

Optimization of thermal energy storage by the thermocline technology

Wanruo Lou

▶ To cite this version:

Wanruo Lou. Optimization of thermal energy storage by the thermocline technology. Thermics [physics.class-ph]. Université de Nantes, 2021. English. NNT: . tel-03337888

HAL Id: tel-03337888 https://theses.hal.science/tel-03337888

Submitted on 8 Sep 2021

HAL is a multi-disciplinary open access archive for the deposit and dissemination of scientific research documents, whether they are published or not. The documents may come from teaching and research institutions in France or abroad, or from public or private research centers. L'archive ouverte pluridisciplinaire **HAL**, est destinée au dépôt et à la diffusion de documents scientifiques de niveau recherche, publiés ou non, émanant des établissements d'enseignement et de recherche français ou étrangers, des laboratoires publics ou privés.





THESE DE DOCTORAT DE

L'UNIVERSITE DE NANTES

ECOLE DOCTORALE N° 602 Sciences pour l'Ingénieur Spécialité : « Energétique-Thermique-Combustion»

Par

« Wanruo LOU»

« Optimisation du stockage de chaleur par la technologie thermocline »

Thèse présentée et soutenue «Nantes», le «29/04/2021»

Unité de recherche: Laboratoire de Thermique et Energie de Nantes (UMR 6607)

Rapporteurs avant soutenance :

Régis OLIVES	Professeur – PROMES, Université de Perpignan-Via Domitia
Kevyn JOHANNES	Maître de conférences – CETHIL, Université Claude Bernard Lyon 1

Composition du Jury :

	-	
Président :	Christian CRISTOFARI	Professeur – SPE, Università di Corsica Pasquale Paoli
Examinateurs :	Pascal STOUFFS	Professeur – LATEP, Université de Pau et des Pays de l'Adour
	Christian CRISTOFARI	Professeur – SPE, Università di Corsica Pasquale Paoli
	Nicolas BAUDIN	Maître de conférences - LteN, Université de Nantes
Dir. de thèse :	Lingai LUO	Directrice de recherche CNRS - LteN, Université de Nantes
Co-dir. de thèse :	Yilin FAN	Chargé de recherche CNRS – LteN, Université de Nantes

Chapter 1. Aim and scope of the thesis	4
1.1. General context	4
1.2. Research Barriers	4
1.3. Research objective	5
1.4. Thesis outline	5
Chapter 2. Thermal energy storage system design: geometry and operating settings	9
2.1. Introduction	. 11
2.2. Indicator	. 15
2.2.1 General classification	. 15
2.2.2 Thermodynamic performance	. 18
2.2.3 Hydrodynamic performance	. 20
2.3. Parameters influence in thermal performance	. 22
2.3.1 Flow rate	. 22
2.3.2 Working temperature	. 28
2.3.3 Flow distributor	. 29
2.3.4 Inlet/outlet location	. 34
2.4 Conclusion	. 35
Chapter 3. Single-tank thermal energy storage systems for concentrated solar power: Flow	
distribution optimization for thermocline evolution management	. 42
3.1 Introduction	. 45
3.2 Methodology for modeling and optimization	. 47
3.2.1 Parameter definition	. 48
3.2.2 Performance indicators	. 50
3.2.3 Heuristic optimality criterion	. 53
3.2.4 Basic principle of the optimization algorithm	. 54
3.2.5 Numerical implementation	. 55
3.3 Simulation and results	. 57
3.3.1 Geometries and simulation parameters	. 57
3.3.2 Optimization of the upper baffle for charging process	. 59
3.3.3 Combination of upper & bottom baffles for cyclic operation (charging &	
discharging processes)	. 62
3.4 Effect of design parameters	. 65
3.4.1 Reynolds number <i>Re</i>	. 65
3.4.2 Aspect ratio H/L	. 67
3.4.3 Cone angle ϑ	. 68
3.4.4 Number of orifices N	. 69
3.4.5 Global porosity of baffle (Φ)	. 70
3.5 Conclusion and perspectives	. 71
Chapter 4. Mitigating the impact of thermal jet in a single-medium thermocline storage tan	c :
Novel optimization strategy for flow distributor	. 78
4.1. Introduction	. 81

Table of contents

4.2. Experiment set-up	85
4.2.1 SMT storage tank and Ring-Opening Plate Distributor (ROPD)	85
4.2.2 Test-rig and experimental procedure	86
4.2.3 CFD Model and validation	89
4.2.4 Parameter definition and performance indicator	
4.2.5 Uncertainty analysis	
4.3. ROPD: CFD-based optimization and experimental verification	
4.3.1 Optimization strategy	
4.3.2. Numerical results	
4.3.3 Experimental verification	102
4.4. Influences of operating parameters: a sensitivity study	106
4.4.1 Local temperature evolutions	106
4.4.2 Energy and exergy efficiencies	110
4.5. Conclusion and prospects	113
Chapter 5. Impact of buoyant jet entrainment on the thermocline behaviors in a sin	ngle-medium
storage tank: fluid flow and heat transfer characterization	122
5.1. Introduction	125
5.2. Methodology	133
5.2.1. Experimental setup and procedures	133
5.2.2 Thermocline storage tank	135
5.2.3 PIV setup	135
5.2.4 CFD parameters	137
5.2.5 Characteristic indicators and dimensionless numbers	139
5.2.6 Uncertainty analysis	141
5.2.7. Comparison between CFD and PIV results	142
5.3. Optimization of the orifice baffle for thermocline control: the CFD results	145
5.3.1. Optimization algorithm	145
5.3.2 Optimization results	147
5.3.3 Comparison on the velocity and temperature profiles	149
5.3.4 Convective vs. diffusive heat transfer: gradient Richardson number	
Rigradient and Pe number	151
5.4. Influences of operating parameters on the thermal jet propagation and on the	ne thermal
performance of the SMT storage tank: experimental results	154
5.4.2 Effect of the injecting flow rate	
5.4.3 Effect of working temperature difference	
Chapter 6. Conclusions and future prospects	102 174
6.1 Thesis summary	
6.2 Perspectives	176
Synthèse de Thèse	177
Reference	
Acknowledgements	

Chapter 1. Aim and scope of the thesis

1.1. General context

The primary bottleneck in technology preventing effective deployment of renewable energy resources is their intermittent nature, thus the energy supply and the demand aren't balanced in time [Chavan et al., 2020]. Thermal energy storage (TES) is proposed, as one potential technology and cost-effective solution in order to solve the mismatch issues and enhance reliability and dispatchability [Suresh and Saini, 2020; Pelay et al., 2017]. Except for solar energy, thermal energy issuing from many variable sources such as solar thermal energy, geothermal energy, fossil-fuel power plants, nuclear power plants, industrial waste heat and biomass can also be employed in the TES system [Alva et al., 2018]. Currently available options of the TES system mainly include two-tank storage and thermocline-based single-tank storage. Over the last decade, thermocline-based single-tank storage system draws more and more attention than the conventional two-tank TES system because of its cost-effectiveness (about 35% cheaper) and space compactness characteristics [Grirate et al., 2016].

1.2. Research Barriers

As for the thermocline-based TES system during the charging/discharging process, the stratification ensures the hot heat transfer fluid (HTF) floating to the upper region of the TES tank and the large density cold fluid sinking to the bottom region. Their heat transfer region with a temperature gradient, named thermocline, is located at their interface. Generally speaking, the temperature gradient of thermocline should be held as sharp as possible in order to enhance a larger stored heat quantity into the upper hot HTF region. In other words, limiting the thermocline volume is the primary technological challenge for achieving greater thermal performance and thermal stratification [Li and Zheng, 2016].

The main barriers for thermocline-based TES technology include:

Homogeneous flow distribution may not be maintained, especially for the single-medium thermocline (SMT) storage tank [Luo et al., 2020]. The inlet thermal jet occupying the large difference of velocity and temperature compared to the calm fluid stored into the SMT must cause the strong mixing of hot and cold fluids and lead to the thermocline degradation, resulting in reduced thermal performances of the storage tank.

Sensible-medium solid fillers are usually chosen as packing mediums since the replaced HTF volume can help increase the cost competitiveness. Since the imperfect heat transfer takes place between the HTF and solid medium, thermal diffusion will

accelerate the thermocline decay (de-stratification). In the real-world packing structure design, there is no agreement for the optimal solution.

Phase change material (PCM) has attracted attention due to its significant latent heat of fusion. However, the severe technology barriers limiting the use of PCM including the high cost [Mostafavi Tehrani et al., 2019], the higher thermal resistance provided by its intrinsically low thermal conductivity [Xu et al., 2015], PCM Leakage [Chandel and Agarwal, 2017].

In real-world commercial and industrial TES appliances, cyclic life is also important in holding these systems from being marketed. For example, these issues exist such as the high temperature and corrosion of the molten salt and thermal ratcheting of tank's wall [Chang et al.,2016; Dahash et al., 2019; Pelay et al.,2020] into a TES integrated CSP plant.

1.3. Research objective

Regarding these above-pointed technology barriers, optimization of thermal energy storage by thermocline technology is our primary objective. We aimed to realize the favorable thermal performance and stratification with help of optimized flow distributors. In this thesis, we tended to create one original optimization algorithm special for flow distributors in order to maintain the thermal stratification of the thermocline region; we were eager to explore deeply the mixing mechanism into the SMT tank, then to minimize the effect of mixing on thermocline degradation by optimized flow distributor. We wanted to conduct the experiment investigation combined with CFD simulation to explore the optimization design on thermal storage system effeteness.

1.4. Thesis outline

The main structure and contents of this thesis are as follows:

Chapter 2

This chapter provides a comprehensive literature review on the SMT storage technology, with a special focus on the influences of tank geometry and operating conditions on thermal performance of the SMT storage tank, in particular, the effect of inlet position, flow distributor design, working temperature range and flow rate. The conventional solutions to reach a calm flow distribution and a stable temperature stratification are also reviewed. A detailed survey about characterization methods for SMT storage system is also conducted.

Chapter 3

This chapter mainly describes a novel CFD-based optimization algorithm employed on the perforated plate distributor. The propagation behavior of thermocline with distributor(s) is studied. We find that its size distribution of orifices directly determines the residence time (passage time) of the thermal front left from the buffer region, thus each orifice size can be adjusted based on its own residence time. After minimizing the standard deviation of all residence times, the thermocline region is smoothed in parallel, and the thermocline area is also minimized. Different impacts such as height/diameter aspect ratio, number of orifices, opening porosity, flow rate and cone angel are considered then we prove this optimization algorithm is effective and robust to improve charging/discharging/overall efficiency and capacity ratio.

Chapter 4

In this chapter, we employ our original optimization algorithm into a CFD model in order to determine the size distribution of the Ring-Opening Plate Distributor (ROPD). Both upper and bottom ROPDs are optimized with help of a novel optimization procedure in series. Transient thermocline volume, as well as velocity/temperature distribution provided by the experimentally validated CFD model, represent the gained favorable thermal performance after the optimization procedure. After that, the feasibility of this original optimization algorithm is validated experimentally by testing a lab-scale cylinder SMT storage tank in which water is regarded as HTF, by measuring the local temperature evolutions of the fluid during both the charging and discharging operations. A parametric study on the influences of operating flow-rate and operating temperature range on the energy and exergy efficiencies is also performed.

Chapter 5

This chapter provides an experimental validation of the inlet orifice baffle-type distributor (OBD) method by using Particle Image Velocimetry (PIV) technique. The completed mechanism of mixing into the main storage region including four evolution regimes: (1) Inertia dominated regime; (2) Transition regime; (3) Buoyancy dominant regime; (4) Lateral dispersion regime is observed although the distributors with uniform orifices are equipped in SMT tank. After the optimized upper distributor equipping, the mixing behavior within the main fluid region doesn't represent obvious anymore. The predicted CFD also provides more detail of entire velocity and thermal information, thus the competing relation between the convection and diffusion heat transfer mechanisms on the degradation of temperature stratification is particularly explored.

Reference

Alva G, Lin Y, Fang G. An overview of thermal energy storage systems. Energy 2018; 144:341–378. https://doi.org/10.1016/j.energy.2017.12.037

Chandel S, Agarwal T. Review of current state of research on energy storage, toxicity, health hazards and commercialization of phase changing materials. Renew Sustain Energy Rev 2017; 67: 581–596. https://doi.org/10.1016/j.rser.2016.09.070

Chang Z, Li X, Xu C, Chang C, Wang Z, Zhang Q, ... Li Q. The effect of the physical boundary conditions on the thermal performance of molten salt thermocline tank. Renew Energ 2016;96: 190–202. https://doi.org/10.1016/j.renene.2016.04.043

Chavan S, Gumtapure V. Numerical and experimental analysis on thermal energy storage of polyethylene/functionalized graphene composite phase change materials. J Energy Storage 2020; 27:101045. https://doi.org/10.1016/j. est.2019.101045.

Dahash A, Ochs F, Janetti MB, 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 Energ 2019;239:296–315. https://doi.org/10.1016/j.apenergy.2019.01.189

Grirate H, Agalit H, Zari N, Elmchaouri A, Molina S, Couturier R, Experimental and numerical investigation of potential filler materials for thermal oil thermocline storage, Sol. Energy 2016;131: 260–274, https://doi.org/10.1016/j.solener.2016. 02.035.

Li G, Zheng X. Thermal energy storage system integration forms for a sustainable future. Renew Sustain Energy Rev 2016;62:736–757. https://doi.org/10.1016/j.rser.2016.04.076

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. https://doi.org/10.1016/j.ces.2014.11.051

Mostafavi Tehrani S, Shoraka Y, Nithyanandam K, Taylor RA. Shell-and-tube or packed bed thermal energy storage systems integrated with a concentrated solar power: A techno-economic comparison of sensible and latent heat systems. Appl Energ 2019;238:887–910. https://doi.org/10.1016/J.APENERGY.2019.01.119

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. https://doi.org/10.1016/j.rser.2017.03.139

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. https://doi.org/10.1002/ep.13388

Suresh C, Saini R. Experimental study on combined sensible-latent heat storage system for different volume fractions of PCM. Sol Energy 2020;212:282–296. https://doi.org/10.1016/j.solener.2020.11.013

Xu B, Li P, Chan C, Tumilowicz E. General volume sizing strategy for thermal storage system using phase change material for concentrated solar thermal power plant. Appl Energ 2015;140:256–268. https://doi.org/10.1016/j.apenergy.2014.11.0

Chapter 2. Thermal energy storage system design: geometry

and operating settings

Abstract

The concept of energy conversion from stored solar energy to electricity can be realized by combination of three basic part, a concentrated solar power plant (CSP), a thermal energy storage system (TES) and power generation block. Thermocline technology has received much attention over the last two decades because of its high cost-effective approach compared to the conventional two-tank storage system. This paper reviews the recent progress in thermocline technology: a systematic characterization manners about thermocline are summarized; key players within geometry and operating conditions on thermal performance are described as well as their own detailed effects.

Nomenclature

Latin letters	
Ср	Heat capacity at constant pressure [J kg ⁻¹ K ⁻¹]
D	Diameter of tank [m]
d	Radius [m]
Ε	Energy [J]
Ex	Exergy [J]
Fr	Froude number
g	Gravitational acceleration [m s ⁻²]
Н	Height of tank [m]
h	Vertical distance of main fluid storage region [m]
Μ	Energy at certain moment [J]
ṁ	Mass flow rate [kg s^{-1}]
Pe	Péclet number
Re	Reynolds number
Ri	Richardson number
Т	Temperature [K]
t	Time [s]
u	Velocity at x-coordinate [m s ⁻¹]
V	Volume [m ³]
x	X-coordinate [m]

у	<i>Y</i> -coordinate [m]									
Ζ	Z-coordinate [m]									
Greek symbols										
α	Thermal diffusivity [m ² s ⁻¹]									
η	Efficiency [-]									
λ	Thermal conductivity [W m ⁻¹ K ⁻¹]									
μ	Viscosity [kg $m^{-1} s^{-1}$]									
ρ	Density [kg m ⁻³]									
σ	Capacity ratio [–]									
Subscripts/sup	erscripts									
bottom	Bottom									
С	Cold temperature									
ch	Charging									
dis	Discharging									
exp	Experiment									
f	Final									
fluid	Fluid									
in	Inlet									
j	Layer/node									
h	Hot temperature									
max	Maximum									
out	Outlet									
str	Perfectly stratified									
top	Тор									
р	Packed-bed									
tank	Thermocline-based TES tank									
stored	Stored									
0	Initial state									
Abbreviations										
CAPEX	Capital expenditure									
CFD	Computational Fluid Dynamics									
CSP	Concentrated solar power									
HTF	Heat transfer fluid									
IR	Infrared measurement									
LIF	Laser-Induced Fluorescence									
MIX	Mix number									
OPEX	Operational expenditure									
ORC	Organic Rankine Cycle									
PCM	Phase change material									
PIV	Particle Image Velocimetry									
TC	Thermocouple									
RTD	Resistance Temperature Detector									

SMT	Single-medium thermocline
Str	Stratification number
TES	Thermal energy storage

2.1. Introduction

The feasibility of solar thermal storage technology has been widely proved, thereby the growing application both in industrial sites and domestic sector has performed an apparent advances and developments in recent years. What can't be neglected was the TES integration with warm water supply, air conditioners, heat pumps and the food transport, even with its using growth in the building construction systems.

A concentrating solar power is comprised of a solar collector, TES integration and power generation block, among which TES systems is the most important for that this thermal storage technology isn't fully-fledged to be built to commercial application. To further develop research in TES technology, several different concepts were proposed. Based on whether heat transfer fluid circling in the solar collector could also serve as thermal media in the containing device or not, these TES integrated CSP systems can be classified into direct and indirect systems. For the indirect concept (Fig.1.C&D), HTF and storage material are separate with direct contact where a heat exchanger is used to transfer the thermal energy between HTF and storage medium by mode of heat conduction.

With respect to the current situation, a great deal of previous research has focus on the TES solution integrated in the CSP electrical power plant, which was conducted by two tanks with molten salt, thermal oil, gas or the other heat transfer fluid (HTF) to carry the thermal energy from solar field then store it in the hot tank.

Figure 1.A shows a simplified schema of direct two-tank TES integration system, the cold fluid at T_c from cold tank will go through of solar field, where the cold HTF could be heated up to high temperature T_h , then will be sent to the hot tank for storage where the HTF in high temperature would be pumped to the Organic Rankine Cycle ORC unit subsequently. This kind of storage was the most proven utility-scale built plant and these tested successful TES plant also encouraged more researcher to improve TES storage performance and cut off the high construction cost. In the aim to reduce the CAPEX and OPEX, in recent years, the concept of one single storage tank and the novel conception of dual media thermocline tank with available waste solid as filler material appeared.



Fig. 1. A. Direct concept for two-tank TES integration. B. Direct concept for single-tank TES integration. C. Indirect concept for two-tank TES integration, single tank. D. Indirect concept for single-tank TES integration.

As depicted as figure 1.B, basic operations during charging and discharging processes for the single-tank thermocline case is very similar to that for the conventional two-tank TES system, but the only difference is that a high-temperature zone and a low temperature zone are stored in only one tank where the two zones are separated by a transition area with a time-dependent gradient in temperature called thermocline. During charging operation, the HTF in high temperature entre into the top of tank, and the original flow of the tank would be extracted from the bottom of the single tank to solar flied. In contradiction to the heating process, the hot fluid extracted from the top of the thermocline-like tank to the heat exchanger for steam generation, then the cold HTF out of the heat exchanger would return to the bottom of the tank during the discharging step. Furthermore, the concept of dual media TES system which cheap solids such as solid waste reduce the used quantity of expensive HTF at approximately 70%, may prevail dominantly in comparison of single media TES system in view of CAPEX value.



Fig. 2. A. Different levels of stratification for same amount of stored heat (a) highly stratified, (b) moderately stratified, while (c) fully mixed storage. (1) Hot region, (2) thermocline, (3) cold region, and (4) uniform temperature [Dahash et al., 2019].

Just as its name implies, thermocline, as one 'line' separates hot and cold fluid within the perfectly stratified case. Hot and cold region stay above and below orderly because of their different densities. Regarding the real-world thermocline-based storage, thermocline represents the interface region instead of line. As shown as the Fig.2 a-b [Dahash et al., 2019], temperature density passes throughout such interface zone. The thermocline propagates at the charging/discharging period due to continuous heat transfer including advection, diffusion, conduction and heat loss to the environment. Concerning the de-stratification performance in Fig.2, the thermocline in Fig.2 (a) exhibits much thinner because of its sharper temperature gradient. The de-stratification phenomenon is more serious thus thermal dead zone is enlarged. Compared to the cases in (a)-(b), no thermocline behavior exists inside the storage tank, which is called the fully mixed statute.

There may be an ocean of research work to improve thermocline TES system's performance, mainly including operating condition and geometrical condition. The cost of filler material affects directly the cost of producing the mechanical power then decides the price of electricity. Researchers expect the novel system using PCM not only to improve the performance of the stratification degree on the thermocline tank but also to result in lower cost estimates than the two-tank TES system. From of this opinion, some thermocline systems combined with latent and sensible filler present. Some interesting investigation about novel design will be reviewed next.

Admitted that original intentions for introducing the packed media in a thermocline container is to cut down volumetric quantity of commercially high priced HTF, in the meanwhile, the industrial solid waste also could be recycled in this widely

accepted TES system. Furthermore, in restrictions of operational research circumstance, the advance in optimization of sensible thermal packed configuration is a little slow.



Fig. 3. Schematic influence factors for thermocline thermal performance

Numerous methods have been tried to classify influence factors of thermal performance for thermocline-like tank and summarized the influence rules in low temperature [Gang,2016] . Among of these methods, the most popular one is divided by operating (ex. velocity, HTF, cyclic operation, operating temperature, cut-off temperature) and geometrical condition (ex. inlet/outlet, flow distributor, tank form, L/D aspect ratio, and insulation). In this paper, we take consideration of packed thermocline cases, we rearranged this classification of these major influence factors. In view of recent journal papers, we surmised that thermodynamic performance is dominated by operating condition while main influence factors for stratification degree are related to geometry. But no research indicated thermodynamic behaviors are totally independent to stratification level yet, there exits several commonly influenced parameters for thermodynamic behavior and stratification degree such as inlet flow rate, inlet/outlet position, operating temperature difference and flow distributor design. More discussion upon these primary effects is given by followed sub-sections.

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

2.2. Indicator

2.2.1 General classification

Literature reviewing suggests various methods to characterize the performance of single-medium TES system. In general, it can be expressed by numerical indicators or graphical manners. If the thermal condition of TES tank is the measured target, these invasive manners, because of the simplicity for instrumental set up, is widely used to detect the temperature values. Concerning that the measurements by thermocline (TC) or resistance Temperature Detector (RTD) could monitor the temperature values at certain preset locations, the non-invasive optical measuring techniques such as Laser-Induced Fluorescence (LIF), infrared measurement (IR) are developed for the temperature maps[Andersen et al., 2007;Al-Habaibeh et al., 2018]. In the meantime, few analytic model is deeply developed for single-medium TES system because of the complexity of flow distribution at that case. Recent analytic models for single-medium TES system includes the plug flow model [Kleinbach et al., 1993], plume entrainment model [Nizami et al., 2013; Zachár et al., 2020] and multi-node model [Bai et al., 2020]. Moreover, the CFD technique for the single-medium TES system is widely used to visualize the transient fields after comparison/validation work with experiment T results. For the velocity measurements, Wang and Davidson successfully performed the velocity field visualization by Particle Image Velocimetry (PIV) technique inside the TES vessel [Wang and Davidson, 2017]. However, due to the instrumental constraints (ex. camera resolution) for the optical measurement, partial inlet jet and uncompleted evolution of mixing motion were monitored. In the meanwhile, CFD technique could make up with that issue. For example, the CFD model validated by experimental temperature profiles were employed to capture the global fields of both temperature and flow [Bouhal et al., 2017; El-Amin et al., 2018].



Fig.4 Graphical methods to characterize thermal performance in SMT tank: (a) analytic model [Zachár,2020];(b) CFD[Chandra and Matuska,2020] (c) Invasive measurement[Kim et al., 2019];(d)LIF[Andersen et al., 2007];(e)IR [Al-Habaibeh et al.,2018]. And the hydronymic performance: (f)CFD [Bouhal et al., 2017]; (g)PIV [Wang and Davidson,2017]; (h)LDV [Hess and Miller, 1982];

Considering the need of TES integrated renewable energy systems, their needs are various. Thus, except the above graphical manner, miscellaneous manners can be evaluated the thermocline and thermal performance. After reviewing the common indicators, table.1 summarizes two categories employed to characterize the performance of TES system. Several dimensionless numbers in fluid mechanics and those indices of intuitive analysis, first-law of thermodynamic, second-law of thermodynamic in thermal performance are listed. For example, it is worthy to note that Froude number, functional dependent on Richardson number, demonstrated the ratio of inlet kinetic energy to potential energy. Owing to its critical number, it usually is regarded as a judging indicator whether the inlet operating condition will lead to a big mixing [Kim et al., 2017]. In recent years, there has been the growing interest in exergy analysis which expresses the maximum quantity of useful work that can be transferred from TES system. Energy analysis obeying the law of energy conservation can be behalf of the quantity but not for the quality. In the application domain of singlemedium TES tank where the flow distributor is more complex than these filler-packed cases, the issue of exergetic loss is the one of the most important barriers for design in optimal TES system. The amount of exergy destruction is generally caused by heat transfer, fluid friction and energy dissipation to the surroundings.

Method of		Indicator	Ref example	Remarks			
characterization							
	Intuitive analysis	Temperature gradient	Fasquelle et al., 2018	-Usually represent the stratification degree versus the axial position or the time. -But not appropriate to perform the comparison with other researchers' TES system.			
		Thermocline thickness	Chandra and Matuska,2020	-Be adopted to the situation having the stable thermocline and regular form of thermocline such as packed-bed TES tank. -But not suitable for the single-medium tank			
		Tail factor	LOU et al.,2020	-Tail factor ∈ [0, 1]. -Be irrespective of energy content. -Represent the difficulty to complete the fully storage/extraction.			
Thermal	1 st law	Energy efficiency (Charging, discharging, overall)	Zanganeh et al., 2015	-Functional dependent on the ratio of the net difference of input & output to the total input energy. -The overall efficiency sometimes expresses the effect of cyclic operation.			
periormance		Capacity ratio	LOU et al.,2020	-Functional dependent on the ratio of the net difference of input & output to the theoretically maximum energy stored inside the TES tank.			
	2 nd law	Exergy/ entropy efficiency	árdenas and Garvey., 2019	-Key indicator for assessing the useful work transferred and extracted during energy release process.			
	Others	Mix number	Wang et al.,2020	-Mix number $\in [0, 1]$. -It decreases as thermal stratification increasing. Bulk indicator for TES tank against time.			
		Stratification number	Bouhal et al., 2017	-Assessment the decay of stratification inside the TES tank both for charging/discharging.			
Hydrodynamic performance		Reynolds number Re	Shaikh et al., 2018	 -Have the different forms for inlet, tank or packed fillers -The basic factor for the determination of flow regime. - Its effect on thermal performance may not be monotonous. 			
		Froude number Fr	Chung et al. 2008 Kaloudis et al.2016	-Fr is usually used to characterize the gravity current for the inlet/ diffuser design. -Recommended range of the inlet Froude number is found less than 2.			
		Péclet number Pe (bulk or local form)	Chandra and Matuska,2020	- As ratio of the convection to the diffusion rates, influences largely the thermocline thickness, especially in SMT.			
		Richardson number Ri	Paiva et al.,2020	-Have the bulk or the gradient form. -The critical value is often set at 0.25, acknowledging that the thermocline is stable while the value is superiors to 1.			

Table.1. Method of performance characterization for single media tank (SMT).

2.2.2 Thermodynamic performance 2.2.1.1 Intuitive analysis

The observation of temperature gradient against time or height holds for the general evaluation of thermocline behavior is one of the most common manner for previous research groups [Advaith et al., 2021]. Especially, such manner is appropriate for the comparison work within their own circumstance at given operating and geometry conditions whereas isn't adapted to compare the thermal performance among the different TES systems. The thermocline thickness, as the most widely used indicator describing the de-stratification state, represents inappropriate for the single-medium TES system. in terms of a single-medium tank, thermocline thermal front isn't usually plat, as shown in fig.5 by the isotherms [Dehghan and Barzegar, 2011], thus the thermocline surface and volume may more favorable for 2D and 3D CFD simulation models, respectively. Furthermore, a dimensionless indicator named tail factor, which is irrespective of the HTF psychical thermal properties, operating and geometry conditions, is calculated from the dimensionless outlet temperature profile against the dimensionless time [Lou et al., 2020].Such indicator, evaluating the thermocline degradation degree, is easy to calculate and to compare among different TES storage tank.

2.2.1.2 First law

The ratio between the integral of net stored energy to the integral of entire inlet energy is named as charging efficiency η_{ch} while the integral of net extracted energy to the integral of initially stored energy is named as discharging efficiency η_{dis} .

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

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

Where T_{in} and T_{out} are the inlet and outlet temperature of the HTF, respectively. C_p is the temperature-dependent specific heat of the HTF and \dot{m} is the mass flow-rate. Cut-off temperature in the end of charging or discharging is dependent on the real-world operating conditions.

