Exergetic Optimization and Performance Evaluation of Multi-Phase Thermal Energy Storage Systems

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Abstract

This study outlines a methodology for modeling and optimizing multi-phase thermal energy storage systems for solar thermal power plant (STPP) operation by incorporating energy and exergy analyses to a TES system employing a storage medium that can undergo multi-phase operation during the charging and discharging period. First, a numerical model is developed to investigate the transient thermodynamic and heat transfer characteristics of the storage system by coupling conservation of energy with an equation of state to model the spatial and temporal variations in fluid properties during the entire working cycle of the TES tank. Second, parametric studies are conducted to determine the impact of key design parameters on both energy and exergy efficiencies. The optimal values must balance exergy destroyed due to heat transfer and exergy destroyed due to pressure losses, which have competing effects. Optimization is utilized to determine parameter values within a feasible design window, which leads to a maximum exergetic efficiency of 87%.

Nomenclature

A	cross-sectional area (m^2)
c	specific heat of fluid ($\text{Jkg}^{-1}\text{K}^{-1}$)
E	energy (J)
Ex	exergy (J)
f	friction factor
h	heat transfer coefficient ($\text{Wm}^{-2}\text{K}^{-1}$)
k	thermal conductivity ($\text{Wm}^{-1}\text{K}^{-1}$)
m	storage fluid mass (kg)
$\dot{m}$	HTF mass flow rate (kgs^{-1})
N_L	number of tube rows
P	pressure (kPa)
P_h	heat transfer perimeter (m)
q	heat rate (W)
T	cross-sectional average temperature ($^{\circ}\text{C}$)
u	internal energy per unit mass (Jkg^{-1})
V	HTF velocity (m/s)
x	distance through the tank (m)
χ	correction factor

η	pump efficiency
ρ	fluid average density (kg m^{-3})
ψ	exergetic efficiency
CSP	concentrating solar power
HTF	heat transfer fluid
STPP	solar thermal power plant
TES	thermal energy storage

Subscripts

<i>b</i>	bypass
<i>c</i>	critical
<i>charge</i>	charging period
<i>discharge</i>	discharging period
<i>gen</i>	generator
<i>i</i>	inner
<i>in</i>	inlet
<i>max</i>	maximum
<i>o</i>	outer
<i>out</i>	outlet
<i>sol</i>	solar field
<i>stor</i>	storage
<i>tank</i>	tank (refers to HTF flow through tank)
<i>tot</i>	total
<i>wall</i>	wall (refers to wall of tube containing storage fluid)

1. Introduction

Thermal energy storage (TES) has been identified as one of the primary solutions to increase utilization of solar energy to bridge the gap between solar availability and peak demand, enable higher capacity factor, and permit flexible dispatchability in solar thermal power plants (STPP) [1]. This paper addresses a proposed single-tank TES system utilizing fluids in their two-phase (liquid-vapor) to supercritical states. In particular, it characterizes the thermodynamic performance of such a system during a full working cycle and explores the sensitivity of first- and second-law efficiencies to key design parameters (mass flow rate, tube geometry, operating temperature).

Current commercially deployed thermal storage systems in parabolic trough STPPs are two-tank sensible heat designs using molten salt as the storage medium [2, 3]. The most substantial challenge of this technology is its high capital cost at \$80/kWh_e [4]. Currently, the TES system operates with a small temperature difference between the hot tank (at 400°C) and the cold tank (at 300°C). The principal objective of the larger study undertaken in our group [9] is to investigate low-cost storage media options by considering operation at higher pressures typically found in the two-phase and supercritical regimes. By thermally cycling the storage fluid across a large temperature differential, the excursion in internal energy can access both sensible heat and latent heat of vaporization. The main challenge to such a system is the high pressure associated with the high temperature state, necessitating a thick-walled storage vessel. To mitigate this cost, the proposed design utilizes a single-tank TES system, effectively halving the required wall material.

Exergy analysis has become a fundamental approach in analyzing the thermodynamic performance of a wide variety of TES systems. While energy analysis is based on the First Law of Thermodynamics and provides information about the quantity of energy, exergy analysis is based on the Second Law of Thermodynamics and considers the quality of energy, which helps determine where available energy is inefficiently used. In fact, the utility of exergy methods are becoming more widely recognized in recent years. There have been many studies that utilize exergy analysis for TES performance evaluation, including PCM systems [5-12], sensible heat storage [13-16], and thermochemical systems [17-19]. However, there exists a wide range of approaches, and some have concluded that a standard convention is required in order to fairly assess TES performance [20-22]. The rationale is that the energy efficiency of a TES system has remained the primary metric for overall performance, but can be an

inadequate measure because it does not take into account all considerations necessary in TES evaluation (e.g., temperatures of the supplied and recovered energy, storage duration, and how nearly the system approaches ideal performance).

