Review

A Review on the Performance Indicators and Influencing Factors for the Thermocline Thermal Energy Storage Systems

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Abstract: Thermal energy storage (TES) system plays an essential role in the utilization and exploitation of renewable energy sources. Over the last two decades, single-tank thermocline technology has received much attention due to its high cost-effectiveness compared to the conventional two-tank storage systems. The present paper focuses on clarifying the performance indicators and the effects of different influencing factors for the thermocline TES systems. We collect the various performance indicators used in the existing literature, and classify them into three categories: (1) ones directly reflecting the quantity or quality of the stored thermal energy; (2) ones describing the thermal stratification level of the hot and cold regions; (3) ones characterizing the thermo-hydrodynamic features within the thermocline tanks. The detailed analyses on these three categories of indicators are conducted. Moreover, the relevant influencing factors, including injecting flow rate of heat transfer fluid, working temperature, flow distributor, and inlet/outlet location, are discussed systematically. The comprehensive summary, detailed analyses and comparison provided by this work will be an important reference for the future study of thermocline TES systems.

Keywords: thermal energy storage (TES); single-tank thermocline; thermal stratification; performance indicator; flow distributor

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1. Introduction

The exploitation of renewable energy sources plays a key role in the climate change mitigation. Among the existing technologies, the solar thermal conversion has shown its feasibility and applicability in various industrial and domestic sectors [1–3]. Taking the concentrated solar power (CSP) as an example, heat is provided by the solar field, and the integration of thermal energy storage (TES) module allows the plant output be not strictly dependent on the solar irradiation in time, which can greatly improve the output stability and dispatchability [4,5]. In fact, the TES module is of critical importance for the solar thermal conversion systems, which significantly affects the overall performance [6,7].

Particularly for the TES integrated CSP systems, there are generally four basic concepts: (a) active direct with two-tank, (b) active indirect with two-tank, (c) active indirect with single-tank, and (d) passive, as shown in Figure 1 [8]. For the direct type, the heat transfer fluid (HTF) circulating within the solar collectors is the same as that within the TES modules. By contrast, in indirect systems, the HTF loops within the solar collectors and the TES modules are separated, and the heat exchange between them is achieved by heat exchangers. Currently, the direct two-tank configuration has been most-widely adopted in the utility-scale CSP plants. In this case, the cold HTF at T_c coming from the cold tank will go through the solar field, where it will be heated up to the high temperature T_h and then sent to the hot tank for storage (charging). For the discharging, the HTF in the hot tank can be pumped to the heat-work conversion units, e.g., Rankine cycle modules for power generation. In order to further improve the performance of TES system and to cut off its

high construction cost, the concept of single-tank thermocline-based TES was proposed and developed. As shown in Figure 1c, the basic operating processes of the charging and discharging in the single-tank thermocline system are similar to those in the two-tank TES case. The only difference is that the high-temperature and the low-temperature HTFs are stored in the single tank, and these two zones are separated by a transition region with a time-dependent temperature gradient called thermocline [9]. During the charging process, the hot HTF enters the tank from the top, while the initial cold HTF stored in the tank is extracted from the bottom and transported to the solar field. For the discharging process, the hot fluid extracted from the top of the thermocline tank is transported to the heat exchangers for the power generation, and then the cold HTF out of the heat exchangers returns to the bottom of the tank. The single-tank thermocline concept greatly reduces the quantity of HTF used in the circulating loops compared to the two-tank configuration, which significantly lowers the capital expenditure [10].

Figure 1. Integration concept of TES system in a solar thermal power plant [8]. (a) Active direct concept for two-tank TES integration; (b) active indirect concept for two-tank TES integration; (c) active indirect concept for single-tank thermocline TES integration; (d) passive concept for TES integration. Reprinted with permission from ref. [8]. 4 October 2021, Elsevier.

The evaluation criteria of the performance (i.e., the performance indicators) are of importance for guiding the design of thermocline TES systems. In fact, researchers have proposed many performance indicators [4], including the charging and discharging efficiencies [11], the capacity ratio [12], the exergy efficiency [13], the tail factor [12], and the stratification number (*Str*) [14]. Nevertheless, a clear classification and analysis on these performance indicators are lacking among the existing literature. Here, we classify them into three categories: (1) ones directly reflecting the quantity or quality of the stored thermal energy; (2) ones describing the thermal stratification level of the hot and cold regions; (3) ones characterizing the thermo-hydrodynamic features within the tank. Note that the performance indicators of Category #1 can be defined based on the first law of thermodynamics (e.g., the charging and discharging efficiencies), the second law of thermodynamics (e.g., the exergy efficiency), or the data analysis (e.g., the tail factor). Moreover,

for the thermocline tanks, the hot and cold regions stay above and below orderly due to their different densities, as shown in Figure 2. Generally, there are three levels of stratification [9]: (a) highly stratified level; (b) moderately stratified level; (c) fully mixed level. Owing to the critical importance of the thermal stratification, several indicators of Category #2 have been proposed to characterize it, such as the stratification number [14] and the MIX number [15]. Furthermore, some other indicators (Category #3) defined according to the thermo-hydrodynamic characteristics are needed to estimate the flow regimes and patterns within the tank [16], such as Re number and Pe number. More explanations on the definitions, functions, and comparisons of these performance indicators will be given in Section 2.

Figure 2. Different levels of stratification in thermocline TES tank [9]. (a) Highly stratified; (b) moderately stratified; (c) fully mixed storage. (1) Hot region; (2) thermocline region; (3) cold region; (4) uniform temperature. Reprinted with permission from Ref. [9]. 4 October 2021, Elsevier.

There have been tremendous researches devoted to investigating the effects of influencing factors on the performance of TES systems [14,15,17–22]. Figure 3 provides a list of these influencing factors in a thermocline TES system, and they actually serve as the design variables that can be adjusted to improve the specific performance indicators. Generally, these influencing factors are classified into [4] (i) the operating conditions (e.g., velocity, HTF, cyclic operation, operating temperature, and cut-off temperature), (ii) the tank geometrical parameters (e.g., inlet/outlet, flow distributor, tank form, aspect ratio, and insulation), and (iii) the packing conditions (e.g., the packed media and packing configuration). The detailed discussions upon some of these influencing factors, including injecting HTF flowrate (velocity), working temperature, inlet/outlet location and flow distributor, will be addressed in Section 3. These four influencing factors above have the dominant effects on the thermal stratification level as well as the performance of the thermocline TES systems, and they are relatively adjustable in practice. However, a comprehensive and comparative analysis on them is still rare, which is the reason why we concentrate on them here.

The present paper could thereby be a useful reference by clarifying the performance indicators and the relevant influencing factors for the thermocline TES systems. Various performance indicators and their influencing factors in the existing literature papers are collected, classified, and analyzed systematically.

Figure 3. Schematic influencing factors (design variables) for thermocline thermal performance.

2. Performance Indicators of Thermocline TES System

Generally, the performance of the TES systems can be characterized by two distinct methodologies: the graphical manners (i.e., the cartographies for the temperature or flow fields within the storage tank) and the performance indicators (i.e., some parameters calculated to describe one or several features of a TES system). The former ones are straightforward and can provide the detailed local features of temperature or velocity distributions [18,23–27], but they cannot directly indicate the global performance of the thermocline tanks. Since it is frequently needed to analyze the performance of a TES module integrated in a whole energy system, more general performance indicators are necessary. Table 1 summarizes three categories of performance indicators for the thermocline TES systems. The detailed explanations and discussions on these performance indicators are given as follows.

Table 1. Summary of common performance indicators for thermocline TES tanks.