In term of capacity ratio (σ), such index is defined as the ratio of real stored thermal energy to the maximum energy storage capacity of the tank. As a result, σ equals to 1 for fully charging process but is smaller than 1 for partial charging process. The definition in a real 3D storage tank is shown in eq.

Capacity ratio
$$\sigma = \frac{E_{stored}}{E_{stored}^{max}} = \frac{E_{in} - E_{out}}{V_{fluid} \cdot (c_{p_h} \cdot T_h \cdot \rho_h - c_{p_c} \cdot T_c \cdot \rho_c)}$$
 (3)

2.2.1.3 Second law

If combined with first and second law of thermodynamics, exergy can describe both the quantity and quality of energy inside the TES tank. The dissipated work due to friction, mixing, heat loss can be expressed by exergy analysis [Carmona et al., 2020; Ma et al., 2020]. In order to assess the benchmark of theoretical thermal performance of energy system, Shah and Furbo defined the equation for exergy efficiency as the portion of experimental exergy sum of tank layers to the exergy of perfectly stratified tank [Shah and Furbo,2003; Chandra and Matuska, 2020]:

$$\eta_{E_{\chi}} = \frac{E_{\chi}(exp)}{E_{\chi}(str)} \tag{4}$$

Where,

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

And j is the layer number, m_j is the mass of control volume. Such efficiency varies from 0 to 1, for 0 meaning the fully mixed state while 1 signifying perfectly stratified state.

2.2.1.4 Others

The stratification number, which was proposed by Fernandez-Seara Jose et al, describes the ratio of the mean of temperature gradients for each radial position at each time interval to the temperature gradients in the beginning of the charging process [Fernandez-Seara et al., 2007]. The stratification number (*Str*) is given by the following equation:

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

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

19

$$\overline{\left(\frac{\partial T}{\partial z}\right)_{max}} = \begin{bmatrix} \frac{T_{max} - T_{min}}{(J-1)\cdot\Delta z} \end{bmatrix}$$
(8)

The two positions of nodes J and j+1 should be equidistant on a radial location and meaning of the parameter was presented as the average stratification number.

On the contrary to the stratification number which concerns the temperature gradient against the height deviation of the tank, The MIX number focus on the instantaneous degraded energy during mixing process. The definition equation is followed [Wang et al., 2020]:

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

Where

$$M_E = \sum_{j=1}^J y_j \cdot E_j \tag{10}$$

$$E_j = \rho_j \cdot c_p \cdot V_j \cdot T_j \tag{11}$$

where y is the distance between the tank bottom and the center of the node j. The value is from 0 to 1, the smaller of MIX, the less mixing extend and better thermal stratification state.

2.2.3 Hydrodynamic performance 2.2.3.1 *Ri*

The Richardson number Ri, the ratio of buoyancy force to mixing forces, having been widely used to assess the thermal stratification. The formula of bulk tank is commonly used:

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

Where the L is the characterization length of storage tank, v is the characteristic velocity and the maxi value of Ri_{bulk} is usually employed to mark the global state for TES system at given operating condition including the working temperature range and flow rate [Yee et Lai, 2001; Njoku et al, 2016]. Sometimes, such indicator is dependent on the outlet temperature at charging duration [Castell et al, 2010; Afshan et al., 2020].

In order to explore deeply the deviation part inside the storage tank, the gradient form is also appeared [Chandra et al., 2020]:

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

As per definition, thermocline sustains better stratification at lower inlet velocity.

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

Table 2. Comment on bulk Richardson number

2.2.3.2 Pe

The Péclet number *Pe* is defined to be the ratio of the convective transport rate to the diffusive transport rate. The bulk formula assessing the competition relation of inertial to diffusion is defined:

$$Pe = \frac{\nu \cdot L}{\alpha}, \alpha = \frac{\lambda}{\rho \cdot c_p} \tag{14}$$

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

Once the previous authors tend to explore more precisely such competition relation. The other formulas appeared. In case of packed-bed TES tank, the superficial and effective péclet number can be used where the characterization length is the particle diameter [Zanganeh et al., 2014; Davenne et al., 2018]. Moreover, regarding the SMT system, the local velocity distribution has significant difference, thus the superficial or mean velocity of the tank can stand for the local competition between the inertial and diffusion.

Concerning that, the grid /element Péclet number is resolved to characterize the hydrodynamic statute inside the control volume [Li et Tao, 2002; Dallaire, 2017; Bayón et Rojas, 2013; Patankar, 1980]:

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

Where δx and δy are the size of the control volume, v stands for the velocity inside this grid, and α is determined by the temperature value of such grid.

2.2.3.3 Fr

Froude number, as indicator for diffuser design or inlet design, represents the characteristic ratio of inertial force to buoyancy force [Kocijel, et al.2020].

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

As per definition, Fr could be the reciprocal of square root of the bulk Richardson number. A growing attention focusing on its recommended maximum value is usually related to the flow distributor design [Chung et al. 2008; Kaloudis et al.2016].

2.3. Parameters influence in thermal performance2.3.1 Flow rate

The effect of the HTF flow rate, acknowledging that determines the primary flow regime inside the TES vessel, has been world widely discussed in different TES systems. In this paper, we listed several published papers which could stand for the impact of this operating parameter on TES systems without packing or with sensible/latent solid fillers. To reveal its inner flow distribution during discharging process, one two-dimensional (2D) transient computational fluid dynamics (CFD) simulation was performed to study the influence of flow rate for a solar hot water storage tank [Dehghan and Barzegar, 2011]. Fig.5 demonstrates the virtual streamlines and isotherms at different inlet Reynolds number. By the means of raising the inlet Reynolds number Re_{in} the global streamlines turns from the buoyancy-controlled motion ($Re_{in} = 100$) to inertial-controlled motion ($Re_{in} = 1000$). The incoming cold HTF arrive upwards to the upper region of the tank, thus the thermocline borders is even close to the outlet port while $Re_{in} = 1000$.

In another published paper related to single-medium TES tank [Yaïci et al., 2013], four various flow rates 0.05, 0.1, 0.15, 0.2 kg s⁻¹ were numerically investigated by a 3D transient CFD simulation. From the displayed graphic temperature contours and outlet temperature profiles, they discovered that de-stratification phenomenon may be more pronounced at low flow rate 0.05 kg s⁻¹ whereas the mixing is more intense at

high flow rates. A competition between the effect of mixing and thermal diffusion is proposed thus an optimal flow rate may exist at given TES structure and working temperature. From the perspective of the comparison work about this parameter flow rate, the graphical temperature distribution displayed at the dimensionless charging /discharging time may be more persuaded.



Fig.5 Streamlines (left) and isotherms (right) inside the tank at t = 20 (a) Re = 100, (b) Re = 200, (c) Re = 400, (d) Re = 600, (e) Re = 800 and (f) Re = 1000 [Dehghan and Barzegar, 2011].

Besides, a mount of previous articles focused on the thermal performance for TES system with sensible solid fillers where certain of them summarized the contrary conclusions. On the one hand, the raised flow rate is found to shorten effective discharging time and decrease the discharging efficiency due the incoming flow penetrated further away along the centerline of the tank at high flow rate [Abdulla and Reddy, 2017; Wu et al., 2014; Yang and Garimella, 2010]. Wu et al. proposed four inner packing structure including the channel-embedded, parallel-plate, rod-bundle and packed-bed then investigated these four structure into the parametric study of inlet flow velocity. Among these, the packed-bed kind reflects the more robust geometrical structure and the discharging efficiency can be kept always at high value. However, the channel-embedded kind occupied the worst thermal performance as varying inlet flow velocity. On the other hand, the decreased transient thermocline thickness is found as inlet velocity going up during discharging process [Wang et al., 2020]. Furthermore, Nandi et al. discovered the relatively stable charging efficiency at laminar zone where Re_{in} is at the range [10,300] whereas, the platform of such efficiency doesn't exist anymore as Re_{in} is superior to 300. The similar trend of discharging efficiency is also gained. For more clarity, the influence of Re_p is also conducted to compare the thermal stratification. Fig. 6(b) shows the increasing trend of stratification number as raising Re_p at [10,100]; the thermocline behavior of storage is still kept while $Re_p \epsilon$ [10,300]; the thermocline behavior is disappeared at the turbulent zone, especially $Re_p > 400$ [Nandi et al.,2018].



Fig.6 Variation of charging efficiency with inlet Reynolds number Re_{in} (a); Change in stratification with Reynolds Number Re_p (b).

Study	N/E	TES	Inner structure	Tank size	Т	Inlet flow	HTF	Conclusion about the effect of flow rate
		form		(m)		rate		
Dehghan	N	Mantle	No packing	H/D=2		$Re_{in}=200,$	water	-low <i>Re_{in}</i> : mixing region is confined in the bottom zone during discharging
and		vertical				400, 500, 600,		-high Re_{in} : mixing region expands directly towards to outlet port
Barzegar, 2011		cylinder				800		
Yaïci et	N	Vertical	No packing	H = 1.194	$T_c = 30$	0.05, 0.1,	water	-Low inlet velocity : thermocline deterioration is more pronounced
al., 2013		cylinder		D = 0.442	$T_{h} = 60$	0.15, 0.2		-High inlet velocity : the mixing is intense
					(°C)	kg s ⁻¹		-Competition relation between the effect of mixing and thermal diffusion resulting
								in an optimal flow rate.
Wang et	Ν	Vertical	Porous bed	H = 5.9	$T_c = 566$	0.002, 0.0025,	Solar	Discharging:
al.,2020		cylinder		D=3	$T_h = 723$	0.0030,	salt	- inlet velocity↑: transient thermocline thickness↓
					(K)	0.0035,		- inlet velocity \uparrow : mechanical stress of the steel wall \downarrow , does better to the structural
						0.0040 m s^{-1}		reliability of TES tank.
								Charging:
								-The effect of inlet velocity on formation of thermocline is relatively small
Nandi et	N	Vertical	Sensible	H = 6.1	$T_c = 290$	<i>Re_{in}</i> =10-1000	Solar	$Re_{in}\epsilon$ [10,300]
al.,2018		cylinder	packed-bed	D =3	$T_h = 390$		salt	-Effect of <i>Re_{in}</i> on charging/discharging efficiency can be ignored
					(°C)			-stratification number \uparrow while Re_p is ϵ [10,100]
								$Re_{in}\epsilon$ [300,1000]
								-Reduction of thermocline behavior of storage with $Re_{in}\epsilon$ [300,400]
								-an unsteady and a chaotic flow, and a turbulent flow model is considered for the
								TES with $Re_{in} > 300$
								- Re_{in} 1: Charging/discharging efficiency \downarrow
Abdulla et	Ν	Vertical	Sensible	H = 12	$T_c = 563$	0.0003373 -	Solar	- Inlet velocity ↑: effective discharging time↓, discharging efficiency↓
Reddy,		cylinder	packed-bed	D = 14.38	$T_h = 663$	0.0007724	salt	
2017					(K)	m s ⁻¹		
Wu et	N	Vertical	- Channel-	H = 15	$T_c = 290$	0.002-0.01	thermal	- inlet velocity : effective discharging time \$\u00e4 for all four inner structures
al.,2014		cylinder	embedded		$T_h = 390$	m s ⁻¹	oil	- inlet velocity [↑] : the main trend of effective discharging efficiency↓
			-Parallel-plate		(°C)			- inlet velocity 1: the inner structure with packed-bed could maintain the high
			- Rod-bundle					effective discharging efficiency, whereas channel-embedded structure has the worst
			- Packed-bed					performance for effective discharging efficiency

Table.3 Parametric study on flow rate within thermocline-based TES system

Yang et	N	Vertical	Sensible	<i>H</i> = 2.77-	$T_c = 250$	$Re_p=1, 5, 10,$	HITEC	- Re_p^{\uparrow} , discharging efficiency \downarrow
Garimella,		cylinder	packed-bed	32.1	$T_h = 450$	20, 30, 50	salt	
2010				D = 2 or 5	(°C)			
Bhagat et	N	Vertical	PCM packed	<i>H</i> = 25	$T_c = 142$	0.00145-	Hyther	- Flow rate1: effective discharging time
Saha,		cylinder	bed	D=12	$T_h = 171$	0.0058	m 600	-Mass flow rate influences the heat transfer rate between HTF and PCM significantly
2016					(°C)	kg s ⁻¹		- flow rate1: volumetric heat transfer coefficient1, overall effectivenes1
Saha et	N	Vertical	PCM packed	<i>H</i> = 0.6	$T_c = 325$	200, 240,	air	- inlet flow rate1: effective charging time↓
Das,2020		cylinder	bed	D=0.25	$T_h = 446$	280		- inlet flow rate1: charging efficiency↓
					(°C)	kg h-1		- inlet flow rate [↑] : second law efficiency [↑]
He et	E, N	main	No packing	H = 1.1	$T_{c} = 37$	0.3, 0.6, 0.9	water	- inlet flow rate1:thermocline thickness1
al.,2019		cylinder	Or	D=0.9	$T_{h} = 70$	m ³ h ⁻¹		- Thicker thermocline appeared within the PCM packed-bed structure compared to
		body	PCM packed		(°C)			the no packing case
			bed					

Table.4 Parametric study on working temperature within thermocline-based TES system

Study	N/E	TES	Inner	Tank size	Temperature	Inlet	HTF	Remarks about working temperature range ΔT
		form	structure	(m)		flow		
Nelson et	E	Vertical	No packing	H = 1.08-	$\Delta T = 10, 12$	7.12 ×	water	$-\Delta T \uparrow$: thermal stratification \uparrow and mixing \downarrow
al1999		cvlinder		1.89	(°C)	10-5		č
,				D=0.54		m ³ s ⁻¹		
Shaikh et	N	Vertical	No packing	H = 1.016	$T_c = 573$	$Re_{tank} =$	Solar	$-\Delta T \uparrow$, penetration length
al., 2018		cylinder	but with	D =0.488	$T_h = 651, 773, 823$	5-100	salt	$-\Lambda T \uparrow$ thermocline thickness \uparrow as the buoyancy driven
			vertical		(K)			mixing becomes dominant
			porous					mixing occomes dominant
			distributor					
Yaïci et	Ν	Vertical	No packing	H = 1.194	$\Delta T = 5,10,15$	0.1	water	$-\Delta T$ doesn't affect the formation of the no matter varying T _h
a., 2013		cylinder		D = 0.442	(K)	kg s ⁻¹		or T _{ini} thermocline during charging process
,				-	()	0		
Sun et	E,N	Vertical	No packing	H = 7	$T_{c} = 10$		water	$-\Delta T \uparrow$, thermocline thickness \uparrow
al.,2018		cylinder		<i>D</i> =4	$T_h = 20, 40, 60, 80$			
					(°C)			
Abdulla et	Ν	Vertical	Sensible	H = 12	$T_c = 613$	0.00033	Solar	$-\Delta T$, thermocline thickness , discharging efficiency
Reddy,		cylinder	packed-bed	D = 14.38	$T_h = 663 - 838$	73-	salt	
2017		-	-		$\Delta T = 50-100$	0.00077		
					(K)	24 m s ⁻¹		

Wang et al.,2020	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 - $\Delta T \uparrow$ effective charging/discharging time
Nandi et al.,2018	N	Vertical cylinder	Sensible packed-bed	H = 6.1 $D = 3$	$T_c = 290 T_h = 390,490,590 (°C)$	<i>Re_{in}=10</i> -1000	Solar salt	Lat r , encentre enarging theorem r is a second gring time r . Laminar regime $Re_{in} \in [10,300]$: -The effect of ΔT on both charging and discharging efficiency is small enough. Turbulent regime $Re_{in} > 300$: $-\Delta T$ \uparrow , charging efficiency \uparrow
Xu et al., 2012	N	Vertical cylinder	Sensible packed-bed	H = 14 D = 8.6	$T_c = 190-340$ $T_h = 390$ (°C)	6.017*1 0 ⁻⁴ m s ⁻¹	solar salt	- ΔT doesn't affect the effective discharging time - The TES tank with the largest ΔT results to the thinner thermocline thickness. However, this confined effect on thermocline propagation isn't considerable.
Yin et al., 2017	E,N	Vertical cylinder	Porous bed	H = 0.6 D = 0.12	$T_c = 290$ $T_h = 350,370,390$ (°C)	0.02 m s ⁻¹	molten nitrate salt	$-\Delta T$ ↑, discharging efficiency \downarrow
Bruch et al.,2017	E	Vertical cylinder	Sensible packed-bed	H = 3 $D = 1$	Various ranges: 100-250, 75-175, 100-200, 75-250, 125-200, 150-250 (°C)	0.001 m s ⁻¹	oil	-Single dimensionless temperature profile has been observed, regardless of temperature set points
Kumar et al., 2016	Ε	Vertical cylinder	sensible Or PCM- combined packed bed	H = 1 $D = 0.4$	$T_c = 35$ $T_h = 60, 70, 80$ (°C)	1, 1.5, 2 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	N	Vertical cylinder	Partial PCM packed bed	H = 1 $D = 0.4$	$T_c = 15, 25, 35 T_h = 60, 70, 80 (°C)$		water	$-\Delta T \uparrow$: time-evolution Richardson number $-\Delta T \uparrow$: charging efficiency \uparrow while a slightly lower storage efficiency is found
Bhagat et Saha, 2016	N	Vertical cylinder	PCM Packed-bed	H = 25 D= 12	$T_c = 142$ $T_h = 180-240$ (°C)	0.00145 -0.0058 kg s ⁻¹	Hyther m 600	 Flow rate effective discharging time keeps the equivalent value ΔT energy stored/released overall effectiveness \

2.3.2 Working temperature

If the buoyancy force is a function of density difference between the inlet jet and surrounding fluid, such jet is called thermal jet. The Fig.7 shows the comparison of the predicted trajectory movement for thermal jet. The positively buoyant jet is expected since the hot jet occupying the lighter density could stay upwards, thus the thermal stratification is in normal order. However, negatively buoyant jet having the heavier density can't be supported by the bottom lighter hot fluid, thus such temperature overturning causes unstable thermocline. The working temperature range is also one of the most important operating factors for than temperature difference ΔT between the hot and cold fluid directly determines the buoyancy force then affects the mixing motion driven by buoyant plume.



Fig.7 Thermal jet

Previous experimental investigation [Al-Habaibeh et al., 2018] have clearly shown the trajectory movement by the means of the water 'snake' due to the temperature difference, including positive ΔT or negative ΔT . Moreover, the paper reviewing on the quantitative effect of temperature difference ΔT within the thermocline-based TES system is given in table.4, including the systems without packing, with the sensible solid fillers and with latent PCM packing, respectively. At first, if the TES system is set as the single-medium unit, we found the results weren't consensus. On the one hand, the high ΔT is regarded as one expected factor for mixing restriction. For example, Nelson et al. indicated that thermal stratification is improved and mixing region is more confined as ΔT is increasing [Nelson et al., 1999]. The penetration length of inlet injection can be shortened by increasing ΔT [Shaikh et al., 2018]. On the other hand, high ΔT is found to accelerate the stratification destruction. For instance, Sun et al depicted that the thermocline thickness is rising as ΔT increasing [Sun et al., 2018]. Moreover, Shaikh et al. established that CFD model where the inlet hot flow entered into the TES vessel from a vertical porous distributor eccentrically placed in the tank [Shaikh et al., 2018]. They discovered that the thermocline propagated more seriously with greater ΔT although the penetration distance of entering jet is decreased. The concluded the reason of de-stratification at high ΔT is because of the dominant buoyancy driven mixing. From the perspective of the effect of ΔT into the single-medium TES tank, the set of flow rate should be considered. The diffusion heat transfer turns to be more important with the greater thermal conductivity coefficient at higher ΔT as long as the momentum-forced injection doesn't control the mixing any more.

In accordance with the thermal performance of TES tank with the sensible packing, the previous results were relatively consensus. The increased thermocline thickness and decreased charging/discharging efficiency are observed as setting a higher ΔT [Abdulla and Reddy, 2017; Wang et al., 2020; Yin et al., 2017; Kumar et al., 2016]. Such similar result is easy to understand since the laminar regime and relatively uniform and calm flow distribution can be normally obtained into the packed-bed region. It was worthy noted that the effect of ΔT was deeply studied by varying Re_{in} in the range of [10, 1000]. The numerical result demonstrated the considerable difference induced by varied ΔT between laminar and transient regimes. At the transient regime, the higher ΔT can improve the charging efficiency, which is correspond to phenomenon in the single-medium TES cases.

Furthermore, the effect of ΔT was also investigated into the TES tank with latent packing fillers [Kumar et al., 2016; Majumdar et al., 2019; Bhagat and Saha, 2016]. Among them, the rising ΔT contributed to the better thermal performance, such as the charging efficiency, Richardson number, the amount of stored/released energy and overall effectiveness. The impacts of ΔT on sensible and latent packed-bed were experimentally discovered to be opposite [Kumar et al., 2016]. The increased temperature driving potential at higher working temperature enhanced instantaneous heat transfer between HTF and PCM fillers.

2.3.3 Flow distributor

Concerns have arisen into the time evolution of flow and temperature distribution into the transient heat charging/discharging process. The incoming injection occupying the difference in the density and velocity in comparison with the surrounding calm fluid in TES vessel must disturb the initial flow state and introduce the expected perturbation. Accordingly, the device named flow distributor helping to rebuild the flow distribution for the purpose of improving the thermal stratification has received much attention in recent years. Until now, there is no general agreement on the classification of the flow distributors. Considering this, we classify these published designs on the basis of their primary functions into three kinds: producing multi-branch flow, producing inversed flux and producing radical flux, respectively. Fig.8 summarizes the several designs of flow distributor.

More precisely, the multi-branch type is usually fabricated with multiple ports in order to divide the inertial force controlled incoming injection into many minor flux. Li et al. performed a CFD analysis to characterize the influence of simple inlet and the shower-type distributor. A higher effective discharging efficiency was obtained by such flow distribution in comparison with the simple inlet for the flow rate range from 5 to 15 L min⁻¹. The mean raised discharging efficiency was about 14% [Li et al., 2014]. A sintered bronze conical diffuser (SBCD) and a conventional inlet elbow were really fabricated into TES tank in order to evaluate the thermal performance in energy storage process [García-Marí et al, 2013]. The sensibility analysis for both structures was conducted by varying flow rate at 6, 10, 16 L min⁻¹. According to the temperature profiles along the centerline of the tank, the SBCD at all flow rates obtained the greater temperature gradient versus dimensionless charging time whereas the tailing phenomenon of the thermocline much more serious with the conventional inlet elbow, especially at 16 L min⁻¹. Besides, according to both mix number and thermocline thickness, the TES geometry with SBCD demonstrated a better thermal performance and more robust character while vary flow rate. Furthermore, perforated plate design in this classification kind has been widely used within packed-bed TES system because the perforated plate could support the solid filler and such shape is simple to fabricate. Owing to such packed-bed structure, the conventional perforated plate is considered to be sufficient producing uniform flow distribution [He et al., 2019; Nandi et al., 2018].

Afrin et al. numerically investigated two different pipe flow distributors where the orientation of orifice openings were set towards to the main fluid storage region in fig.8 (c) or to upper/bottom port in fig.8 (g). The velocity maps generated by these two different orientations are displayed in fig.9 the left velocity field is much more uniform than that in the right figure. It implied that the concept of multi-branch design may not be sufficient, and the velocity direction from the distributor is also important. In recent years there has been considerable interest in obstacle design into the single-medium tank. Our second classification type of distributor focus on generating inversed flux by means of installing the flat obstacle, inclined obstacle, moving plat or adjusting the orientation of orifice opening.



[Afrin et al., 2013](c); [Bouhal et al.,2017](e,f); [Afrin et al., 2013](g); [Taher et al.,2019](h); [Shah et al.,2005](i); [Gajbhiye et al.,2018](j); [Dragsted et al.,2017] (k); [Al-Azawii et al.,2020] (l)

Taher et al., proposed three different obstacle configurations, as shown in the Fig.10 [Taher et al., 2019]. The simple right angle plate (config.1) had a thick thermocline region. Two eddy currents were discovered in the static zone above simple right angle plate. The inertial force controlled motion was still maintained above the plate then stretching along the sidewalls of the tank was taken place. The mixing even expanded to the half of the storage tank. According to the config.2, the supplementary plate installed laterally adjusted the direction of flow jet thus the incoming cold flow return back to the centerline of the TES vessel. This change had the potential to obtain more stable flow filed. A novel passively moving plate (config.3) was placed at axis of spherical TES tank where dynamic mesh was used to update the computational domain

as plate moving. Such design can prevent the direct contact between the momentumcontrolled inlet jet and the calm ambient fluid inside the storage tank.



Fig.9 Velocity contour along the vertical direction [Afrin et al., 2013]

Chandra et al. numerically investigated three geometries of inlet & diffuser devices: a shower-type diffuser, a slotted inlet and a simple inlet, respectively. Among these three configurations, the slotted one performed the best than the shower-type one and simple one at large flow rate of 800 L h⁻¹ whereas on obvious improvement was gained with this slotted inlet device at low flow rate of 200 L h⁻¹. Their efficiency difference turned from 8.7% at 800 L h⁻¹ to merely 1-2% at 200 L h⁻¹. They predicted that slotted typed inlet can represent the best thermal performance. In the author's opinion, the reason why the slotted device is regarded as the optimal choice may be because of the direction of such slot. The direction of inlet flow leaving from such slotted inlet device is switched to be reversed from the primary flow direction of discharging process [Chandra and Matuska, 2020]. Their conclusion is almost correspondent to that of Li et al [Li et al., 2014].

Regarding the third type of distributor, the primary target of these designs is aimed at generation radical 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 single-medium TES tank [Gajbhiye et al., 2018]. For this primary function of this design, it can be said that porous flow distributor makes it possible to allow the inlet flow to ooze on the basis of the matching density instead of inertial force. Furthermore, in order to deeply study the difference of axial and radical flow, Al-Azawii et al. experimentally investigated the packed-bed TES system with the incoming axial flow or with the radical flux. The perforated tube is immersed all the length of packed-bed where the axial flow inside the perforated tube can be interrupted by zero/one/two inserted perforated plate(s). These extra plates provide a resistance to the hot air flowing axially and force the air flowing out from the perforated openings of the immerged tube. The charging efficiency can be increased from 75.3% to $80.3\pm2.8\%$ by the conventional avail flow and radical flow with two extra plates [Al-Azawii et al., 2020]. Besides, Dragsted et al. experimentally investigated four stratifiers including one design made of the flexible polymer with opening all along the side and total length of such stratifier. This flexible material can avoid counter flow from ambient to stratifier [Dragsted et al., 2017].In comparison of other two conventional solid stratifiers producing radical flux, this flexible distributor performed better regarding the stratification efficiency at low flow rate. All three stratifiers producing radical flow were worse than a simple rigid pipe whose opening was only in the top. Assuming that the inlet position may play important role in the thermal stratification, we also review the papers concerning the effect of inlet position in the sub-section 3.4.



Fig.10 Evolution of the temperature contours during the discharge for different configurations [Taher et al., 2019].

2.3.4 Inlet/outlet location

As before mentioned in subsection, the incoming flow have the large difference of density and velocity in comparison to the fluid stored inside the storage tank. Accordingly, a free shear layer is formed causing a jet at the inlet port where the vortex, mixing, entrainment are subsequently inside the storage vessel, as displayed in Fig.11 [Dahash et al., 2019]. The buoyancy-driven plume and inertial-controlled jet will coexist and further extend to a larger region. If the momentum motion of the incoming jet is large enough to impinge itself against the opposite side wall, the mixing would be propagate upwards and downwards [Toyoshima et al., 2013]. These aforesaid possibilities may occur near the outlet, thus inlet/outlet is often recommend to be suited close to upper or bottom wall of storage tank.



Fig. 11 Schematic representation of the water flow, mixing phenomena, water inlet/outlet positions and effective water zone for a thermally stratified TES with direct charging/discharging ports [Dahash et al., 2019].

For example, the location of side inlet/outlet port because of simple structure has been studied experimentally [Assari et al., 2018] and numerically [Yaïci et al., 2013; Farmahini-Farahani, 2012] in the past decade. Regarding the optimal inlet/outlet location, the experimental and numerical conclusions represent a good agreement. Farmahini-Farahani obtained the highest dimensionless exergy when the inlet and outlet port were set as close as the upper and bottom walls. In the research of Yaïci et al., the buoyant impact on mixing isn't very obvious for the case 1 where the vertical distance between inlet port and upper wall is just 20 mm, as shown in Fig.12(c). As raising such distance from case 2 to case 4, buoyancy-driven plume occupied more and more important portion into mixing flow. Subsequently, the impinged jet continues to expand upward and downwards thus dead zone is enlarged. The similar result was also obtained within a horizontal storage tank where four different locations of side inlet/outlet ports were experimentally investigated, as shown in Fig. 12(a). They

recommended that the introduction of hot water was from the highest position and cold water outflow was from the lowest position. Besides, incoming and outgoing flow were introduced and extracted by the immersed pipes in Tank A in Fig.12 (b) [Kim et al., 2017; Kim et al., 2019]. The energy storage /extraction represents inappropriate within Tank A since the thermocline propagated severely owing to internal thermal conduction between the immerged pipes and stored fluid. Concerning such issue and the stability of operating condition, they modified the configuration tank A to B and C. Time-evolution temperature profiles monitored by thermocouples indicated that tank B occupying the shortened immerged inlet/outlet pipes performed better than case A.