Previous researchers have investigated optimum specifications for key parameters in designing a wide range of TES systems [23-26]. Pioneering works by Bejan [27-29] utilized exergy analysis to evaluate the performance of a sensible heat storage (SHS) system and identify an optimum charging time and an optimum number of thermal units. Other studies have extended Bejan's analysis by modeling the entire charge-discharge cycle and concluded that a typical optimized system destroyed about 70-90% of the supplied exergy content, as reported by Krane [30] and Domanski and Fellah [31]. Aceves-Saborio et al. [32] presented a versatile lumped model for characterizing latent heat storage systems, with emphasis on the potential for a wide range of applications. In Rosen's work [16], the importance of modeling spatial variations of temperature in TES systems is demonstrated. Dincer [20, 33] investigated the viability of different SHS media and defined useful criteria for evaluating exergetic performance. Gunnewiek, Nguyen, and Rosen [34] studied the optimum discharge period for an SHS system for an entire working cycle. Erek and Ezar modeled the effect of charging processes on the storage process [35].

Additionally, optimization methods have emerged as a useful tool that can assist in energy forecasting and resource allocation, design and operation strategies, energy supply and demand modeling, and emission reduction. Baños et al. conducted a comprehensive review of research efforts that utilize a wide range of optimization schemes in renewable energy systems to obtain solutions to maximize energy resource allocation and reliability, and minimize system cost [36]. From the perspective of solar thermal power plant design, optimization can determine the optimal sizing, component specifications, and plant operation.

The objective of the present study is the thermodynamic evaluation and optimization of the overall performance of a TES system utilizing a storage medium undergoing operation in the liquid, two-phase, vapor, or supercritical regimes. This investigation extends the analysis of a prior study investigating the optimum operating parameters for the same system, as reported in [37, 38]. As in the prior study, this paper uses the thermodynamic model that improves fidelity by calculating the internal energy and temperature as functions of both time and space (distance through the tank, x). As mentioned in the previous work, naphthalene is used as a storage fluid in this instance, and is solely an example of using an equation of state to model any representative storage fluid of interest.

2. System model

The TES system consists of the storage fluid encapsulated in tubes connected end-to-end, within a common enclosure through which the heat transfer fluid (HTF) flows, creating an internal heat exchanger within the storage tank. In a large-scale system, multiple tanks can be connected in series to comprise the total energy storage capacity. The schematic shows the operation during discharging, when the HTF flows between storage and the steam generator. During charging, the HTF is directed to flow between TES and the solar field

Figure 1: Single-tank system configuration with tubes encapsulating the storage fluid (not drawn to scale).

The TES tank interacts with the solar field and the power block, as shown in Figure 1. In the charging loop, thermal power from the solar field is transferred to the encapsulated storage fluid via heat transfer from the HTF. The discharging loop reverses this process and removes energy from the thermal storage unit to the HTF for power generation. In this study, we analyze the charging and discharging periods individually with respective control volume inputs and outputs, as indicated in Figure 2. The system under study is a parabolic trough CSP plant with a 50 MW_e steam Rankine cycle.

Figure 2: Basic thermal energy storage system components, with the defined control volume under study for a) the charging loop, and b) the discharging loop.

Since the storage fluid temperature is high at the beginning of the discharging period, the HTF temperature will also be higher than the optimal value required by the steam generator; this value is taken as 390°C based on a performance study of a two-tank TES system performed by Kolb [3]. To reduce the HTF temperature to its design value, the HTF exiting the thermal storage unit is mixed with the HTF exiting the steam generator through a bypass loop.

The tubes encapsulate a fixed mass of storage fluid; because volume remains constant, the average density, or loading, ρ also remains constant. Thus, the discharging period effectively follows an isochoric (i.e. constant specific volume) process.

Figure 3: P-v diagram showing vertical lines of constant average fluid density.

The charging period begins at an initial two-phase state corresponding to low temperature, pressure, and internal energy, and ends at a final state corresponding to high temperature, pressure, and internal energy. Conversely, the discharging period reverses this process as energy is extracted from the TES system. At any instant in time, the internal energy and density can be related to the storage fluid temperature and pressure:

$$u = u(T_{stor}, P), \quad \rho = \rho(T_{stor}, P) \quad (1,2)$$

Thus with fixed density, if the storage fluid temperature is known, pressure can be evaluated from Eq. (2) and its internal energy can then be determined from Eq. (1). The Peng-Robinson equation of state and enthalpy departure function are implemented in the model, which have been shown to provide reasonable predictive capabilities for a wide variety of fluids [39]. Additional needed inputs for Eq. (1) are the fluid critical temperature and pressure, acentric factor, and the ideal gas specific heat and saturation pressure as functions of temperature. In this paper,

naphthalene is modeled as an example of storage fluid, though any fluid can be investigated if its thermodynamic properties, namely density and internal energy, are known as a function of temperature and pressure.

