Storing wind power by direct thermal energy conversion and thermal energy storage

Abstract
Thermal energy storage (TES) has emerged as a competent and reliable system for concentrated solar power plants. TES has low capital costs and high efficiency (ratio of energy discharged and energy stored). In this study, the possibility of integrating thermal energy storage with a wind turbine has been modeled and discussed. In conventional wind farms, the extracted power output is less than the power in the wind to prevent the charge but by employing thermal energy storage, maximum wind energy within the limits of Betz law can be cropped for later use. This paper determines the efficiency and output of such a system. The model developed in this study gives a steady power output

Introduction

The main tasks of storage are buffering during transient weather conditions, dispatch ability, improving the annual capacity factor of the power plant, even distribution of electricity and achieving full load operation of the steam cycle at high efficiency. On the other hand the major requirements of storage system are high energy density of the storage material, good heat transfer between heat transfer fluid and the storage medium, mechanical and chemical stability of storage material, chemical compatibility of heat transfer fluid, heat exchanger and storage medium, reversibility of charging/ discharging cycles, low thermal losses and ease of control. There are various approaches to achieve storage of power. In this paper, thermal energy storage have been discussed.

Recent studies have shown that thermal energy systems are cost effective and efficient for concentrated solar power plants. The cost of the overall system is reduced and the availability of the solar power is considerably improved with TES [1][2]. The three main approaches for thermal energy storage are *sensible heat storage, latent heat storage* of phase change materials (PCM) and *thermochemical storage* by reversible chemical reactions [1]. The sensible heat storage, a storage medium is used which is one and the same as the liquid heat transfer fluid in either or both of the two flow loops. The simultaneous collection, transfer and storage of heat is termed *direct thermal storage*. On the other hand, *indirect thermal storage system* involves a separate storage medium from the collector heat transfer fluid, incorporating some form of heat exchange arrangement [1].

Solar Energy Generating Systems (SEGS) in California, with the combined capacity from three separate locations at 354 MW is the second largest solar thermal energy facility in the world. The SEGS-1 facility employs two-tank mineral oil storage system with a capacity of 115MWh [2].

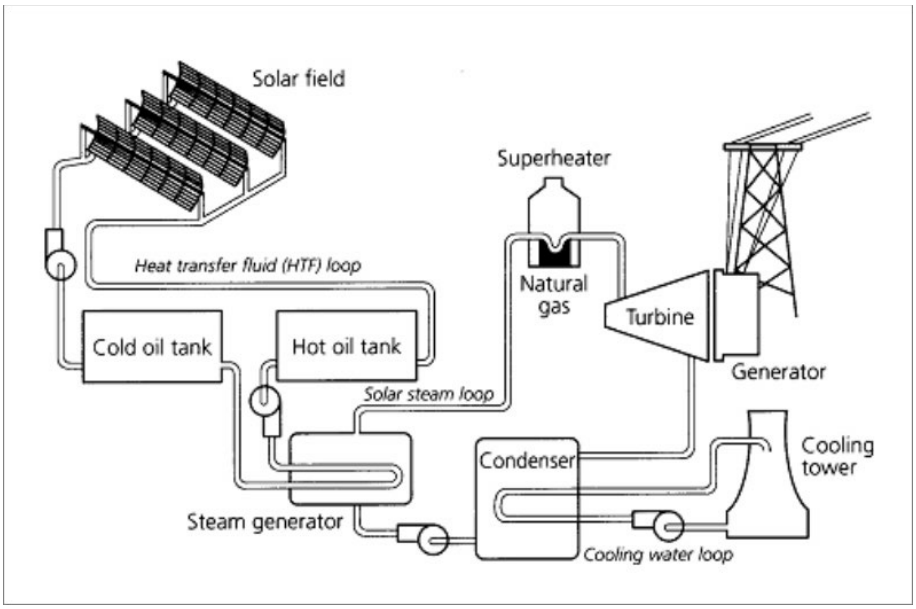


Figure 1- Process Scheme of SEGS-1 [2]

SEGS also uses a two tank molten salt storage at one of its facilities as shown in Fig.2 that uses a nitrate salt mixture which acts as storage and also as the heat transfer fluid. This system has a capacity of 105MWh [2][3].