Fig.12 Inlet-outlet locations effect in horizontal water storage tank [Assari et al., 2018](a); Piping and configuration of solar salt thermal storage tank A,B and C [Kim et al., 2019](b); Effect of inlet location via temperature contours[Yaïci et al.,2013](c)

2.4 Conclusion

This work provides a comprehensive literature review on thermocline thermal performance. Graphic manners, thermodynamic and hydrodynamic performance indicators were summarized. The indicator normalization is recommended for evaluation thermocline behavior and thermal performance. The gradient or grid formula gives more detail information about hydrodynamic performance during dynamic
operating duration. In order to reduce the impact of perturbated operating condition (e.g., uncontrollable and variable inlet high temperature in different tests of parametric study), those important indicators (e.g., η_{ch} in chapter 3, 4, and 5) were resolved by normalized temperatures. CFD technology, as the assistance tool, may make up with the disadvantage of invasive measurements and optic measurement while the analytic models for single-medium storage case still have wide development space. Previous optic measurement still sustains the local measurement of injection jet instead of at the complete thermocline evolution during charring/discharging period. In this thesis (chapter 5), optic method was used to be characterize time and space evolution of inlet thermal jet during charging period. Moreover, the mixing mechanism determined by the thermal advection-diffusion is worthy to be deeply explored for SMT tank. We discussed that relation of heat transfer via one modified Péclet number in chapter 5. The compromise of optimal flow rate is also dependent on the also operating conditions. The effect of operating parameters are various for storage systems without packing, with sensible /latent packing. The packing mode, pack-bed, represents its better robust character within the sensibility analysis compared to the other packing structure with high porosity. The published previous flow distributors can be divided into 3 types by their primary function: multi-branch, inversed flux and radical flux. In chapter 3, 4 and 5, the SMT system with the plate distributors (perforated, slotted) are numerically and experimentally tested. We supposed that plat distributor could concentrate the several advantages of those three types of flow distributors. For example, perforated plate distributor could divide the main thermal jet into multiple branch via orifices, produce inversed flux because of the solid part located in the plate.

Reference

Abdulla A, Reddy KS, 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. https://doi.org/10.1016/j.ijthermalsci.2017.07.004

Afshan M, Selvakumar A, 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. https://doi.org/10.1016/j.renene.2019.10.172

Afrin S, Kumar V, Bharathan D, Glatzmaier C, Ma Z, Computational analysis of a pipe flow distributor for a thermocline based thermal energy storage system. J Sol Energy Eng 2013;136(2):021010. https://doi.org/10.1115/1.4024927

Al-Azawii M, 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. https://doi.org/10.1016/j.applthermaleng.2020.115804

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. https://doi.org/10.1016/j.apenergy.2018.04.014

Andersen E, Furbo S, Fan J. Multilayer fabric stratification pipes for solar tanks. Sol Energy 2007;81:1219-26. https://doi.org/10.1016/j.solener.2007.01.008

árdenas B, Garvey S. A load-based approach for optimizing a packed-bedthermal store. J Energy Storage 2019;25:100835. https://doi.org/10.1016/j.est.2019.100835

Assari M, Basirat Tabrizi H, Savadkohy M, Numerical and experimental study of inletoutlet locations effect in horizontal storage tank of solar water heater. Sustain. Energy Techn 2018;25:181–190. https://doi.org/10.1016/j.seta.2017.12.009

Bai Y, Wang Z, Fan J, Yang M, Li X, Chen L, Yang J, Numerical and experimental study of an underground water pit for seasonal heat storage. Renew Energy 2020;150:487–508. https://doi.org/10.1016/j.renene.2019.12.080

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 Transfer 2013;60(1):713–721. https://doi.org/10.1016/j.ijheatmasstransfer.2013.01.047

Bhagat K, Saha S. Numerical analysis of latent heat thermal energy storage using encapsulated phase change material for solar thermal power plant. Renew Energy 2016;95:323–336. https://doi.org/10.1016/j.renene.2016.04.018

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. https://doi.org/10.1016/j.solener.2017.08.061

Bruch A, Molina S, Esence T, Fourmigué J, Couturier R. Experimental investigation of cycling behaviour of pilot-scale thermal oil packed-bed thermal storage system. Renew Energy 2017;103:277–285. https://doi.org/10.1016/j.renene.2016.11.029

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. Energ Convers Manage 2020;222:113189. https://doi.org/10.1016/j.enconman.2020.113189

Castell A, Medrano M, Solé C, Cabeza L. Dimensionless numbers used to characterize stratification in water tanks for discharging at low flow rates. Renew Energy 2010;35(10):2192–2199. https://doi.org/10.1016/j.renene.2010.03.020

Chandra Y, Matuska T. Numerical prediction of the stratification performance in domestic hot water storage tanks. Renew Energy 2020;154:1165–1179. https://doi.org/10.1016/j.renene.2020.03.090

Chung J, Cho S, Tae C, Yoo H. The effect of diffuser configuration on thermal stratification in a rectangular storage tank. Renew Energy 2008;33(10):2236–2245. https://doi.org/10.1016/j.renene.2007.12.013 Dahash A, Ochs F, Janetti M, 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. https://doi.org/10.1016/j.apenergy.2019.01.189

Dallaire J. Modélisation des changements de phase solide-liquide: effect de la variation de la densité avec la phase et du couplage thermomécanique. Université Laval, 2017.

Davenne T, Garvey S, Cardenas B, Rouse J. Stability of packed bed thermoclines. J Energy Storage 2018; 19:192–200. https://doi.org/10.1016/j.est.2018.07.015

Dehghan A, Barzegar A. Thermal performance behavior of a domestic hot water solar storage tank during consumption operation. Energ. Convers. Manage 2011;52(1):468–476. https://doi.org/10.1016/j.enconman.2010.06.075

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. https://doi.org/10.1016/j.solener.2017.03.008

Farmahini-Farahani M. Investigation of four geometrical parameters on thermal stratification of cold water tanks by exergy analysis. Int JExergy 2012;10(3):332–345. https://doi.org/10.1504/IJEX.2012.046814

El-Amin M, Al-Ghamdi A. Experiments and numerical simulation for a thermal vertical jet into a rectangular water tank. Results Phys 2018;10:680–692. https://doi.org/10.1016/j.rinp.2018.07.016

Fasquelle T, Falcoz Q, Neveu P, Hoffmann J. A temperature threshold evaluation for thermocline energy storage in concentrated solar power plants. Appl Energy 2018;212:1153–1164. https://doi.org/10.1016/j.apenergy.2017.12.105

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. https://doi.org/10.1016/j.solener.2017.12.062

Li G. Sensible heat thermal storage energy and exergy performance evaluations. Renew Sust Energ Rev 2016;53:897–923. https://doi.org/10.1016/j.rser.2015.09.006

García-Marí E, Gasque M, Gutiérrez-Colomer R, Ibáñez F, González-Altozan P. A new inlet device that enhances thermal stratification during charging in a hot water storage tank. Appl Therm Eng 2013;61(2):663–669. https://doi.org/10.1016/j.applthermaleng.2013.08.023

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. https://doi.org/10.1016/j.applthermaleng.2018.10.051

Kleinbach E, Beckman W, Klein S. Performance study of one dimensional models for stratified thermal storage tanks, Sol Energy, 1993;50(2):155–166. https://doi.org/10.1016/0038-092X(93)90087-5

Kaloudis E, Grigoriadis D, Papanicolaou E. Numerical simulations of constant-in flux gravity currents in con fi ned spaces : Application to thermal storage tanks. Int J Therm Sci 2016;108:1–16. https://doi.org/10.1016/j.ijthermalsci.2016.04.018

Kim D, Yoon S, Kim Y, Song C, Lee K, Choi J. Experimental studies of the discharge performance of single-medium TES for CSP applications. Appl Therm Eng 2017;127:499–507. https://doi.org/10.1016/j.applthermaleng.2017.08.057

Kim D, Yoon S , Kim Y, Lee K, Choi J. Experimental studies on the charging performance of single-tank single- medium thermal energy storage. Appl Therm Eng 2019;149:1098–1104. https://doi.org/10.1016/j.applthermaleng.2018.12.062

Kumar G, Nagarajan D, Chidambaram L, 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. https://doi.org/10.1016/j.energy.2016.09.014

Li S, Zhang Y, Li Y, Zhang X. Experimental study of inlet structure on the discharging performance of a solar water storage tank. Energ Buildings 2014;70:490–496. https://doi.org/10.1016/j.enbuild.2013.11.086

Li Z, Tao W. A new stability-guaranteed second-order difference scheme. Numer Heat Tran B-Fund 2002;42(4):349–365. https://doi.org/10.1080/10407790190053987

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. https://doi.org/10.1016/j.est.2020.101749

Ma Z, Li M, Zhang K, Yuan F. Novel designs of hybrid thermal energy storage system and operation strategies for concentrated solar power plant. Energy 2020;216:119281. https://doi.org/10.1016/j.energy.2020.119281

Majumdar R, Saha S. Effect of varying extent of PCM capsule filling on thermal stratification performance of a storage tank. Energy 2019;178:1–20. https://doi.org/10.1016/j.energy.2019.04.101

Nandi B, Bandyopadhyay S, Banerjee R. Numerical modeling and analysis of dual medium thermocline thermal energy storage. J. Energy Storage,2018;16:218–230. https://doi.org/10.1016/j.est.2018.01.020

Nelson J, Balakrishnan A, Srinivasa Murthy S. Experiments on stratified chilled-water tanks: Expériences menées avec des reservoirs d'accumulation d'eau glacée à stratification. Int J Refrig 1999;22(3):216–234. https://doi.org/10.1016/S0140-7007(98)00055-3

Nizami D, Lightstone M, Harrison S, Cruickshank C. Negative buoyant plume model for solar domestic hot water tank systems incorporating a vertical inlet. Sol Energy 2013;87:53–63. https://doi.org/10.1016/j.solener.2012.10.001

Paiva B, Siegle E, Schettini C. Channel curvature effects on estuarine circulation in a highly stratified tropical estuary: The São Francisco river estuary (Brazil). Estuar Coast Shelf S 2020;238:106723. https://doi.org/10.1016/j.ecss.2020.106723

Saha S, Das R. Exergetic and performance analyses of two-layered packed bed latent heat thermal energy storage system. Int J Energ Res 2020;44(3):2208–2225. https://doi.org/10.1002/er.5081

Shah L, Andersen E, Furbo S. Theoretical and experimental investigations of inlet stratifiers for solar storage tanks. Appl Therm Eng 2005;25(14–15):2086–2099. https://doi.org/10.1016/j.applthermaleng.2005.01.011 Shah L, Furbo S. Entrance effects in solar storage tanks. Sol Energy 2003;75(4):337–348. https://doi.org/10.1016/j.solener.2003.04.002

Shaikh W, Wadegaonkar, A Kedare, S. B Bose, M. Numerical simulation of single media thermocline based storage system. Sol Energy 2018;174:207–217. https://doi.org/10.1016/j.solener.2018.08.084

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(1):613–621. https://doi.org/10.2298/TSCI160720173S

Taher M, Bouchahm N, Guerira B, Bensaci C. On the thermal stratification inside a spherical water storage tank during dynamic mode. Appl Therm Eng 2019;159: 113821. https://doi.org/10.1016/j.applthermaleng.2019.113821

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 Fl 2013;44:403–413. https://doi.org/10.1016/j.ijheatfluidflow.2013.07.009

Wang S, Davidson J. Performance of a rigid porous-tube stratification manifold in comparison to an inlet pipe. Sol Energy. 2017;146:298-308. https://doi.org/10.1016/j.solener.2017.02.045

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. https://doi.org/10.1016/j.est.2019.101010

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. https://doi.org/10.1016/j.applthermaleng.2020.115010

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. https://doi.org/10.1016/j.apenergy.2013.08.044

Xu C, Wang Z, He Y, Li X, Bai F. Parametric study and standby behavior of a packedbed molten salt thermocline thermal storage system. Renew Energy 2012;48:1–9. https://doi.org/10.1016/j.renene.2012.04.017

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(1–2):152–163. https://doi.org/10.1016/j.applthermaleng.2013.07.001

Yang Z, Garimella S. Thermal analysis of solar thermal energy storage in a molten-saltthermocline.SolEnergy2010;84(6):974–985.https://doi.org/10.1016/j.solener.2010.03.007

Yee C, Lai F. Effects of a porous manifold on thermal stratification in a liquid storage tank. Sol Energy 2001;71(4):241–254. https://doi.org/10.1016/s0038-092x(01)00043-3

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. https://doi.org/10.1016/j.applthermaleng.2016.08.2 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(1):316–320. https://doi.org/10.1016/j.applthermaleng.2014.05.020

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. https://doi.org/10.1016/j.apenergy.2014.07.110

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. https://doi.org/10.1016/j.solener.2019.11.008

Chapter 3. Single-tank thermal energy storage systems for concentrated solar power: Flow distribution optimization for thermocline evolution management

Abstract

Concentrated Solar Power (CSP) technology captures solar radiation and converts it into heat for electricity production. It has received an increasing attention because integrated thermal energy storage (TES) systems can largely enhancing the reliability and the dispatchability. Over the last decade, low-cost single storage tank based on the thermocline technology becomes an alternative to commonly-used two-tank TES system. However, the improper inlet/outlet manifolds may cause the strong mixing of hot and cold fluids and disturb the temperature stratification, resulting in reduced thermal performances of the storage tank.

This study aims at solving the flow maldistribution problem in the single-tank thermocline storage system by appropriately structuring the inlet/outlet manifolds. The technical solution is based on the insertion of optimized perforated baffles in the manifolds. 2D Computational fluid dynamics simulations were performed to calculate the transient flow and temperature profiles in the storage tank during the charging and discharging operations. The optimal size distribution of orifices on the upper baffle has been determined for homogenizing passage times of the thermal front, so as to enhance the temperature stratification. A novel intermediate evaluation indicator was introduced to characterize the real-time thermal behavior, which could reduce the computational cost of the optimization problem by a factor of 6 at least. Numerical results shown that the proposed optimization algorithm could significantly improve the thermal performances, indicated by the increased values of charging/discharging efficiency, the capacity ratio and the overall efficiency, ex., the fully charging efficiency be increased by 29% by comparing the unstructured manifold geometry and the one with optimized baffles. The parametric study on certain geometry and operating factors also demonstrated that the flow distribution optimization method was robust, effective and efficient.

Keywords: Concentrated Solar Power (CSP); Thermal Energy Storage (TES); Temperature stratification; Thermocline evolution; Thermal front; Flow distribution

This chapter is published as

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. https://doi.org/10.1016/j.est.2020.101749

Nomenclature

Latin letters	
Ср	Heat capacity at constant pressure [J kg ⁻¹ K ⁻¹]
d	Size of orifice [m]
Ε	Energy [J]
е	Thickness [m]
Н	Height of thermocline tank [m]
h	Height of the middle flow zone [m]
L	Length of tank [m]
l	Size of upper/bottom manifolds [m]
'n	Mass flow rate [kg s ⁻¹]
Ν	Number of orifice [–]
Р	Position of horizontal cutting-section [-]
Re	Reynolds number [–]
S	Surface area [m ²]
<i>S</i> *	Dimensionless surface of thermocline area [-]
Т	Temperature [K]
T^*	Dimensionless temperature [-]
t	Time [s]
t^*	Dimensionless time [-]
$\overline{t^*}$	Dimensionless mean value of all passage times [-]
v	Velocity [m s ⁻¹]
v^*	Dimensionless velocity [-]
\bar{v}	Mean velocity [m s ⁻¹]
W	Port size [m]
Χ	X-coordinate [m]
X^*	Dimensionless X-coordinate [-]
Y	<i>Y</i> -coordinate [m]

Greek symbols

α	Proportion of orifice size [–]
γ	Relaxation factor [–]
Е	Tail factor [–]
η	Efficiency [–]
θ	Cone angle [°]
λ	Thermal conductivity [W m ⁻¹ K ⁻¹]
μ	Viscosity [kg m ^{-1} s ^{-1}]
ρ	Density [kg m ⁻³]
σ	Capacity ratio [–]
Φ	Global porosity of baffle [–]

Subscripts/superscripts

ch	Charging
dis	Discharging
f	End moment for fully charging/discharging
Fin	Intermediate stop moment
Н	High temperature
i	Orifice index
in	Inlet
ideal	Ideal case
j	Optimization step
L	Low temperature
max	Maximum
out	Outlet
overall	Overall charging-discharging process
port	Inlet/outlet port
stored	Stored
tank	Thermocline-based TES tank
tot	Total void fluid area
unit	Unit of baffle
x	Threshold coefficient [-]
0	Initial state

Abbreviations

CFD	Computational Fluid Dynamics
CSP	Concentrated solar power
GHG	Greenhouse gas
HTF	Heat transfer fluid
TES	Thermal energy storage
STD	Standard derivation

3.1 Introduction

Nowadays, the increasing energy consumption worldwide, the higher price of fossil fuels and the environmental impacts of greenhouse gas (GHG) emission stimulate the use of renewable resources as the alternative [Nicolas et al., 2019]. Solar energy conversion by Concentrated Solar Power (CSP) technology has a great potential within the future energy scenario because the integrated thermal energy storage (TES) systems can largely enhance the reliability and the dispatchability, allowing the production of electricity on demand [Ellingwood et al., 2020; Pelay et al., 2019; Zhang et al., 2016]. About half of the CSP plants currently in operation includes a TES system while this percentage rises up to more than 80% for those planned and under construction [Pelay et al., 2017]. Progress in TES technologies thus plays a key role for the massive deployment of CSP plants regarding their cost-effectiveness and competitiveness [Li, 2016; Ortiz et al., 2019].

In today's most common CSP plants, the TES system is typically based on the sensible heat storage (mostly molten salts) using a two-tank assembly in which hot and cold heat transfer fluids (HTFs) are located in two separate tanks [Alva et al., 2018]. Thermocline TES technology on the other hand, consists in using only one tank instead of two. Both hot and cold HTFs are stored within the same tank but naturally separated from each other by the buoyancy force, with the presence of a temperature stratification zone called the *thermocline*. Compared to conventional two-tank TES systems, the single-tank thermocline storage is a more cost competitive option (about 35% cheaper) [Grirate et al., 2016; Motte et al., 2015], i.e., the reduced amount of high-priced HTF by about 70% because of using cheap solid or industrial waste as energy storage material [Esence et al., 2017; Galione et al., 2015]. As a result, the interest for this technology is rapidly growing worldwide with a variety of new numerical and experimental developments being reported in the recent literature [Ahmed et al., 2019; Keilany et al., 2020; Pizzolato et al., 2017; Vigneshwaran et al., 2019].

The thermocline zone inside the storage tank is not motionless but evolves along with the dynamic operation of the TES system. During the charging (e.g., daytime), the hot HTF heated up in the solar field is introduced at the top of the tank and the thermocline zone moves downward. During the discharging (e.g., night), the cold HTF injected at the bottom of the tank pushes out the hot HTF (the stored heat) toward the power generation cycle and the thermocline zone moves upward. It is expected that the temperature stratification inside the storage tank should not be disturbed or degraded by the injected fluid flow during repeated charging-discharging cycles [Bruch et al., 2017; Zhao et al., 2017]. In real-world engineering however, the sudden expansion from the inlet port to the tank body may cause higher fluid velocity in the middle and lower velocity on the periphery of the tank. The flow non-uniformity, the local vortices and

recirculation will cause the strong mixing of hot and cold HTFs and degrade the temperature stratification, resulting in the lower overall efficiency and the reduced storage capacity of the storage tank [Li, 2016]. Therefore, how to characterize and control the dynamic thermocline behaviors when injecting the HTF during charging/discharging is actually one of the major challenges for the design and operation of the thermocline TES tank in CSP plants.

The vast majority of the earlier studies aim at achieving a uniform inlet velocity distribution or a horizontal flow front (velocity piston flow) leaving from the inlet port. For this reason, structured manifolds or fluid distributors are usually introduced for the inlet/outlet zones of the storage tank instead of simple injecting/outflowing ports. Afrin et al. [2013] proposed a pipe flow distributor with numerous sub-pipes to homogenize the fluid velocity distribution on the horizontal surface of a cylindrical TES tank. The effects of the number of sub-pipes, the locations/number of the holes on the sub-pipes were numerically studied. Kim et al. [2017] experimentally tested a multiport jet type inlet diffuser combined with a simple return outlet tube (without diffuser) in a singlemedium TES for CSP application. They found that the diffuser was actually required for both inlet and outlet to alleviate the temperature deviation. In practical, the installation of perforated baffles at upper and bottom parts of the storage tank has been proposed by many researchers to approach the radially homogeneous velocity distribution over the cross-sections [Ahmed et al., 2019; Geissbühler et al., 2016; Yin et al., 2014; Zavattoni et al., 2017]. The conical or circular manifold with uniform orifices baffle was the most commonly used inlet/outlet structure in both numerical and experimental investigation to make sure the fluid can be spread equally [De and Reuter, 2019; He et al., 2019; Nandi et al., 2018]. One exception is the work by Wang et al. [2015] which compared seven flow velocity profiles produced by different configurations of the inlet/outlet flow distributor. Their numerical results indicated that compared to uniform velocity profile, certain non-uniform flow profiles could lead to the reduced thermocline thickness and smaller entropy generation. This is actually in line with some observations reported in the literature [Kumar and Singh, 2019; Milman et al., 2012; Wei et al., 2015a] in that the optimal flow distribution is usually not uniform but obeys certain trend subjected to a defined optimization objective and constraints. However, controversy still exists on the relation between the optimal velocity profile and the expected thermocline evolution.

The full understanding of the complex transient transport phenomena (fluid flow, heat transfer, etc.) inside the single-medium thermocline storage tank is still lacking on the basis of above literature survey. In particular, the relation between the optimal velocity profile of HTFs produced by the inlet distributor and the minimum thermocline thickness is unclear. Furthermore, certain of the most widely used performance indicators like thermocline thickness, stratification number seem not to be totally adapted for characterizing real-world single-medium TES tanks. Moreover, investigations on effective solutions to maintain the temperature stratification during dynamic charging/discharging operations are needed.

The present study seeks to fill the research gap by appropriately structuring the inlet/outlet manifolds, using perforated baffles having optimized size distribution of orifices. Computational Fluid Dynamics (CFD) simulations were performed to characterize the transient fluid flow and heat transfer behaviors inside the storage tank. The main objectives and originalities of this paper are then threefold: (1) to reveal the fact that instead of uniform velocity profiles on the cross-sections, the flat thermal front (thermally piston flow) renders the highest level of temperature stratification in the storage tank; (2) to manage the HTF flow distribution by using optimized orifice baffles, so as to enhance the thermal performance of the thermocline storage tank; and (3) to propose and develop a heuristic algorithm coupled with a novel optimality criterion that can effectively determine the optimal size distribution of the orifices on the baffle.

It should be noted that the flow distribution management strategy coupled with novel optimized baffled fluid distributor/collector has been proposed and proven to be really effective in our earlier studies [Luo et al., 2015; Wei et al., 2015b; Wei et al., 2016; Wei et al., 2017]. But in the current study, this method is for the first time applied to a more complicated situation (TES tank) to tackle the dynamic thermocline behavior optimization problem. The contributions of this paper are important because it will expand the limited literature and provide additional insights to issues involving the flow distribution optimization and thermocline evolution management in the storage tank. The results obtained may be used for the large deployment of the thermocline TES technology in future CSP plants.

The rest of the paper is organized as follows. The novel optimality criterion and the basic principles of the heuristic algorithm are described in detail in section 2. A 2D numerical benchmark is then introduced and optimization results are presented and compared in section 3. The influences of various design and operational parameters are analyzed and discussed in section 4. Finally in section 5, main findings and perspectives are summarized.

3.2 Methodology for modeling and optimization

This section presents the basic principles of the CFD-based heuristic algorithm for optimizing the thermal performance of the thermocline storage tank. The following assumptions and simplifications have been made for this study.

- Incompressible Newtonian fluid;

- 2D flow domain;
- No solid filler in the storage tank;
- The storage tank is well-insulated, i.e., negligible heat loss to the environment;

- Constant injecting HTF temperatures for charging (T_H) and for discharging (T_L) , respectively;

- Homogeneous temperature profile in the storage tank at the initial-state (t_0) .

3.2.1 Parameter definition

Figure 1 shows the schematic view of the thermocline storage tank used as a representative model for the current study. It is a simplified 2D domain of H in height and L in length, having single inlet port and outlet port in the center. The void volume for the fluid is noted as S_{tot} . Two perforated baffles are installed at the upper and bottom parts of the storage tank, dividing the whole domain into three consecutive zones: the upper manifold from the upper port to the upper baffle; the middle flow zone and the bottom manifold from the bottom baffle to the bottom port. The height of middle flow zone is h, with three cutting-sections (P1, P2, P3) marked for monitoring.

The width of the perforated baffles equals to the width of the storage tank and their thickness is *e*. The upper baffle is divided into *N* control units of identical length (L_{unit}) , indexed by *i* from 1 to *N* from the left side to the right side. For each control unit, there is one orifice of width d_i arranged in the middle, as shown in Fig. 1. Note that α_i is defined as the local porosity of *i*th control unit and Φ the global porosity of the baffle. Assuming that every orifice can't exceed the boundary of its control unit $(d_i < L_{unit})$, the following geometric relations can be written:

$$L_{unit} = \frac{L}{N} \tag{1}$$

$$\alpha_i = \frac{d_i}{L_{unit}} < 1 \tag{2}$$

$$\Phi = \frac{\sum d_i}{L} = \frac{\sum \alpha_i}{N} \tag{3}$$



Figure 1. Schematic view of a 2D thermocline storage tank equipped with upper and bottom orifice baffles

For the charging process, the storage tank is supposed to be fully filled with the cold HTF at homogeneous temperature T_L at the initial state t_0 . Once the charging begins, the hot HTF at constant temperature (T_H) is injected into the upper port at constant flow-rate \dot{m}_H . It flows from top to bottom, passing consecutively through the upper manifold zone, the middle flow zone and the bottom manifold zone. At the same time, the cold HTF is forced out from the bottom port so that the thermal energy is stored in the tank. The discharging process is actually the reverse operation in which the cold HTF $(T_L; \dot{m}_L)$ is injected into the bottom port and flows from the bottom to the top to release the stored thermal energy.

We define T_x as a temperature value within the range of T_L to T_H , calculated by Eq. (4).

$$T_x = T_L + x\% \times (T_H - T_L) \tag{4}$$

The threshold coefficient at x=20 is used to define the temperature borders or fronts $[T_{20}, T_{80}]$ that enclose the thermocline region in the storage tank. The evolution of the hot front T_{80} and the cold front T_{20} during the charging or discharging operations determines then the shape of the thermocline zone as well as the level of temperature stratification. We define S(t) as the surface area enclosed by $[T_{20}, T_{80}]$ at time t. Then the dimensionless surface area of the thermocline zone $S^*(t)$ can be obtained by dividing S(t) by the total void fluid domain (S_{tot}) , shown in Eq.(5):

$$S^*(t) = \frac{S(t)}{S_{tot}} \tag{5}$$

Compared to multi-medium TES tank, the thermocline shape of single-medium TES tank is normally more complex due to the lack of velocity/temperature

homogenization by sensible/latent fillers. Therefore, thermocline surface area, rather than the thermocline thickness, is used here as the indicator.

3.2.2 Performance indicators

Some global performance indicators commonly used in the literature [Zanganeh et al., 2015] are introduced to indicate the thermal performance of the storage tank, including the charging/discharging efficiency (η), the overall efficiency ($\eta_{overall}$) and the capacity ratio (σ). Their definitions are shown in Eqs. (6-9).

$$\eta_{ch} = \frac{E_{in} - E_{out}}{E_{in}} = \frac{\int_{t_0}^{t_f} \dot{m} \, c_p \, (T_{in} - T_{out}) dt}{\int_{t_0}^{t_f} \dot{m} \, c_p \, (T_{in} - T_0) dt} \tag{6}$$

$$\eta_{dis} = \frac{E_{out} - E_{in}}{E_{stored}} = \frac{\int_{t_0}^{t_f} \dot{m} c_p (T_{out} - T_{in}) dt}{E_{stored}}$$
(7)

$$\eta_{overall} = \eta_{ch} \eta_{dis} \tag{8}$$

Where T_{in} and T_{out} are the inlet and outlet temperature of the HTF, respectively. *Cp* is the temperature-dependent specific heat of the HTF and \dot{m} is the mass flow-rate. t_f is the end time for fully charging/discharging process ($T_{out} = T_{99}$ for charging and $T_{out} = T_{01}$ for discharging).

Capacity ratio
$$\sigma = \frac{E_{stored}}{E_{store}^{max}} = \frac{E_{in} - E_{out}}{S_{tot}(Cp_H T_H \rho_H - Cp_L T_L \rho_L)}$$
 (9)

The capacity ratio (σ) indicates the ratio of real stored thermal energy to the maximum energy storage capacity of the tank. As a result, σ equals to 1 for fully charging/discharging operations but is smaller than 1 for partial charging/discharging. In this study, the cut-off outlet temperature is set as T_{20} for the charging process.