		Indicator	Remarks
Category #1	1st law	Energy efficiency (charging, discharging, and overall)	-Functional dependent on the ratio of the net difference between input and output to the total input energy. -The overall efficiency sometimes expresses the effect of cyclic operation.
		Capacity ratio	-Functional dependent on the ratio of the net difference between input and output to the theoretically maximum energy stored inside the TES tank.
	2st law	Exergy/entropy efficiency	-Key indicator for assessing the useful work transferred and extracted during energy release process.
	Data analysis	Tail factor	-Vary between 0 and 1. -Independent on the thermal properties of HTF, the geometries, and the operating conditions. -Represent the capacity/difficulty to complete the fully energy storage/extraction.
Category #2		Temperature gradient	-Usually represent the stratification degree versus the axial position or the time. -May not be practical for the comparison between different TES systems.
		Thermocline thickness	-Suitable for the situation of stable thermocline evolution and regular thermocline form such as in packed-bed TES tanks. -Sometimes not suitable for single-medium thermocline (SMT) tanks.
		Mix number	-Vary between 0 and 1. -A bulk indicator for TES tank against time.
		Stratification number	-Assessment of the stratification decay inside the TES tank both for charging and discharging.
Category #3		Richardson number Ri	-In bulk or gradient form. -The critical value is often set at 0.25 (cf. Table 2); stable thermocline while the value is superiors to 1.
		Péclet number Pe	-The ratio of the convection to the diffusion rates, influencing largely the thermocline thickness, especially in SMT tanks. -In bulk or local form.
		Froude number Fr	-Usually used to characterize the gravity current for the inlet diffuser design. -Recommended range of the inlet Fr is usually found smaller than 2.
		Reynolds number Re	-Different forms depending on the choice of characteristic length. -A basic factor for the determination of flow regime.

2.1. Category #1

The indicators defined obeying the law of energy conservation (1st law) can characterize the quantity of the stored thermal energy. The ratio between the integral of net stored energy to the integral of entire inlet energy is called the charging efficiency η_{ch} , while the integral of net extracted energy to the integral of initially stored energy is the discharging efficiency η_{dis} ,

$$\eta_{ch} = \frac{E_{in} - E_{out}}{E_{in}} = \frac{\int_0^t \dot{m} C_p \{T_{in}(t) - T_{out}(t)\} dt}{\int_0^t \dot{m} C_p \{T_{in}(t) - T_0\} dt} \quad (1)$$

$$\eta_{dis} = \frac{E_{out} - E_{in}}{E_{stored}} = \frac{\int_0^t \dot{m} c_p \{T_{out}(t) - T_{in}(t)\} dt}{E_{stored}} \quad (2)$$

where T_{in} and T_{out} are the time-dependent inlet and outlet temperature of the HTF, respectively, C_p is the specific heat of the HTF and $\dot{m}$ is the mass flow-rate. Note that the cut-off temperature in the end of charging or discharging is dependent on the real-world operating conditions. An appropriate cut-off temperature must be chosen to avoid the overheating/sub-cooling of HTF and guarantee a high storage capacity. Several methods about the selection of the cut-off temperature were proposed [28] and the threshold values $[T_{20}, T_{80}]$ are usually adopted [17].

The capacity ratio (σ) is the ratio of the actual stored thermal energy to the maximum energy storage capacity [12],

$$\sigma = \frac{E_{stored}}{E_{stored}^{max}} = \frac{E_{in} - E_{out}}{V_{fluid} \cdot (c_{ph} \cdot T_h \cdot \rho_h - c_{pc} \cdot T_c \cdot \rho_c)} \quad (3)$$

σ equals to 1 corresponds to a fully charging process. Note that the indicators based on 1st law merely focus on the quantity of the stored energy, while the energy quality issue cannot be well addressed.

Exergy (entropy) can describe both the quantity and quality of energy stored in the TES tanks, with combining 1st and 2nd laws [4]. The work dissipation due to friction, mixing, and heat loss can be addressed by the entropy or exergy analysis [29–31]. For instance, in order to assess the deviation from an actual TES tank to the perfectly stratified tank, Shah and Furbo [21] formulated the exergy efficiency as the ratio of the actual exergy to the exergy of the perfectly stratified tank:

$$\eta_{Ex} = \frac{E_x(exp)}{E_x(str)} \quad (4)$$

where,

$$E_x = \sum_{j=1}^J m_j \cdot c_p \cdot (T_j - T_c) - \sum_{j=1}^J m_j \cdot c_p \cdot T_c \cdot \ln\left(\frac{T_j}{T_c}\right) \quad (5)$$

and j is the layer number and m_j is the mass of control volume. Such efficiency varies from 0 to 1, with 0 meaning the fully mixed state and 1 referring to the perfectly stratified state.

For the TES integrated CSP plants, the outlet temperature of the HTF from the TES tank is a crucial parameter that influences the overall performance and thus must be carefully monitored and controlled during the operations. During the charging process, the cold HTF extracted from the tank should not be overheated to avoid the inconvenient control operations, including the solar field defocusing and the adjustment of the HTF mass flow rate [32,33]. By contrast, in the discharging process, the sub-cooling of the hot HTF from the storage tank to the power generation module should also be avoided [28]. In practice, a high-steep slope for the outlet temperature curve is expected to have a positive effect on the capacity ratio. Thus, as a data analysis, a dimensionless indicator called the tail factor is calculated from the dimensionless outlet temperature profile against the dimensionless time [12]. The tail factor is independent on the thermal properties of HTF, the geometries, and the operating conditions. To some degree, it reflects the quality of the stored thermal energy, and importantly it is convenient to calculate and compare among different storage tanks.

2.2. Category #2

The evolution of temperature gradient against time or height serves as the general evaluation of thermocline behavior which is one of the most common manners [34]. Nevertheless, the observation of temperature gradient is not convenient to compare the thermal performance among different TES tanks. The thermocline thickness, describing the de-stratification state, is inappropriate for the single-medium thermocline (SMT) tank, since the thermocline front is not always flat, as illustrated in Figure 4 [35]. The thermocline surface or volume calculated based on 2D or 3D CFD simulations could be more suitable.

Figure 4. Streamlines (left) and isotherms (right) inside the tank [35]. (a) $Re = 100$; (b) $Re = 200$; (c) $Re = 400$; (d) $Re = 600$; (e) $Re = 800$; (f) $Re = 1000$. Reprinted with permission from ref. [35]. 4 October 2021, Elsevier.

The stratification number (Str), which was proposed by Fernandez-Seara et al. [36], describes the ratio between the mean temperature gradients for each radial position at each time interval and the temperature gradients at the beginning of the charging process. The Str is given by,

$$Str(t) = \frac{(\partial T / \partial z)_t}{(\partial T / \partial z)_{max}} \quad (6)$$

$$\left(\frac{\partial T}{\partial z} \right)_t = \frac{1}{J-1} \left[\sum_{j=1}^{J-1} \left(\frac{T_{j+1}^t - T_j^t}{\Delta z} \right) \right] \quad (7)$$

$$\left(\frac{\partial T}{\partial z} \right)_{max} = \left[\frac{T_{max} - T_{min}}{(J-1) \cdot \Delta z} \right] \quad (8)$$

These two positions of nodes j and $j+1$ should be equidistant on the radial location.

On the contrary to Str that concerns the temperature gradients against the height of tank, the mix number (MIX) focuses on the instantaneous degraded energy during the mixing process [15]:

$$MIX = \frac{M_{str} - M_{exp}}{M_{str} - M_{fullymixed}} \quad (9)$$

where

$$M = \sum_{j=1}^J y_j \cdot E_j \quad (10)$$

$$E_j = \rho_j \cdot c_p \cdot V_j \cdot T_j \quad (11)$$

where y is the distance between the tank bottom and the center of node j . MIX varies between 0 and 1. A smaller MIX refers to the weaker mixing degree, and thus the better thermal stratification state.