Table 1: System specifications of the example TES system.

Storage fluid properties	
Critical temperature, T_c	478°C
Critical pressure, P_c	4070 kPa
Acentric factor, ω	0.309
Molecular weight	128.17 kgkmol ⁻¹
Effective specific heat of storage fluid, c_{stor} (from Peng-Robinson)	2.18 – 2.98 kJkg ⁻¹ K ⁻¹
Thermal conductivity of storage fluid, k_{stor}	0.122 Wm ⁻¹ K ⁻¹
Fixed system parameters	
Specific heat of tube wall material, c_{wall}	0.5 kJkg ⁻¹ K ⁻¹
Thermal conductivity of tube wall material, k_{wall}	16.3 Wm ⁻¹ K ⁻¹
Density of tube wall material, ρ_{wall}	7990 kgm ⁻³
Ultimate tensile strength of tube wall, F_{tu}	291 MPa
Specific heat of HTF, c_{HTF}	2.446 kJkg ⁻¹ K ⁻¹
Thermal conductivity of HTF, k_{HTF}	0.098 Wm ⁻¹ K ⁻¹
Density of HTF, ρ_{HTF}	1060 kgm ⁻³
Tube length, L	12 m
Baffle spacing	2.4 m
Baffle cut	25%
Inner Nusselt number, $(hD)_i/k_{stor}$	6.92
HTF mass flow rate to generator $\dot{m}_{gen}$	547 kgs ⁻¹
Dead-state temperature, T_0	25°C
Storage capacity	1621 MWh _t
Varied system parameters	
Storage fluid mass (kg)	Baseline value: 2.49×10 ⁷ kg Range: 1.96×10 ⁷ – 2.66×10 ⁷ kg (Adjusted to meet 1621 MWh _t)
Outer Nusselt number, $(hD)_o/k_{HTF}$	271 – 556
Maximum system pressure	1.78 – 12.2 MPa

2.1. Energy analysis

The TES system and discharging loop are modeled with a set of cross-sectionally averaged energy and mass balances, as given below.

Energy balance of the storage fluid:

$$(\rho c A)_{stor} \frac{\partial T_{stor}}{\partial t} = -(h P_h)_i (T_{stor} - T_{wall}) + (k A)_{stor} \frac{\partial^2 T_{stor}}{\partial x^2} \quad (3)$$

Energy balance of the tube wall material:

$$(\rho c A)_{wall} \frac{\partial T_{wall}}{\partial t} = (h P_h)_i (T_{stor} - T_{wall}) - (h P_h)_o (T_{wall} - T_{HTF}) + (k A)_{wall} \frac{\partial^2 T_{wall}}{\partial x^2} \quad (4)$$

Energy balance of the HTF flowing through the storage tank:

$$(\rho c A)_{HTF} \frac{\partial T_{HTF}}{\partial t} + \dot{m}_{tank} c_{HTF} \frac{\partial T_{HTF}}{\partial x} = (h P_h)_o (T_{wall} - T_{HTF}) + (k A)_{HTF} \frac{\partial^2 T_{HTF}}{\partial x^2} \quad (5)$$

The equations also assume that the bulk mean and cross-sectional average temperatures are identical for the HTF; this assumption has been shown to be quite accurate from previous CFD studies using FLUENT investigating buoyancy-driven flows for tubes in different orientations [40, 41]. In addition, the temperature difference across the tube wall is neglected due to the tube wall thermal resistance being small compared with the overall thermal resistance. The selected value of h_i is a conservative value assuming solely conduction from FLUENT simulations of HTF flowing parallel to a single tube, through an outer cylindrical annulus control volume of radius 0.001 m, utilizing the same values for fluid properties (Table 1), mass flow rate (Table 3), and the value of h_o is obtained from Zukauskas [42]:

$$\overline{Nu}_o = \frac{(hD)_o}{k_{HTF}} = C Re_{max}^m Pr_{HTF}^{0.36} \left(\frac{Pr}{Pr_s} \right)^{1/4}, \text{ for } \begin{cases} N_L \geq 20 \\ 0.7 \leq Pr \leq 500 \\ 1000 \leq Re_{max} \leq 2 \times 10^6 \end{cases} \quad (6)$$

$$\text{where } \begin{cases} C = 0.35 \left(\frac{S_T}{S_L} \right)^{0.2} \text{ and } m = 0.6 \text{ (for } 10^3 \leq Re_{max} \leq 10^5 \text{)} \\ C = 0.021 \text{ and } m = 0.84 \text{ (for } 2 \times 10^5 \leq Re_{max} \leq 2 \times 10^6 \text{)} \end{cases} \quad (7)$$