For a CSP plant with integrated TES system, the outlet HTF temperature from its attached TES system seems to be one of the most important parameters that should be closely monitored and controlled during dynamic operations. During charging, the outlet HTF from the TES system toward the solar field should not be overheated so as to avoid the inconvenience of control operations such as the solar field defocusing or the HTF mass flow variation [Fasquelle et al., 2018a; Sanz et al., 2019]. During discharging, the subcooling of the outflowing HTF from the storage tank to the power generation cycle should also be prevented [Modi and Pérez-Segarra, 2014]. Particularly for the thermocline storage tank, an appropriate cut-off temperature has to be defined to prevent the overheating/subcooling on one hand, and to guarantee a high storage capacity on the other hand. Several methods about the selection of the cut-off temperature are proposed [Modi and Pérez-Segarra, 2014] and the threshold values $[T_{20}, T_{80}]$ are commonly used [Fasquelle et al., 2018b]. In real-world engineering however, a high-steep slope for the T_{out} curve is strongly expected to have a more flexible range of the cut-off temperature as well as a higher capacity ratio.

In order to unfold the information on the slop of the T_{out} curve that cannot be indicated by the efficiencies or the capacity ratio, a new performance indicator, namely the tail factor (ε), is proposed and introduced here. It is defined as the slope of the dimensionless HTF outlet temperature (T_{out}^*) and the dimensionless time, as shown and explained in Eqs. (10; 11) and Fig. 2.



Figure 2. Definition of tail factor $\mathcal E$ for the charging (a) and the discharging (b) in a thermocline storage tank

$$\varepsilon_{ch} = \frac{T_{99}^* - T_{20}^*}{t_{799}^* - t_{720}^*} \tag{10}$$

$$\varepsilon_{dis} = \frac{T_{80}^* - T_{01}^*}{t_{T_{01}^*}^* - t_{T_{80}^*}^*} \tag{11}$$

where the dimensionless temperature T_x^* is calculated by Eq. (12):

$$T_{\chi}^* = \frac{T_{\chi}}{(T_H - T_L)} \tag{12}$$

 $t_{T_x}^*$ is the dimensionless time calculated by Eq. (13), where t_{T_x} indicates the moment when the HTF outlet temperature reaches T_x .

$$t_{T_{X}}^{*} = \frac{t_{T_{X}}}{t_{ideal}}$$
(13)

The denominator t_{ideal} in Eq. (13) is the overall passage time for an ideal storage tank with perfect temperature stratification (thermally piston flow without heat exchange), calculated by Eq. (14).

$$t_{ideal} = \frac{S_{tot}}{v_{in}W_{in}} \tag{14}$$

where v_{in} is the velocity of the inlet HTF.

The higher value of ε means that the shape of T_{out}^* curve as a function of time is closer to a step line, implying that the thermocline storage tank is operated close to the ideal condition. The definition of tail factor excludes the influences of the used HTF type and their thermo-physical properties (e.g., Cp, ρ), thereby more direct and more convenient to calculate. By comparing the ε value of real charging/discharging process with that of ideal case ($\varepsilon = \infty$), the level of temperature stratification can then be quantified.

One may also notice that to calculate these performance indicators, the fully charging or discharging process should be simulated. The employment of an intermediate parameter having certain connections to these global indicators may be beneficial to ease the calculation burden.

3.2.3 Heuristic optimality criterion

Most of the studies in the literature aim at achieving uniform velocity profile on the horizontal surface of the thermocline storage tank. Here we propose that a *thermally piston flow with flat thermal front* could be more beneficial. However, this is difficult to achieve even at the presence of the perforated baffle, as explained in Fig. 3 for a typical charging process.



Figure 3. Schematic view on the evolution of the thermal front during the charging process

At t_0 time state, the storage tank is full of the cold HTF at homogeneous temperature T_L . Once the charging begins, the hot HTF at T_H will be injected into the upper port, getting in contact the cold HTF in the upper manifold. A temperature transition zone will then occur due to the heat transfer between the hot and cold HTFs. The hot front (T_{80}) will firstly pass through the middle orifices (i=N/2; N/2+1) facing to the upper port and then extend towards the orifices situated on the side (e.g., i=1; N). Generally for orifice *i*, its mean fluid temperature passing through ($T_i(t)$) increases over time from T_L to T_{20} then to T_{80} , and finally reaches T_H . We define the passage time t_i as the time when the hot front (T_{80}) passes through the orifice *i*:

$$T_i(t_i) = T_{80} (15)$$

Evidently for different orifices, their passage times are not identical. Middle orifices have shorter passages time compared to those on the side, due to the shorter

distance to the upper port and relatively higher velocity magnitude. The dimensionless mean passage time for all the orifices can be calculated by Eq. (16).

$$\overline{t^*} = \frac{\sum_{i=1}^N t_i^*}{N} = \frac{\sum_{i=1}^N t_i}{N t_{ideal}}$$
(16)

We define t_{Fin}^* as the moment when the hot front T_{80} totally passes through the upper manifold, the corresponding dimensionless thermocline surface area being $S^*(t_{Fin}^*)$, as shown in Fig. 3. $S^*(t_{Fin}^*)$ is actually an intermediate value that will still evolve along with the subsequent charging process. The relation between this intermediate indicator and some global performance indicators will be discussed later.

The basic idea of the optimization method is to adjust the size distribution of perforated orifices on the baffle so that the thermal front T_{80} could pass through every orifice at almost the same time. This thermally piston flow can consequently minimize the S^* value at t_{Fin}^* moment so that the temperature stratification in the storage tank can be maintained. Here we propose a heuristic optimality criterion, suggesting that the thermally piston flow could be realized when the passage times (t_i^*) for orifices are identical (Eq. 17):

$$t_i^* = \overline{t^*} (i=1, 2... N)$$
 (17)

With this optimality criterion in hand, an original CFD-based optimization algorithm has been developed.

3.2.4 Basic principle of the optimization algorithm

The optimization algorithm is developed to determine the optimal size distribution of orifices on the perforated baffle so as to homogenize the passages times of the thermal front. As explained in Fig. 4, the local porosity of *i*th orifice (α_i) is varied simultaneously by comparing its passage time (t_i^*) and the mean value of all passage times ($\overline{t^*}$). Specifically for *j* optimization step, if $t_{i,j}^*$ is greater than $\overline{t_j^*}$, the size of orifice *i* will be enlarged to get a relatively smaller passage time $t_{i,j+1}^*$ at step *j*+1. Otherwise, if $t_{i,j+1}^*$ is lower than $\overline{t_j^*}$, the value of of $\alpha_{i,j+1}$ will be reduced to lengthen the passage time $t_{i,j+1}^*$ of the thermal front. This variation rule from *j* to *j*+1 step for *i*th orifice is written in Eq. (18).

$$\Delta \alpha_{i,j} = \alpha_{i,j+1} - \alpha_{i,j} = \gamma \left(t_{i,j}^* - \overline{t_j^*} \right) \tag{18}$$

Where γ is a relaxation factor chosen to ensure a fast and stable convergence. Following this variation rule, one may easily verify that although the local porosity α_i varies from one iteration to the next, the global porosity of the baffle Φ is kept constant (Eq. 19).

$$\sum_{i=1}^{N} \Delta \alpha_{i,j} = \sum_{i=1}^{N} (\alpha_{i,j+1} - \alpha_{i,j}) = \sum_{i=1}^{N} \alpha_{i,j+1} - \sum_{i=1}^{N} \alpha_{i,j} = \Phi_{j+1} - \Phi_j = 0$$
(19)



Figure 4. Basic principle of the optimization algorithm for enlarging or reducing the size of an orifice

The degree of closeness between the optimized results and the optimality criterion is quantified by the standard deviation of the dimensionless passages times $(STD_{t_i}^*)$ defined as follows:

$$\text{STD}_{t_i^*} = \sqrt{\frac{1}{N-1} \sum_{i=1}^{N} (\frac{t_i^* - \overline{t^*}}{\overline{t^*}})^2}$$
(20)

3.2.5 Numerical implementation

An automatic program is developed to realize the iterative procedure of the heuristic algorithm. Its major steps are schematized in the flow chart of Fig. 5 and are described in detail below.

(1) Input the geometry of the storage tank (height, length, cone angle, etc.) and general operational conditions (used HTF, their physical properties, initial temperature state, operating temperatures, inlet flow-rate, etc.).

(2) Definition of the parameters relative to the upper & bottom perforated baffles, such as the location and thickness, the orifice number N and the global porosity Φ , the size distribution of orifices.

(3) Mesh generation of the whole simulation domain.

(4) CFD simulation of the transient flow and temperature profiles in the storage tank from t_0 to t_{Fin}^* .

(5) Calculation of the dimensionless thermocline surface area $S^*(t_{Fin}^*)$ (Eq. (5)); calculation of the passage time t_i^* for each orifice. Modification of the baffle configuration by comparing t_i^* and $\overline{t^*}$ according to the variation rule in Eq. (18).

(6) Regeneration of the mesh for the updated simulation domain; recalculation of the transient flow and temperature profiles in the storage tank from t_0 to t^*_{Fin} .

(7) Check the stable tolerance of the algorithm. If the tolerance is satisfied, then the iteration procedure is terminated. If not so, the procedure goes back to step 2 for recurrence. The result is considered to be stable when the variation of $\Delta S^*(t_{Fin}^*)$ is smaller than 1% from *j* to *j*+1 iteration step (Eq. 21).

$$\Delta S^*(t_{Fin}^*) = \frac{S_{j+1}^* - S_j^*}{\frac{1}{2} \left(S_{j+1}^* + S_j^*\right)} < 0.01$$
(21)

(8) Continue the full charging (or discharging) process until the T_{out}^* reaches T_{99}^* (or T_{01}^*). Export the final baffle configurations and the values of the global performance indicators (η , σ , ε).



Figure 5. Flow chart of the optimization algorithm

3.3 Simulation and results

In this section, the simulation parameters and results for a storage tank are presented to showcase the feasibility and the effectiveness of the proposed optimization algorithm.

3.3.1 Geometries and simulation parameters

A 2D simulation domain as shown in Fig. 1 has been created as the benchmark case to represent the storage tank. Its main dimensions are listed in Table 1. An upper and a bottom baffle were installed in the storage tank to regulate the thermal front

behaviors. Their dimensions are also listed in Table 1. There are 10 orifices on each baffle (N=10), the global porosity (Φ) being 0.3. Identical size of orifices ($\alpha_{i,0} = \Phi, i = 1, 2, ..., N$) was set for both baffles at the starting point (j=0). Recall that for the charging process, the size distribution of orifices on the upper baffle will evolve according to the optimization algorithm while that on the bottom baffle remains unchanged, and vice versa for the discharging process. A reference case with empty upper baffle for the charging is also included for comparison, noted hereafter as step j=-1. The influence of the upper & bottom baffle combinations on the thermocline behaviors will be discussed later.

	Table 1	. Main	geomet	ric dim	ensions fo	r the be	nchmark	storag	ge tank	(unit: m	ım)
Parameter	Н	h	L	W	N (-)	е	<i>e</i> _{port}	l_1	l_2	l_3	l_4
Value	200	140	200	10	10	5	3	20	35	60	5

The working HTF used is the solar salt. Its thermophysical properties using polynomial fitting correlations are listed in Table 2.

	rable.2. Thermo-physical properties of the fiff [fiormann et al., 2010]									
Material	Property	Unit	Fitting correlation $T(^{\circ}C)$							
Solar salt	Density ρ	kg m ⁻³	1818.5 (700 K)							
(600 K<7<800 K)	Specific heat Cp	J kg ⁻¹ K ⁻¹	1443 + 0.172T							
	Thermal conductivity λ	$W m^{-1}K^{-1}$	0.443 + 0.00019T							
	Dynamic viscosity μ	$kg \ m^{-1} \ s^{-1}$	$(22.174 - 0.12T + 2.281 \times 10^{-4} T^2)$							
			$-1.474 \times 10^{-7} T^3)/1000$							

Table.2. Thermo-physical properties of the HTF [Hoffmann et al., 2016]

Constant velocity normal to the inlet port surface was set as the inlet boundary condition. The outlet boundary condition was set as pressure-outlet with zero static pressure. All the walls for the storage tank were defined as non-slip and adiabatic. The operational pressure was fixed at 101 325 Pa. For the charging process, the fluid zone was initialized as T_L =600 K (fully occupied by the cold HTF). Hot HTF (T_H =800 K) was injected into the upper port at 0.00288 m·s⁻¹. The Reynolds number in the tank equals to 50, based on the mean temperature of hot and cold HTF (700 K). Vice versa for the discharging process, the fluid zone was initialized as T_H =800 K and cold HTF (T_L =600 K) was injected into the bottom port.

Fluid flow and temperature profiles in the storage tank were calculated by using Ansys FLUENT code (version 17.2). Finite-volume method with collocated grid at vertex was applied for numerical discretization. Second-order upwind differential scheme was applied for momentum and standard method for pressure. SIMPLE method was used for the pressure–velocity coupling in order to correct the pressure field oscillations [Rodrigues and Lemos, 2020]. Simulations were performed under transient state with fixed time step of 0.2 s. The solution was considered to be converged when (i) normalized residuals were smaller than 10^{-5} for mass and momentum equations and 10^{-7} for energy equation, and (ii) the inlet static pressure become constant (less than 0.5% variation).

For each iteration step of the optimization algorithm, MATLAB R2016b was used for data post-processing of the computed flow and temperature profiles from FLUENT, to calculate the size variation of each orifice according to Eq. (18) and to pass the renewed geometric coordinates to Ansys Workbench for a new CFD simulation.

The structured model mesh was refined to raise the calculation accuracy and to speed up the convergence. A grid independence test was performed with the increased number of total elements from 30 k to 400 k. It can be observed in Fig. 6 that the difference in $\Delta S^*(t_{Fin}^*)$ is less than 1% when the number of elements is higher than 200 k. Therefore, the mesh with 200 k elements has been used in the following study for a compromise between accuracy and computational time.



Figure 6. Grid independence test based on the dimensionless thermocline surface area $S^*(t_{Fin}^*)$

3.3.2 Optimization of the upper baffle for charging process

Figure 7 shows the evolution of the dimensionless thermocline surface area (S^*) and the standard deviation of passages times ($STD_{t_i^*}$) as a function of the optimization step. It can be seen that the values of $STD_{t_i^*}$ are 0.598 for the no upper baffle case (j=-1) and 0.224 for the uniform orifice baffle case (j=0), indicating non-identical passage times. When the optimization algorithm proceeds from one step to the next, both the values of $STD_{t_i^*}$ and S^* decrease rapidly, indicating the coherence of the proposed optimality criterion. Finally convergence ($\Delta S^*(t_{Fin}^*) < 0.01$) can be reached after 6 optimization steps, showing that the CFD-based optimization algorithm is effective and fast approaches the optimum.



Figure 7. Dimensionless thermocline surface *S*^{*} and standard deviation of passages times as a function of the optimization step

Figure 8 gives a comparison on the evolution of temperature profile in the storage tank from t_0 to t_{Fin}^* for 3 cases (without upper baffle; uniform orifice upper baffle; optimized upper baffle). When no baffle is installed in the upper manifold (*j*=-1), hot HTF injected in the center upper port causes higher fluid velocity in the middle and lower velocity on the periphery of the storage tank (also shown by case 1 in Fig. 10). The thermal front expands radially and the temperature stratification is totally disturbed due to the strong mixing of hot and cold HTFs. At t_{Fin}^* moment, the temperature of HTF at the bottom port (T_{out}^*) has already reached T_{80}^* . The temperature transitional zone (the thermocline) propagates to the whole height of the storage tank, implying low thermal performance and reduced storage capacity.

The insertion of a uniform orifice upper baffle (j=0) serves as an additional flow resistance to block the downward jet flow of hot HTF, leading to the alleviated velocity peak in the middle of the tank (case 2 in Fig. 10). The thermocline zone propagates slower in the vertical (Y) direction but instead extends toward the lateral (X) direction in the upper manifold. Nevertheless, the thermal front still shows a parabolic shape which is far from the expected thermally piston flow. At t_{Fin}^* moment when the hot front T_{80}^* totally passes cross the upper manifold, the cold front T_{20}^* almost touches the bottom baffle.



Figure 8. Comparison on the evolution of the temperature profiles in the storage tank from t_0 to t^*_{Fin} during the charging process

By running the optimization algorithm, the sizes of middle orifices become smaller while those near the walls are enlarged. Consequently, the injected hot HTF is further guided toward the lateral walls in the upper manifold (case 3 in Fig. 10), so does the thermocline zone. The hot HTF tends to occupy the whole upper manifold before crossing through the optimized upper baffle. Almost flat hot and cold fronts can be observed at $\frac{1}{2} t_{Fin}^*$ on Fig. 8, indicating that quasi-thermally piston flow has been achieved. After that, the thermocline zone still evolves and exhibits a double-hump shape at the t_{Fin}^* moment, due to the relatively higher velocities of the hot HTF near the side walls during the whole charging process. This characteristic of negative parabolic shaped velocity profile is also clearly shown at the case 3 in Fig. 10 . Nevertheless, the surface area of the thermocline zone is the smallest after optimization, indicated by the smallest S^* value (0.012) compared to the without baffle case ($S^*=0.32$) or uniform orifice baffle case ($S^*=0.20$). Figure 9 shows the T_{out}^* curves as a function of dimensionless time t^* for different cases. It can be observed that the curve for the non-upper baffle case has a long tail (ε =0.137), indicating that it takes very long time for T_{out}^* to reach T_{99}^* . On the contrary, the slope of curve for the optimized upper baffle case is steeper (ε =0.607), implying smaller mixing effect of hot and cold HTFs in the storage tank. Also indicated on Fig. 9 are values of some intermediate or global indicators introduced to evaluate the performance of the thermocline storage tank. Note that in addition to the three tested cases, the values for the ideal condition are also listed for comparison. Among the three tested cases, the optimized upper baffle case always presents the best values, showing the validity of the proposed optimality criterion and the effectiveness of the developed optimization algorithm.



Figure 9. Comparison on different performance indicators for the tested cases

Another remark is that the intermediate parameter is in close connection to the global performance indicators: minimizing the $S^*(t_{Fin}^*)$ leads to maximize the values of tail factor (ε), fully charging efficiency (η) and the capacity ratio (σ). Instead of simulating the full charging (discharging) process, each iteration can be stopped at t_{Fin}^* time. The whole calculation time can then be reduced by a factor of 6 at least, without influencing the optimization results.

3.3.3 Combination of upper & bottom baffles for cyclic operation (charging & discharging processes)

Note that the upper baffle has been optimized for the charging process when uniform orifice baffle is always used for the bottom manifold. However, the bottom baffle is also a matter of concern regarding the cyclic operation when the discharging process is included in particular. To complete the study, the bottom baffle is also optimized by running the algorithm during the discharging process when the optimized upper baffle is installed, as shown by Case 4 in Fig. 10. One may observe that the optimal orifice size distributions of bottom baffle is slightly differed from that of the upper baffle, mainly due to the impact of the gravity effect. However the global trend is the same: larger orifices close to the lateral walls whereas smaller orifices in the middle facing the upper/bottom ports.

The charging, discharging and overall efficiencies for different combinations of upper & bottom baffles are presented in Table 3. Not surprisingly, the combination of optimized upper & bottom baffles provides the best performances. One interesting finding by comparing case 3 (optimized upper baffle & uniform orifice bottom baffle) and case 4 (optimized upper & bottom baffles) is that the installation of optimized bottom baffle favors both the charging and the discharging efficiencies. This may be explained by the optimum velocity profile achieved by case 4, i.e., negative parabolic shaped velocity profiles can be guaranteed for all three horizontal cutting surfaces (*P1*, *P2* and *P3* indicated in Fig. 10) by using the combination of optimized upper & bottom baffles. Higher velocity near to the lateral walls would aide to overcome the frictional forces near to side wall, rendering flatter progress of thermal fronts inside the storage tank.



Figure 10. Terminal velocity profiles in the storage tank and at different height positions for different combination of upper & bottom baffles cases (charging process)

	bottom baffles										
	Case 1	Case 2	Case 3	Case 4							
	-Without upper baffle	-Uniform upper baffle	-Optimized upper baffle	-Optimized upper baffle							
	-Uniform bottom baffle	-Uniform bottom baffle	-Uniform bottom baffle	-Optimized bottom baffle							
Charging	0.17	0.41	0.48	0.54							
Discharging	0.25	0.42	0.50	0.55							
Overall	0.04	0.17	0.24	0.30							

By comparing case 2 and case 4 in Fig. 10, another important conclusion is that *uniform velocity profile at horizontal surfaces doesn't necessarily lead to better thermocline performances*. The results of this study support the finding of Wang et al. [2015] in that higher lateral velocities favor the temperature stratification inside the

storage tank. But we go one step further because the optimal velocity profiles are deduced by using the proposed optimization algorithm, rather than empirically imposed.

3.4 Effect of design parameters

In this section, the effects of design and operational parameters on the performances of the thermocline storage tank will be systematically evaluated, so as to test the robustness and effectiveness of the proposed optimization algorithm. These parameters include the Reynolds number (*Re*), the aspect ratio (*H/L*), the cone angle (θ) , the number of orifices (*N*) and the global porosity of the baffle (Φ). While one design variable is tested, other parameters are kept the same as those used for the benchmark study in section 3. Note that for simplification reason, only the upper baffle has been optimized during the charging process when uniform orifice bottom baffle is always used for this parametric study.

3.4.1 Reynolds number Re

Controversy exists in the literature regarding the effect of Re (or injected HTF flow-rate) on the temperature stratification inside the storage tank. Some researchers [Bayón and Rojas, 2013] reported that an increasing inlet velocity could lead to an increased discharging efficiency, a thinner thermocline region and a shorter discharging time. Others observed the opposite performance that a higher Re would reduce the effective discharging efficiency and time [Wu et al., 2014]. The quantitative relation between thermal performance and Re was still undetermined.

In this study, *Re* numbers based on the width of the storage tank L (Eq. 21) from 10 to 200 have been tested, covering the most used range for thermocline storage tanks in CSP application [Rodrigues and Lemos, 2020].

$$Re_{tank} = \frac{\rho \bar{v}_{tank} L}{\mu} \tag{21}$$

Note that the mean temperature between the hot and cold HTFs (700 K) is used to calculate the ρ and μ values. The temperature profiles inside the storage tank at t_{Fin}^* time at 4 different Re_{tank} numbers (10; 50; 100; 200) are shown in Fig. 11. With the increasing value of Re_{tank} , the thermocline zone approaches more the bottom port because of the stronger inertial force. The thermal fronts also become less flat, confirming that higher Re_{tank} disturbs the temperature stratification. When uniform orifice upper baffle is used, the thermocline region for $Re_{tank}=100$ or 200 has even reached the bottom port at t_{Fin}^* . A decreased trend of charging efficiency versus the increasing Re was observed, as shown in Table 4. By examining the values of performance indicators before and after optimization for each tested Re_{tank} (Table 4), one can clearly observe that the S^{*} value could be largely reduced while the values of ε , η and σ significantly augmented. When *Re* is very small, the time for heat transfer between the hot and cold HTFs is longer, resulting in larger S^* value and small σ value. In this regard, the conduction may play an effective role of the heat transfer. But the higher the Re_{tank} , the double-hump shape of the thermal fronts at t_{Fin}^* becomes more visible, as shown in Fig. 11. It is more difficult to flatten out the irregular thermal fronts due to the stronger inertial force which is typically in response to convective fluid movement. Therefore, there may be a way to find a critical *Re* in range of 50 to 100 to get a maximum σ by coordinate the dominant mechanism of heat transfer. Furthermore, Fig.12 represents the various final size distribution of orifices after optimization under different Retank where higher Reynold number occupies relatively smaller opening area near the centerline of storage tank and larger opening area near the lateral walls. That trend also is considered to be logical that the forced convection of high Reynolds number via upper inlet port located at centerline the should be restricted.



Figure 11. Effect of Re_{tank} on the temperature profile at t_{Fin}^* time (Re_{tank} =10; 50; 100; 200)



Figure 12. Effect of Re_{tank} on the optimized size distribution of orifices at t^*_{Fin} time ($Re_{tank}=10$; 50; 100; 200)

<i>Re_{tank}</i>	τ	Jniform o	rifice up	per baff	fle	Optimized orifice upper baffle					
	t_{Fin}^*	<i>S</i> *	3	σ	η	t^*_{Fin}	<i>S</i> *	3	σ	η	
10	0.47	0.31	0.53	0.65	0.46	0.28	0.20	0.62	0.73	0.50	
50	0.56	0.20	0.43	0.70	0.41	0.31	0.14	0.61	0.81	0.48	
100	0.72	>0.19	0.37	0.71	0.39	0.41	0.14	0.53	0.83	0.44	
200	0.98	>0.13	0.27	0.68	0.31	0.54	0.15	0.45	0.84	0.41	

Table 4. Effect of Retank on the performance of the thermocline storage tank

3.4.2 Aspect ratio H/L

The aspect ratio (H/L) is also an important design parameter to consider. Three values of aspect ratio (H/L=0.5; 1 and 2) have been tested and results are shown in Fig. 12 and Table 5.



Fig.13. Effect of aspect ratio on the temperature profile at t_{Fin}^* (*H/L*=0.5; 1 and 2)

Before optimization of the upper baffle, the thermocline zone shows a regular parabola shape. For small aspect ratio (H/L=0.5), the height of the storage tank is not sufficient for the thermocline zone to develop. It spreads easily to the bottom manifold at t_{Fin}^* time and influences greatly the T_{out}^* , indicated by the small value of ε (0.22) in Table 5. In this sense, a slender shape storage tank would be more advantageous for the propagation of the thermal fronts, as also reported by Mertens et al. [2014].

After optimizing the upper baffle, the surface area of thermocline zone (S^*) for each H/L value can be reduced to a great extent. The thermocline zones for all the tested cases stay above the bottom manifold at t_{Fin}^* time, without interference to the T_{out}^* .

Again the thermal performance of the charging process can be improved, indicated by the augmented values of various performance indicators listed in Table 5.

	Table	5. Effect	or aspece	Tauo on	the peri	ormanee	of the th	ici mocm	ne stora	ge tank
пла	ι	Jniform c	orifice up	oper baff	Optimized orifice upper baffle					
H/L	t_{Fin}^{*}	<i>S</i> *	3	σ	η	t_{Fin}^{*}	<i>S</i> *	3	σ	η
0.5	1.7	>0.23	0.22	0.30	0.22	0.45	0.22	0.33	0.70	0.34
1	0.56	0.20	0.43	0.70	0.41	0.31	0.14	0.61	0.81	0.48
2	0.43	0.14	0.45	0.80	0.44	0.28	0.10	0.53	0.82	0.45

Table 5. Effect of aspect ratio on the performance of the thermocline storage tank

Regarding the ε and η , better results are achieved for H/L=1 than H/L=2. This is because the optimization algorithm aims at minimizing S^* the at t_{Fin}^* time state while the thermocline area will further evolve due to the incomplete fully charging. This propagation of the thermocline zone at the lower half of the middle fluid zone is less controlled by the optimized upper baffle. The higher level of mixing between the hot and cold HTFs results a higher heat exchange, leading to lower values of the global performance indicators for H/L=2.

3.4.3 Cone angle θ

Cone angle is a relatively new design parameter for thermocline storage tank when considering the truncated conical tank buried partially or totally underground so that the lateral earth would prevent the thermal deformation of the tank body during the cyclic operations. The upper diameter of truncated conical tank is expected to be larger than the bottom diameter (positive cone angle as shown in Fig. 14) so that larger proportion of the upper volume could be used to store the hot HTF, rendering a higher storage density [Zanganeh et al., 2015]. Three different cone angles (θ =-15°; 0°; +15°) have been tested and results are reported in Fig. 14 and Table 6.

From Fig. 14 one may observe that the temperature profiles at t_{Fin}^* time differ from the three θ values without optimization. The thermocline zone for $\theta=15^\circ$ case has already touched the bottom manifold because the converging shape makes the thermal fronts in the middle advance much faster than at the side. By optimizing the upper baffle, similar double-hump shape of the thermocline zone could be achieved. The results in Table 6 also show that conical shape with positive angle ($\theta=15^\circ$) performs better than the negative angle ($\theta=-15^\circ$) when optimized upper baffle is used. This is because its upper flow area is relatively larger than others, thus more effectively controlled by the optimized upper baffle. However, the bottom baffle should also be carefully designed for the discharging process to render an optimal overall efficiency.