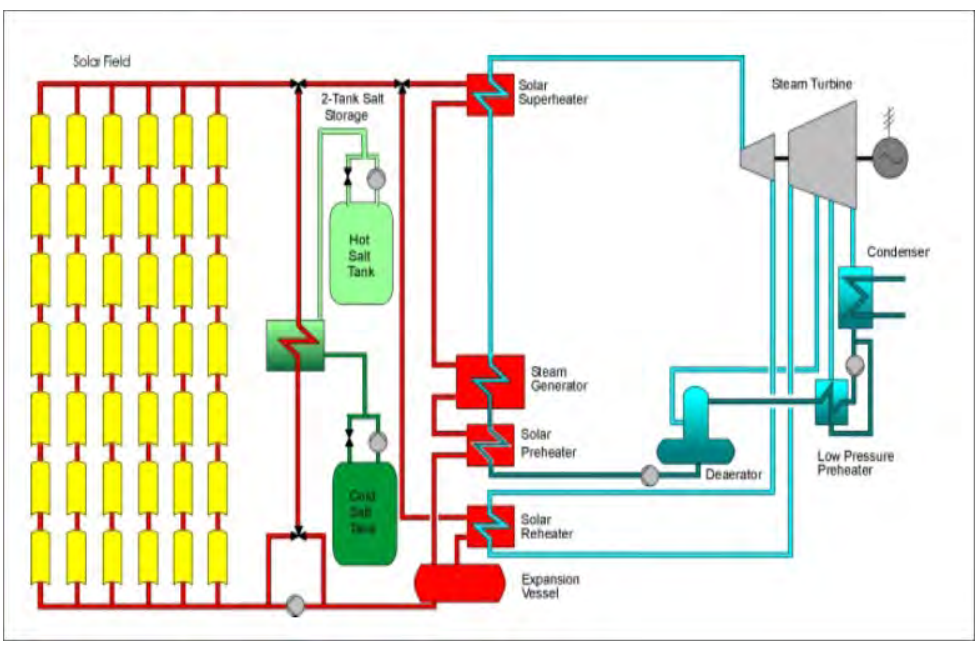


Figure 2- Process scheme of SEGS molten salt storage system [2]

Phase change materials (PCM) have also been frequently discussed for thermal energy storage. These materials store heat associated with phase transition. The amount of heat stored in these materials is proportional to the material mass, the fraction of material that undergoes the phase change, and the material heat of fusion [1]. Since the phase-transition enthalpy of PCMs are usually much higher (100- 200 times) than sensible heat, latent heat storage has much higher

storage density than sensible heat storage. However, the main disadvantage of latent heat storage is its low thermal conductivities, which mostly fall into the range of 0.2 W/(m K) to 0.7 W/(m K), and therefore relative heat transfer enhancement technologies must be adopted [4]. Luz International Ltd. after completion of SEGS (I-V) highlighted the need for advanced, cost-effective and reliable technologies to fully utilize the high operating temperature of the solar field (397 degree Celsius). In seeking a solution to this need, Luz developed two conceptual designs of TES systems utilizing phase change materials in the temperature range of interest for a SEGS plant [5].

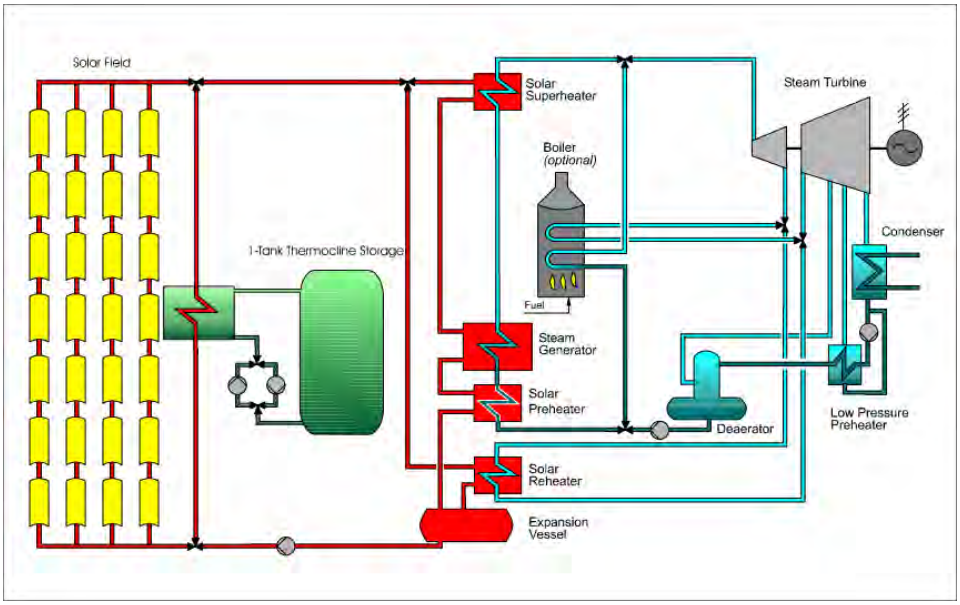


Figure 3- Process Scheme for SEGS for PCM storage [2]

Special chemicals can absorb/release a large amount of thermal energy when they break/form certain chemical bonds during endo- thermal/exothermal reactions. Based on such characteristics, the storage method making use of chemical heat has been invented. Suitable materials for chemical heat storage can be organic or inorganic, as long as their reversible chemical reactions involve absorbing/releasing a large amount of heat. When designing a chemical storage system, three basic criteria need to be considered: excellent chemical reversibility, large chemical enthalpy change and simple reaction conditions [4]