2.3. Category #3

Richardson Number (Ri)

The Richardson number Ri is the ratio of buoyancy force to mixing force [37]. The formula for the bulk tank is commonly used:

$$Ri_{bulk} = \frac{g \cdot \beta \cdot (T_{top} - T_{bottom}) \cdot L}{v^2} \quad (12)$$

where L is the characterization length of storage tank and v is the characteristic velocity. The value of Ri_{bulk} is usually employed to mark the global stratification state of thermocline tank under the given operating condition (e.g., working temperature range and flow rate) [38,39]. Sometimes, it is dependent on the outlet temperature at the charging process [40,41]. Generally, the stratification level is higher at larger Ri_{bulk} . Some more detailed comments

on this indicator are given in Table 2. In order to explore the local thermal stratification inside the tank, the gradient form of Ri number was also derived [18]:

$$Ri_{gradient} = \frac{\frac{g}{\rho_0} \cdot \frac{\partial \rho}{\partial z}}{\left(\frac{\partial v}{\partial z}\right)^2} \quad (13)$$

Table 2. Comments on the bulk Richardson number for indicating the thermal stratification level.

Reference	TES Tank Structure	Ri Range	Remarks
Yee and Lai, 2001 [38]	SMT with or without porous manifold	0.01, 1, 100	- $Ri = 0.01$: inertial force dominates and the mixing caused by entrainment does not involve any stratification. - $Ri = 1$: forced and natural convection are in competition. - $Ri = 100$: thermal stratification becomes more evident.
Njoku et al., 2016 [39]	SMT	10, 10^2 , 10^3 , 10^4	-Both exergy efficiency and entropy generation number generally increase with increasing Ri . -At low Ri , the buoyancy forces generated are not sufficient to completely suppress radial temperature gradients thus entropy largely enlarging.
Castell et al., 2010 [40]	SMT	0–6	-Definition of Ri determined by the temperature different of tank top and bottom, is varied against time. -The tendency of Ri is basically inversed to that of MIX number.
Afshan et al., 2020 [41]	SMT with or without PCM balls	0–32	-Varied Ri against time. -The tendency of Ri is correspondent to that of stratification number for both TES systems with or without PCM balls.

According to its definition, we can estimate that the thermocline tank sustains a better stratification level with a lower inlet velocity that leads to a lower characteristic velocity.

Péclet number (Pe)

The Péclet number Pe is defined as the ratio between the convective and the diffusive transport rates. The bulk formula is given by,

$$Pe = \frac{v \cdot L}{\alpha}, \quad \alpha = \frac{\lambda}{\rho \cdot c_p} \quad (14)$$

where the characterization length is normally the storage tank diameter, v is the mean velocity, and α is the thermal diffusivity.

Furthermore, the concept of the Pe can be extended by choosing different characteristic lengths, velocities, or properties. For instance, in case of packed-bed TES tanks, the superficial and effective Pe is adopted, where the characteristic length is the particle diameter [42,43]. Regarding the SMT system, the local velocity distribution can vary significantly; thus, the superficial or mean velocity cannot stand for the local competition between convection and diffusion. In this regard, the grid/element form of Pe is applied to reflect the hydrodynamic state inside a single control volume [44], which is given by,

$$Pe_{\delta x} = \frac{v \cdot \delta x}{\alpha}, \quad Pe_{\delta y} = \frac{v \cdot \delta y}{\alpha} \quad (15)$$

where δx and δy are the lengths of the control volume, v stands for the velocity at the node, and α is dependent on the node temperature.

Froude number (Fr)

The Froude number Fr , as an indicator usually used for the diffuser design or inlet design, represents the characteristic ratio between the inertial and buoyancy forces:

$$Fr = \frac{v}{\sqrt{L \cdot g \cdot (\rho_c - \rho) / \rho}} \quad (16)$$

Fr is the reciprocal of square root of the bulk Richardson number. A growing attention focusing on its recommended maximum value is frequently related to the design of flow distributors [19,45].

Reynolds number (Re)

The Reynolds number Re , as an important indicator of different flow regimes, reflects the ratio between inertial to viscous forces, which is given by:

$$Re = \frac{\rho v L}{\mu} \quad (17)$$

in which μ is the dynamic viscosity of the fluid. Note that one single tank usually holds different Re values at the inlet, within the vessel body, and between the packed fillers. The flow between the fillers turns to be laminar with a low Re number [46], while a high- Re -number turbulent flow is frequently identified within the inlet region. Moreover, its influence on the thermal performance of TES tanks may not be monotonous [16].

3. Influencing Factors for the Thermocline TES System

3.1. Flow Rate

The injecting flow rate of HTF determines the primary flow patterns inside the TES vessel. Table 3 gives the influence of flow rate on the performance of thermocline TES tanks. For instance, Dehghan et al. [35] performed a 2D transient CFD simulation to study the influence of flow rate on a solar hot water storage tank, as shown in Figure 4. With the increasing inlet Re numbers (Re_{in}), the streamlines turn from the buoyancy-controlled pattern to the inertial-controlled pattern. In another paper on the SMT tank [47], the authors discovered that the de-stratification phenomenon is obvious at the low flow rate (0.05 kg s^{-1}), and the mixing is enhanced at the high flow rate. Thus, an optimal flow rate could exist for the given structure and working temperature, due to the competition between mixing and thermal diffusion.

Table 3. Study on the influence of flow rate for thermocline-based TES systems.

Refs	N/E	TES Form	Inner Structure	Tank Size (m)	T	Inlet Flow Rate	HTF	Remarks on the Effect of Flow Rate
Dehghan and Barzegar, 2011 [35]	N	Mantle vertical cylinder	No packing	$H/D = 2$		$Re_{in} = 200, 400, 500, 600, 800$	water	-Low Re_{in} : mixing region is confined in the bottom zone during discharging. -High Re_{in} : mixing region expands directly towards the outlet port.
Yaïci et al., 2013 [47]	N	Vertical cylinder	No packing	$H = 1.194$ $D = 0.442$	$T_c = 30$ $T_h = 60$ (°C)	0.05, 0.1, 0.15, 0.2 kg s^{-1}	water	-Low inlet velocity: thermocline deterioration is more pronounced. -High inlet velocity: the mixing is intense. -Competition between the effect of mixing and thermal diffusion resulting in an optimal flow rate.
Abdulla and Reddy, 2017 [48]	N	Vertical cylinder	Sensible packed-bed	$H = 12$ $D = 14.38$	$T_c = 563$ $T_h = 663$ (K)	0.0003373– 0.0007724 m s^{-1}	Solar salt	-Inlet velocity $\uparrow$: effective discharging time $\downarrow$, discharging efficiency $\downarrow$.

Table 3. Cont.