Fig.14. Effect of cone angle on the temperature profile at t_{Fin}^* ($\theta = -15^\circ$; 0° ; $+15^\circ$)

					e perior				Stor "Be	
θ	U	niform o	orifice up	oper baff	Optimized orifice upper baffle					
	t_{Fin}^{*}	<i>S</i> *	3	σ	η	t_{Fin}^*	S^*	3	σ	η
-15°	0 44	0.15	0.42	0.68	0.40	0.24	0.11	0.50	0.78	0 4 4

0.41

0.41

0.31

0.41

0.14

0.16

0.61

0.65

0.81

0.81

0.48

0.50

Table 6. Effect of cone angle on the performance of the thermocline storage tank

3.4.4 Number of orifices N

0.56

0.76

0.20

0.21

0.43

0.42

0.70

0.70

0°

+15°

The number of orifices (N) on the baffle is also an important design & optimization factor since it determinates how many small sub-streams entering the middle flow zone could be produced. Comparative studies were performed with N value of 6, 10 and 20, while the total porosity of the upper baffle was fixed at Φ =0.30. Results for the uniform orifice baffle case and optimized baffle case are reported in Fig. 15 and Table 7.

When uniform orifice upper baffle is used, the $S^*(t_{Fin}^*)$ value decreases with the increasing *N*. In fact, the width of orifice (3 mm) is much smaller for *N*=20 than that for *N*=6 (10 mm), indicating higher flow resistance of the upper manifold that impedes the expansion of thermocline zone passing through. In particular, the mean local velocity traversed the upper baffle with 20 orifices must be larger than that with 6 orifices, which contributes to smaller amount of conductive term versus the time accumulation. Vice versa, for small number of orifices, the hot front traversed together with much more accumulated conductive term of the heat transfer [Mohamad et al., 2019; Ansys theory Guide, 2019]. This results in the thicker thermocline zone and it

even touches already the bottom port at t_{Fin}^* moment. As a result, higher number of orifices having identical size is beneficial to maintain the temperature stratification.

When the optimized upper baffle is installed, the thermal fronts could be largely flattened for all three cases tested. Although the $S^*(t_{Fin}^*)$ value is reduced by half from 0.202 (*N*=6) to 0.107 (*N*=20), almost the same values of global performance indicators ($\varepsilon \approx 0.60$; $\eta \approx 0.48$; $\sigma \approx 0.80$) could be achieved when using the optimized upper baffle. This implies that the proposed optimization algorithm is capable of achieving the optimum performance of the thermocline storage tank regardless of the number of orifices initially set up.



Fig.15. Effect of orifice number (N) on the temperature profile at t_{Fin}^* (N=6; 10; 20)

						1				0
• •	τ	Jniform o	orifice up	pper baff	le	Optimized orifice upper baffle				fle
IN	t_{Fin}^{*}	<i>S</i> *	3	σ	η	t^*_{Fin}	<i>S</i> *	3	σ	η
6	0.79	>0.23	0.41	0.68	0.40	0.49	0.20	0.59	0.80	0.48
10	0.56	0.20	0.43	0.70	0.41	0.31	0.14	0.61	0.81	0.48
20	0.45	0.15	0.48	0.73	0.44	0.24	0.11	0.60	0.80	0.48

Table 7. Effect of orifice number N on the performance of the thermocline storage tank

3.4.5 Global porosity of baffle (Φ)

Three global porosity values (Φ =0.1; 0.3; 0.5) have been tested. The results obtained are shown in Fig. 16 and Table 8 and briefly commented as follows.



Fig.16. Effect of global porosity (Φ) on the temperature profile at t_{Fin}^* time (Φ =0.1; 0.3; 0.5)

In this paper, the global porosity is the proportional fluid length to whole length of the baffle (*L*). Lower global porosity is beneficial to maintain the temperature stratification when uniform orifice upper baffle is used, again because of the higher flow resistance that impedes the natural propagation of the thermocline zone. On the other hand, the larger porosities may have more improving space in thermal performance. After optimization, similar shape of the thermocline zone can be reached at t_{Fin}^* , leading to reduce their differences for thermal performance of the storage tank. The larger porosities have more improved space for thermal performance, including ε , σ , η .

		0	1	,	1					0
Φ	Uniform orifice upper baffle					Optimized orifice upper baffle				
	t_{Fin}^{*}	<i>S</i> *	3	σ	η	t_{Fin}^{*}	<i>S</i> *	3	σ	η
0.1	0.48	0.18	0.50	0.73	0.45	0.37	0.15	0.57	0.81	0.47
0.3	0.56	0.20	0.43	0.70	0.41	0.31	0.14	0.61	0.81	0.48
0.5	0.80	>0.22	0.36	0.67	0.37	0.36	0.15	0.56	0.81	0.46

Table 8. Effect of global porosity (ϕ) on the performance of the thermocline storage tank

3.5 Conclusion and perspectives

In the present study, the inlet/outlet manifolds of a thermocline storage tank are optimized, providing quasi-thermally piston flow inside the tank so as to maintain the temperature stratification during charging or discharging process. The design of upper and bottom manifolds is based on the insertion of perforated baffles. An original optimization algorithm is developed to determine the optimal orifice size distribution on the baffle, using a novel optimality criterion of equal passage time of hot front for
every orifice. Based on the results obtained, main conclusions can now be summarized as follows:

- Uniform velocity distribution on horizontal cross-sections doesn't necessarily lead to the highest temperature stratification. Flat thermal front (thermally piston flow) is favorable to enhance the thermal performance of the thermocline storage tank.
- Equalizing the passage times of the hot front (optimality criterion) for different orifices can effectively minimize the surface area (S^*) of the thermocline zone at t_{Fin}^* , leading consequently to the significantly improved values of global performance indicators (tail factor, charging/discharging efficiency, capacity ratio).
- The combination of optimized upper and bottom baffles provides the highest charging, discharging and overall efficiencies.
- The tail factor (ε) can well indicate the performance of the thermocline storage tank. The values of ε can provide additional information on the slope of the outlet HTF temperature curve, which may not be appropriately quantified by other commonly-used performance indicators.
- A series of parametric studies performed demonstrates that the proposed baffle insertion solution and the associated optimization algorithm are effective, efficient, robust and simple to implement. For all the tested cases, the performance improvement of the thermocline storage tank using our method is significant.

It should be noted that the efficiency of the optimization method depends largely on the correct simulation of the transient flow and temperature profiles during charging & discharging processes. The validation by laboratory experiment of a thermocline storage tank equipped with or without optimized upper & bottom baffles is our ongoing work.

In the meantime, filling the storage tank with solid materials is now commonly considered in studies for CSP plants since it makes possible to the cost reduction of the TES system [Martin et al., 2018]. The optimization of thermocline behaviors at the presence of solid fillers by considering both the thermal diffusion the fluid-solid interactions is also the topic of our current work.

Finally, the application fields of the energy-efficient thermocline units are rather vast and promising, and not limited to high temperature TES in CSP plants. Developing and integrating thermocline storage tanks with high energy efficiency, controllable and

stable for long-time use and cost-effective for TES or energy recovery in other residential or industrial sectors [Chandra and Matuska, 2019; Han et al., 2009; Li, 2016] is the direction of our future work.

Acknowledgement

This work is supported by the French ANR within the project OPTICLINE (ANR-17-CE06-0013)

References

Afrin S, Kumar V, Bharathan D, Glatzmaier G, Ma Z. Computational analysis of a pipe flow distributor for a thermocline based thermal energy storage system. J Sol Energy Eng 2013;136:021010. https://doi.org/10.1115/1.4024927

Ahmed N, Elfeky K, Qaisrani, Mumtaz A, Wang Q. Numerical characterization of thermocline behaviour of combined sensible-latent heat storage tank using brick manganese rod structure impregnated with PCM capsules. Sol Energy 2019;180:243-256. https://doi.org/10.1016/j.solener.2019.01.001

Alva, G., Lin, Y., Fang, G. An overview of thermal energy storage systems. Energy 2018;144: 341-378. https://doi.org/10.1016/j.energy.2017.12.037

Ansys Fluent Theory Guide, version 19.1, ANSYS Incorporated; 2019. https://www.ansys.com

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. https://doi.org/10.1016/j.ijheatmasstransfer.2013.01.047

Bruch A, Molina S, Esence T, Fourmigué J F, Couturier R. Experimental investigation of cycling behaviour of pilot-scale thermal oil packed-bed thermal storage system. Renew Energ 2017;103:277–285. https://doi.org/10.1016/j.renene.2016.11.029

Chandra Y, Matuska T. Stratification analysis of domestic hot water storage tanks: Acomprehensivereview.EnergyBuild2019;187:110-131.https://doi.org/10.1016/j.enbuild.2019.01.052

De J, Reuter H. A review of performance modelling studies associated with open volumetric receiver CSP plant technology. Renew Sust Energ Rev 2018;82: 3848–3862. https://doi.org/10.1016/j.rser.2017.10.086

Ellingwood K, Mohammadi K, Powell K. A novel means to flexibly operate a hybrid concentrated solar power plant and improve operation during non-ideal direct normal irradiation conditions. Energy Convers Manag 2020;203:112275. https://doi.org/10.1016/j.enconman.2019.112275

Esence T, Bruch A, Molina S, Stutz B, Fourmigué J. A review on experience feedback and numerical modeling of packed-bed thermal energy storage systems. Sol Energy 2017;153:628-654. https://doi.org/10.1016/j.solener.2017.03.032

FasquelleT, Falcoz, Q, Neveu, P, Hoffmann J. A temperature threshold evaluation for thermocline energy storage in concentrated solar power plants. Appl Energy 2018a;212:1153–1164. https://doi.org/10.1016/j.apenergy.2017.12.105

Fasquelle T, Falcoz Q, Neveu P. Numerical simulation of a 50MWe parabolic trough power plant integrating a thermocline storage tank. Energy Convers Manag 2018b;172:9–20. https://doi.org/10.1016/j.enconman.2018.07.006

Galione P, Pérez-Segarra CD, Rodríguez I, Torras S, Rigola J. Numerical evaluation of
multi-layered solid-PCM thermocline-like tanks as thermal energy storage systems for
CSP applications. Energy Proc 2015;69:832–841
https://doi.org/10.1016/j.egypro.2015.03.099

Geissbühler L, Kolman M, Zanganeh G, Haselbacher A, Steinfeld A. Analysis of industrial-scale high-temperature combined sensible/latent thermal energy storage. Appl Therm Eng 2016;101:657–668. https://doi.org/10.1016/j.applthermaleng.2015.12.031

Grirate H, Agalit H, Zari N, Elmchaouri A, Molina S, Couturier R. Experimental and numerical investigation of potential filler materials for thermal oil thermocline storage. Sol Energy 2016;131:260–274. https://doi.org/10.1016/j.solener.2016.02.035

Han Y, Wang R, Dai Y. Thermal stratification within the water tank. Renew Sust Energ Rev 2009;13:1014–1026. https://doi.org/10.1016/j.rser.2008.03.001

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. https://doi.org/10.1016/j.applthermaleng.2018.10.051

Hoffmann J, Fasquelle T, Goetz V, Py X. A thermocline thermal energy storage system with filler materials for concentrated solar power plants: Experimental data and numerical model sensitivity to different experimental tank scales. Appl Therm Eng 2016;100:753–761. https://doi.org/10.1016/j.applthermaleng.2016.01.110

Keilany M, Milhé M, Bézian J, Falcoz Q, Flamant G. Experimental evaluation of vitrified waste as solid fillers used in thermocline thermal energy storage with parametric analysis. J Energy Storage 2020;29:101285

Kim D, Ho Seok, Kim Y, Song C, Lee K, Choi J. Experimental studies of the discharge performance of single-medium TES for CSP applications. Appl Therm Eng 2017;127:499–507. https://doi.org/10.1016/j.applthermaleng.2017.08.057

Kumar S, Singh P. A novel approach to manage temperature non-uniformity in minichannel heat sink by using intentional flow maldistribution. Appl Therm Eng 2019.163:114403. https://doi.org/10.1016/j.applthermaleng.2019.114403

Li G. Sensible heat thermal storage energy and exergy performance evaluations. Renew Sust Energ Rev 2016;53:897–923. https://doi.org/10.1016/j.rser.2015.09.006

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. https://doi.org/10.1016/j.ces.2014.11.051 Martin C, Bonk A, Braun M, Odenthal C, Bauer T. Investigation of the long-term stability of quartzite and basalt for a potential use as filler materials for a molten-salt based thermocline storage concept. Sol Energy 2018; 171: 827–840. https://doi.org/10.1016/j.solener.2018.06.090

Mertens N, Alobaid F, Frigge L, Epple B. Dynamic simulation of integrated rock-bed thermocline storage for concentrated solar power. Sol Energy 2014; 110: 830–842. https://doi.org/10.1016/j.solener.2014.10.021

Milman O, Spalding D, Fedorov V. Steam condensation in parallel channels with nonuniform heat removal in different zones of heat-exchange surface. Int J Heat Mass Transf 2012;55:6054–6059. https://doi.org/10.1016/j.ijheatmasstransfer.2012.06.016

Modi A, Pérez-Segarra C. Thermocline thermal storage systems for concentrated solar power plants: One-dimensional numerical model and comparative analysis. Sol Energy 2014;100:84–89. https://doi.org/10.1016/j.solener.2013.11.033

Motte F, Falcoz Q, Veron E, Py X. Compatibility tests between Solar Salt and thermal storage ceramics from inorganic industrial wastes. Appl Energy 2015;155:14–22. https://doi.org/10.1016/j.apenergy.2015.05.074

Nandi B, Bandyopadhyay S, Banerjee R. Numerical modeling and analysis of dual medium thermocline thermal energy storage. J Energy Storage 2018;16: 218–230. https://doi.org/10.1016/j.est.2018.01.020

Nicolas L, Falcoz Q, Pham Minh D, Hoffmann JF, Meffre A, Nzihou A, Goetz V. Flexibility and robustness of a high-temperature air/ceramic thermocline heat storage pilot. J Energy Storage 2019;21:393–404. https://doi.org/10.1016/j.est.2018.11.034

Ortiz C, Valverde J, Chacartegui R, Perez-maqueda L, Giménez P. The Calcium-Looping (CaCO₃/CaO) process for thermochemical energy storage in Concentrating Solar Power plants. Renew Sust Energ Rev 2019;113:109252. https://doi.org/10.1016/j.rser.2019.109252

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. https://doi.org/10.1016/j.energy.2018.10.163

Pelay U, Luo L, Fan Y, Stitou D, Rood M. Thermal energy storage systems for concentrated solar power plants. Renew Sust Energ Rev 2017;79:82–100. https://doi.org/10.1016/j.rser.2017.03.139

Pizzolato A, Donato F, Verda V, Santarelli M, Sciacovelli A. CSP plants with thermocline thermal energy storage and integrated steam generator – Techno-economic modeling and design optimization. Energy 2017;139:231–246. https://doi.org/10.1016/j.energy.2017.07.160

Rodrigues F, Lemos M. Transient performance of a thermocline energy storage system using the two-energy equation model. Int J Heat Mass Transf 2020;150:119323. https://doi.org/10.1016/j.ijheatmasstransfer.2020.119323 Sanz J, Nuñez F, Zaversky F. Benchmarking analysis of a novel thermocline hybrid thermal energy storage system using steelmaking slag pebbles as packed-bed filler material for central receiver applications. Sol Energy 2019;188:644–654. https://doi.org/10.1016/j.solener.2019.06.028

Vigneshwaran K, Sodhi G, Muthukumar P, Guha A, Senthilmurugan S. Experimentaland numerical investigations on high temperature cast steel based sensible heat storagesystem.ApplEnergy2019;251:113322.https://doi.org/10.1016/j.apenergy.2019.113322

Wang L, Yang Z, Duan Y. Influence of flow distribution on the thermal performance of dual-media thermocline energy storage systems. Appl Energy 2015;142:283–292. https://doi.org/10.1016/j.apenergy.2014.12.024

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. https://doi.org/10.1016/j.cherd.2016.06.026

Wei M, Fan Y, Luo L, Flamant G. Fluid flow distribution optimization for minimizing the peak temperature of a tubular solar receiver. Energy 2015a;91:663–677. https://doi.org/10.1016/j.energy.2015.08.072

Wei M, Fan Y, Luo L, Flamant G. CFD-based evolutionary algorithm for the realization of target fluid flow distribution among parallel channels. Chem Eng Res Des 2015b;100:341–352. https://doi.org/10.1016/j.cherd.2015.05.031

Wei M, Fan Y, Luo L, Flamant G. Design and optimization of baffled fluid distributor for realizing target flow distribution in a tubular solar receiver. Energy 2017;136:126–134. https://doi.org/10.1016/j.energy.2016.04.016

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.
https://doi.org/10.1016/j.apenergy.2013.08.044

Yin H, Ding J, Yang X. Experimental research on thermal characteristics of a hybrid thermocline heat storage system. Appl Therm Eng 2014;62:293-301 https://doi.org/10.1016/j.applthermaleng.2013.09.018

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. https://doi.org/10.1016/j.apenergy.2014.07.110

Zavattoni S, Geissbühler L, Barbato M, Zanganeh G, Haselbacher A, Steinfeld A. (2017). High-temperature thermocline TES combining sensible and latent heat - CFD modeling and experimental validation. In: SolarPACES 2016 International conference, Abu Dhabi, UAE; 2016. https://doi.org/10.1063/1.4984449

Zhang H, Baeyens J, Cáceres G, Degrève J, Lv Y. Thermal energy storage: Recent developments and practical aspects. Prog Energy Combust Sci 2016;53:1-40. https://doi.org/10.1016/j.pecs.2015.10.003

Zhao B, Cheng M, Liu C, Dai Z. Cyclic thermal characterization of a molten-salt packed-bed thermal energy storage for concentrating solar power. Appl Energy 2017; 195:761–773. https://doi.org/10.1016/j.apenergy.2017.03.110

Chapter 4. Mitigating the impact of thermal jet in a singlemedium thermocline storage tank: Novel optimization strategy for flow distributor

Abstract

This paper presents a numerical and experimental study on the impact of thermal jet in single-medium thermocline storage tanks. Special focus is given on how to maintain the temperature stratification inside the storage tank during the dynamic charging or discharging operation, by using a new type of fluid diffusing device, namely the Ring-Opening Plate Distributors (ROPDs). For this purpose, a novel CFD-based optimization strategy has firstly been developed, capable of determining the optimal sizes of the ring-shape openings under given operating condition. This optimization method has been validated by experimental testing results of a lab-scale cylinder water storage tank and the effectiveness of the ROPDs on mitigating the impact of thermal jet has been highlighted.

Then, the influences of the inlet flow rate and the temperature difference on the local transient temperature profiles and on the energy and exergy efficiencies of the storage tank have been experimentally investigated. The stability range of operational parameters, characterized by the bulk Froude (Fr) number, has been determined for maintaining quasi-undisturbed advance of the thermocline. Beyond this stability range, the optimized ROPDs have shown to be capable of maintaining high charging/discharging efficiency of the storage tank. The findings of this work may help to provide design guidelines on flow diffusers and determine adapted operational conditions for practical applications of SMT storage tanks with optimized performance.

Keywords: Single-medium thermocline (SMT); Thermal energy storage (TES); Ring-Opening Plate distributor (ROPD); Thermal jet; Temperature stratification; optimization.

This chapter is to be submitted to Applied Thermal Engineering as

Lou W, Xie B, Aubril J., Fan Y, Luo L* and Arrivé A. Mitigating the impact of thermal jet in a single-medium thermocline storage tank: Novel optimization strategy for flow distributor.

Nomenclature

Latin letters					
Ср	Heat capacity at constant pressure [J kg ⁻¹ K ⁻¹]				
D	Diameter of tank [m]				
d	Diameter of inlet/outlet port [m]				
Ε	Energy [J]				
Ex	Exergy [J]				
е	Thickness of the baffles [m]				
Fr	Froude number				
g	Gravitational acceleration [m s ⁻²]				
Н	Height of thermocline tank [m]				
h	Vertical distance of main fluid storage region [m]				
k	Turbulent kinetic energy [m ² s ⁻²]				
L	Distance between the upper and bottom baffles[m]				
Ν	Number of openings on the plate distributor				
Р	Pressure [Pa]				
'n	Mass flow rate [kg s^{-1}]				
Pe	Péclet number				
Q	Flow rate [L min ⁻¹]				
R	Uncertainty				
Re	Reynolds number				
S	Unit in series				
S	Surface [m ²]				
Т	Temperature [K]				
t	Time [s]				
t^*	Dimensionless time [–]				
\overline{t}	Mean value of all passage times [K]				
v	Velocity [m s ⁻¹]				
V	Volume [m ³]				
$ar{v}$	Mean velocity [m s ⁻¹]				
W	Width of the ring-shape opening [m]				
X	Measured variables				
x	X-coordinate [m]				
у	<i>Y</i> -coordinate [m]				
Greek symbols					
α	dimensionless surface of each ring-shape opening [-]				

- γ Relaxation factor [-]
- ε Rate of dissipation of turbulent kinetic energy [m² s⁻³]
- η Efficiency [–]

λ	Thermal conductivity [W m ⁻¹ K ⁻¹]
μ	Viscosity [kg $m^{-1} s^{-1}$]
ρ	Density [kg m ⁻³]

Subscripts/superscripts

bottom	Bottom		
С	Cold temperature		
ch	Charging		
des	Destruction		
Ε	Energy		
Ex	Exergy		
fin	Final		
h	Hot temperature		
i	Ring-shape opening index		
in	Inlet		
ini	Initial time		
j	Optimization step		
manifold	Manifold region		
max	Maximum		
min	Minimum		
normal	Normal velocity		
out	Outlet		
upper	Upper		
residence	Residence time		
stored	Stored		
tank	Thermocline-based TES tank		
total	Total amount		
x	Threshold coefficient		
0	Reference		
Abbreviations			
ABS	Acrylonitrile butadiene styrene		
CFD	Computational Fluid Dynamics		
cDAQ	Compact data acquisition system		
DMT	Dual-medium thermocline		
HTF	Heat transfer fluid		
TC	Thermocouple		
RNG	Re-Normalization Group methods		
ROPD	Ring-Opening Plate Distributor		
SMT	Single-medium thermocline		
STD	Standard derivation		
TES	Thermal energy storage		

4.1. Introduction

Transforming the world energy system in conformity with sustainability goals requests the prompt uptake of renewables within a majority of energy demand domains. Thermal energy storage (TES) has the considerable potential to realize high shares of renewable energy integration in different sectors, such as power generation, industries and buildings [source: Celsiuscity]. Beneficial features of TES including variable storage capacity and duration (e.g., seasonal capability), flexible decoupling of supply/demand, adaptable integration manners and life cycle advantages have attracted special attentions from various energy markets. According to the International Renewable Energy Agency (IRENA)'s Paris Agreement-aligned transforming energy scenario, it's expected that the installed TES capacity will be increased threefold in the next 10 years, from 234 GWh in 2019 to at least 800 GWh in 2030 [source: IRENA, 2020].

Current available TES systems can be classified into two-tank or single-tank concept, differentiate themselves by involving two individual tanks or only one tank for storing and circulating the heat transfer fluids (HTFs). For the latter, hot and cold HTFs are separated naturally by the density difference, with the presence of a temperature transition zone between them called the *thermocline*. A techno-economic analysis [Grirate et al., 2016] showed that the single-tank TES technology, when associated with low-cost sensible fillers, could cut off more than 35% of the capital investment compared to its counterpart (two-tank concept). In this regard, the singletank thermocline TES technology has gained more attention globally than ever before, in line with the EU's long-term strategy of achieving carbon neutrality by 2050. Under this strategy, national energy and climate plans (NECPs) for 2021-2030 outline how the member states intend to address energy efficiency, renewables, greenhouse gas emissions reduction, etc. For instance, this draft of the EU's upcoming renewable energy directive confirms the bloc's objective of sourcing 38-40% of its energy from renewables by 2030 [source: European Commission]. The operation of a thermoclinebased storage tank is generally composed of three stages: the charging (energy storage), the stand-by and the discharging (energy release). As for the charging, the hot HTF is introduced into the top port of the storage tank, driving the cold HTF out from the bottom port. The opposite fluid direction is proceeded for the discharging operation. The interval time period between the charging and the discharging is called the standby. A temperature stratification zone, the thermocline, exists at the interface of hot and cold HTFs where the temperature gradient is formed. Besides the heat loss from the storage tank to the ambient, the overall performance of the thermocline storage tank depends largely on the level of temperature stratification [Ortega-Fernández et al., 2017]. To achieve a highly-efficient charging/discharging operation, the thermocline decay (de-stratification) should be avoided to the maximum extent possible and to maintain the thermocline volume as small as possible [Gajbhiye et al., 2018].

Nevertheless, the heat transfer always takes place at the hot and cold interface because of the lack of material insulation that adiabatically separates two HTFs. Especially during the dynamic charging or discharging process, the thermal jet, i.e., the inflowing HTF jet with different velocity and temperature (density) will disturb the calm fluid initially stored inside the thermocline tank. Compared to the diffusion heat transfer due to the temperature gradient, the convection heat transfer caused by this sudden injection and the subsequent fluid mixing could be a dominant factor that degrades the temperature stratification under certain circumstances. The impact of this thermal jet could be more prominent for the single-medium thermocline (SMT) storage tanks than dual-medium thermocline (DMT) storage tanks. Without the buffering effect of solid fillers, the mixing region of hot and cold HTFs could occupy a large volume of the SMT storage tank, sometimes even the whole volume under high discharging flow rate condition [Dehghan and Barzegar, 2011]. Given the fact that SMT storage tanks are commonly employed for the space heating and cooling of both individual buildings and districts/communities due to their volume compactness, simple construction and easy maintenance [Advaith et al., 2020], a large number of researches have been devoted to study the impacts of thermal jet on the thermocline degradation and to propose mitigation measures.

A large amount of researches have been devoted to investigate the effect of process parameters (e.g., flow rate, temperature difference) on overall performance of the SMT storage tank in order to determine the suitable operating conditions [Li et al., 2016]. Manu et al. [2016] systematically investigated the simulated instantaneous response of thermocline to inlet temperature and velocity after the partial charging period. Disturbances at different velocities or temperatures issued from the inflowing thermal jet on thermocline evolutions of the SMT tank. They found that not all imposed perturbations would cause the breakdown of initially stable thermocline, and long duration of disturbance who was longer than the critical time value induced destabilized the thermocline. Shaikh et al. [2018] numerically tested three different charging temperatures (T_h =651.15 K, 773.15 K and 823.15K) for a SMT storage tank of the comparison on the velocity contours indicated that the penetrated distance decreased while the thermocline thickness enlarged with the increasing inlet charging temperature. They assumed the raison is that the buoyancy driven mixing becomes dominant after the inlet porous tube. However, Advaith et al. [2021] express the experimental results that the larger temperature difference (larger density difference) of operating range could guarantee the more effective thermal stratification and sustain a stable

thermocline for longer time. The initial formation of thermocline during the charging of a cold water tank has been performed by Yoo et al., [1986] by means of flow visualization. Momentum-dominated flow has been observed at a Froude number (Fr, as defined in Eq.13) higher than 2, provoking the jet entrainment phenomenon at the very beginning of the charging process and the subsequent unexpected mixing.

After knowing the limitations of simple inlet/outlet ports dealing with the thermal jet in the SMT storage tanks, efforts have been made on the design and implementation of structured flow diffusing devices as alternative. In regard, different shapes of flow distributor have been proposed and tested including tubular distributors [e.g., Dragsted et al., 2017; Chandra et al., 2020], baffle plates [e.g., Bouhal et al., 2017; Kocijel et al., 2020; Taher et al., 2019] and porous distributors [e.g., Wang and Davidson, 2017]. The general strategy is, with the help of specially structured flow distributors, to produce multi-branch streams [e.g., Li et al., 2014], inversed flow [e.g., Bouhal et al., 2017] or radial flow [e.g., Al-Azawii et al., 2020], so as to alleviate the disturbances brought by the strong momentum-dominated flow. Karim [2011] introduced two octagonal diffusers with different opening areas into a rectangular chilled-water tank. Their experimental results showed that the diffuser having the larger opening area but smaller Fr number (0.5) resulted a worse temperature stratification. The unequal pressure drop at such low Fr number and thereby the non-uniform flow might be the reason of the performance degradation. Taher et al. [2019] conceived a new moving plate applicable to a spherical SMT tank via turbulent CFD model (inlet Reynolds number is over 4000). Time-dependent Richardson number (the reciprocal of the squared Froude number) was used to evaluate thermal stratification for SMT systems equipped with fixed plate baffles/ moving plate. Their CFD results using dynamic mesh showed that the moving plate could largely reduce the mixing motions and the convection heat transfer caused by the momentum-controlled jet.. Experimental investigation inside the SMT system equipped with a inlet tube stratifier revealed that Richardson number is the best option than péclet number, Reynolds number to define the thermal stratification, which is in good agreement with the instantaneous temperature profile via different inlet flow rate, reflecting the real behaviour of energy release process[Castell et al., 2010].Furthermore, Chung et al. [2008] numerically investigated four design parameters including inlet Reynolds number, Froude number, area ratio of diffuser to SMT cross-section and diffuser configuration including radial plate type, radial adjusted plate type, and H-beam type. Their results presented that the contribution of Fr could be negligible whereas Reynold number and distributor shape governed the thermal performance of energy storage, such as instantaneous thermocline thickness. Although Froude number directly affects the characteristics of thermal jet and thermal stratification, the acceptable working range of Froude number, Reynolds number together with other conditions in the operating strategy of SMT system isn't established yet. Chandra and Matuska [2020] investigated experimentally validated CFD model to simulate a SMT system with slotted, perforated diffuser or simple inlet device with examination of two fundamental operational variables, inlet Reynolds number and working temperature range. Slotted diffuser represented outstanding performance at larger inlet flow rate whereas there is little difference in MIX number and discharging efficiency at smaller inlet flow rate. The performance improvement of the proposed distributors shapes seemed much more meaningful according to the practical operating conditions.