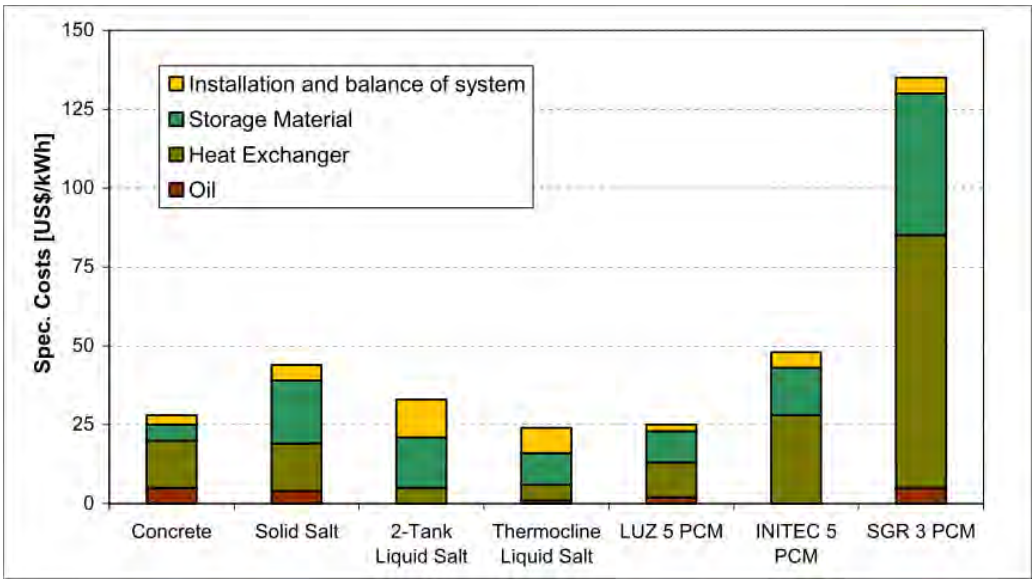


Figure 4- Specfic costs of different TES systems [2]

Three storage options offer favorable cost are two tank molten salt storage or one tank salt storage thermocline, concrete storage and phase change material [2].

By all existing research studies, it has been proved that thermal energy storage is a cost effective way of storing energy for intermittent power sources. Since wind power is highly intermittent, backup thermal storage systems deserve attention. The energy costs of the wind with backup thermal, the wind with battery energy storage and Wind Powered Thermal Energy System (WTES), which employs heat generator and thermal energy storage system were compared by Okazaki et. al. It seems WTES becomes the most economical system in these three systems although the estimation is in its initial stages. WTES becomes much more attractive when it is constructed besides CSP and/or bio-mass plant since many parts of elements can be shared [6]. Siemens and Apple have also shown interest in the development of a similar wind thermal energy systems[7][8].

Overview of the system

A thermal energy storage system mainly consists of three parts, the storage medium, heat transfer mechanism and containment system. The thermal energy storage system modeled in this work uses the two-tank-direct configuration where the heat transfer fluid (HTF) also acts as the energy storage medium. This requires two separate tanks, but eliminates the need for an additional heat exchanger to transfer heat from the collection HTF to the storage medium. The fluid is stored at its lower temperature in a cold tank, heated by mechanical work from wind turbine, and then stored at an elevated temperature in the hot tank. The stored energy in the hot tank is delivered to the load by pumping the HTF through the boiler. In this model, it is assumed that saturated liquid water is fed to the boiler and it exits as saturated steam. In this exchange, the HTF returns to its lower temperature and is pumped back to the cold tank. The output power is represented by the flow rate of the saturated steam generated in the boiler [9].

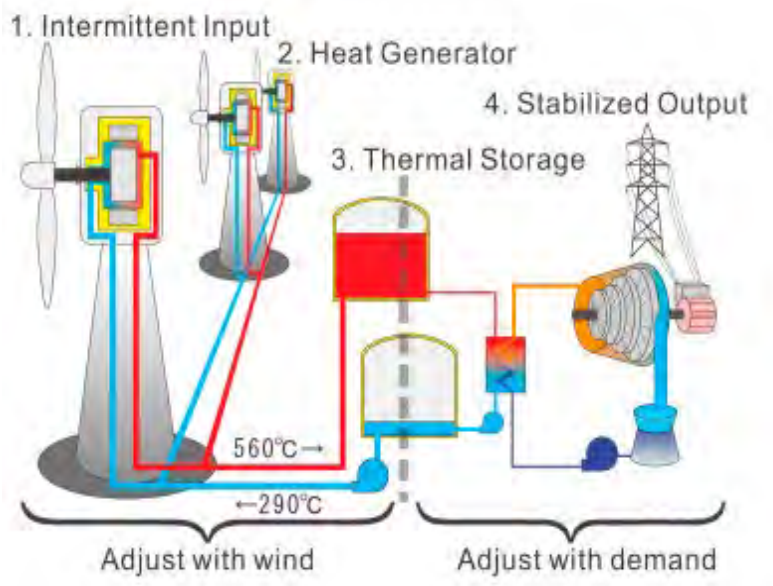


Figure 5- Wind Thermal Energy System using two tank molten salt technology [6]