C and m are for staggered tube configuration, and Re_{max} is based on the maximum fluid velocity within the tube bank. For a staggered arrangement,

$$S_D = \left[S_L^2 + \left(\frac{S_T}{2} \right)^2 \right]^{1/2} < \frac{S_T + D}{2} \quad (8)$$

$$V_{max} = \begin{cases} \frac{S_T}{2(S_D - D)} \bar{V} \\ \frac{S_T}{S_T - D} \bar{V} \end{cases} \quad (9)$$

where S_T , S_L and S_D are the transverse, longitudinal, and diagonal pitch, respectively. The storage tubes and tank are assumed to be the same length, and the boundaries at the tank inlet and outlet are assumed adiabatic. With regard to initial temperature conditions, the storage tank is initially assumed to be fully discharged. Then, cyclic operation is performed until the temperatures within the tank achieve a steady periodic condition.

The value of c_{stor} , the effective specific heat of the storage fluid, is defined as

$$c_{stor} = \frac{\Delta u}{\Delta T} \quad (10)$$

where Δu is the change in internal energy corresponding to an incremental change in temperature, ΔT , which is operationally taken across one time step. The effective specific heat is calculated at each spatial and time step, and is termed “effective” because it includes latent heat effects.

The inlet condition for the HTF is different for the charging and discharging loops. For the charging period, the inlet condition for the TES tank is governed by solar irradiation. The required solar field size is informed by a user-specified storage capacity (Table 1), which is held constant for all cases. In this study, the model assumes a subdivided solar field as described by Rovira, in which a portion of the solar field is dedicated to directly feed the power block for nominal plant operation during the day, and another portion that is dedicated to charging the TES system [43]. For a required energy output during the discharging period, the energy input during the charging period must be equal, therefore the solar field area dedicated to storage can be expressed as

$$A_{sol} = \frac{E_{tot}}{\int_0^{t_{charge}} \dot{Q}_{sol} dt} \quad (11)$$

where $\dot{Q}_{sol}$ is the solar irradiance as a function of time during the charging period. The solar field area for one loop is calculated by dividing the total area by the number of loops $N_{loops} = 120$, which is a standard solar field layout as described by Montes [44]. During charging, the mass flow rate is varied to achieve a constant inlet HTF temperature of 400°C, and the minimum and maximum mass flow rate through a single loop is set to 1 kg/s and 12 kg/s, respectively, based on the inputs from System Advisor Model [45] which are informed from practical considerations.

Table 2: Empirical steam generator specifications for an Andasol-type plant [3].

Steam generator specifications	
Mass flow rate, $\dot{m}_{gen}$	547 kg/s ⁻¹
Inlet temperature, $T_{gen,in}$	390°C
Outlet temperature, $T_{gen,out}$	289°C
Power output, q_{gen}	50 MW _e

Assuming well-insulated piping and tank such that heat losses are negligible, the equality of the HTF temperature between the tank and solar field during the charging period is

$$T_{tank,in} = T_{sol,out} \quad (12)$$

$$T_{tank,out} = T_{sol,in} \quad (13)$$

with similar expressions between the tank and steam generator during the discharging period. Once the discharge model is solved for a specified period of time (12 h), the quantity of thermal energy removed from storage, E_{tot} , can be calculated:

$$E_{tot} = (m\Delta\bar{u})_{stor} + (mc\Delta\bar{T})_{wall} + (mc\Delta\bar{T})_{HTF} \quad (14)$$

where the overbar represents the volumetric average and Δ denotes the difference between the initial and final states. Note that E_{tot} also represents the quantity of solar energy that had to be diverted to storage during the charging period. Then, for the discharging period, the power block specifications of an Andasol-type plant are modeled [46], as given in Table 2. Specifically, the value of E_{tot} needed to generate the desired turbine power of 50 MW_e for 12

hours at the turbine efficiency of 37% is $E_{tot} = 50 \text{ MW} \times 12 \text{ h} / 0.37 = 1621 \text{ MWh}_t$. The storage fluid mass is iterated upon to achieve the desired storage capacity.

Pressure losses in both energy and exergy analyses of TES systems cannot be evaluated accurately if design geometry is not known or is not within the scope of work. Pump work is often neglected in part because of this, but in many cases, it has been shown to be a significant energy cost and is an important consideration [30, 34]. Additionally, it has been demonstrated that the pressure loss has a greater effect in exergy analysis [22, 34].