Refs	N/E	TES Form	Inner Structure	Tank Size (m)	T	Inlet Flow Rate	HTF	Remarks on the Effect of Flow Rate
Wu et al., 2014 [49]	N	Vertical cylinder	-Channel-embedded -Parallel-plate -Rod-bundle -Packed-bed	$H = 15$	$T_c = 290$ $T_h = 390$ (°C)	$0.002\text{--}0.01 \text{ m s}^{-1}$	thermal oil	-Inlet velocity ↑: effective discharging time ↓ for all four inner structures. -Inlet velocity ↑: the main trend of effective discharging efficiency ↓. -Inlet velocity ↑: the inner structure with packed-bed could maintain the high effective discharging efficiency, whereas channel-embedded structure has the worst performance for effective discharging efficiency.
Yang and Garimella, 2010 [50]	N	Vertical cylinder	Sensible packed-bed	$H = 2.77\text{--}32.1$ $D = 2 \text{ or } 5$	$T_c = 250$ $T_h = 450$ (°C)	$Re_p = 1, 5, 10, 20, 30, 50$	HITEC salt	- Re_p ↑, discharging efficiency ↓.
Nandi et al., 2018 [51]	N	Vertical cylinder	Sensible packed-bed	$H = 6.1$ $D = 3$	$T_c = 290$ $T_h = 390$ (°C)	$Re_{in} = 10\text{--}1000$	Solar salt	$Re_{in} \in [10, 300]$ -Effect of Re_{in} on charging/discharging efficiency can be ignored. -Stratification number ↑ while Re_p is $\in [10, 100]$ $Re_{in} \in [300, 1000]$ -Reduction of thermocline behavior of storage with $Re_{in} \in [300, 400]$. -An unsteady and a chaotic flow, and a turbulent flow model should be considered for the TES with $Re_{in} > 300$. - Re_{in} ↑: Charging/discharging efficiency ↓.
Wang et al., 2020 [52]	N	Vertical cylinder	Porous bed	$H = 5.9$ $D = 3$	$T_c = 566$ $T_h = 723$ (K)	$0.002, 0.0025, 0.0030, 0.0035, 0.0040 \text{ m s}^{-1}$	Solar salt	Discharging: -Inlet velocity ↑: transient thermocline thickness ↓. -Inlet velocity ↑: mechanical stress of the steel wall ↓, better for the structural reliability of TES tank. Charging: -The effect of inlet velocity on the formation of thermocline is relatively small.
Bhagat and Saha, 2016 [53]	N	Vertical cylinder	PCM packed-bed	$H = 25$ $D = 12$	$T_c = 142$ $T_h = 171$ (°C)	$0.00145\text{--}0.0058 \text{ kg s}^{-1}$	Hytherm 600	-Flow rate ↑: effective discharging time ↓. -Mass flow rate significantly influences the heat transfer rate between HTF and PCM. -Flow rate ↑: volumetric heat transfer coefficient ↑, overall effectiveness ↑.
He et al., 2019 [54]	E, N	Main cylinder body	No packing or PCM packed-bed	$H = 1.1$ $D = 0.9$	$T_c = 37$ $T_h = 70$ (°C)	$0.3, 0.6, 0.9 \text{ m}^3 \text{ h}^{-1}$	water	-Inlet flow rate ↑: thermocline thickness ↑. -Thicker thermocline appeared within the PCM packed-bed structure compared to the no packing case.
Saha and Das, 2020 [55]	N	Vertical cylinder	PCM packed-bed	$H = 0.6$ $D = 0.25$	$T_c = 325$ $T_h = 446$ (°C)	$200, 240, 280 \text{ kg h}^{-1}$	air	-Inlet flow rate ↑: effective charging time ↓; charging efficiency ↓; second law efficiency ↑.

Additionally, several previous articles investigated the thermal performance of TES systems with sensible solid fillers [35,47–55], which reached the contrary conclusions. On the one hand, the increasing flow rate has been found to decrease the discharging efficiency, since the incoming flow at a high flow rate can penetrate far away along the centerline of the tanks [48–50]. On the other hand, the reduced transient thermocline thickness was found with the increasing inlet velocity during the discharging process [15]. Nandi et al. [51] found the stratification number could be enhanced with the increasing particle-based Reynolds number Re_p from 10 to 100, but the charging efficiency is nearly unchanged with the increasing Re_{in} (from 10 to 300) at the laminar regime.

3.2. Working Temperature

The buoyancy force induced by the HTF injection depends on the density difference between the inlet jet (called the thermal jet) and surrounding fluids. As shown in Figure 5, the thermal jet of hot fluid with lower density will maintain the thermal stratification, which is called positive buoyant jet; by contrast, the thermal jet of cold fluid with higher density can lead to the unstable thermocline, called negative buoyant jet. Apparently, the temperature difference ΔT between the hot and cold fluids determines the buoyancy force of the thermal jets and then affects the fluid mixing driven by the buoyant plumes, and thus the working temperature range can be an important operating parameter. The experimental investigation of Al-Habaibeh et al. [24] has clearly shown the trajectory movement by the means of the water ‘snake’ due to the temperature difference, including the positive ΔT or the negative ΔT . Table 4 summarizes the effects of ΔT for the various thermocline-based TES systems, including the systems without the packing, with the sensible solid fillers, and with latent PCM packing, respectively. For the SMT tanks, the conclusions are not consistent. On the one hand, the high ΔT is regarded to be able to control the mixing phenomenon, and thus improve the stratification [56]. Moreover, the penetration length of inlet injection could also be shortened with the increasing ΔT [16]. On the contrary, the high ΔT was found to accelerate the stratification destruction, i.e., the increased thermocline thickness with the increasing ΔT [57]. Shaikh et al. [16] found that the thermocline propagates more rapidly with a larger ΔT , though the penetration length of thermal jet is reduced. Such de-stratification should be attributed to the mixing driven by the buoyancy. Therefore, the setting of working temperature range should be considered carefully.

Figure 5. The schematics for the thermal jet.

Table 4. Study on the influence of working temperature for thermocline-based TES systems.

Reference	N/E	TES Tank Shape	Inner Packing	Tank Size (m)	Temperature	Inlet Flow	HTF	Remarks
Shaikh et al., 2018 [16]	N	Vertical Cylinder	No packing but with vertical porous distributor	$H = 1.016$ $D = 0.488$	$T_c = 573$ $T_h = 651, 773, 823$ (K)	$Re_{tank} = 5-100$	Solar salt	$-\Delta T \uparrow$, penetration length $\downarrow$. $-\Delta T \uparrow$, thermocline thickness $\uparrow$ as the buoyancy driven mixing becomes dominant.
Yaïci et al., 2013 [47]	N	Vertical Cylinder	No packing	$H = 1.194$ $D = 0.442$	$\Delta T = 5, 10, 15$ (K)	0.1 kg s^{-1}	water	$-\Delta T$ does not affect the formation of the thermocline during charging process regardless of varying T_h or T_{mi} .
Abdulla and Reddy, 2017 [48]	N	Vertical Cylinder	Sensible packed-bed	$H = 12$ $D = 14.38$	$T_c = 613$ $T_h = 663-838$ $\Delta T = 50-100$ (K)	$0.0003373-0.0007724 \text{ m s}^{-1}$	Solar salt	$-\Delta T \uparrow$, thermocline thickness $\uparrow$, discharging efficiency $\downarrow$.
Nandi et al., 2018 [51]	N	Vertical Cylinder	Sensible packed-bed	$H = 6.1$ $D = 3$	$T_c = 290$ $T_h = 390, 490, 590$ ($^{\circ}\text{C}$)	$Re_{in} = 10-1000$	Solar salt	Laminar regime $Re_{in} \in [10, 300]$: -The effect of ΔT on both charging and discharging efficiency is not obvious. Turbulent regime $Re_{in} > 300$: $-\Delta T \uparrow$, charging efficiency $\uparrow$.
Wang et al., 2020 [52]	N	Vertical Cylinder	Porous bed	$H = 5.9$ $D = 3$	$T_c = 506-626$ $T_h = 723$ (K)	0.003 m s^{-1}	Solar salt	Both charging and discharging processes $-\Delta T \uparrow$, thermocline thickness $\uparrow$. $-\Delta T \uparrow$, effective charging/discharging time $\downarrow$.
Bhagat and Saha, 2016 [53]	N	Vertical Cylinder	PCM packed-bed	$H = 25$ $D = 12$	$T_c = 142$ $T_h = 180-240$ ($^{\circ}\text{C}$)	$0.00145-0.0058 \text{ kg s}^{-1}$	Hytherm 600	-Flow rate $\uparrow$, effective discharging time keeps the equivalent value. $-\Delta T \uparrow$, energy stored/released $\uparrow$, overall effectiveness $\uparrow$.
Nelson et al., 1999 [56]	E	Vertical Cylinder	No packing	$H = 1.08-1.89$ $D = 0.54$	$\Delta T = 10, 12$ ($^{\circ}\text{C}$)	$7.12 \times 10^{-5} \text{ m}^3 \text{ s}^{-1}$	water	$-\Delta T \uparrow$: thermal stratification $\uparrow$ and mixing $\downarrow$.
Sun et al., 2018 [57]	E, N	Vertical Cylinder	No packing	$H = 7$ $D = 4$	$T_c = 10$ $T_h = 20, 40, 60, 80$ ($^{\circ}\text{C}$)		water	$-\Delta T \uparrow$, thermocline thickness $\uparrow$.
Yin et al., 2017 [58]	E, N	Vertical Cylinder	Porous bed	$H = 0.6$ $D = 0.12$	$T_c = 290$ $T_h = 350, 370, 390$ ($^{\circ}\text{C}$)	0.02 m s^{-1}	molten nitrate salt	$-\Delta T \uparrow$, discharging efficiency $\downarrow$.
Kumar et al., 2016 [59]	E	Vertical Cylinder	sensible Or PCM-combined packed-bed	$H = 1$ $D = 0.4$	$T_c = 35$ $T_h = 60, 70, 80$ ($^{\circ}\text{C}$)	$1, 1.5, 2 \text{ L min}^{-1}$	water	$-\Delta T \uparrow$: time-evolution Richardson number $\uparrow$ for both packing systems. Sensible packed-bed: $-\Delta T \uparrow$: charging efficiency $\downarrow$. Combined sensible and latent packed-bed: $-\Delta T \uparrow$: charging efficiency $\uparrow$.
Majumdar et al., 2019 [60]	N	Vertical Cylinder	Partial PCM packed-bed	$H = 1$ $D = 0.4$	$T_c = 15, 25, 35$ $T_h = 60, 70, 80$ ($^{\circ}\text{C}$)		water	$-\Delta T \uparrow$: time-evolution Richardson number $\uparrow$. $-\Delta T \uparrow$: charging efficiency $\uparrow$ while a slightly lower storage efficiency is found.