Nevertheless, a great number of studies on this topic still rest on the numerical or experimental testing of certain type of diffuser structures under certain process conditions, working especially on selecting one or several geometric parameters (e.g., nozzle position and diameter, number of sub-pipes, etc.). Very few studies involve the structure optimization of flow distributors to make them more adaptable to different storage tank geometries and operating conditions.

The above literature survey indicates that more systematic investigations with experimental feedback are still needed on the impacts of thermal jet as well as the migration measures for SMT storage tanks. In particular, the separate influence of the operating parameters (e.g., inlet flow rate; working temperature difference) as well as their combined effects on the temperature stratification need further clarification, as controversy still exists in the published literature. Moreover, the design and structure optimization of fluid flow distributors for the SMT storage tanks deserve particular attention. In fact, the use of appropriate flow diffusing devices could render high charging & discharging efficiencies of the storage tank being operated under a wide range of working conditions. However, studies involving both the simulation, optimization and experimental verification have been rarely reported in the literature. There is still much room for improvement.

As a continuation and extension of our previous study [Lou et al., 2020], this work has firstly presented the structure optimization of a new type of fluid diffusing device, namely the Ring-Opening Plate Distributors (ROPDs), to mitigate the degradation of temperature stratification in the SMT storage tank due to the thermal jet. We strive to bridge the CFD-based optimization procedure for both upper and bottom ROPDs and the experimental testing results of a lab-scale cylinder SMT storage tank for both the charging and discharging operations. Then, a systematic experimental investigation into the charging/discharging performances of the cylinder TES tank equipped with or without ROPDs has been performed. The influences of the inlet flow rate and the temperature difference on the local transient temperature profiles and on the energy and exergy efficiencies were compared and analyzed. The stability range of operational parameters, characterized by the bulk *Fr* number, has been determined for maintaining quasi-undisturbed advance of the thermocline. Beyond this stability range,

the optimized ROPDs have shown to be capable of maintaining high charging/discharging efficiency of the storage tank. The findings of this work may help to provide design guidelines on flow diffusers and determine adapted operational conditions for practical applications of SMT storage tanks with optimized performance.

4.2. Experiment set-up4.2.1 SMT storage tank and Ring-Opening Plate Distributor (ROPD)

The SMT prototype subjected to experimental testing is a vertical cylindrical storage tank having an inner diameter (*D*) of 194 mm and inner height (*H*) of 390 mm, the *H/D* ratio thus being 2.0. The upper or bottom port (i.d.: 10 mm) is situated in the center of the upper or bottom cover of the tank, respectively (Fig. 1), rendering it an axisymmetric geometry that could be simplified into 2D in the CFD-based optimization. The tank body was made of polycarbonate (ABAQUEPLAST), having a good temperature resistance (up to 125°C) and a low thermal conductivity ($\lambda = 0.2$ W m⁻¹K⁻¹) to reduce the heat loss to the ambient. The storage tank was fabricated in-house, by firstly sticking together the lateral wall (3 mm in thickness) and the upper & bottom covers (12 mm in thickness) using a transparent adhesive (Araldite Polyurethane 2028) and then being sealed by a silicone adhesive (Loctite 5366).

Two plates with ring-shape openings also made of polycarbonate were installed into the cylindrical storage tank, dividing the inner tank body into three parts as shown in Fig. 1(a): the upper manifold (30 mm in height), the middle storage zone (318 mm in height) and the bottom manifold (30 mm in height). The disc-like ring-opening plates have an overall dimension of 194 mm in diameter and 6 mm in thickness. Each plate has three ring-shape openings, their widths subjected to optimization as will be described in detail in section 3.2 and Fig. 7. The upper and bottom plates were supported by small cubes fixed at the lateral wall. The translucent silica gel (Loctite 5366) was used to seal up the border between the plates and the tank wall so that the fluid can pass from one zone to another only through the ring openings. The volume of the middle storage zone is 9.4 L while the total internal volume of the storage tank is 11.2 L.

Twelve thermocouples TC (RS PRO type K 1/0.3mm; ± 0.35 K) were used to measure the fluid temperature at different locations inside the storage tank, as shown in Fig. 1(b). Among them, TC_1/TC_8 right facing the upper/bottom ports were mainly used to monitor the inflowing fluid temperature before touching the upper/bottom plate during the charging or discharging operation, respectively. TC_2-7 also located on the central axis were installed to measure the axial fluid temperature variation, evenly spaced 63.6 mm between one another. All the TCs were attached to a slender I-shape

plastic support. The support containing the TCs at their proper positions was then fixed at the upper and the bottom plates, with its end passing through the disc-like plate via an extra hole (6 mm in diameter) in the center. The crevice between the plates and the TC support were then filled with the silica gel for the sealing purpose.



Figure 1. Geometry and dimensions of the cylindrical SMT storage tank for laboratory testing. (a) schematic view; (b) vertical cross-section and arrangement of the thermocouples (Unit: mm)

4.2.2 Test-rig and experimental procedure

The experimental test-rig shown in Fig. 2 is composed of the cylindrical SMT storage tank, hot and cold circuits, a feed tank and the control & data acquisition system. The HTF used in this study is tap water. Both water tanks (THERMOR Steatis 100 L) was equipped with a closed loop driven by a fluid circulator (GRUNDFOS ALPHA120-50N150) to ensure the temperature-homogenous cold/hot fluid status in the water tank. A built-in heater strip (1200 W maximum) was installed inside the (hot) water tank 1 as the heat source while a cryo-thermostat (LAUDA Proline RP 845)

connected to a home-made helical-coil heat exchanger inside the (cold) water tank 2 was used as the cold source. Temperature sensors PT100 (RS 762-1134; ± 0.35 K) were installed inside both water tanks to measure the water temperature inside. The output fluid from each water tank was firstly filtered by a 20 µm filter (HNP Mikrosysteme F-MI2-T), and then delivered by a gear pump (TUTHILL Pump P series, 0.067-7.63 L min⁻¹) to the cylindrical SMT storage tank (or to the feed tank). The flow rate of each circuit was measured by a flow meter (Kobold DPM 1150 G2, 0.05-5 L min⁻¹; $\pm 1\%$ of full scale) installed downstream of the pump. The feed tank (300L) was used to store and recycle the HTFs. Two additional platinum resistance thermometers PT100 (RS 891-9145; ± 0.35 K) were imbedded into the upper and bottom ports of the SMT storage tank to measure and monitor the inlet/outlet temperature of the HTF.

Both the charging and discharging operations were investigated. Each operation was composed of several steps, briefly described as follows (charging as example).

(1) Preparation: before any charging process, the water quantity is ensured to be sufficient inside both water tanks. The closed loops for water heating and cooling were then turned on.

(2) Filling-up: once the temperature set points in water tanks were reached, the gear pump 2 was started to deliver the cold water into the SMT storage tank from the bottom to the top.

(3) Preheating: once the SMT storage tank was full of the cold water at T_c , the gear pump 1 was started to deliver the hot water to preheat the piping between the hot water tank and the SMT storage tank. The purge water was guided to the feed tank via the 3-way electric valve (ASCO SCG327B001) at the top (inlet) port of the SMT storage tank. The purpose of this step was to eliminate the possible temperature and flow-rate fluctuations of the inflowing hot water during the short start-up period due to the existing (cold) water in the connection pipe. This step usually lasted 10 minutes to about 1 hour depending on the operating flow rate.

(4) Once the constant flow rate (\dot{m}_h) and preset temperature (T_h) had been reached, the electrical value at the top (inlet) port was switched on to inject the hot water into the SMT storage tank. The charging operation continued till the tank was fully occupied by the hot water.

The discharging process follows basically the same protocol, except that the bottom port was switched to be the inlet whereas the upper port became the outlet.

The tested charging/discharging flow rate (Q_{in}) ranged between 0.3 and 1.5 L min⁻¹, corresponding to an inlet Reynolds number (Re_{in} defined in Eq. 15) from 1368

to 6831 and an average tank Reynolds number (Re_{tank} defined in Eq. 16) from 50 to 251. The tested temperature of the hot water T_h ranged from 323.15 K to 343.15 K while the cold water temperature T_c was kept at 293.15 K. Note that charging and discharging processes were performed separately, without addressing the cyclic operation issue. The tank body as well as the pipes and valves for entering the hot fluid were well-insulated by foamglass to reduce the heat loss at a minimum level. The tightness of the setup was verified before every test to avoid the water leakage. A LabVIEW platform developed in-house, together with the data acquisition system (NI cDAQ 9178) and the modules was used to control and regulate the output of two gear pumps (NI 9265), the heater strip (NI 9265), the cryo-thermostat and all electrical valves (NI 9474), , and to record the real-time measurement data of the PT100 (NI 9217), TCs (NI 9213) and the flowmeters (NI 9209) during the whole charging or discharging process.



Fig.2 Test-rig for the cylindrical SMT storage tank. (a) Photo view; (b) Schematic diagram

4.2.3 CFD Model and validation

In this work, CFD simulations were performed to calculate the transient velocity & temperature characteristics in the storage tank which were then used for the configuration optimization of the ROPDs. 2D-axisymetric fluid domain (Fig. 3) adapted to cylindrical TES tanks was employed to represent the geometry of the tested prototype. Water was used as the working HTF. Its temperature-dependent thermophysical properties using polynomial fitting correlations are listed in Table 1.

Pro	operty Unit	Fitting correlation T (K)
Density	ρ kg m ⁻³	$739.57 + 1.9908T - 0.0038T^2$
Specific	the heat C_p J kg ⁻¹ K ⁻¹	$5438.6 - 8.07T + 0.0129T^2$
Therma	l W m ⁻¹ K ⁻¹	$-0.7888 + 0.0077T - 1 \times 10^{-5}T^{2}$
conduct	ivity λ	
Viscosit	ty μ N s m ⁻²	$0.0215 - 1.198 \times 10^{-4} T + 1.701 \times 10^{-7} T^2$

Table.1. Thermo-physical properties of water (293.15 K<T<333.15 K) [engineering toolbox]

ANSYS FLUENT 19.1 was used to solve the Navier–Stokes equations and the energy equation. Pressure-based solver was used in numerical calculations. The operational pressure was fixed at 101325 Pa. The gravity effect was considered while the viscous heating effect was neglected. These open papers stated that the flow inside the thermocline-based SMT tank is naturally turbulent when the inlet Reynolds number is over 4000 [Castell et al.,2010; Taher et al., 2019]. In this investigation, the k- ε RNG t turbulent model is selected to study the flow behavior and thermal stratification. Such model is recommended by other authors who had the approximative operating condition (Reynolds number, HTF kind, working temperature) [Assari et al., 2018]. The SIMPLE algorithm was applied for the pressure-velocity coupling. The standard spatial discretization scheme was used for pressure, while the second-order upwind scheme was used for the discretization of momentum, energy, k and ε equations.

The initial conditions and the boundary conditions were set as follows, taking the charging process as example (Fig. 3).

(i) The initial conditions were zero velocities and uniform temperature.

$$v_x = 0; v_y = 0; T = T_c$$
 (1)

(ii) At inlet, hot HTF (T_h) was injected into the upper port at t_0 moment (the charging started):

$$v_y|_{y=in} = -v_{in} \tag{2}$$

89

$$T_{y}|_{y=in} = T_{h} \tag{3}$$

Fully developed velocity profile [Laloui et Loria, 2020] written in a complied user defined function was used for v_{in} , as shown in Eq. (4):

$$v_{in} = v_{in_{max}} \left[1 - \frac{(x^2)}{\left(\frac{d}{2}\right)^2} \right], x \in [0, \frac{d}{2}]$$
 (4)

Where d is the diameter of the inlet and $v_{in_{max}}$ is twice of the mean inlet velocity.

(iii) Zero gauge pressure outlet at the (bottom) outlet port:

$$P_{out} = 0 \text{ Pa} \tag{5}$$

(iv) Axis boundary for *y*-axis (*x*=0):

$$v_{normal} = 0 \tag{6}$$

(v) Other walls were all set as adiabatic (i.e., negligible heat loss to the environment), including top, bottom and lateral walls of TES tank as well as upper and bottom plates. For example, at the lateral wall:

$$\frac{\partial T}{\partial x}\Big|_{x=\frac{D}{2}} = 0 \tag{7}$$

The setting of boundary conditions for the discharging process was almost the same, except that the positions of inlet and outlet were interchanged. The initial condition for the fluid domain was zero velocities and uniform temperature at T_h .



Fig.3 2D axisymmetric fluid domain, the mesh and the boundary conditions. Charging process for example.

Simulations were performed under transient state with fixed time step of 0.001 s. The solution was considered to be converged when (i) normalized residuals were smaller than 10^{-5} for mass, momentum, k and ε equations and 10^{-7} for energy equation, and (ii) the inlet gauge pressure become constant (less than 0.5% variation).

The structured quad mesh was used, with inflation layers and additional local refinement for parts with large flux gradient. A grid independence study was performed with the total number of elements increasing from 0.2 million to 0.38 million. The comparison on the calculated mean passage time \bar{t} (Eq. 26) and on the inlet gauge pressure was shown to be inferior to 0.011% when the number of elements was higher than 0.2 million. After considering both the calculation time and precision, the mesh used in this study had approximately 0.23 to 0.25 million elements relying on different ROPD configurations. Simulations were then carried out by a workstation with 2.2 GHz Intel processors (Intel Xeon CPU E5-2630 v4) and 32.0 GB RAM. An average

calculation time of 3 days was required for each simulation of fully charging or discharging. Compared to full 3D simulation of the SMT storage tank in another study reported in the next chapter (18 days for each simulation), the calculation time has been significantly reduced due to the simplification from 3D to axisymmetric 2D model.

The used CFD model was validated by comparing with our experimental results on the cylindrical SMT storage tank (conditions: T_h =333.15 K, T_c =293.15 K, $Q_{in} = 1$ L min⁻¹, initial upper & bottom ROPDs, charging process). The temperature profiles at 8 positions of the central axis measured by TC_1-8 were compared with the simulated data, as shown in Fig. 4. In general, good agreement on the global tendency of the thermocline movement between the simulation and experimental results could be observed.



Fig.4 Comparison of the simulation and experimental results on the evolution of temperature profiles inside the cylindrical TES tank. Conditions: T_h =333.15 K, T_c =293.15 K, $Q_{in} = 1$ L min⁻¹, initial upper & bottom ROPDs, charging

4.2.4 Parameter definition and performance indicator

The theoretical charging/discharging time of the storage tank ($t_{residence}$) is defined as the fluid residence time (plug flow) of the storage tank, calculated by the ratio of the total tank volume ($V_{tank}=11.2$ L) and the charging/discharging volume flowrate (Q_{in}):

$$t_{residence} = \frac{V_{tank}}{Q_{in}} \tag{8}$$

Similarly, the theoretical charging/discharging time of the upper/bottom manifold $(t_{manifold})$ could be calculated by Eq. (9).

$$t_{manifold} = \frac{V_{manifold}}{Q_{in}} \tag{9}$$

Where $V_{manifold}$ the volume of the upper/bottom manifold (0.89 L).

The dimensionless charging/discharging time t^* is calculated as the proportion of the real operating time t of energy storage/release to the theoretical residence time of the tank ($t_{residence}$).

$$t^* = \frac{t}{t_{residence}} \tag{10}$$

Similarly, the instantaneous temperature value is dimensionalized for easier comparison.

$$T^* = \frac{T - T_0}{T_h - T_c} \tag{11}$$

Where T_0 is the reference or dead-state temperature [Haller et al., 2009; Mawire et Taole, 2013], which was set to be 293.15 K for this study.

The dimensionless thermocline volume $V^*_{thermocline}$ is defined as the volume proportion of the storage tank occupied by the thermocline, as shown in Eq. (12). Here the threshold values $[T_{20}; T_{80}]$ were used to identify the temperature borders or fronts of the thermocline zone [Hoffmann et al., 2017], calculated by Eqs. (13) and (14).

$$V^*_{thermoclin} = \frac{V_{[T_2;T_{80}]}}{V_{tank}}$$
(12)

$$T_{20} = T_c + 20\% \times (T_h - T_c)$$
(13)

$$T_{80} = T_c + 80\% \times (T_h - T_c) \tag{14}$$

The Reynolds number at the inlet port and at the tank body can be calculated by Eqs. 15 and 16, respectively.

$$Re_{in} = \frac{\rho \bar{\nu}_{in} d}{\mu} \tag{15}$$

$$Re_{tank} = \frac{\rho \bar{v}_{tank} D}{\mu} \tag{16}$$

93

For simplification, the physical properties (ρ and μ) of water at 333.15 K were used to calculate the Re_{in} whereas Re_{tank} is estimated by the mean temperature (313.15 K) of charging process.

The bulk Froude number (Fr), sometimes used as an indicator to evaluate the performance of flow diffusors for SMT storage tanks [Kocijel et al.2020], is calculated by Eq. (17).

$$Fr = \frac{\bar{v}_{in}}{\sqrt{dg(\rho_c - \rho_h)/\rho_c}} \tag{17}$$

Where ρ_c is the density of cold fluid at 293.15 K, ρ_h is the density of hot HTF ranging from 323.15 K to 343.15 K.

Performance indicators based on both first and second law of thermodynamics are introduced to evaluate the performance of the SMT storage tank, including the charging/discharging efficiency (η) , the charging/discharging exergy (Ex) and the discharging exergy efficiency (η_{ex}) .

The energy storage or charging efficiency (η_{ch}) presents the ratio of the effective storage energy to the overall inflowing energy to the storage tank [Zanganeh etal., 2015]:

$$\eta_{ch} = \frac{E_{in} - E_{out}}{E_{in}} = \frac{\int_0^t \dot{m} C_p (T_{in} - T_{out}) dt}{\int_0^t \dot{m} C_p (T_{in} - T_0) dt}$$
(18)

Where \dot{m} is the mass flow rate and E is the transported energy through the inlet/outlet port.

The energy release or discharging efficiency (η_{dis}) indicates the ratio between the released energy from the storage tank and the overall stored energy:

$$\eta_{dis} = \frac{E_{out} - E_{in}}{E_{stored}} = \frac{\int_0^t \dot{m} c_p \left(T_{out} - T_{in}\right) dt}{E_{stored}} \tag{19}$$

For charging, the inlet exergy (Ex_{in}) and outlet exergy (Ex_{out}) are defined as [Cheng et Zhai, 2018]:

$$Ex_{in} = \int_0^t \dot{m} c_p \left[(T_{in} - T_0) - T_0 \left(\ln \frac{T_{in}}{T_0} \right) \right] dt$$
(20)

$$Ex_{out} = \int_0^t \dot{m} c_p \left[(T_{out} - T_0) - T_0 \left(\ln \frac{T_{out}}{T_0} \right) \right] dt$$
(21)

94

The transferred exergy during the charging or discharging is calculated by Eq. (22) or Eq. (23), respectively.

$$Ex_{ch} = Ex_{in} - Ex_{out} = \int_0^t \dot{m} c_p \left[(T_{in} - T_{out}) - T_0 \left(\ln \frac{T_{in}}{T_{out}} \right] \right] dt$$
(22)

$$Ex_{dis} = Ex_{out} - Ex_{in} = \int_0^t \dot{m} c_p \left[(T_{out} - T_{in}) - T_0 \left(\ln \frac{T_{out}}{T_{in}} \right) \right] dt$$
(23)

The maximum transferable exergy for the discharging process (Ex_{max}) is calculated by Eq. (24):

$$Ex_{max} = \int_0^t \dot{m} c_p \left[(T_{max} - T_0) - T_0 \left(\ln \frac{T_{max}}{T_0} \right) \right] dt$$
(24)

Where T_{max} is estimated as the mean value of all thermocouples inside the TES tank at the beginning of the discharging process.

The exergy efficiency η_{ex} is calculated by Eq. (25) [Carmona et al., 2020; Roos and Haselbacher, 2021].

$$\eta_{ex} = \frac{Ex_{dis}}{Ex_{max}} \tag{25}$$

4.2.5 Uncertainty analysis

An uncertainty analysis was performed following the method of Moffat [1988]. The experimental result was assumed to be obtained from a set of measurements using a data interpretation program represented by

$$R = R(X_1, X_2, X_3, \dots, X_N,)$$
(26)

Where X_i were these measured variables, i.e., the volume flow rate Q, the inlet/outlet temperature T_{in}/T_{out} and the the initial temperature of the tank T_c .

The effect of each measurement uncertainty for the one variable would be:

$$\delta R_{X_i} = \frac{\partial R}{\partial X_i} \delta X_i \tag{27}$$

Where one more independent variables are presented in the function R, the individual terms are combined by a root-square equation.

$$\delta R = \left\{ \sum_{i=1}^{N} \left(\frac{\partial R}{\partial X_i} \delta X_i \right)^2 \right\}^{\frac{1}{2}}$$
(28)

Then the relative uncertainty is calculated by:

Uncertainty =
$$\frac{\delta R}{R} = \left\{ \sum_{i=1}^{N} \left(\frac{1}{R} \frac{\partial R}{\partial X_i} \delta X_i \right)^2 \right\}^{\frac{1}{2}}$$
 (29)

Kalapala and Devanuri also gave the good example for calculation of uncertainty in energy stored or discharged [Kalapala and Devanuri,2020]. In our condition, as for η_{ch} , the indepent variables in the experiments includig mass flow rate, inlet and outlet temperature:

For example, the uncertainty of Fr can be calculated using the method [Kalapala and Devanuri,2020; Moffat,1988] given by Eq.30.

$$\frac{\delta Fr}{Fr} = \frac{\partial Fr}{\partial \bar{v}_{in}} \frac{\delta \bar{v}_{in}}{Fr}$$
(30)

Flow meters were calibrated for flow rate $0.05 - 5 \text{ Lmin}^{-1}$ with the accuracy at 1%. Therefore, the uncertainty Fr number is estimated to be 1%.

4.3. ROPD: CFD-based optimization and experimental verification

This section introduces firstly the optimization procedure for upper and bottom ROPDs. Then the numerical results for different optimization steps are displayed and discussed. An experimental verification is provided at last to showcase the effectiveness of the optimization method.

4.3.1 Optimization strategy

The optimization strategy for the upper and bottom ROPDs is mainly based on the CFD-based algorithm proposed and developed in our earlier study [Lou et al., 2020]. One minor adaptation is the transposition from 2D to axisymmetric 2D to better fit the cylindrical geometry and the ring-opening configuration. The basic idea of the optimization is to adjust the surface areas of the ring openings on the disc-like plate so that the thermal front could pass through every opening at almost the same time during a charging/discharging process. Here we define the passage time t_i as the time interval between t_0 (starting point) and the moment when the thermal front (T_{80} for charging and T_{20} for discharging, respectively) just traverses the i^{th} opening of the (upper or bottom) plate. The mean passage time (\bar{t}) for all ring-shape openings can be calculated by Eq. 31.

$$\bar{t} = \frac{\sum_{i=1}^{N} t_i}{N} \tag{31}$$

Where *N* is the number of openings on the plate, being 3 in this study.

Therefore, the optimality criterion can be written as:

$$t_i = \bar{t} \ (i = 1, 2, 3) \tag{32}$$

suggesting that the thermally piston flow could be realized when the passage times (t_i) for all openings are identical.

The dimensionless surface of each ring-shape opening α_i is calculated as:

$$\alpha_i = \frac{s_i}{s_{total}} = \frac{s_i}{\pi D^2/4}$$
(33)

Where s_i is the surface area of the *i*th ring-shape opening, and s_{total} is the cross-sectional area of the storage tank.

An iterative program is written such that α_i is modified according to the deviation of its passage time from the mean value $(t_i - \bar{t})$, as shown in Eq. (34). Specifically for *j* step, the α_i will be reduced at step j + 1 if $t_{i,j}$ is found smaller than \bar{t}_j . Otherwise, α_i will be enlarged at step j + 1 if $t_{i,j}$ is observed larger than the mean \bar{t}_i at step *j*.

$$\Delta \alpha_{i,j} = \alpha_{i,j+1} - \alpha_{i,j} = \gamma \left(t_{i,j} - \bar{t}_j \right) \tag{34}$$

Where γ is an adjusting factor chosen to ensure a fast and stable convergence. With this variation rule, the sum of the surface areas for the all three openings is kept constant from one iteration to the next. In practice, the standard deviation of passages times (STD_{t_i}) is monitored for each iteration, indicating the closeness of the t_i values.

$$STD_{t_i} = \sqrt{\frac{1}{N-1} \sum_{i=1}^{N} (\frac{t_i - \bar{t}}{\bar{t}})^2}$$
(35)

Major steps for the optimization strategy is schematically shown in Fig. 5 and briefly described below.

97

S1: A tank geometry having simple inlet and outlet ports (without any ROPD) is firstly simulated as a reference case for comparison. Without the adapted structure to properly handle the momentum-dominated flow, the thermal jet is supposed to cause strong convective heat transfer, resulting in the degradation of temperature stratification.

S2: The optimization procedure begins with the storage tank equipped with two initial ROPDs, representing the tank geometry as shown in Fig. 1(a). The initial configuration of the ROPD has three openings with identical surface area (as illustrated in Fig. 7).

S3: The optimization algorithm described above is executed to optimize the opening configuration on the upper ROPD. The optimization is considered to be complete when the decline of thermocline volume $V_{thermclin}^*$ (cf. Eq. 12) at t_{fin}^* moment is smaller than 1% from *j* to j + l iteration. t_{fin}^* indicates actually the moment when the hot front (T_{80} in this study) has totally passed the upper manifold.

S4: The same combination of ROPDs at the final iteration of S3 (optimized upper one and initial bottom one) is simulated for the discharging process.

S5: The optimization algorithm is executed again for adjusting the size distribution of ring openings on the bottom plate subjected to the discharging process. Using this optimization procedure, the combination of the optimized upper and bottom ROPDs could be determined.



Fig. 5. CFD-based optimization strategy for upper and bottom ROPDs.

4.3.2. Numerical results

The effectiveness of the optimization algorithm and procedure was firstly examined by a case study under the operating condition of T_h =333.15 K, T_c =293.15 K, and $Q_{in} = 1$ L min⁻¹. Six iterations were needed to optimize the upper ROPD while five iterations were needed for the bottom ROPD. Fig. 6 shows the decreased values of $V_{thermocline}^*$ and $STD_{t_i^*}$ along with the optimization iteration, showing that the proposed algorithm is rather effective. The initial and optimized geometry and dimensions of the ROPDs are presented in Fig. 7.



Fig. 6. Dimensionless thermocline volume $V_{thermocline}^*$ at t^*_{fin} and standard deviation of passage times STD $_{t_i}$ versus the optimization iteration. (a) Upper ROPD during the charging; (b) bottom ROPD during the discharging. Condition: T_h =333.15 K, T_c =293.15 K, $Q_{in} = 1$ L min⁻¹

Provide Provid	OF	pening centerline location r ₁ , r ₂ , r ₃] (mm)	Opening width [<i>w</i> ₁ , <i>w</i> ₂ , <i>w</i> ₃] (mm)	Opening surface [s ₁ , s ₂ , s ₃] (mm ²)
	Initial ROPD	D per [16.2, 48.5, 80.8] om	[10.378, 3.460, 2.076]	[1054, 1054, 1054]
	Optimized upper ROPD		[2.512, 3.642, 3.566]	[254, 1105, 1803]
Opening 3	Optimized bottom ROPD		[2.362, 3.644, 3.570]	[240, 1110, 1812]
e.g., Initial ROPD				

Fig. 7. Geometry and dimensions of the initial and optimized ROPDs. Condition: T_h =333.15 K, T_c =293.15 K, $Q_{in} = 1 \text{ L min}^{-1}$

In order to provide a deeper insight into the fluid flow and heat transfer behaviors subjected to different inlet/outlet structures, the virtual streamslice superimposed over velocity contours and the fluid temperature cartographies at $t_{manifold}^*$ moment of charging (or discharging) process are displayed in Fig. 8. Five combinations of upper & bottom diffusers are actually simulated and compared, including simple inlet/outlet ports (S1 for charging), initial upper & bottom ROPDs (S2 for charging), optimized upper & initial bottom ROPDs (S3 for charging), optimized upper & initial bottom ROPDs (S4 for discharging) and optimized upper & bottom ROPDs (S5 for discharging).

Vortices are clearly visible on the velocity field of S1, due to the momentum dominated flow injected from the upper port. The density difference due to the temperature difference between the hot and cold fluids induces the buoyancy force that will interact with the momentum force, resulting in the rising of hot fluid in the form of plumes. The buoyant plumes of hot fluid will then be cooled down by the surrounding cold fluid via the convective heat transfer. This jet entrainment phenomenon characterized by the appearance of buoyant plumes will be visualized and analyzed in detail in the next chapter. It can be observed from Fig. 8 (S1) that the thermal jet could penetrate into a large depth of the tank (y=-120 mm) and the thermocline occupies a large volume (almost 30%) of the storage tank.