Naturally, each TES system will model pressure drop differently based on respective tank geometry, flow regime, etc. For PCM systems utilizing shell-and-tube exchangers, several studies have adopted Kern's Method [47], or the Bell-Delaware Method [48]. Moreover, varying assumptions can be made in respective analyses to simplify the model for versatility. For a sensible storage system utilizing a heat exchanger, Krane approximated pressure drop using flow past a tube [30]. For first formulations, the Darcy-Weisbach equation and Friedel correlation have also been used. For this analysis, design geometry of the TES tank is modeled as crossflow over a large tube bank (using baffles in Figure 1). The tube bundle arrangement is based on standard design for shell-and-tube heat exchangers [49] and is kept constant. A constant isentropic pump efficiency of 0.93 is assumed. The pressure drop for a tube bank with a staggered configuration is given in [50]:

$$\Delta p = N_L \chi \left(\frac{\rho_{HTF} V_{max}^2}{2} \right) f \quad (15)$$

where N_L is the number of tube rows, V_{max} is the maximum velocity of the HTF, χ is the correction factor, and f is the friction factor; the latter two are given in charts as a function of Reynolds number and tube configuration [45]. The exergy destruction due to a pressure drop Δp is written as

$$\dot{W}_{pump} = \frac{\dot{m}_{HTF} \Delta p}{\rho_{HTF} \eta} \quad (16)$$

where this is both an energy and exergy cost. This is because exergy is defined as “useful work” [51]; in governing principles, work done to the system is equivalent in both the first and second law analyses.

2.2. Exergy analysis

The exergy supplied to and recovered from the TES tank can be evaluated solely from consideration of the inlet and exit conditions of the HTF. The exergy supplied to the TES tank during the charging period, assuming an incompressible fluid with constant specific heat and equivalent pressure states at the inlet and outlet due to the presence of the pump, can be expressed as

$$Ex_{charge} = \int_0^{t_{charge}} (\dot{m}c)_{HTF} \left[(T_{HTF,in} - T_{HTF,out}) - T_0 \ln \left(\frac{T_{HTF,in}}{T_{HTF,out}} \right) \right] dt \quad (17)$$

And the exergy recovered from the TES tank during the discharging period is similarly denoted as

$$Ex_{discharge} = \int_{t_{charge}}^{t_{discharge}} (\dot{m}c)_{HTF} \left[(T_{HTF,out} - T_{HTF,in}) - T_0 \ln \left(\frac{T_{HTF,out}}{T_{HTF,in}} \right) \right] dt \quad (18)$$

It should be noted that by defining the control volume across the TES tank, any exergy destruction that occurs between the storage fluid and HTF in the tank is still accounted for, as it will ultimately manifest itself in the outlet HTF temperature.

Exergetic efficiency can be expressed in a multitude of ways, depending on the analysis [21]. In this work, the analysis of a full working cycle allows exergetic efficiency to be expressed as

$$\psi = \frac{\int_{t_{charge}}^{t_{discharge}} (\dot{m}c)_{HTF} \left[T_{HTF,out}(t) - T_{gen,out} - T_0 \ln \left(\frac{T_{HTF,out}(t)}{T_{gen,out}} \right) \right] dt - E_{pump}}{\int_0^{t_{charge}} (\dot{m}c)_{HTF} \left[T_{sol,out} - T_{HTF,in}(t) - T_0 \ln \left(\frac{T_{sol,out}}{T_{HTF,in}(t)} \right) \right] dt} \quad (19)$$

where t_{charge} and $t_{discharge}$ are the charge and discharge time, respectively, and E_{pump} is the pumping energy during a full charge-discharge cycle. The pump consumption is equivalent to the irreversibility, or exergy destruction, due to pressure drop within the system, in addition to the non-isentropic nature of the pump. The subsequent section of the study is the investigation of several parameters on system performance. The effects of charging mass flow rate, tube bundle geometry, and storage and yield temperature are studied and described in detail in the next section.