Modelling WTES

In a wind turbine, the kinetic energy in the wind turns the propeller-like blades around a rotor. The rotor is connected to the main shaft, which spins a generator to create electricity. The maximum theoretical value of the coefficient of performance is 0.593, a value determined by a fluid mechanics constraint known as the Betz limit. Actual coefficients of performance are less than this limit due to various aerodynamic and mechanical losses. The performance coefficient Cp of the turbine is the mechanical output power of the turbine divided by wind power and a function of wind speed, rotational speed, and pitch angle (beta).

However, the concept used in this study is to use the mechanical work of the rotor instead of converting it into electricity. The rotating energy is converted to the thermal energy at the top of the tower directly. This thermal energy can be stored as sensible heat in the thermal energy storage system. The stored thermal energy is transferred by the heat transfer fluid (HTF) to produce steam that drives the turbine generator when required [6].

$$Power\ by\ the\ wind = P = \frac{\rho A v^3}{2}$$

Where, ρ is the density of air, A is the cross-sectional area, v is the velocity of the wind

$$Power\ by\ the\ blade = P_b = P_w \times C_p$$

C_p is the performance coefficient determined by Betz limit

$$Tip\ speed\ ratio = \lambda = \omega R/v$$

Where ω is the rotational speed, R is the length of the blade, v is the velocity of the wind. Solving for rotational speed,

$$\omega = \frac{\lambda v}{R}$$

By calculating rotational velocity, torque and work done can be determined.

$$Torque = \tau_r = \frac{P_b}{\omega}$$

$$W = \tau_r \theta = I\alpha\ \omega^2$$

$$W = \frac{1}{2} I\ \omega^2$$

Where θ is the rotation by the shaft, I is the moment of inertia. Moment of inertia for three cylindrical blades is given by:

$I = 3\ (\frac{1}{3} ML^2) = ML^2$

Finally work is given by, W= $ML^2\ \omega^2$

For simplicity the shape of the turbine blade has been considered a cylindrical rod. The actual shape of the blade is complex and varies among different manufacturers.

Figure 6- A three-blade horizontal wind turbine

Next step is to convert mechanical work into heat. This can be accomplished using a heat generator. The heat generator has lighter weight and lower cost than that of the electric generator since the heat generator is a kind of simple brake in principle [6]. The mechanical equivalent of heat states that motion and heat are mutually interchangeable and that in every case, a given amount of work would generate the same amount of heat, provided the work done is totally converted to

heat energy. By first law of thermodynamics, $\Delta U = Q + W$. For no change in internal energy, all the work done on a system will be converted to heat.

The heat generated can be then moved to the base utility for molten salt storage. There might be some heat losses during the transfer which have not been taken into consideration for this study. A molten salt mixture of nitrates 60 % $NaNO_3$ 40% KNO_3 with a melting point of 221°C was chosen. The temperature of both tanks at all times should be above the melting point to avoid solidification. The molten salt acts as heat transfer fluid (HTF) to heat liquid water to steam which then drives a steam turbine to generate electricity.

In the current model, a three blade horizontal axis wind turbine was modeled for wind speed nea Columbia river gorge, Oregon, US for a hub height of 100m. The input wind speed to the model is as shown in Fig.7. For a co-efficient performance of 0.4, blade length of 37m and density of air 1.2041kg/m^3, the output power of such a turbine would be varying with the cube of winds speed and therefore highly variable. This intermittency creates problems of stability and power scheduling in the grid. Ideally, using thermal storage system the output power should be a constant number, storing the energy during peak wind speeds and delivering the stored energy during the less windy durations.