The installation of the initial upper ROPD could restrict, to some extent, the impact of the inflowing thermal jet within the upper manifold, where the vortices and the flow recirculation could be clearly seen (Fig. 8 (S2)). Nevertheless, a small stream of the momentum-dominated hot flow 'leaks' from the upper manifold into the middle storage zone through the first opening close to the central axis of the tank. This "leakage" will cause the invasion of the thermal front to a certain depth of the middle storage zone (y=-72 mm) at $t^*_{manifold}$ moment while a large part of the upper manifold is still occupied by the cold fluid, as shown in the temperature cartography of Fig. 8 (S2).

This disruptive flow leakage could be largely eliminated by optimizing the configuration of the upper ROPD, i.e., by narrowing the first ring opening near the central axis while widening the other two. The unexpected factors disturbing the temperature stratification (e.g., forced convection, buoyancy-driven eddy advection, instabilities of density/velocity, etc.) are by and large limited within the upper manifold. Therefore, the thermocline zone shows a thin and flattened shape, maintaining the stable temperature stratification, as displayed in Fig. 8 (S3).

Shown in Fig. 8 (S4) is actually the inverse (discharging) operation of the same ROPDs combination for S3. But the fluid and temperature fields are similar to those of S2 because the bottom ROPD for managing the cold inflowing jet has not been optimized. The 'leaking' stream of the inflowing jet still invades the middle storage zone but the (cold) thermal front penetrates with a smaller depth (y=-328 mm) compared to that for charging (S2). This is because the gravity force could better partly alleviate the strong inertia force of the entering jet.

Fig. 8 (S5) displays the flow and temperature fields of the storage tank equipped with the optimized bottom ROPD for the discharging operation, analogous to the inverse of that shown in Fig. (S3). Again, the optimized ring-opening configuration could result in a regular and flat thermocline, thereby successfully maintaining the temperature stratification at a high level.



Fig. 8. Velocity contour, streamslice and temperature cartography of the 2D axisymmetric fluid domain at $t^*_{manifold}$ moment for different combinations of upper and bottom ROPDs. Condition: T_h =333.15 K, T_c =293.15 K, $Q_{in} = 1$ L min⁻¹

In order to explore in depth the effect of ROPD configuration on the alleviation of the thermocline decay, the time-evolution of the dimensionless thermocline volume $v_{thermocline}^{*}$ for each combination of ROPDs is plotted in Fig. 9 and briefly commented as follows. Logically, the $v_{thermocline}^{*}$ for S1 case is the largest at almost all time. The increasing $v_{thermocline}^{*}$ curve for S4 is very similar to that for S2 but with a slight smaller value at the same t^{*} moment. This is because of the positive contribution of the gravity effect as discussed above. The $v_{thermoclin}^{*}$ curves for S3 and S5 increase first and then stabilize around the $t_{manifold}^{*}$ moment, indicating that the thermocline expansion due to the inflowing thermal jet could be largely limited in the manifold. After $t_{manifold}^{*}$, the thermocline enclosed by the thermal fronts $[T_{20}; T_{80}]$ traverses the optimized ringopening plate and arrives calmly and stably into the middle storage zone.



Fig. 9 Time-evolution of the dimensionless thermocline volume $V_{thermocline}^*$ for different tested combinations of ROPDs. Condition: T_h =333.15 K, T_c =293.15 K, $Q_{in} = 1$ L min⁻¹

Moreover, concerning the maximum gauge pressure (Condition: T_h =333.15 K, T_c =293.15 K, $Q_{in} = 1 \text{ L min}^{-1}$) at inlet port with two initial flow distributors is 43 Pa. the increased value of pressure drop is at less 3% in comparison of the SMT without any flow distributor.

4.3.3 Experimental verification

To verify the numerical results obtained, initial and optimized ROPDs were fabricated in-house and experimental testing has been performed using the cylinder storage tank. Three combinations of ROPDs were tested and compared including S1 (without ROPDs), S2 (initial upper & bottom ROPDs) and S5 (optimized upper & bottom ROPDs), under the same condition for the optimization ($T_h = 333.15$ K, $T_c = 293.15$ K, $Q_{in} = 1$ L min⁻¹) for both charging and discharging operations. The temperature profiles measured by TC_2-6 at the axis of the middle storage zone for the three tested combinations during the charging process are plotted in Fig. 10, all showing the typical shape of S-curve as displayed by the top explicative image. Four periods of a thermocline charging operation could then be identified based on the climbing trend

of the T^* curve, including (i) initial charging period [t_0 to $t^*(T^* = 0.2)$]; (ii) thermocline period [$t^*(T^* = 0.2)$ to $t^*(T^* = 0.8)$]; (iii) tailing period [$t^*(T^* = 0.8)$] to $t^*(T^* = 0.99)$] and (iv) fully charging period [$t^* > t^*(T^* = 0.99)$]. The temporal demarcation depends actually the threshold temperatures of the thermocline and in this study it is [T_{20} ; T_{80}].

The temperature profile of the perfectly stratified thermal plug-flow is also added in Fig. 10, providing a comparison between ideal and real-world operations. The early rise and the existing slope of the T^* curves at the thermocline period for S1, S2, S5 compared to the ideal profile indicate clearly the existence of the thermocline with a certain thickness due to the inevitable heat transfer between the hot and cold fluids. Note that the rising time $t^*(T^* = 0.2)$ implies the advance speed of (cold) thermal front whereas the duration of the thermocline period (or the slope of T^*) indicates the thickness of the thermocline at the central axis. In this regard, by comparing the T^* curves of the three ROPD combinations, it can be inferred that the extent of stratification degradation are not identical. On one hand, the T^* profile for S5 shows the latest rising moment among the three tested combinations, indicating that the calm cold environment in the middle storage zone could be held for the longest period before the arrival of the thermocline. On the other hand, its slope is also the steepest, implying the fastest passage of the thermocline with the highest temperature gradient.



Fig. 10. Dimensionless temperature T^* versus dimensionless charging time t^* for different tested combinations of ROPDs (S1, S2 and S5). Condition: T_h =333.15 K, T_c =293.15 K, $Q_{in} = 1$ L min⁻¹

The discharging operation has also been performed for these three combinations, and similar general trend could be observed (cf. Fig. A1 in Appendix A). Some more explanations may be found in Appendix A.

In Figs. 11a and 11b plotted the normalized outlet temperature T_{out}^* , also called the cut-off temperature, versus t^* and the charging efficiency η_{ch} versus T_{out}^* for the three tested combinations of ROPDs. Both the steeper slope of the T_{out}^* curve and the higher values of η_{ch} for S5 case indicate the improved thermal performance of the storage tank by equipping the optimized ROPDs. Fig. 11c shows the transferred exergy Ex_{ch} during the charging process versus T_{out}^* . It is worth noting that the Ex_{ch} rises very fast at the final stage of the charging for S1, i.e., when $T_{out}^* > 0.8$. Apparently, the pronounced tailing phenomenon, i.e., very slow rise of the T_{out}^* greatly impacts the charging performance of the storage tank. On one hand, the convective heat transfer due to fluid mixing will cause the irregular and distorted shape of the thermocline, making it much more difficult to timely drain out the cold fluid through the bottom port. On the other hand, the largely prolonged real charging time needed to reach the fully charged condition (tailing period as explained in top image of Fig. 10) results in the non-negligible heat loss to the ambient despite the insulation. Therefore, the fully charged condition of the SMT storage tank is neither easy nor cost-effective to achieve when the impact of the inflowing thermal jet (fluid mixing) cannot be appropriately handled.



Fig. 11 Comparison on the charging performance of the SMT storage tank with different combinations of ROPDs. (a) Dimensionless outlet temperature T_{out}^* versus dimensionless charging time t^* ; (b) Charging efficiency η_{ch} versus T_{out}^* ; (c) Transferred exergy Ex_{ch} versus T_{out}^* . Condition: T_h =333.15 K, T_c =293.15 K, $Q_{in} = 1$ L min⁻¹



Fig. 12. Comparison on the discharging performance of the SMT storage tank with different combinations of ROPDs. (a) Dimensionless outlet temperature T_{out}^* versus dimensionless discharging time t^* ; (b) Discharging efficiency η_{dis} versus T_{out}^* ; (c) Transferred exergy Ex_{dis} versus T_{out}^* . Condition: T_c =293.15 K, T_h =333.15 K, $Q_{in} = 1$ L min⁻¹.

Figs. 12a and 12b show T_{out}^* versus the dimensionless discharging time t^* and the discharging efficiency η_{dis} versus T_{out}^* for the three tested combinations of ROPDs.

Similar conclusion could be drawn, i.e. the discharging performance of the storage tank can be largely improved by the installation of the optimized ROPDs. The prominent mixing motion for S1 case causes significant convection heat transfer, thereby expanding the thermocline and lowing the discharging efficiency. The transferred exergy Ex_{dis} versus T_{out}^* is plotted on Fig. 12c, showing that for a given T_{out}^* , the Ex_{dis} value for S5 is always the highest among the three tested combinations. No important rising of Ex_{dis} at the final stage of discharging despite the still existing tailing phenomena for the S1 case. This is because the outlet HTF in the ending of discharging process has lower exergy per unit mass compared with outlet HTF in the latest stage of charging process.

In conclusion, both the numerical and experimental results show that the optimized ROPDs are really effective on the improvement of the charging/discharging performance of the SMT storage tank.

4.4. Influences of operating parameters: a sensitivity study

Additional charging and discharging experiments have then been performed for the same SMT storage tank and the three combinations of ROPDs (S1, S2 and S5) under different flowrate (Q_{in} =0.3, 0.5, 1.0 and 1.5 L min⁻¹) and hot fluid temperature (T_h = 323.15, 333.15 and 343.15 K) conditions. The aim of this part of study is on one hand, to clarify the separate influence of the operating parameter as well as their combined effects on the temperature stratification, and on the other hand, to test the robustness of the optimized ROPDs subjected to varied operating conditions. The obtained experimental results are reported and discussed in the following sub-sections.

4.4.1 Local temperature evolutions

The normalized value of the temperature measured by TC_6, TC_4 and TC_2 are plotted against t^* in Figs. 13 (for different T_h) and 14 (for different Q_{in}), monitoring the movement of thermocline in different locations of the middle storage zone during the charging operation.

By examining the Fig. 13, one may observe that all solid T^* curves (S5) have the largest slope, the dashed curves (S2) come as the second, and the cross marker curves (S1) are the flattest. The stabilization of the temperature stratification by the ROPDs could be again highlighted. The impacts of the T_h (the temperature difference between the hot and cold fluids) on the charging performance of the storage tank are not the same when different ROPDs are installed. Higher T_h leads to relatively later rise and steeper slope of the T^* curve for S1 and S2. This is because the larger density difference due to the larger temperature difference results in the larger buoyancy force, contributing positively to balance out the momentum effect of the inflowing thermal jet. Detailed analysis and discussion on the effect of temperature difference on the buoyant plumes may be found in the next chapter. It can also be observed from Fig. 13 that the T^* curves of S5 case are almost overlapping, indicating that when optimized ROPDs are used, the temperature difference has little impact on the thermocline behaviors in the middle storage zone of the tank. The T^* curves for the discharging operation analogous to Fig. 13 is shown in Fig. A2 of the Appendix. Similar tendency and findings could be reached thus will not be further commented here.



Charging process

Fig. 13 Effect of the inlet temperature T_h on the thermocline advance in the middle storage zone of the tank equipped with different combinations of ROPDs. (a) T^* profile at TC_6; (b) T^* profile at TC_4; (c) T^* profile at TC_2. Condition: T_c =293.15 K, $Q_{in} = 1$ L min⁻¹, charging.
The sensitivity/robustness of the optimized ROPDs over the operating flow rate is shown in Fig. 14. In general, the low charging flow rate (Q_{in}) renders a steeper slope of the T^* curve, indicating smaller impacts of the thermal jet and more regular and calmer thermocline evolution. For this condition, the benefit of ROPDs on the charging performance improvement is marginal, this aspect will be quantified later when analyzing some global performance indicators. For higher flow rates of our tested range $(Q_{in} > 1.0 \text{ Lmin}^{-1}; Re_{tank} > 167)$, simple inlet/outlet port (S1) is apparently incapable of handling the strong momentum-force of the inflowing thermal jet, indicated by the early rising as well as the lengthened tailing period shown in Fig. 14.

For $Q_{in} > 0.5 \text{ Lmin}^{-1}$, the T^* curves of S1 case rise always the earliest, followed by those of S2 case while those of S5 case rise the latest. This is in line with the simulation results as indicated by the temperature cartography in Fig. 8. Nevertheless for $Q_{in}=0.3 \text{ Lmin}^{-1}$, later rise of S2 than S5 and S1 has been observed by examining Fig. 14. The streamline of inlet thermal jet at smaller flow rate 0.3 Lmin⁻¹ must be more roundabout than that at larger flow rate. This is because that inflowing turbulence dampened quickly and no obvious vertical mixing appeared at lower flow rate. Buoyancy-driven eddy advection makes the horizontal mixing much more predominant for $Q_{in}=0.3 \text{ Lmin}^{-1}$. Accordingly, the size distribution of annular openings of S2 exacerbates further the difficulties for heat transfer issuing from the inlet thermal jet having less inertia force.

For S5 case, the ranking of rising time is 0.3 < 0.5 < 1.0 < 1.5 L min⁻¹. This is because, with the optimized size of ring-openings (narrowed opening close to the center and widened close to the lateral wall), the central vertical injecting flow is obstructed by the upper plate but directed horizontally to fully occupy the upper manifold. And the higher the Q_{in} , the stronger the inertia force and thereby the injecting flow is more likely to pass earlier through the lateral opening. The (cold) thermal front will presents a curved (parabolic) shape instead of the flat shape, resulting in the delayed advance of the thermal front at the central axis.

For S1 case, the ranking of rising time is firstly 1.5 < 1.0 < 0.3 < 0.5 L min⁻¹ (TC_6) and later 1.5 < 0.3 < 1.0 < 0.5 L min⁻¹ (TC_2). This is due to the interaction of inertia force and the buoyancy force and their relative strengths throughout the movement of the thermocline.

For S2 case, the ranking of rising time is 1.0 < 1.5 < 0.3 = 0.5 L min⁻¹ while the case at 0.5 L min⁻¹ has the steeper slope. We supposed that the interaction between the flow distributor and inlet flow rate must determine the evolution of thermal jet and its streamline inside the SMT system. For example, The raison why rising time at 1.5 L

 min^{-1} is later than that of 1.0 L min⁻¹ is that the lateral dispersion (regime 4 in chapter 5) issuing from the opening 1 may much serious than that at 1.0 L min⁻¹.

More information and detailed discussions on the shape evolution of thermocline subjected to different opening configurations under certain *Re* condition may be found in our earlier study [Lou et al., 2020]

The T^* curves for the discharging operation analogous to Fig. 14 is shown in Fig. A3 of the Appendix. Similar tendency but with small differences may be found, as being further commented in the Appendix. In conclusion, the effectiveness of the optimized ROPDs is thereby robust for both the lower and higher working flow rates under our tested range.



Fig. 14. Effect of the charging flow rate Q_{in} on the thermocline advance in the middle storage zone of the tank equipped with different combinations of ROPDs. (a) T^* profile at TC_6; (b) T^* profile at TC_4; (c) T^* profile at TC_2. Condition: T_h =333.15 K, T_c =293.15 K, charging

4.4.2 Energy and exergy efficiencies

To further explore the global performance of the SMT storage tank, energy and exergy efficiencies are analyzed and discussed in this sub-section. To facilitate the comparison, the thermal threshold is taken to be 20%, i.e., the cut-off $T_{out}^* = 0.80$ for the charging and $T_{out}^* = 0.20$ for the discharging, respectively.

Fig. 15 shows the calculated charging or discharging efficiency of the storage tank base on different combinations of ROPDs and under various testing conditions. The bulk *Fr* numbers are also calculated and shown in Fig. 15 for analysis. Among the three tested combinations of ROPDs, S5 is shown to be the best that always renders the highest charging or discharging efficiency throughout all tested ranges of flow-rate and working temperature differences. Compared to S2, η_{ch} could be improved by up to 3.6% where as η_{dis} could be improved by up to 2.9% (condition: T_h =323.15 K, T_c =293.15 K, $Q_{in}=1.0 \text{ Lmin}^{-1}$) by optimizing the size of the ring-openings. Compared to S1, the improvement is much more significant especially under high flow rate condition, reaching 14.5% for η_{ch} and 19.8% for η_{dis} , respectively (condition: T_h =323.15 K, T_c =293.15 K, Q_{in} =1.5 L min⁻¹). The use of initial ROPDs (S2) is beneficial compared to simple inlet/outlet ports (S1) at relatively high flow rate ($Q_{in} \ge 1 \text{ Lmin}^{-1}$) but less performant than optimized ROPDs (S5). Under relatively low flow rate ($Q_{in} \leq 0.5$ L min⁻¹) however, the initial ROPDs might provide counterproductive work (additional flow disturbance caused by the ring-opening plates), indicated by lower values of η_{ch} compared to those of S1.

In Fig. 15, the separate effect of the flow rate or the temperature difference on the global performance of the SMT storage tank can also be figured out. For the charging, the impact of working temperature difference of HTFs is straightforward for all three combinations: a larger temperature difference favors the level of temperature stratification, accompanied with a higher η_{ch} . For the discharging however, the influence of the working temperature difference seems less significant, except for S1 case at relatively high flow rate ($Q_{in} \ge 1 \text{ Lmin}^{-1}$) under which the difference can still be noticed.

Regarding the flow rate, it shows great influence for S1 for both charging and discharging operations. For $Q_{in} \ge 1 \text{ Lmin}^{-1}$, the charging or discharging efficiency of S1 falls sharply. Relatively marginal influence of flow rate may be observed for S2 and S5 especially for the discharging operation. The installation of ROPDs, especially after configuration optimization, could efficiently manage the flow and temperature disturbances brought by the strong momentum effect of the inflowing thermal jet.



Fig. 15 Charging efficiency (a) and discharging efficiency (b) of the SMT storage tank equipped with different combinations of ROPDs. Conditions: T_h =323.15, 333.15, and 343.15 K, T_c =293.15 K, Q_{in} = 0.3, 0.5, 1.0 and 1.5 L min⁻¹

Moreover, by examining the values of bulk Fr number, some hints may be obtained on the ranges of operating parameters for the SMT storage tank for stable and calm thermocline evolution. Note that the Fr number considers both the inertia force (inlet flow rate) and the buoyancy force (density/temperature difference). For small Frnumbers (e.g., Fr<3), the flow and temperature disturbances induced by the inflowing thermal jet are relatively small so that simple inlet & outlet ports could work well. Nevertheless, for higher Fr number conditions (e.g., Fr>3), additional structuration of the inlet & outlet port/manifold has to be considered to alleviate the negative impact of the momentum-dominated thermal jet on the thermocline decay. Furthermore, it seems that the η_{dis} value (and also the η_{Ex} as will be shown in Fig. 16) of the storage tank with simple inlet & outlet ports decreases monotonously with the increasing Fr number. For η_{ch} value however, its correlation with the Fr number is less straightforward but the global trend is similar. In short, the higher the Fr number, the geometry adaptation of the flow diffusers is more needed, even indispensable, by using some optimization algorithms as done in this study. In that way, the SMT storage tank could be operated under some harsh working conditions with a good performance.

Fig. 16 shows the calculated exergy efficiency η_{Ex} (cf. Eq. 25) of the storage tank, evaluating the usefulness of the stored energy in the TES tank by considering both the energy quantity and quality [Advaith et al., 2020]. Similar rankings of three ROPD combinations and similar influences of flow rate or working temperature difference as

those for η_{dis} could be observed thus will not be repeated here. A noticeable feature is that $Q_{in} = 1 \text{ Lmin}^{-1}$ ($Re_{tank} = 167$, the design point) turns out to be the optimal working flow rate for the optimized ROPDs from the viewpoint of exergy efficiency. While for S1 and S2, the optimal working flow rate is still $Q_{in} = 1 \text{ Lmin}^{-1}$. In addition, better performance could achieved by S5 under a higher working temperature difference ($T_h = 343.15$ K) than under its design point ($T_h = 333.15$ K), highlighting again the benefit of larger temperature difference of working HTFs in practice.



Fig. 16. Exergy efficiency η_{Ex} of the SMT storage tank for different combinations of ROPDs during the discharging process. Conditions: T_h =323.15, 333.15, and 343.15 K, T_c =293.15 K, $Q_{in} = 0.3$, 0.5, 1.0 and 1.5 L min⁻¹

4.5. Conclusion and prospects

In this study, CFD simulations and experimental measurements have been performed for a cylindrical SMT storage tank subjected to charging or discharging operations. ROPDs with or without optimized configurations have been installed in the tank and their ability on the alleviation of disturbances induced by the thermal jet has been tested. Main conclusions are summarized as follows:

- The optimizing algorithm proposed in [Lou et al., 2020] has been experimentally validated and proved to be effective, efficient, robust and simple to implement.
- A larger working temperature difference between the HTFs favors the level of temperature stratification for the charging process. For the discharging, the influence of working temperature difference is less obvious within the range of our tested conditions.
- The charging or discharging efficiency of the storage tank with simple inlet/outlet ports falls sharply when the inlet flow rate is higher than 1 L min⁻¹ (*Re_{tank}*>167). When ROPDs are used, the influence of the inlet flow rate could be limited to a certain extent.
- The combined effect of inlet flow rate (inertial force) and working temperature difference (buoyancy force) could be characterized by the bulk *Fr* number. A stable range of the operating parameters for the SMT storage tank (with simple inlet/out ports) has been determined. For small *Fr* numbers (e.g., *Fr*<3), the flow and temperature disturbances induced by the inflowing thermal jet are relatively small. For higher *Fr* number conditions (e.g., *Fr*>3), additional structuration of the inlet & outlet port/manifold and geometry adaptation are essential to maintain the good performance of the SMT storage tank operated under some harsh working conditions.
- A series of parametric studies performed demonstrates that the combination of the optimized ROPDs could effectively mitigate the strong momentum effect of thermal jet thereby significantly improve the global performances of storage tank for all the tested cases. For example, compared to the simple inlet/outlet ports, the exergy efficiency (η_{Ex}) could be improved from 56.3% to 84.2% (condition: T_h =323.15 K, T_c =293.15 K, $Q_{in} = 1.5$ L min⁻¹).

As prospects, a deeper understanding on the fluid mechanisms of the thermal jet (jet entrainment phenomena) and its influences on the heat transfer characteristics is very much needed. Therefore, the CFD simulations and the experimental measurements of local velocity and/or temperature fields by using intrusive (e.g., TC array in both axial and radial coordinates) or non-intrusive optical-based methods (e.g., particle image velocimetry, infrared camera, etc.) are our on-going work. The extension of the optimization algorithm [Lou et al. 2020] to real 3D for perforated orifice baffle distributors fitted for rectangular SMT storage tanks is also our future work. Moreover, the effectiveness of diffuser designs under (partial) cyclic operations is certainly an interesting aspect to be tested. The effect of inlet fluctuation on the performance of the

SMT storage tank issuing from its integration to the real environment will also be addressed.

Acknowledgement

This work is supported by the French ANR within the project OPTICLINE (ANR-17-CE06–0013) and by the Chinese Scholarship Council (CSC) with the scholarship for Ms. Baoshan XIE (No. 201908430177).

Appendix 4A: Temperature profiles during discharging

The temperature profiles measured by TC_3-7 at the axis of the middle storage zone for the three tested combinations during the discharging process are plotted in Fig. 4A-1 (condition: T_h =333.15 K, T_c =293.15 K, $Q_{in} = 1$ L min⁻¹). Similarly, four periods of a thermocline discharging operation could be identified based on the declining trend of the T^* curve, namely initial discharging period, thermocline period, tailing period and fully discharging period. The T^* profile for S1 displays the earliest decline and the longest tailing among the three tested combinations, showing the serious degradation of the temperature stratification. Strong fluctuation of temperature measurement at TC_3 and TC_4 can be observed from Fig. A1, implying the transient fluid temperature due to the fluid mixing and flow instabilities. Another difference with respect to the charging operation (Fig. 10) is that the T^* curves for S2 and S5 are closer to each other.



Fig. 4A-1. Dimensionless temperature profiles T^* versus dimensionless discharging time t^* for different tested combinations of ROPDs (S1, S2 and S5). Condition: T_h =333.15 K, T_c =293.15 K, $Q_{in} = 1$ L min⁻¹

Fig. 4A-2 shows the temperature profiles at TC_3, TC_5 and TC_7 for three tested ROPD combinations (S1, S2 and S5) with different initial temperature T_h . Different T_h values have clear influence on the thermocline evolution of the storage tank with simple inlet & outlet port (S1): larger temperature difference results in the later decline time and the steeper slope. For other two ROPD combinations (S2 and S5), the influence of T_h on their T^* curves is less noticeable.



Fig. 4A-2. Effect of the initial temperature T_h on thermocline advance inside the SMT storage tank for different combinations of ROPDs. (a) T^* profile at TC_7; (b) T^* profile at TC_5; (c) T^* profile at TC_3. Condition: T_c =293.15 K, $Q_{in} = 1$ L min⁻¹, discharging.

Fig. 4A-3 shows the temperature profiles at TC_3, TC_5 and TC_7 for three tested ROPD combinations (S1, S2 and S5) with different discharging flow rate Q_{in} . For $Q_{in} > 0.5 \text{ L min}^{-1}$, the *T** curves of S1 case decline always the earliest, followed by those of S2 case while those of S5 case decline the latest. For $Q_{in} = 0.3 \text{ L min}^{-1}$, the difference is not obvious. For S1 and S2 cases, the ranking of decline time is $1.5 < 1.0 < 0.5 < 0.3 \text{ L min}^{-1}$. For S5 case however, the *T** curve at $Q_{in} = 0.3 \text{ L min}^{-1}$ declines the latest while the difference between other three is difficult to tell.



Fig. 4A-3. Effect of the discharging flow rate Q_{in} on the thermocline advance in the middle storage zone of the tank equipped with different combinations of ROPDs. (a) T^* profile at TC_7; (b) T^* profile at TC_5; (c) T^* profile at TC_3. Condition: T_h =333.15 K, T_c =293.15 K, discharging

Reference

Advaith S, Parida D, Aswathi K, Dani N, Chetia U, Chattopadhyay K, Basu S. Experimental investigation on single-medium stratified thermal energy storage system. Renew Energ 2020;164:146–155. https://doi.org/10.1016/j.renene.2020.09.092 Assari M, Basirat Tabrizi H, Savadkohy 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. https://doi.org/10.1016/j.seta.2017.12.009 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. https://doi.org/10.1016/j.enconman.2020.113189

Castell A Medrano, Solé C, Cabeza L. Dimensionless numbers used to characterize stratification in water tanks for discharging at low flow rates. Renew Energ 2010;35:2192–2199. https://doi.org/10.1016/j.renene.2010.03.020

Celsiuscity. https://celsiuscity.eu/thermal-energy-storage/

Cheng X, Zhai X. Thermal performance analysis and optimization of a cascaded packed bed cool thermal energy storage unit using multiple phase change materials. Appl Energ 2018;215:566–576. https://doi.org/10.1016/j.apenergy.2018.02.053

Chung JD, Cho SH, Tae CS, Yoo H. The effect of diffuser configuration on thermal stratification in a rectangular storage tank. Renew Energy 2008;33:2236–45. https://doi.org/10.1016/j.renene.2007.12.013.

Dehghan A, Barzegar A. Thermal performance behavior of a domestic hot water solar storage tank during consumption operation. Energy Convers Manag 2011;52(1):468–476. https://doi.org/10.1016/j.enconman.2010.06.075

Engineering toolbox. https://www.engineeringtoolbox.com

European Commission. https://ec.europa.eu/info/energy-climate-changeenvironment/implementation-eu-countries/energy-and-climate-governance-and-

reporting/national-energy-and-climate-plans_en#the-process

Kalapala L, Devanuri J. Energy and exergy analyses of latent heat storage unit positioned at different orientations–An experimental study. Energy 2020; 194:116924. https://doi.org/10.1016/j.energy.2020.116924

Karim MA. Experimental investigation of a stratified chilled-water thermal storagesystem.ApplThermEng2011;31(11–12):1853–1860.https://doi.org/10.1016/j.applthermaleng.2010.12.019

Kocijel L, Mrzljak V, Gla V. Pressure drop in large volumetric heat storage tank radialplatediffuser.JEnergyStorage2020;29:101350.https://doi.org/10.1016/j.est.2020.101350.