As for the TES tanks with sensible packing, the previous conclusions can be relatively consistent. The increasing thermocline thickness and decreasing charging/discharging efficiencies have been observed as setting a larger ΔT [15,48,52,58,59], since the laminar pattern and relatively uniform and calm flow distribution can be obtained in the packed-

bed region. Actually, the effects of ΔT are opposite for the sensible and latent packed-bed systems [53,59,60]. For the TES tanks with latent packing fillers, the increasing ΔT can lead to the better thermal performance, including the charging efficiency, Ri number, the amount of stored/released energy and overall effectiveness. Meanwhile, the higher working temperature can also enhance the heat transfer between HTF and PCM fillers.

3.3. Flow Distributor

The management of heat and mass transfer by fluid flow is a key issue for the design, the operation and the optimization of many industrial processes and energy conversion systems [61,62]. Flow equidistribution is a classical goal for most of the studies reported in the literature. Different types of flow equalization devices are proposed, as summarized in Refs. [63–65].

Interests have arisen into the time evolution of flow and temperature distributions during the heat charging/discharging process in thermocline tanks. The injected fluid usually holds the significantly-different densities and velocities, compared to the calm fluid within the TES vessel, which will disturb the initial flow state. A device called flow diffuser or distributor that can rebuild the expected flow distribution to improve the thermal stratification has received much attention recently. Until now, there is no general agreement on the classification of flow distributors. Here, we classify the existing designs into three types on the basis of their primary functions: (1) producing multi-branch flow; (2) producing inversed flux; (3) producing radial flux, as illustrated in Figure 6.

Figure 6. Classification and examples of flow distributor used in thermocline TES tank. (a) shower type [66]; (b) conical porous diffuser [20]; (c) perforated plate [67]; (d) incline flat plate [14]; (e) overflow tubular [67]; (f) moving plate [68]; (g) compound-pipe stratifier [69]; (h) porous pipe [70]; (i) perforated tube [71]. Reprinted with permission from refs. [14,20,67–71]. 4 October 2021, Elsevier.

The multi-branch type is usually fabricated with the multiple ports to divide the inertial-force-controlled injected flow into many minor streams [51,54,66–72]. A CFD

analysis of Li et al. [66] showed that the higher effective discharging efficiency (about 14% in average) could be obtained with the shower-type inlet distributor compared to the simple-inlet case (flow rate ranges from 5 to 15 L min⁻¹). Furthermore, the perforated plate design has been widely used as the multi-branch type distributor in the packed-bed TES systems, since it can support the solid filler and be simple to fabricate. Owing to the natural buffering effect of the packed solid fillers, the conventional perforated plate is considered to be sufficient to produce a uniform flow distribution [51,54].

In addition, Afrin et al. [72] numerically investigated two different pipe flow distributors, where the orientations of orifice openings were set towards the main fluid storage and to the upper/bottom port, respectively. They found that the flow direction from the distributor could have a noticeable influence on the velocity distribution. In this regard, there has been considerable interests in the second type of distributors that generate the inversed flux by the means of installing the flat obstacle, the inclined obstacle, the moving plat, or adjusting the orientation of orifice opening. Chandra et al. [18] numerically investigated three geometries of inlet diffuser: a shower-type diffuser, a slotted inlet, and a simple inlet. For the slotted inlet, the direction of inlet flow is switched to be reversed from the primary flow injection. This geometry could thereby improve the efficiency by 8.7% at a large flow rate (800 L h⁻¹). Similar conclusion has also been reported by Li et al. [66]. However, no obvious improvement (only 1–2%) was found for this slotted inlet device at low flow rate (200 L h⁻¹).

Regarding the third type of distributors, the primary target is to generate radial flux. For example, a perforated, eccentrically mounted, vertical, and porous flow distributor consist of an inner porous pipe and a concentric outer pipe was fabricated and installed into the SMT tank [70], as shown in Figure 6h. Furthermore, in order to unfold the differences between the axial and radial flows, Al-Azawii et al. [71] experimentally investigated the packed-bed TES systems with the incoming axial flow and with the radial flux, respectively. The perforated tube was immersed into the packed-bed tank, where the axial flow inside the perforated tube could be interrupted by the plates, and in this case the charging efficiency was enhanced from 75.3% to 80.3 ± 2.8%. Moreover, Dragsted et al. [73] experimentally investigated four stratifiers, including one made of the flexible polymer. This flexible material can avoid the counter flow from the surrounding to the stratifier, resulting in a better thermal stratification at low flow rate.

Furthermore, an original CFD-based optimization algorithm was developed by Lou et al. [12] to determine the optimal flow distribution and restricted thermocline propagation manner using a SMT tank at high temperature as an example. A practical method for homogenizing residence times of the thermal front in order to flatten the thermocline zone was proposed, based on the insertion of a geometrically optimized perforated. The feasibility of optimization algorithm has then been validated experimentally by testing of a lab-scale cylinder SMT storage tank at low temperature, by measuring the local temperature evolutions of the fluid during both the charging and discharging operations. The results showed that optimized distributors can significantly improve the energy and exergy efficiencies under large range of operating conditions.

3.4. Inlet/Outlet Location

Due to the variations of densities and velocities of the injected fluid, a free shear layer is always formed to cause a jet at the inlet port, where the vortex, mixing, and entrainment appear and, subsequently, move inside the storage vessel [9]. The buoyancy-driven plume and the inertial-controlled jet will coexist and further extend to a larger region inside the tank. If the momentum motion of the jet is strong enough to impinge itself against the opposite side wall, the mixing will propagate upwards and downwards [74]. The identical phenomenon may also occur near the outlet port. Therefore, the suitable location of the side inlet/outlet can be important [75,76]. Farmahini-Farahani [76] found that the TES tank could have the highest dimensionless exergy, when the inlet and outlet ports are set close to the upper and bottom walls, respectively. As shown in Figure 7a, the

influence of different locations of side inlet/outlet ports was experimentally investigated for a horizontal storage tank by Assari et al. [75]. The authors recommended to inject the hot water from the highest position and extract the cold water from the lowest position. Kim et al. [77,78] also studied the position of the incoming and outgoing flows by using the immersed pipes, as shown in Figure 7b. The time-evolution temperature profiles monitored by the thermocouples indicated that Tank B with the shortened immersed inlet/outlet pipes performed better. In the work by Yaïci et al. [47] (Figure 7c), the buoyant effect on mixing was not that strong, when the vertical distance between the inlet port and the upper wall is small (20 mm in Case 1). As increasing such distance from Case 2 to Case 4, the buoyancy-driven plume showed more and more profound impact, when compared to the mixing flow. Subsequently, the impinging jet continued to expand upward and downwards, leading to the enlarged dead zone.