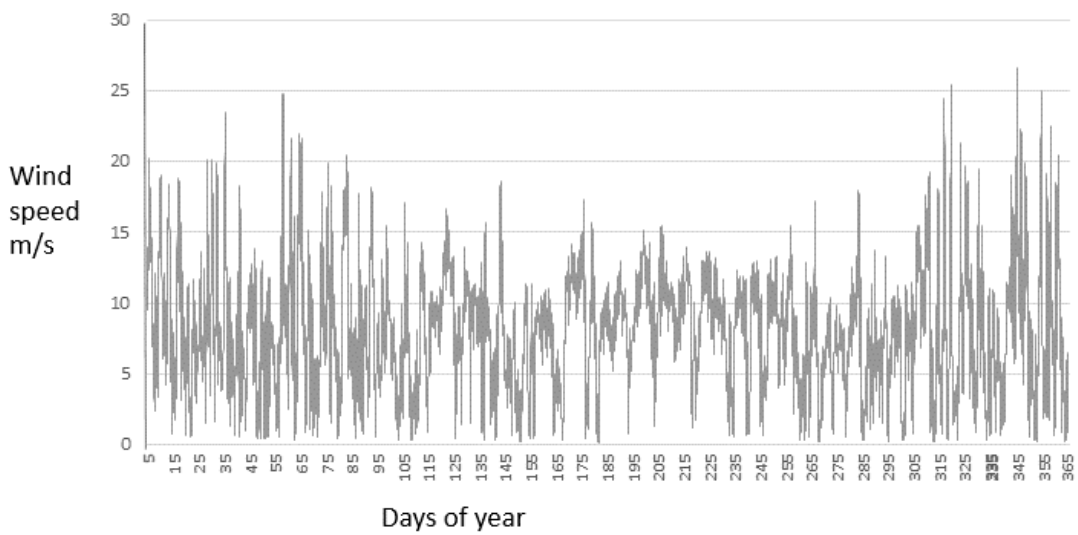


Figure 7- Wind speed time series for the year 2006 in Oregon,US at 100m height

The ratio of the tangential speed of the blade tip and wind velocity is given by tip speed ratio (TSR). This varies with material, structure and design of the blade. If the tip speed ratio is too high, the blades of the turbine are moving too fast and acting like solid wall against the wind. Alternatively, if the tip speed ratio is too low, the blades are moving too slow and the wind is passing without being utilised. Tip speed ratio is chosen such that the blades do not pass through turbulent air while efficiently capturing the non-turbulent winds. A well designed rotor would have

a tip speed ratio of 6 to 7. Co-efficient of performance C_p used in the wind power calculation is still relevant since the turbine losses and mechanical losses need to be accounted for.

Overall turbine efficiency, $C_{p=} \eta_a \times \eta_m \times \eta_e$

Where η_a= efficiency of turbine (blade aerodynamic efficiency), η_m= mechanical efficiency taking into account the power lost to bearing and gear tooth friction and η_e= electrical efficiency accounting for power lost to windings and electrical resistance

Since, in this study we are not using the generator part of the turbine electrical efficiency does not need to be accounted for. Therefore, $C_{p=} \eta_t \times \eta_m$

Theoretically, by using the principles of brake or exerting work on molten salt storage tank (stirring) all of the work done can be converted to heat. The heat generated in Joules is shown in Fig. 8.

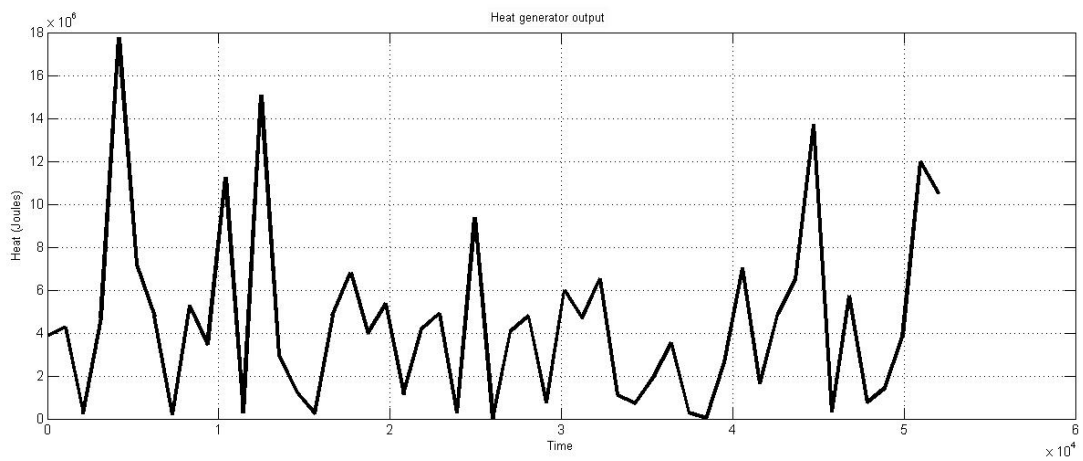


Figure 8- Heat generated by the heat generator in joules in an year

The volume of the salt plays an important role in determining the temperature of the molten salt. For this model, the volume was kept at $7m^3$ to raise the temperature of the molten salts from 265ºC (538.15 K) to 551.85ºC (825K). This volume and temperature have been adjusted according to the existing technology and to exchange enough heat from the molten salt to run the steam turbine and avoid condensation inside the steam turbine. The temperature of the hot tank of molten salt is shown in Fig. 9

