
LEC	levelized electricity cost	<i>TD</i>	theoretical diagram
ORC	organic Rankine cycle	<i>TI</i>	theoretical isentropic
P	pump	<i>TM</i>	thermodynamic
PP	payback period		total
PTC	parabolic trough collector	<i>V</i>	volume
SE	screw expander		water