2.3. Simulation parameters

In this analysis, the sole constraint will be energy output – each TES case must discharge 1621 MWh_i in order to fairly compare each case. Using energy output as an objective constraint is a rational approach for comparing TES systems with varying parameters in order to determine optimal operating values, such that exergy destruction is minimized while maintaining the same output. The TES system model will determine the required storage fluid mass, while the charging period will take this as an input. Certain parameters are varied to evaluate impact on overall system performance; the operating range for each of these parameters is shown in Table 3. However, it should be noted that many parameters play a role in optimizing any TES system and in this analysis, several key variables were maintained constant:

- 1) Storage fluid: In this paper, the TES system model examines naphthalene as an example storage fluid, although other fluids are being considered [52]. The storage fluid properties play a key role in energy density. Additionally, the encapsulation geometry can also change to spheres, which affects the trade-off between heat transfer and pressure drop.
- 2) Heat transfer fluid: Therminol VP-1 is used as the HTF in this study; however, molten salt as the HTF would allow for higher temperature operation and consequently higher Carnot efficiency power blocks. Also, air, steam, and supercritical CO₂ have been investigated in literature as possible HTF for advanced power cycles in CSP.
- 3) Tank length: The tank length is kept constant in this study at a technically feasible value – however, a longer tank may improve thermal stratification in the tank which can improve exergetic performance.

Table 3: Design and operating parameters under study.

Parameter	Design window	Baseline value
Storage fluid loading (kgm ⁻³)	500 – 700	600
Mass flow rate, $\dot{m}_{charge}$ (kgs ⁻¹)	200 – 1000	600
Maximum storage temperature, $T_{stor,max}$	300 – 500	500
Tube radius (m)	0.02 – 0.05	0.025
Tube pitch	1.15 – 1.35	1.25
Power block design inlet temperature (°C)	300 – 390	390

Weather data from Daggett, California, USA were used to bring realistic operating conditions to the analysis [53]. The solar direct normal irradiation (DNI) data measured for a typical August 1 day in Daggett, California are presented in Figure 4. This is used across all the different cases.

Figure 4: Solar irradiation curve for Daggett, CA measured for a sunny day used for numerical simulation [53].

The system model was implemented in MATLAB using the Peng-Robinson equation of state and enthalpy departure function calculated for naphthalene using data of Barrow and McClellan [54], and the saturation pressure was modeled using the Pitzer correlation [55]; the model is discussed in more detail in [37]. The governing equations were discretized explicitly in space and time. The Courant number was 0.15, which was chosen after careful examination that further time and spatial step refinement did not show noticeable change in the results. The residual error tolerance is converged to 10^{-5} to achieve a steady periodic temperature profile in the tank.

3. Numerical results and discussion

3.1. Design point simulation

The temperature profile of the storage fluid in the tank during the charging period is shown in Figure 5a, where τ is the fractional time. The spatial gradient within the tank shows the TES tank remains colder at the outlet. The degree of spatial temperature variation will vary based on the mass flow rate: higher mass flow rates will lead to a more uniform temperature distribution. During the discharging period, the HTF flow direction is reversed as the cold HTF extracts energy from the tank, a cold front advances through the tank. This is typical of thermocline systems, in order to maintain the thermal gradient in the tank generated during the charging period. The system is designed to adjust the storage fluid mass such that the temperature at the exit of the tank reaches the design value for the turbine inlet temperature.

Figure 5: Temperature profile within the tank during a) charging and b) discharging.

3.2. Effect of storage fluid loading

Next, the impact of storage fluid loading is evaluated. Varying the storage fluid loading corresponds to traversing the P - v diagram across different phase regimes. Higher storage fluid loading is beneficial because it corresponds to larger energy storage density, i.e. the required thermal energy can be stored in a smaller volume. However, increasing storage fluid loading also has the negative impact of increasing maximum system pressure (holding maximum temperature fixed); this impacts either the tube radius, or the number of tubes needed to meet the energy storage capacity requirement.

Figure 6: Effect of storage fluid loading on exergy destruction.

Figure 6 illustrates the effect of storage fluid loading on exergetic performance. The variation of storage fluid loading directly dictates the phase of the storage fluid during operation. Higher storage fluid loading leads to higher energy density, but at the expense of higher pressure. Therefore, it is clear that there is an optimum storage fluid loading that balances these competing effects. For the system under study, closer inspection reveals that operating between saturated liquid and two-phase mixture is the optimal loading. Intuitively, this makes sense because sub-cooled liquid results in higher pressures which necessitates a larger tube radius (increased exergy destroyed due to heat transfer), while operating in the liquid-vapor phase with constant pressure does not yield any appreciable benefits in thermophysical properties.

3.3. Effect of varying operation mode

As mentioned previously, the results shown thus far assume an operation mode that achieves constant inlet HTF temperature by varying the HTF mass flow rate. The following parametric study investigates the alternative operation mode of constant HTF mass flow rate with varying inlet HTF temperature. In Figure 7, the transient behavior at different locations in the charging loop is presented for varying mass flow rate. The results of Figure 7 illustrate the competing effects that must be balanced.

(a) $\dot{m}_{HTF} = 400 \text{ kg/s}$

(b) $\dot{m}_{HTF} = 600 \text{ kg/s}$

(c) $\dot{m}_{HTF} = 800 \text{ kg/s}$

Figure 7: The transient behavior of the TES tank during the charging period, for different cases with increasing HTF mass flow rate.