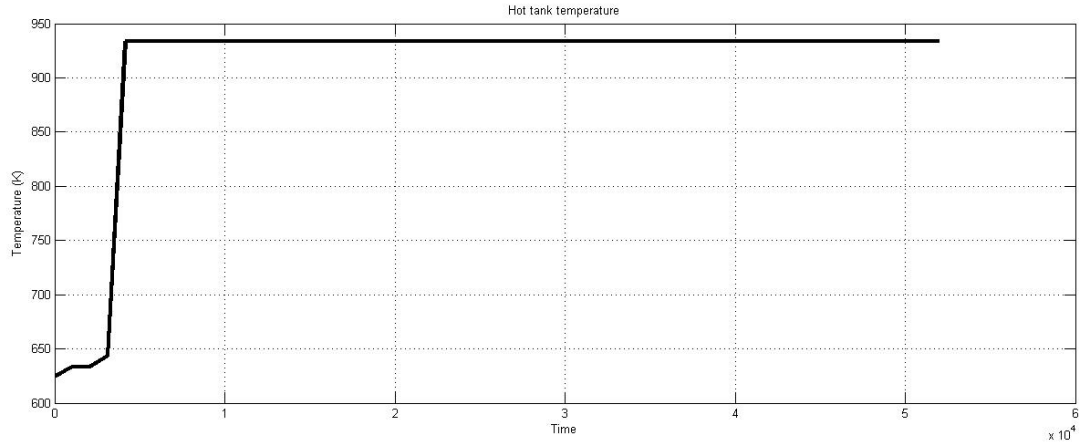


Figure 9- Maxium temperature achieved in the thermal storage hot tank

The high efficiency of molten salt storage systems ensure minimum losses of energy at this step. The thermal energy stored Q depends upon the volume of the tank V, heat capacity of the salts c_p and temperature difference dT between the two tanks. $Q = \rho_s V c_p dT$. Salt slurry must be easy to pump and heat transfer coefficient must exceed 150 W/m^{2}K for the system to function. Lesser the heat- coefficient, larger the size of required heat exchanger. Log mean temperature difference was calculated for the heat exchanger using the temperature of the hot tank for heating the water from 25 ^{o}C to 312 ^{o}C. The parameters of molten salt and heat exchanger have been listed in Table 1.

Table 1- Selected parameters of molten salt tank storage and heat exchanger

Heat capacity of nitrate salts	1.6 kJ/kg K
Cold tank temperature	538 K
Density of salts	1870 kg/m^3
Heat transfer co-efficient	355 W/m^{2}K
Area of heat exchanger	40 m^2
Mass flow rate	12,600 kg/hour

Counter flow heat exchanger was chosen for this model as shown in Fig.10.

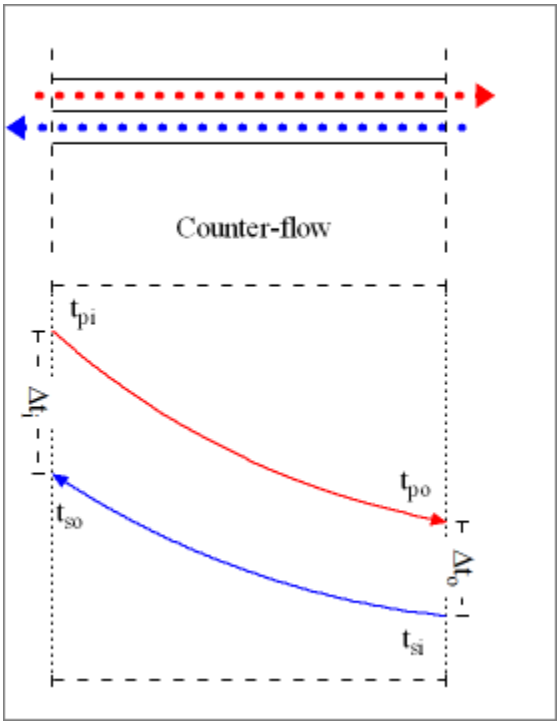


Figure 10- Counter flow configuration of heat exchangers

Heat rate oh heat exchanger was determined as shown in Fig.11.