Figure 7. (a) Inlet-outlet location effect in the horizontal water storage tank [75]; (b) piping and configuration of solar salt thermal storage tank A, B, and C [78]; (c) effect of inlet location via temperature contours [47]. Reprinted with permission from refs. [47,75,78]. 4 October 2021, Elsevier.

4. Conclusions

This work provides a literature review on the performance indicators and the influencing factors for the thermocline TES systems. The three categories of performance indicators, i.e., (1) ones directly reflecting the quantity or quality of the stored thermal energy, (2) ones describing the thermal stratification level, and (3) ones characterizing the thermo-hydrodynamic features, are analyzed in detail. The selection of the performance indicators for a specific thermocline tank is, actually, a tradeoff between the convenience of characterization (calculation or measurement) and the amount of information. Apparently, the charging/discharging efficiency, the capacity ratio, and the tail factor in Category #1, and the bulk dimensionless numbers in Category #3 can be conveniently calculated based on the measurements, providing an overall evaluation of the performance. Nevertheless,

the charging/discharging efficiency, the capacity ratio, and the tail factor should be regarded as the output of the design of TES tank, and they can hardly tell the designers which factor should be responsible and which part can be improved when the thermal performance is unsatisfactory, since they do not give any detailed information inside the tank. Similarly, the information on the thermo-hydrodynamic features provided by the bulk dimensionless numbers can be rather coarse or even misleading, since as mentioned above these dimensionless numbers can vary significantly in the different regions within a tank. In fact, the gradient or grid formula gives more detailed information about thermo-hydrodynamic performance within the tank, but usually require dynamic CFD simulations or experimental visualization to obtain detailed information on the local velocity and temperature profiles. These indicators, once obtained, can be very helpful for improving the thermocline tank design. Moreover, the indicator normalization is recommended for evaluating the performance.

In addition, four influencing factors, i.e., flow rate, working temperature, flow distributor, and inlet/outlet location, are discussed systematically. The essential remarks are as follows: (1) the choice of the optimal flow rate is highly dependent on the tank geometry and temperature difference; (2) the effects of operating parameters are varying for the storage systems in the different conditions, such as without packing, with sensible or latent packing; (3) there are generally three types of flow distributors classified by their primary functions: the multi-branch, the inversed flux and the radial flux; (4) the inlet/outlet tube is recommended to be located close to the upper or bottom wall of the storage tank.

This paper can serve as an essential reference that contributes to the development of the well-designed thermocline TES systems with the well-handled operating conditions, towards realizing their favorable thermal stratification and thermal performance and for their future applications in different residential, industrial, and commercial sectors.

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References

1. Tian, Y.; Zhao, C.Y. A review of solar collectors and thermal energy storage in solar thermal applications. *Appl. Energy* **2013**, *104*, 538–553. [\[CrossRef\]](#)
2. N'Tsoukpoe, K.E.; Liu, H.; Le Pierrès, N.; Luo, L. A review on long-term sorption solar energy storage. *Renew. Sustain. Energy Rev.* **2009**, *13*, 2385–2396. [\[CrossRef\]](#)
3. Tatsidjodoung, P.; Le Pierrès, N.; Luo, L. A review of potential materials for thermal energy storage in building applications. *Renew. Sustain. Energy Rev.* **2013**, *18*, 327–349. [\[CrossRef\]](#)
4. Li, G. Sensible heat thermal storage energy and exergy performance evaluations. *Renew. Sustain. Energy Rev.* **2016**, *53*, 897–923. [\[CrossRef\]](#)
5. Pelay, U.; Luo, L.; Fan, Y.; Stitou, D.; Castelain, C. Integration of a thermochemical energy storage system in a Rankine cycle driven by concentrating solar power: Energy and exergy analyses. *Energy* **2019**, *167*, 498–510. [\[CrossRef\]](#)
6. Pelay, U.; Luo, L.; Fan, Y.; Stitou, D. Dynamic modeling and simulation of a concentrating solar power plant integrated with a thermochemical energy storage system. *J. Energy Storage* **2020**, *28*, 101164. [\[CrossRef\]](#)