As mentioned previously, based on the defined control volume (i.e. around the TES tank), exergy destroyed through the thermal interaction between the HTF and storage fluid inside the tank will exhibit its overall effect in the outlet HTF temperature. Therefore, we observe that the difference between the inlet and outlet HTF temperature must be minimized because it dictates the exergy destroyed through this mechanism. This is primarily achieved by increasing mass flow rate, which can be seen in Figure 7; similar results have been presented by Erek and Dincer

[56, 57] and by Mawire et al. [14]. However, the increase in mass flow rate clearly results in higher pressure drop and thus, higher exergy destroyed due to viscous effects. The total exergy destroyed will then observe a minimum by balancing this trade-off that occurs for varying mass flow rate.

It becomes clear that these parameters are fundamental to plant operation and warrant detailed investigation (i.e. the HTF exiting the solar field must not exceed a given temperature for thermal stability purposes, and the mass flow rate must not exceed a value for pump work to be reasonable). The results of varying mass flow rate on total exergy destruction and exergetic efficiency will be illustrated in detail in the following section.

3.4. Effect of tube radius and pitch

It should be noted that balancing exergy destruction associated with heat transfer and with pressure drop is also dependent on many design geometry parameters, such as baffle spacing, tube radius, tube or tank length, and tube pitch among others. To investigate the optimum design geometry in the system under study, tube pitch and tube radius are selected as representative variables for the design geometry space, while other parameters, such as baffle spacing and cut, are taken from nominal values in heat exchanger design [49, 58].

Similar TES systems have also been studied to optimize encapsulated tubes and sphere size and spacing to balance pressure losses with adequate heat transfer behavior. Several studies for packed beds reported that increasing the radius of the storage material lessens the pressure drop, but thermal performance of the system declines due to lower surface area available for heat transfer [59, 60]. The system under study is also subject to this trade-off. The influence of tube radius and tube pitch on heat transfer and pressure drop are shown in Figure 8.

Figure 8: Effect of tube radius and pitch on heat transfer coefficient and pressure drop.

The examination of tube pitch indicates that pressure drop varies more markedly than the heat transfer coefficient as tube pitch decreases, similar to the results reported for the parametric study of baffle spacing [61]. The tube pitch is constrained to a design window to maintain a feasible tank volume, as well as ensure that the transient heat transfer behavior is sufficient enough to allow for a reasonable charge and discharge time. These values are informed from the heat exchanger design [49].

3.5. Effect of storage and yield temperature

With recent advances in CSP, the yield temperature at which energy is extracted is consistently increasing to higher temperatures in order to increase Carnot efficiency. However, operating the entire system at higher temperature increases entire plant cost because high temperature materials must be utilized in plant components such as the solar field and TES system [62]. In the system under study, energy storage capacity is held constant across all cases. Therefore, storing at 400°C requires more storage fluid mass (i.e. higher number of tubes) than at 500°C. This also leads to increased pump work for a given mass flow rate. Moreover, the exergy destruction due to a finite temperature difference is also higher in the 400°C case, which requires a higher thermal mass. This results in a higher degree of temperature nonuniformity within the tank – the difference between the inlet and outlet HTF temperature becomes comparatively larger during the charging period.

Additionally, it is imperative to consider the operation of the power block in concert with the entire system. Various power block configurations implemented in STPPs have been studied, such as sliding pressure and temperature [46, 63], scaling empirical models [24], and varying extraction points [44]. A typical 50 MW_e steam Rankine cycle is modeled as the usual size for parabolic trough STPPs. Similar to the SEGS VI power block, sliding pressure operation is allowed for flexibility from nominal operation and evaluated as a design variable using transfer functions developed by Patnode [24], and are detailed in Table 4. The design value for the inlet HTF temperature is 390°C, and the minimum allowable temperature is 300°C. The storage temperature is kept at 10°C above the turbine inlet temperature, which impacts the charge and discharge time. It is important to note that the practical limit of Therminol VP-1 is 400°C, therefore the cases with temperatures exceeding this are hypothetical. Still, the general result can provide insight into the trade-offs between power block specifications and TES design. The effect of storage and yield temperature is shown in Figure 9.

Table 4: Specifications of sliding pressure operation for $\dot{m} = 547 \text{ kgs}^{-1}$ for the 50 MW_e Rankine cycle.