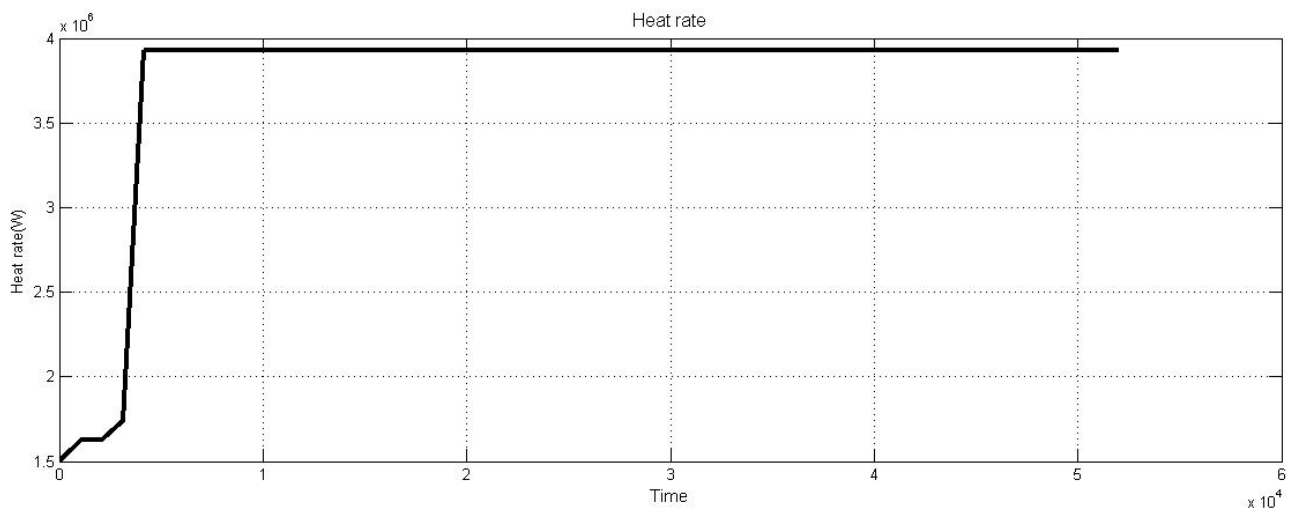


Figure 11- Heat rate by the heat exchanger

Mass flow rate was determined as 3.2 kg/s or 12,600 kg/hour required for optimum output from steam turbine

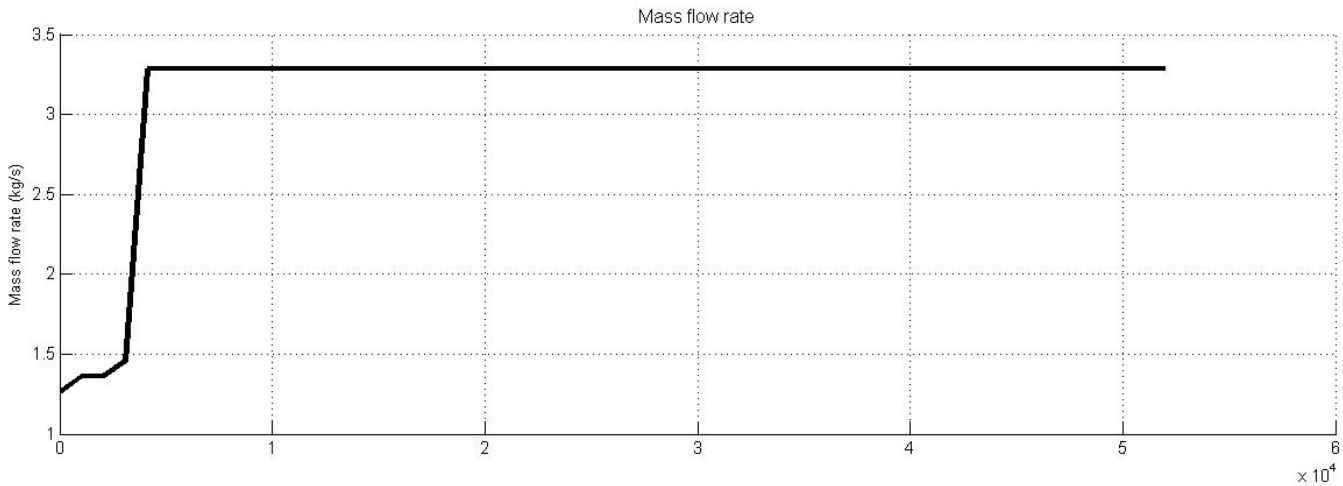


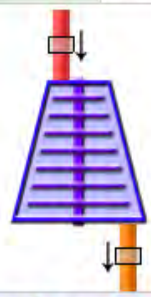
Figure 12-Mass flow rate of water

Using the steam system modeller tool SSMT by US department of energy [10], the steam turbine was modeled. The following parameters of the steam turbine were set to ensure optimised functioning and no condensation in the turbine.