7. Pelay, U.; Azzaro-Pantel, C.; Fan, Y.; Luo, L. Life cycle assessment of thermochemical energy storage integration concepts for a concentrating solar power plant. *Environ. Prog. Sustain. Energy* **2020**, *39*, e13388. [\[CrossRef\]](#)
8. Pelay, U.; Luo, L.; Fan, Y.; Stitou, D.; Rood, M. Thermal energy storage systems for concentrated solar power plants. *Renew. Sustain. Energy Rev.* **2017**, *79*, 82–100. [\[CrossRef\]](#)
9. Dahash, A.; Ochs, F.; Janetti, M.B.; Streicher, W. Advances in seasonal thermal energy storage for solar district heating applications: A critical review on large-scale hot-water tank and pit thermal energy storage systems. *Appl. Energy* **2019**, *239*, 296–315. [\[CrossRef\]](#)
10. Cascetta, M.; Petrollese, M.; Oyekale, J.; Cau, G. Thermocline vs. two-tank direct thermal storage system for concentrating solar power plants: A comparative techno-economic assessment. *Int. J. Energy Res.* **2021**, *45*, 12. [\[CrossRef\]](#)
11. Zanganeh, G.; Pedretti, A.; Haselbacher, A.; Steinfeld, A. Design of packed bed thermal energy storage systems for high-temperature industrial process heat. *Appl. Energy* **2015**, *137*, 812–822. [\[CrossRef\]](#)
12. Lou, W.; Fan, Y.; Luo, L. Single-tank thermal energy storage systems for concentrated solar power: Flow distribution optimization for thermocline evolution management. *J. Energy Storage* **2020**, *32*, 101749. [\[CrossRef\]](#)
13. Cárdenas, B.; Garvey, S.D. A load-based approach for optimizing a packed-bed thermal store. *J. Energy Storage* **2019**, *25*, 100835. [\[CrossRef\]](#)
14. Bouhal, T.; Fertahi, S.; Agrouaz, Y.; El Rhafiki, T.; Kousksou, T.; Jamil, A. Numerical modeling and optimization of thermal stratification in solar hot water storage tanks for domestic applications: CFD study. *Sol. Energy* **2017**, *157*, 441–455. [\[CrossRef\]](#)
15. Wang, Z.; Zhang, H.; Dou, B.; Zhang, G.; Wu, W.; Zhou, L. An experimental study for the enhancement of stratification in heat-storage tank by equalizer and PCM module. *J. Energy Storage* **2020**, *27*, 101010. [\[CrossRef\]](#)
16. Shaikh, W.; Wadegaonkar, A.; Kedare, S.B.; Bose, M. Numerical simulation of single media thermocline based storage system. *Sol. Energy* **2018**, *174*, 207–217. [\[CrossRef\]](#)
17. Fasquelle, T.; Falcoz, Q.; Neveu, P.; Hoffmann, J.-F. A temperature threshold evaluation for thermocline energy storage in concentrated solar power plants. *Appl. Energy* **2018**, *212*, 1153–1164. [\[CrossRef\]](#)
18. Chandra, Y.P.; Matuska, T. Numerical prediction of the stratification performance in domestic hot water storage tanks. *Renew. Energy* **2020**, *154*, 1165–1179. [\[CrossRef\]](#)
19. Chung, J.D.; Cho, S.H.; Tae, C.S.; Yoo, H. The effect of diffuser configuration on thermal stratification in a rectangular storage tank. *Renew. Energy* **2008**, *33*, 2236–2245. [\[CrossRef\]](#)
20. García-Marí, E.; Gasque, M.; Gutiérrez-Colomer, R.P.; Ibáñez, F.; González-Altozano, P. A new inlet device that enhances thermal stratification during charging in a hot water storage tank. *Appl. Therm. Eng.* **2013**, *61*, 663–669. [\[CrossRef\]](#)
21. Shah, L.J.; Furbo, S. Entrance effects in solar storage tanks. *Sol. Energy* **2003**, *75*, 337–348. [\[CrossRef\]](#)
22. Hua, Y.-C.; Zhao, T.; Guo, Z.-Y. Transient thermal conduction optimization for solid sensible heat thermal energy storage modules by the Monte Carlo method. *Energy* **2017**, *133*, 338–347. [\[CrossRef\]](#)
23. Andersen, E.; Furbo, S.; Hampel, M.; Heidemann, W.; Müller-Steinhagen, H. Investigations on stratification devices for hot water heat stores. *Int. J. Energy Res.* **2008**, *32*, 255–263. [\[CrossRef\]](#)
24. Al-Habaibeh, A.; Shakmak, B.; Fanshawe, S. Assessment of a novel technology for a stratified hot water energy storage—The water snake. *Appl. Energy* **2018**, *222*, 189–198. [\[CrossRef\]](#)
25. Wang, S.; Davidson, J.H. Performance of a rigid porous-tube stratification manifold in comparison to an inlet pipe. *Sol. Energy* **2017**, *146*, 298–308. [\[CrossRef\]](#)
26. Hess, C.F.; Miller, C.W. An experimental and numerical study on the effect of the wall in a thermocline-type cylindrical enclosure—I Experiments. *Sol. Energy* **1982**, *28*, 145–152. [\[CrossRef\]](#)
27. Zachár, A. Analytic solution for convection dominant heat transport induced by buoyant jet entrainment inside hot fluid storage tanks. *Sol. Energy* **2020**, *195*, 239–248. [\[CrossRef\]](#)
28. Modi, A.; Pérez-Segarra, C.D. Thermocline thermal storage systems for concentrated solar power plants: One-dimensional numerical model and comparative analysis. *Sol. Energy* **2014**, *100*, 84–93. [\[CrossRef\]](#)
29. Carmona, M.; Rincón, A.; Gulfo, L. Energy and exergy model with parametric study of a hot water storage tank with PCM for domestic applications and experimental validation for multiple operational scenarios. *Energy Convers. Manag.* **2020**, *222*, 113189. [\[CrossRef\]](#)
30. Ma, Z.; Li, M.-J.; Zhang, K.M.; Yuan, F. Novel designs of hybrid thermal energy storage system and operation strategies for concentrated solar power plant. *Energy* **2021**, *216*, 119281. [\[CrossRef\]](#)
31. Tarlet, D.; Fan, Y.; Roux, S.; Luo, L. Entropy generation analysis of a mini heat exchanger for heat transfer intensification. *Exp. Therm. Fluid Sci.* **2014**, *53*, 119–126. [\[CrossRef\]](#)
32. Bai, Y.; Wang, Z.; Fan, J.; Yang, M.; Li, X.; Chen, L.; Yuan, G.; Yang, J. Numerical and experimental study of an underground water pit for seasonal heat storage. *Renew. Energy* **2020**, *150*, 487–508. [\[CrossRef\]](#)
33. El-Amin, M.F.; Al-Ghamdi, A. Experiments and numerical simulation for a thermal vertical jet into a rectangular water tank. *Results Phys.* **2018**, *10*, 680–692. [\[CrossRef\]](#)
34. Advait, S.; Parida, D.R.; Aswathi, K.T.; Dani, N.; Chetia, U.K.; Chattopadhyay, K.; Basu, S. Experimental investigation on single-medium stratified thermal energy storage system. *Renew. Energy* **2021**, *164*, 146–155. [\[CrossRef\]](#)
35. Dehghan, A.A.; Barzegar, A. Thermal performance behavior of a domestic hot water solar storage tank during consumption operation. *Energy Convers. Manag.* **2011**, *52*, 468–476. [\[CrossRef\]](#)