Inlet HTF temperature (°C)	Outlet HTF temperature (°C)	Turbine gross output (MW _e)	Efficiency (%)
300	250	20	29.2
350	274	36	34.6
390	289	50	36.9

Figure 9 shows as the yield temperature decreases below the design value, energetic and exergetic efficiency both decrease. It is also clear that exergy differs more markedly than energy. Naturally, the most important aspect is the lower Carnot efficiency at lower operating temperatures. Under closer inspection, the lower inlet HTF temperature for the power block leads to increased flow rate through the bypass loop. This causes additional thermal mixing, which increases the exergy destroyed due to heat transfer as well as pressure losses.

Figure 9: Energy and exergy efficiency for different power block configurations.

3.6. Exergetic optimization

The goal of constructing a system is often to achieve the maximum efficiency at the lowest cost within the boundaries of the technical and economic constraints. Optimization problems can be ill-conditioned with large gradients, which necessitate higher computing resources. Numerous studies have applied optimization schemes to renewable energy system design and operation such as the Nelder-Mead Simplex method, Lagrangian relaxation, and sequential linear or quadratic programming. Depending on the characteristics of the optimization problem, conventional optimization methods may not be feasible; for optimization problems that are NP-hard, polynomial time algorithms require exponential computational time which proves difficult and impractical [36]. To solve this, many researchers have applied heuristic approaches to solar thermal optimization problems including particle swarm [64], genetic algorithms [65], and artificial neural networks [66].

In the previous section, the effect of design parameters was explored by individually varying each parameter from the baseline case. Now, to obtain an optimized solution, the optimization problem is solved using sequential linear programming. In addition to the design constraints, the system optimization is constrained to meet certain requirements corresponding to system performance, such as a minimum charge time, maximum discharge time, energy storage density, and exergetic efficiency. The formulation of the optimization problem is subject to

$$\begin{aligned}
0 &\leq t_{charge} \leq t_{charge,crit} \\
0 &\leq t_{discharge} \leq t_{discharge,crit} \\
E_{out} &= \eta \cdot W_{turb} \cdot t_{discharge} \\
\psi - \psi_{crit} &\geq 0
\end{aligned} \tag{20}$$

The optimization problem utilizes a simplex search method that, given positive convexity, ensures an increase in the objective function. The simplex is defined by k -dimensional space with $k+1$ vertices, where k is the number of design parameters under study. The initial guess for the TES system configuration creates $k+1$ vertices, and the primary vertex is selected by the highest objective function value. The numerical model is called for each iteration to calculate the objective function and apply the design constraints. Then, upon convergence, the optimal parameters and best value of the objective function are those corresponding to the primary vertex. The objective function in this study is the exergetic efficiency; the model can be easily modified to accommodate a multi-objective function defined by the user.

Table 5 outlines the design parameters corresponding to the optimal system design and operation obtained from the parametric studies, as well as the optimization algorithm.

Table 5: Optimal values for the design parameters evaluated by parametric study and optimization.

Parameter	Parametric study	Optimization
Storage fluid loading (kgm^{-3})	600	600
Charging mass flow rate, $\dot{m}_{charge}$ (kgs^{-1})	250	210
Maximum storage temperature, $T_{stor,max}$	425	400
Tube radius (m)	0.027	0.022
Tube pitch	1.25	1.12
Power block design inlet temperature	390	390
Exergetic efficiency	84.1	87.9

The results shown in Table 5 illustrate that the exergetic performance can be increased from 84.1% to 87.9% by utilizing optimization schemes, with the respective optimal values for the design parameters. This increase in performance allows the TES system to achieve higher capacity factors and operate within the given set of constraints. However, this will often not correspond with a cost optimum [34], which is explored in the field of thermoeconomics by applying cost to exergy analysis, for which many methodologies have been developed [20].

Conclusion

This paper presents a thermodynamic model that evaluates the system performance of a single-tank TES system utilizing storage fluids encapsulated in tubes. Energy and exergy analyses were conducted to evaluate overall system performance as the storage fluid traverses between liquid, two-phase, vapor, or supercritical state. Results generated from the thermodynamic model have been presented using an example TES system utilizing naphthalene contained within stainless steel tubes – it is worth noting that there are certainly other storage and containment material combinations worthy of investigation, some of which are still being explored in this research effort. Using exergy-based optimization, it was shown there is an optimum set of design and operating parameters that balance the trade-off between exergy destroyed due to heat transfer and exergy destroyed due to pressure losses. Additionally, a maximum exergy efficiency of 87% was evaluated, for a given example TES system, by utilizing optimization. For further work, the thermodynamic model can be extended to perform thermoeconomic analysis of the system. Additionally, the system can be modeled using various fluids in combination with cost data to determine the most cost-effective storage fluid candidate. Moreover, different HTFs can be modeled in tandem with advanced power cycles in order to evaluate the performance of even more varied power block configurations.

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