Table 2- Steam turbine model

Solve for:		
Outlet Properties		
Inlet Steam		
Pressure*	1000	psig
Temperature *	900	K
Turbine Properties		
Selected Turbine Property	Mass Flow	
Mass Flow *	12600	kg/hr
Isentropic Efficiency *	80	%
Generator Efficiency *	95	%
Outlet Steam		
Pressure*	100	psig
* Required	Enter	[reset]

Inlet Steam		Mass Flow	12,600 kg/hr
Pressure	1,000.0 psig	Sp. Enthalpy	3,714.4 kJ/kg
Temperature	900.0 K	Sp. Entropy	7.163 kJ/kg/K
Phase	Gas	Energy Flow	46,802 MJ/hr



Isentropic Efficiency	80.0 %
Energy Out	7,062 MJ/hr
Generator Efficiency	95.0 %
Power Out	1,863.6 kW

Outlet Steam		Mass Flow	12,600 kg/hr
Pressure	100.0 psig	Sp. Enthalpy	3,153.9 kJ/kg
Temperature	619.2 K	Sp. Entropy	7.403 kJ/kg/K
Phase	Gas	Energy Flow	39,740 MJ/hr

$$Inlet\ energy\ flow = specific\ enthalpy \times mass\ flow$$

$$\dot{\epsilon}_i = h_i \dot{m}_r = 2{,}984.6 \text{ kJ/kg} * 12{,}600 \text{ kg/hr} = 46{,}802 \text{ MJ/hr}$$

$$Isentropic\ efficiency = \left(\frac{(inlet\ specific\ enthalpy - outlet\ specific\ enthalpy)}{inlet\ specific\ enthalpy - ideal\ outlet\ specific\ enthalpy}\right)$$

$$\eta_{is} = \frac{h_i - h_o}{h_i - h_{ideal}}$$

$$Outlet\ specific\ enthalpy = inlet\ specific\ enthalpy - isentropic\ efficiency \times (inlet\ specific\ enthalpy - ideal\ outlet\ specific\ enthalpy)$$

$$h_o = (h_i - \eta_{is})(h_i - h_{ideal})$$

= 3714.4 kJ/kg – 0.80 * (3714.4 kJ/kg - 2,758.6 kJ/kg)]

= 3,153.9 kJ/kg

Assumptions: Inlet mass flow is equal to equal outlet mass flow.

$$Energy\ out\ =\ (inlet\ specific\ enthalpy\ -\ outlet\ specific\ enthalpy)\ \times\ mass\ flow$$

$$\dot{\epsilon}_o\ = (h_i - h_o)m_r$$

= 7,062 MJ/hr

$$\text{Power output}\ =\ \text{energy out}\ \times\ \text{generator efficiency}$$

$$P_0 = \dot{\epsilon}_o \times\ \eta_{gen}$$

Power output = 1836.6kW

The output steam temperature is 619.2K which is enough to generate another 900 kW using the same system and re-pressurizing the outlet steam.

Overall efficiency of the system

Since we know the efficiency of the steam turbine (isentropic efficiency and generator efficiency), yet to be determined is the efficiency of the heat exchanger.

Efficiency of heat exchanger:

T = hot fluid temperature

$$t\ = cold\ fluid\ temperature$$

$$\overline{T}\ =\ average\ temperature\ of\ the\ hot\ fluid\ T$$

$$\overline{t}\ =\ average\ temperature\ of\ the\ cold\ fluid$$

$$NTU\ =\ number\ of\ transfer\ units$$

U = overall heat transfer coefficient, W/m^2 K

$$Cr\ =\ capacity\ ratio\ Cr = Cmin/Cmax$$

$$A\ =\ heat\ exchanger\ surface\ area, m2$$

$$Heat\ capacity\ rate\ of\ the\ cold\ fluid\ =\ Cc\ =\ (\dot{m}\,c_p)_c$$

$$rate\ of\ heat\ transfer\ =\ q\ =\ Cc\ (t2\ -\ t1)$$

$$Heat\ capacity\ rate\ of\ the\ hot\ fluid = Ch = \frac{q}{T_2 - T_1}$$

$$q\ =\ UA\ (\overline{T}\ -\ \overline{t}\ )\ =\ \eta\ Cmin\ NTU\ (\overline{T}\ -\ \overline{t}\ )$$

$$NTU\ \eta =\ q\ /\ Cmin\ (\overline{T}\ -\ \overline{t}\ )$$

$$\eta = \tanh NTU \frac{\sqrt{1 + {C_r}^2}}{2} \Big/ NTU \frac{\sqrt{1 + {C_r}^2}}{2})$$

Efficiency of the heat exchanger can be calculated using the above equations. For this model, the efficiency is 0.9.

Overall efficiency of the system

$$Overall\ efficiency\ of\ the\ system = (Eff.of\ the\ wind\ turbine\ and\ rotor) * (Eff.of\ heat\ generator) * (Eff.of\ TES) * (Eff.of\ heat\ exchanger) * (Isentropic\ eff.) * (Generator\ eff.)$$

$\eta' = C_p \times \eta_{hg} \times \eta_{TES} \times \eta_{he} \times \eta_{is} \times \eta_{gen}$ = 0.5 (1) (0.9) (0.7) (0.8) (0.95) = 0.2394

The power output is a steady value of 2736 kW or 2.7 MW from a single wind turbine in Oregon, US.

Conclusion

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