36. Fernández-Seara, J.; Uhía, F.J.; Sieres, J. Experimental analysis of a domestic electric hot water storage tank. Part I: Static mode of operation. *Appl. Therm. Eng.* **2007**, *27*, 129–136. [\[CrossRef\]](#)
37. Paiva, B.P.; Siegle, E.; Schettini, C.A.F. Channel curvature effects on estuarine circulation in a highly stratified tropical estuary: The São Francisco river estuary (Brazil). *Estuar. Coast. Shelf Sci.* **2020**, *238*, 106723. [\[CrossRef\]](#)
38. Yee, C.K.; Lai, F.C. Effects of a porous manifold on thermal stratification in a liquid storage tank. *Sol. Energy* **2001**, *71*, 241–254. [\[CrossRef\]](#)
39. Njoku, H.O.; Ekechukwu, O.V.; Onyegegbu, S.O. Comparison of energy, exergy and entropy generation-based criteria for evaluating stratified thermal store performances. *Energy Build.* **2016**, *124*, 141–152. [\[CrossRef\]](#)
40. Castell, A.; Medrano, M.; Solé, C.; Cabeza, L.F. Dimensionless numbers used to characterize stratification in water tanks for discharging at low flow rates. *Renew. Energy* **2010**, *35*, 2192–2199. [\[CrossRef\]](#)
41. Afshan, M.E.; Selvakumar, A.S.; Velraj, R.; Rajaraman, R. Effect of aspect ratio and dispersed PCM balls on the charging performance of a latent heat thermal storage unit for solar thermal applications. *Renew. Energy* **2020**, *148*, 876–888. [\[CrossRef\]](#)
42. Zanganeh, G.; Commerford, M.; Haselbacher, A.; Pedretti, A.; Steinfeld, A. Stabilization of the outflow temperature of a packed-bed thermal energy storage by combining rocks with phase change materials. *Appl. Therm. Eng.* **2014**, *70*, 316–320. [\[CrossRef\]](#)
43. Davenne, T.R.G.; Garvey, S.D.; Cardenas, B.; Rouse, J.P. Stability of packed bed thermoclines. *J. Energy Storage* **2018**, *19*, 192–200. [\[CrossRef\]](#)
44. Bayón, R.; Rojas, E. Simulation of thermocline storage for solar thermal power plants: From dimensionless results to prototypes and real-size tanks. *Int. J. Heat Mass Transf.* **2013**, *60*, 713–721. [\[CrossRef\]](#)
45. Kaloudis, E.; Grigoriadis, D.G.E.; Papanicolaou, E. Numerical simulations of constant-influx gravity currents in confined spaces: Application to thermal storage tanks. *Int. J. Therm. Sci.* **2016**, *108*, 1–16. [\[CrossRef\]](#)
46. Flueckiger, S.M.; Yang, Z.; Garimella, S.V. Review of molten-salt thermocline tank modeling for solar thermal energy storage. *Heat Transf. Eng.* **2013**, *34*, 787–800. [\[CrossRef\]](#)
47. Yaïci, W.; Ghorab, M.; Entchev, E.; Hayden, S. Three-dimensional unsteady CFD simulations of a thermal storage tank performance for optimum design. *Appl. Therm. Eng.* **2013**, *60*, 152–163. [\[CrossRef\]](#)
48. Abdulla, A.; Reddy, K.S. Effect of operating parameters on thermal performance of molten salt packed-bed thermocline thermal energy storage system for concentrating solar power plants. *Int. J. Therm. Sci.* **2017**, *121*, 30–44. [\[CrossRef\]](#)
49. Wu, M.; Li, M.; Xu, C.; He, Y.; Tao, W. The impact of concrete structure on the thermal performance of the dual-media thermocline thermal storage tank using concrete as the solid medium. *Appl. Energy* **2014**, *113*, 1363–1371. [\[CrossRef\]](#)
50. Yang, Z.; Garimella, S.V. Thermal analysis of solar thermal energy storage in a molten-salt thermocline. *Sol. Energy* **2010**, *84*, 974–985. [\[CrossRef\]](#)
51. Nandi, B.R.; Bandyopadhyay, S.; Banerjee, R. Numerical modeling and analysis of dual medium thermocline thermal energy storage. *J. Energy Storage* **2018**, *16*, 218–230. [\[CrossRef\]](#)
52. Wang, G.; Yu, S.; Niu, S.; Chen, Z.; Hu, P. A comprehensive parametric study on integrated thermal and mechanical performances of molten-salt-based thermocline tank. *Appl. Therm. Eng.* **2020**, *170*, 115010. [\[CrossRef\]](#)
53. Bhagat, K.; Saha, S.K. Numerical analysis of latent heat thermal energy storage using encapsulated phase change material for solar thermal power plant. *Renew. Energy* **2016**, *95*, 323–336. [\[CrossRef\]](#)
54. He, Z.; Wang, X.; Du, X.; Amjad, M.; Yang, L.; Xu, C. Experiments on comparative performance of water thermocline storage tank with and without encapsulated paraffin wax packed bed. *Appl. Therm. Eng.* **2019**, *147*, 188–197. [\[CrossRef\]](#)
55. Saha, S.K.; Das, R.B. Exergetic and performance analyses of two-layered packed bed latent heat thermal energy storage system. *Int. J. Energy Res.* **2020**, *44*, 2208–2225. [\[CrossRef\]](#)
56. Nelson, J.E.B.; Balakrishnan, A.R.; Srinivasa Murthy, S. Experiments on stratified chilled-water tanks. *Int. J. Refrig.* **1999**, *22*, 216–234. [\[CrossRef\]](#)
57. Sun, J.; Hua, J.; Fu, L.; Zhang, S. Experimental study of a large temperature difference thermal energy storage tank for centralized heating systems. *Therm. Sci.* **2018**, *22*, 613–621. [\[CrossRef\]](#)
58. Yin, H.; Ding, J.; Jiang, R.; Yang, X. Thermocline characteristics of molten-salt thermal energy storage in porous packed-bed tank. *Appl. Therm. Eng.* **2017**, *110*, 855–863. [\[CrossRef\]](#)
59. Kumar, G.S.; Nagarajan, D.; Chidambaram, L.A.; Kumaresan, V.; Ding, Y.; Velraj, R. Role of PCM addition on stratification behaviour in a thermal storage tank—An experimental study. *Energy* **2016**, *115*, 1168–1178. [\[CrossRef\]](#)
60. Majumdar, R.; Saha, S.K. Effect of varying extent of PCM capsule filling on thermal stratification performance of a storage tank. *Energy* **2019**, *178*, 1–20. [\[CrossRef\]](#)
61. Luo, L. General Introduction. In *Heat and Mass Transfer Intensification and Shape Optimization*; Springer: London, UK, 2013; pp. 1–18.
62. Luo, L.; Wei, M.; Fan, Y.; Flamant, G. Heuristic shape optimization of baffled fluid distributor for uniform flow distribution. *Chem. Eng. Sci.* **2015**, *123*, 542–556. [\[CrossRef\]](#)
63. Siddiqui, O.K.; Zubair, S.M. Efficient energy utilization through proper design of microchannel heat exchanger manifolds: A comprehensive review. *Renew. Sustain. Energy Rev.* **2017**, *74*, 969–1002. [\[CrossRef\]](#)
64. Wei, M.; Boutin, G.; Fan, Y.; Luo, L. Numerical and experimental investigation on the realization of target flow distribution among parallel mini-channels. *Chem. Eng. Res. Des.* **2016**, *113*, 74–84. [\[CrossRef\]](#)

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65. Rebrov, E.V.; Schouten, J.C.; de Croon, M.H.J.M. Single-phase fluid flow distribution and heat transfer in microstructured reactors. *Chem. Eng. Sci.* **2011**, *66*, 1374–1393. [[CrossRef](#)]
 66. Li, S.; Zhang, Y.; Li, Y.; Zhang, X. Experimental study of inlet structure on the discharging performance of a solar water storage tank. *Energy Build.* **2014**, *70*, 490–496. [[CrossRef](#)]
 67. Wei, M. *Nouvelles Méthodes Pour L'optimisation de la Distribution des Fluides et Leurs Applications Dans les Systèmes des Centrales Solaires à Concentration (CSP)*; Université de Nantes: Nantes, France, 2015.
 68. Bouzaher, M.T.; Bouchahm, N.; Guerira, B.; Bensaci, C.-E.; Lebbi, M. On the thermal stratification inside a spherical water storage tank during dynamic mode. *Appl. Therm. Eng.* **2019**, *159*, 113821. [[CrossRef](#)]
 69. Shah, L.J.; Andersen, E.; Furbo, S. Theoretical and experimental investigations of inlet stratifiers for solar storage tanks. *Appl. Therm. Eng.* **2005**, *25*, 2086–2099. [[CrossRef](#)]
 70. Gajbhiye, P.; Salunkhe, N.; Kedare, S.; Bose, M. Experimental investigation of single media thermocline storage with eccentrically mounted vertical porous flow distributor. *Sol. Energy* **2018**, *162*, 28–35. [[CrossRef](#)]
 71. Al-Azawii, M.M.S.; Jacobsen, D.; Bueno, P.; Anderson, R. Experimental study of thermal behavior during charging in a thermal energy storage packed bed using radial pipe injection. *Appl. Therm. Eng.* **2020**, *180*, 115804. [[CrossRef](#)]
 72. Afrin, S.; Kumar, V.; Bharathan, D.; Glatzmaier, G.C.; Ma, Z. Computational analysis of a pipe flow distributor for a thermocline based thermal energy storage system. *J. Sol. Energy Eng. Trans. ASME* **2014**, *136*, 021010. [[CrossRef](#)]
 73. Dragsted, J.; Furbo, S.; Dannemand, M.; Bava, F. Thermal stratification built up in hot water tank with different inlet stratifiers. *Sol. Energy* **2017**, *147*, 414–425. [[CrossRef](#)]
 74. Toyoshima, M.; Okawa, S. An effect of a horizontal buoyant jet on the temperature distribution inside a hot water storage tank. *Int. J. Heat Fluid Flow* **2013**, *44*, 403–413. [[CrossRef](#)]
 75. Assari, M.R.; Basirat Tabrizi, H.; Savadkoohy, M. Numerical and experimental study of inlet-outlet locations effect in horizontal storage tank of solar water heater. *Sustain. Energy Technol. Assess.* **2018**, *25*, 181–190. [[CrossRef](#)]
 76. Farahani, M.F. Investigation of four geometrical parameters on thermal stratification of cold water tanks by exergy analysis. *Int. J. Exergy* **2012**, *10*, 332. [[CrossRef](#)]
 77. Kim, D.H.; Yoon, S.H.; Kim, Y.; Song, C.H.; Lee, K.H.; Choi, J.S. Experimental studies of the discharge performance of single-medium TES for CSP applications. *Appl. Therm. Eng.* **2017**, *127*, 499–507. [[CrossRef](#)]
 78. Kim, D.H.; Yoon, S.H.; Kim, Y.; Lee, K.H.; Choi, J.S. Experimental studies on the charging performance of single-tank single-medium thermal energy storage. *Appl. Therm. Eng.* **2019**, *149*, 1098–1104. [[CrossRef](#)]