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. https://doi.org/10.1016/j.solener.2017.12.062

Grirate H, Agalit H, Zari N, Elmchaouri A, Molina S, Couturier R. Experimental and numerical investigation of potential filler materials for thermal oil thermocline storage. Sol Energy 2016;131:260–274. https://doi.org/10.1016/j.solener.2016.02.035

Haller M, Cruickshank C A, Streicher W, Harrison S, Andersen E, Furbo S. Methodsto determine stratification efficiency of thermal energy storage processes - Review andtheoreticalcomparison.SolEnergy2009;83:1847-1860.https://doi.org/10.1016/j.solener.2009.06.019

Hoffmann J, Fasquelle T, Goetz V, Py X. Experimental and numerical investigation of a thermocline thermal energy storage tank. Appl Therm Eng 2017; 114:896–904. https://doi.org/10.1016/j.applthermaleng.2016.12.053

International Renewable Energy Agency. https://www.irena.org/publications/2020/Nov/Innovation-outlook-Thermal-energystorage

Laloui L, Loria AR. (2020) Analysis and Design of Energy Geostructures: Theoretical Essentials and Practical Application. https://doi.org/10.1016/B978-0-12-816223-1.00008-4

Li G. Sensible heat thermal storage energy and exergy performance evaluations. Renew Sustain Energy Rev 2016;53:897–923. https://doi.org/10.1016/j.rser.2015.09.006.Chandra YP, Matuska T. Stratification analysis of domestic hot water storage tanks: A comprehensive review. Energy Build 2019;187:110–31. https://doi.org/10.1016/j.enbuild.2019.01.052.

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. https://doi.org/10.1016/j.est.2020.101749

Manu K, Deshmukh P, Basu S. Rayleigh-Taylor instability in a thermocline based thermal storage tank. Int J Therm Sci 2016;100:333–345. https://doi.org/10.1016/j.ijthermalsci.2015.10.016

Mawire A, Taole S. A comparison of experimental thermal stratification parameters for an oil/pebble-bed thermal energy storage (TES) system during charging. Appl Energ 2011;88:4766–4778. https://doi.org/10.1016/j.apenergy.2011.06.019

Moffat R. Describing the uncertainties in experimental results. Exp Therm Fluid Sci 1988;1:3e17. https://doi.org/10.1016/0894-1777(88)90043-X.

Ortega-Fernández I, Zavattoni S, Rodríguez-Aseguinolaza J, D'Aguanno B, Barbato M. Analysis of an integrated packed bed thermal energy storage system for heat recovery in compressed air energy storage technology. Appl Energ 2017;205:280–293. https://doi.org/10.1016/j.apenergy.2017.07.039

Roos P,Haselbacher A. Thermocline control through multi-tank thermal-energystoragesystems.ApplEnerg2021;281:115971.https://doi.org/10.1016/j.apenergy.2020.115971

Shaikh W, Wadegaonkar A, Kedare SB, Bose M. Numerical simulation of single media thermocline based storage system. Sol Energy 2018;174:207–17. https://doi.org/10.1016/j.solener.2018.08.084.

Taher M, Bouchahm N, Guerira B, Bensaci C. On the thermal stratification inside a spherical water storage tank during dynamic mode. Appl Therm Eng 2019;159:113821. https://doi.org/10.1016/j.applthermaleng.2019.113821

Wang S, Davidson JH. Performance of a rigid porous-tube stratification manifold in comparison to an inlet pipe. Sol Energy 2017. https://doi.org/10.1016/j.solener.2017.02.045. Yoo J, Wildin MW, Truman CR. Initial formation of a thermocline in stratified thermal storage tanks.ASHRAE Trans 1986; 92:280–292

Zanganeh G, Pedretti A, Haselbacher A, Steinfeld A. Design of packed bed thermal energy storage systems for high-temperature industrial process heat. Appl Energ 2015;137:812–822. https://doi.org/10.1016/j.apenerg

Chapter 5. Impact of buoyant jet entrainment on the thermocline behaviors in a single-medium storage tank: fluid flow and heat transfer characterization

Abstract

To realize high heat storage effectiveness, thermal stratification within a thermocline-based storage system needs to be guaranteed. Especially for the singlemedium thermocline (SMT) storage system, momentum-dominated thermal jet introduced from the inlet port may penetrate deeply into the bottom calm cold fluid and cause the unexpected mixing phenomenon. In order to enhance the thermal performance within the SMT storage system, this study tends to deeply explore the mixing mechanism issuing from the inlet thermal jet then to minimize the effect of mixing via the optimized orifice baffle-type distributor (OBD).

At first, with help of the velocity visualization technique, we could observe the instantaneous velocity distribution. Thermal jet, occupying large difference of density and temperature compared with the surrounding fluid, is found that is engulfed in penetration process while the entrained fluid results in expanded mixing phenomenon. Formed eddies and vortices also cause the eddy advection into thermocline. Eddy advection caused by viscous shear will enlarge the upper mixing zone. Once the extensive eddy-compensated overturning circulation occurs into the closed TES vessel, it's risky to provoke the unstable thermal stratification. Secondly, the original optimization algorithm is used to adjust the size distribution of upper OBD. Numerical results showed that the proposed optimization algorithm could significantly minimize the transient thermocline volume and improve the thermal stratification. The gradient Richardson number also proved that the stably stratified thermocline could be gained after a series of optimization steps for upper OBD. Further, the element Péclet number gives more detail on convective and diffusive transport for both unstable and stable thermocline. Finally, Particle Image Velocimetry (PIV) measurement proved that the issue of mixing effect is minimized by optimized OBD; bulk Richardson number, thermal efficiency and capacity ratio are also increased with the optimized OBD; a brief series of parametric study complete the effect of flow rate and working temperature range on the flow field and thermal performance.

Keywords: Thermal Energy Storage (TES); Single-Medium Thermocline (SMT); Jet entrainment; Buoyant plume; Particle Image Velocimetry (PIV); Temperature Stratification This chapter is to be submitted to Energy Conversion and Management as

Lou W., Nicolas B., Roux S., Fan Y., Luo L*. Impact of buoyant jet entrainment on the thermocline behaviors in a single-medium storage tank: fluid flow and heat transfer characterization

Nomenclature

Latin letters	
Ср	Heat capacity at constant pressure [J kg ⁻¹ K ⁻¹]
D	Diameter of TES tank[m]
d	Diameter of inlet/outlet port [m]
Ε	Energy [J]
е	Thickness of the baffles [m]
Fr	Froude number
G	Generation of turbulent kinetic energy [J kg ⁻¹]
g	Gravitational acceleration [m s ⁻²]
Н	Height of thermocline tank [m]
h _j	Length of penetration [m]
k	Turbulent kinetic energy [m ² s ⁻²]
L	Distance between the upper and bottom baffles [m]
'n	Mass flow rate [kg s ⁻¹]
Ν	Number of orifice [–]
N^2	Squared Brunt-Väisälä frequency [-]
Q	Flow rate [L min ⁻¹]
q	Heat flux per square meter [w m ⁻²]
Р	Pressure [Pa]
Pe	Péclet number [–]
r	Radius of orifice on the baffle [m]
R	Experimental result
Re	Reynolds number [–]
Ri	Richardson number [–]
S	Surface area [m ²]
S^2	Squared vertical shear production [s ⁻²]
Т	Temperature [K]
t	Time [s]
t^*	Dimensionless time [–]
\overline{t}	Mean value of all passage times [K]
v	Velocity [m s ⁻¹]
V	Volume [m ³]
\bar{v}	Mean velocity [m s ⁻¹]
W	Inner Width of the squared tank [m]
X	Measured variables
x	X-coordinate [m]

123

У	<i>Y</i> -coordinate	[m]
Ζ	Z-coordinate	[m]

Greek symbols

α	Thermal diffusivity [1	$m^2 s^{-1}$]
---	------------------------	----------------

- β Thermal expansion coefficient [-]
- γ Relaxation factor [-]
- ε Rate of dissipation of turbulent kinetic energy [m² s⁻³]
- η Efficiency [-]
- λ Thermal conductivity [W m⁻¹ K⁻¹]
- μ Viscosity [kg m⁻¹ s⁻¹]
- ρ Density [kg m⁻³]
- σ Capacity ratio [-]
- ω Rate of dissipation of turbulent kinetic energy [s⁻¹]

Subscripts/superscripts

b	Buoyancy
ch	Charging
h	Hot temperature
i	Orifice index
in	Inlet
ini	Initial time
j	Optimization step
С	Cold temperature
max	Maximum
normal	Normal direction
out	Outlet
stored	Stored
tank	TES tank
x	X-coordinate
у	Y-coordinate
Ζ	Z-coordinate
0	Initial state

Abbreviations

CFD	Computational Fluid Dynamics
cDAQ	Compact data acquisition system
CSP	Concentrated solar power
DMT	Dual-medium thermocline
HTF	Heat transfer fluid
OBD	Orifice baffle-type distributor
PIV	Particle Image Velocimetry

TC	Thermocouple
RANS	Reynolds-averaged Navier-Stokes equations
RTD	Resistance Temperature Detector
RNG	Re-Normalization Group methods
RSM	Reynolds stress model
SMT	Single-medium thermocline
SST	Menter's Shear Stress Transport
TES	Thermal energy storage
UDF	User-defined functions

5.1. Introduction

Thermocline-based energy storage system, as one of the advanced thermal energy storage (TES) technologies, has received growing interest in recent years [Roos and Haselbacher, 2021; Zachár, 2020]. It consists in using only one storage tank containing both hot and cold heat transfer fluids (HTFs) inside but separated from each other by the density difference, with the presence of a temperature transition zone called the thermocline. Compared to the conventional two-tank TES concept, the thermocline system shows the features such as less auxiliary components needed and adjustable energy capacity, highly beneficial for efficient and cost-effective energy conversion and management [Calvet et al., 2013; Flueckiger et al., 2014]. Various application areas of single-tank thermocline TES system have thereby been identified and attempted, covering a wide range of operating temperature. For example, thermocline storage tanks have been proposed and developed for low-temperature (e.g., <100 °C with water as the HTF) space heating/cooling of the residential buildings [Hosseinnia et al., 2021] or sunlight greenhouses [Esen and Yuksel, 2013], for medium-temperature (e.g., 100– 400 °C with gases as the HTF) waste heat recovery of furnace and paper pulp industries [Brückner et al., 2015], and for high temperature (e.g., $> 400^{\circ}$ C with molten salt or air as the HTF) thermal management and storage of the power plants (e.g., concentrated solar power) [Geissbühler et al.,2019]. A great number of researches have also been devoted to improve the efficiency, the controllability and the stability of the thermocline storage tanks for their long-time use in various sectors.

Depending on the presence (or not) of the solid fillers in the storage tank, the single-tank thermocline TES system can be subdivided into single-medium thermocline (SMT) and dual-medium thermocline (DMT). In SMT system only the HTF itself serves as the heat storage medium whereas in DMT, in addition to the HTF, solid fillers (sensible, latent, or a combination of both) are also used to store heat. A considerable amount of investigations have been conducted for DMT systems due to the potential higher energy storage capacity and the cost competitiveness, i.e., the potential of using low-cost recycled materials (e.g., solid industrial by-products or wastes) [Motte et al.,

2015]. Nevertheless, lots of technical barriers remain to be overcome including the thermal ratcheting, the chemical incompatibility between the fillers and the HTF, the maintenance complexity, etc. In contrast, SMT systems are generally free of these problems because of the simple implementation and relatively easy operation compared to its counterpart [Advaith et al., 2020]. Moreover, it has been reported [Mira-Hernández et al., 2015] that both the first and second law efficiencies of the SMT system could be higher than those of the DMT concept under the same operating conditions. Gajbhiye et al. [2018], based on their experimental results, have also concluded that the diffusion heat transfer inside the SMT could be much lower than that inside the DMT. These findings have renewed the interest on the development and application of SMT storage systems in recent years.

The thermocline zone that separates the hot HTF (at the top of the tank) and the cold HTF (at the bottom of the tank) moves downward (or upward) during the dynamic charging (or discharging) operation. It is expected that the temperature transition zone-the thermocline-between the hot and cold HTFs occupies as small volume of the storage tank as possible even during the repeated charging-discharging cycles. This is because a higher level of thermal stratification leads to improved storage capacity and energy efficiency of the system. But without the buffering effect of solid fillers inside the tank, the fluid flow behaviors in the SMT storage tank are much more complex and the calm and stable temperature stratification becomes more difficult to be maintained than in the DMT tank. For a charging operation for example, a certain amount of hot HTF having higher temperature and velocity is suddenly injected into the TES vessel initially occupied by the calm cold HTF. The entering thermal jet will induce complex and unsteady flow patterns, named as the jet entrainment [Lai and Lee, 2012; Punetha et al., 2018; García-Alba et al., 2020], as displayed in Fig. 1 and briefly described below.

(1) Momentum-dominated regime: the inflowing jet injected from upper inlet port is governed by the high inertia force.

(2) Transition regime: inertia force is dampened by the shear stress produced by buoyancy force due to the local density difference.

(3) Buoyancy-dominated regime: eddy advection is driven by the buoyancy force, producing rising fluid streams in the form of a plume. This special flow pattern is also named as buoyant plume in fluid dynamics. The rising plumes with higher temperature are gradually cooled down by the surrounding cold fluid in this region.

(4) Lateral dispersion regime: the flow goes downwards when the density (temperature) difference disappears compared to the surrounding fluid.

The distance between the injecting inlet to the point where the vertical velocity is equal to zero is defined as the jet penetration length h_j (cf. Fig. 1). Note that the boundary between different flow regimes as well as the value of h_j are not static but evolve along with the charging time.



Fig. 1. Schematic of different flow regimes in a SMT storage tank induced by the jet entrainment

The density and velocity differences at the interface of the hot inflowing fluid and the cold surrounding fluid will trigger the Rayleigh-Taylor and Kelvin-Helmholtz instabilities, generating the stretched vortex and local turbulences that enhance the fluid mixing [Zachár, 2020; Altuntop et al., 2006; Peltier et Caulfiel, 2003]. Therefore, the time evolution of thermocline could be largely influenced by the jet entrainment phenomenon under certain circumstances. The temperature stratification could also be disturbed, or even damaged by the entering thermal jet and the resulted buoyant plumes [Haller et al., 2009; He et al., 2019].

The fluid entrainment is in fact frequently encountered in many unconfined spaces such as the fuel gas storage [Kuznetsov et al., 2015], marine outfalls [Aristodemo et al., 2015], plane jet propagation [Aristodemo et al., 2015] and tidewater glaciers [Straneo et al. 2015]. Specifically for confined SMT storage tanks with

conventional vertical or horizontal inlet port(s), many researches have been performed to study the thermo-hydrodynamic behaviors of the thermal jet and the induced buoyant plumes [Karbasi et al., 2017; Dehghan and Barzegar, 2011]. Some studies focusing on the SMT storage tank using water as the HTF are summarized in Table 1. These studies distinguish themselves by the presence of analytical [e.g., Bai et al., 2020], numerical [e.g., Chandra et al., 2020] or experimental results [e.g., Wang and Davidson, 2017], by the use of simple conventional [e.g., Assari et al., 2018] or structured inlet flow distributors [Wang et al., 2020], and by different measurement techniques. Nizami et al. [2013] developed a novel quasi-1D model for the jet entrainment with abovementioned four flow regimes, each with their own hydrodynamic features. By including certain correlations like eddy diffusivity in the governing energy equation, their model showed a good agreement with previous experimental results of [Oliveski et al., 2003]. Other theoretical models have also been proposed for modeling the charging process of the SMT storage tank, including the multi-node model [Bai et al., 2020] and the plume entrainment model [Nizami et al., 2013; Zachár et al., 2020]. The predicted flow perturbation clearly showed that the buoyant plumes could destroy the temperature stratification [Nizami et al., 2013]. Nevertheless, most theoretical models are based on the simplification of uniform and unidirectional flow [Mira-Hernández et al., 2014] while in reality, the velocity distribution is far more complex due to the sudden expansion and the flow mixing.

Another class of studies focuses on the employment of structured flow diffusing devices to replace the simple inlet/outlet ports so as to mitigate the impacts of the inflowing thermal jet. In this regard, different shapes of flow distributor have been proposed and tested including tubular distributors [e.g., Dragsted et al., 2017; Chandra et al., 2020], baffle plates [e.g., Bouhal et al.,2017; Kocijel et al., 2020; Taher et al., 2019; Lou et al., 2020] and porous distributors [e.g., Wang et al., 2020]. A detailed state-of-the art survey on these flow distributors can be found in the previous chapter 2. The main purpose of these specially designed distributors is to counterbalance the strong inertia force of the inflowing jet by producing multi-branch streams [e.g., Li et al., 2014], inversed flow [e.g., Bouhal et al., 2017] or radial flow [e.g., Al-Azawii et al., 2020], thereby limiting the effect of the momentum effect within a certain depth. Nevertheless, the efforts are focused on the numerical or experimental testing of certain type of diffuser structures while the structure optimization of such distributors are generally lacking.

In terms of experimental measuring techniques, most of the studies implemented intrusive temperature sensors within the body of the storage tank to monitor the temperature evolution at several fixed points. However, information on the local velocity profiles, essential for characterizing the jet entrainment phenomenon, cannot be acquired with this method. Very few studies used non-intrusive, opticalbased measurement methods like Particle Image Velocimetry (PIV) and/or Laser Induced Fluorescence (LIF) to construct the velocity and/or temperature fields of the HTF inside the storage tank. A Denmark laboratory investigated both theoretical (CFD) and experimental study (PIV, LIF and TC) on the effect of inlet pipe diffuser including the rigid stratifier with the 'non-return valves' and the rigid stratification pipe with circular openings [Andersen et al., 2008; Shah et al., 2005]. However, it's very regretful that their study on different flow distributor lacked of a good systematicness and completeness: their distributors seemed not to be effective at flow rate < 5 L min⁻¹ while that volume flow rate range is widely used, e.g., into domestic application; predicted temperature field via CFD simulation at certain local positions seems to be correspondent quantitatively to instantaneous experimental results both TC and LIF measurements whereas no comparison of velocity field was performed; because of extensive costs of dynamic temperature/velocity field measurements, only regional field at certain moment was represented where complete spatial and time evolution of entering thermal jet was lacked. . Besides the LIF, Infrared camera has also be used to obtain qualitative or quantitative information of dynamic temperature field [Al-Habaibeh et al., 2018]: but the obtained low-resolution (160×120 pixels) IR cartography could not satisfy the need to quantitatively characterize the thermal stratification phenomenon of inner SMT system; via a flexible fin plastic tube via named water snake, forced buoyancy jet was observed obviously but crudely. There is still much room for improvement, especially in quantitative and instantaneous experimental measurements .

From the above literature survey, one may find that the real-world experimental investigation on the effects of the jet entrainment and buoyant plumes is still lacking, especially when it comes to the thermocline evolution inside the SMT storage tank during the charging/discharging operation. Fine modeling and visualization of the complex (transient) flow patterns are especially needed to identify the main heat transfer mechanisms resulting in the thermocline decay. Moreover, investigations on the structural optimization of the flow distributors to maintain the temperature stratification are required, as the current available solutions are still far from being sufficient and effective.

The present study deals with the impact of buoyant jet entrainment on the thermocline behaviors in the SMT storage tank using a combination of modeling, experimental, analysis and optimization methodologies. The main objectives of this study are then threefold: (1) to visualize and characterize the thermo-hydrodynamic behaviors of the injecting flow (jet entrainment) and different flow patterns; (2) to evaluate the impacts of thermal jet on the convective and diffusive heat transfer mechanisms, and on the thermocline evolution inside a SMT storage tank; and (3) to better manage the temperature stratification by optimizing the inlet flow diffuser so as

to achieve high energy efficiency of the storage tank. Firstly, a 3D numerical investigation was performed to optimize the inlet orifice baffle-type distributor (OBD) of the storage tank, using the optimizing algorithm developed in our earlier study [Lou et al., 2020]. The interactions between the momentum effect and the buoyancy effect on the local velocity and temperature profiles were examined and explained in detail. In particular, the gradient Richardson number and element Péclet number were analyzed and discussed to reveal the effects of convective and diffusive heat transfer on the thermocline decay. Then, the OBDs (initial and optimized) were fabricated and installed into a lab-scale water storage tank. The evolution of the local velocity profiles and flow patterns during the charging operation (hot HTF injecting) were monitored and measured by the PIV method, where the influences of the injecting flow rate and the temperature difference have been particularly addressed. The global performances of the storage tank (characterized by the bulk Richardson number, capacity ratio and charging efficiency) using the initial (uniform) OBD or optimized OBD, were also measured and compared.

Study	Тур	Geometr	Dimension ^b	Inlet diffuser	<i>T</i> (K)	Re _{in}	Re _{tank}	CFD	Experimental	Remarks
Abdelhak et al.,	e" N	y horizontal	(m) H=0.87	-90° bended pipe	T _h	4595	220	$k-\varepsilon$ (better	Model validated	-Vertical tank had better temperature
2015		/Vertical	D=0.48		=338			than $k-\omega$)	by literature data	stratification compared to horizontal one.
		cylinder			$ _{=288}^{I_c}$					recirculation.
Chandra et al.,	N	Vertical	H=1.905 D=0.55	-Perforated pipe	T_h	1400-6440	32-128	k - ε standard	Model validated	-Slotted-type distributor could maintain the
2020		cynnder	<i>D</i> -0.55	-Simple pipe	$ = -353 \\ T_c \\ = 293$				by inclature data	turbulent inflow & low temperature difference
Zachár, 2020	N	Vertical	H=0.27	-Simple pipe	T_h	784	45	SST $k-\omega$ /	Model compared	-The developed jet entrainment model gave
		cynnder	<i>D</i> =0.22		$ -323 \\ T_c \\ =288$			Entrainmen t model	data	transfer is dominated by convective term
Panthalookaran	N, E	Vertical	H=0.59 D=0.31	-Immersed pipe	T_h	1500-3500	22.8-57	k-e RNG/	TC	-CFD models were calibrated by re-
ct al., 2008		cynnder	<i>D</i> =0.51		$ = -343 \\ T_c \\ = 293$			Teanzaore		and the coefficient of dissipation term
Wang et al., 2017 & 2020	N, E	Vertical	H=0.6 D=0.357	-Equalizer via perforated	T_h	1862-	53-477	k-ε	RTD	-The bulk <i>Ri</i> number and MIX number decreased first and then increased with the
2017 & 2020		eynnder	2 0.557		$\begin{bmatrix} T_c \\ =283 \end{bmatrix}$	10,00				increasing injecting flow rate
Assari et al., 2018	N, E	Horizontal	H=1 D=0.4	-Simple inlet	T_h	3750-7500	125-250	k-ε RNG	TC	-Location of the inlet tube largely affected
2010		cynnder	0.1		$\begin{vmatrix} 355 \\ T_c \\ = 293 \end{vmatrix}$					the temperature strainearon
El-Amin et al., 2018	N, E	Vertical cuboid	H=0.47 Cross-section:	-Immersed pipe	T_h	2390-3970	49-71	<i>k-ε</i> realizable	TC	-More stable stratification with larger temperature difference between hot and cold
			0.256 ×0.283		$\begin{vmatrix} T_c \\ = 291 \end{vmatrix}$					HTFs
Gajbhiye et al., 2018	Е	Vertical cylindrical	H=1 D=0.485	-Porous distributor	T_h =358		54-88		TC	-Thermocline thickness decreased with increasing macroscopi <i>Pe</i> number
					T_c					

Table 1. Selected studies on the thermocline evolution inside the SMT storage tank with water as the HTF

131

					=303					-Vertical porous flow distributor could mitigate the heat diffusion
Bai et al., 2020	N, E	Vertical cylinder	H=5.7 D=28.5	-Simple inlet with horizontal round plate	$\begin{vmatrix} T_h \\ = 363 \\ T_c \\ = 283 \end{vmatrix}$		45	Multi-node model/Plug flow model	TC/RTD	-Heat loss coefficient along the side wall was larger compared with bottom and top walls.
Shah et al., 2005	N, E	Vertical cuboid	H=0.9 Cross-section: 0.4×0.4	-Pipe with lockable openings	$ \begin{array}{c} T_h \\ =316 \\ T_c \\ =293 \end{array} $		60	Laminar	PIV/TC	-Stratifier worked more efficiently for larger flow rates
Andersen et al., 2008	N, E	Vertical cylinder	H=1.53 D=0.7	-Pipe with lockable openings -Pipe with circular openings -Fabric stratifier	$ \begin{array}{c} T_h \\ = 323 \\ T_c \\ = 293 \end{array} $		60-300	<i>k-ε</i> realizable	PIV/LIF/TC	-Fabric stratifier leaded to the higher temperature gradient than that of other rigid inlet pipes
Wang and Davidson, 2017	E	Vertical cuboid	H=1 Cross-section: 1.02×1.02	-Inletpipe -Pipe with diffuser -Porous-tube manifold	$ \begin{array}{c} T_h \\ = 323 \\ T_c \\ = 293 \end{array} $	1680-7730	97		PIV/TC	-Porous tube manifold eliminated largely the fluid mixing due to inflowing jet

a. N stands for numerical study and E stands for experimental study

b. *H*: height of the tank; *D*: inner diameter of the tank

5.2. Methodology

5.2.1. Experimental setup and procedures



(a)



Fig. 2. Test rig for the lab-scale SMT storage tank. (a) photo view and (b) schematic diagram

The experimental test rig shown in Fig. 2 is composed of a thermocline storage tank, hot and cold fluid circuits, a feed tank and the control & data acquisition system. Tap water was used as the HTF in this study. Both hot and cold water tanks (THERMOR Steatis 100L) were equipped with a closed loop driven by a fluid circulator (GRUNDFOS ALPHA120-50N150) to ensure the homogenous fluid condition (e.g., temperature, seeding concentration) at the stand-by state. A built-in heater strip connected to a 1200 W power supply was used as the heat source for the hot water tank and a cryo-thermostat (LAUDA Proline RP 845) was used as the cold source for the cold water tank via an internal helical-coil heat exchanger. The water temperature in each water tank was measured by a PT100 temperature sensor (RS 762-1134; $\pm 0.3^{\circ}$ C). The outflowing fluid from each water tank was filtered by a 20 µm filter (HNP Mikrosysteme F-MI2-T). Two gear pumps (TUTHILL Pump P series, 0.067-7.63 L min⁻¹) were used to deliver and adjust the flow rate of hot or cold water from the water tank toward the thermocline storage tank (or to the feed tank). The flow rate of each circuit was measured by a calibrated rotating vane flow meter (Kobold DPM 1150 G2, 0.05-5 L min⁻¹ with an accuracy of 1% of full scale). The feed tank (300 L) served as a container to store and recycle the HTFs as well as the tracer particles for the PIV measurement.

Only the charging operation was investigated, i.e., hot water was injected into the cold water environment to provoke the jet entrainment and the thermocline evolution. Each test was composed of several steps, briefly described as follows.

(1) Preparation: a certain amount of water was heated up (or cooled down) in the hot (cold) water tank to the temperature set point.

(2) Filling up: once the target temperatures were reached, the gear pump 2 was started to deliver the cold water into the thermocline storage tank from the bottom to the top.

(3). Preheating: once the thermocline storage tank was full of the cold water, the gear pump 1 was started to deliver the hot water to preheat the piping between the hot water tank and the thermocline storage tank. The purge water was guided to the feed tank via the 3-way electric valve (ASCO SCG327B001) at the top inlet port of the thermocline storage tank.

(4). When the desired inlet testing condition (constant flow rate and preset temperature) had been reached, the electrical value at the top inlet port was switched on to inject the hot water into the thermocline storage tank. The charging operation continued till the thermocline storage tank was fully occupied by the hot water. The inlet and out temperatures of the thermocline storage tank during the whole charging process were measured and monitored by two PT100 (RS 891-9145; ± 0.35 °C) sensors additionally calibrated between 0 °C and 80 °C before the experiments. The evolution of fluid velocity profiles inside the storage tank was captured and recorded by the PIV facility, described in detail in the following subsection 2.3.

The connections of the piping system were designed such that parts can be easily modified and repaired. The tightness of the setup was verified before every test to avoid the water leakage. The charging flow rate ranged between 0.3 and 1.5 L min⁻¹, corresponding to an inlet Reynolds number (Re_{in} as defined in Eq. 9) from 1368 to 6831 and an average tank Reynolds number (Re_{tank} as defined in Eq. 10) from 39 to 197. The tested inlet temperature of the hot water T_h ranged from 323.15 to 343.15 K while the cold water temperature T_c was kept at 293.15 K.

A LABVIEW platform developed in-house, together with the data acquisition system (NI cDAQ 9178) was used to control the heater strip, the cryo-thermostat, gear pumps and all the electric valves, and to record the measurement data of the flowmeters and the temperature sensors.

5.2.2 Thermocline storage tank

Figure 3 shows the schematic view of the thermocline storage tank fabricated in the laboratory LTEN. It is a cuboid tank with an inner square cross section of 194 mm in width (W) and length (L), and a total inner height (H) of 390 mm. The tank body is made of polycarbonate (ABAQUEPLAST) plates with a thickness of 6 mm, having a good temperature resistance (up to +125°C) and a good transparency (>90% for visible lights) for PIV measurement. The lateral walls of the tank and the upper & bottom covers were firstly sticking together by a transparent adhesive glue (Araldite Polyurethane 2028) and then sealed by a silicone adhesive glue (Loctite 5366).

The thermocline storage tank has a single upper and bottom port in the center of the upper and bottom cover, respectively, their inner diameter (d) being 10 mm. Two perforated baffles made of polycarbonate were installed into the tank, dividing the inner tank body into three parts (shown in Fig. 3): upper manifold from the upper port to the upper baffle, the middle storage zone and the bottom manifold from the bottom baffle to the bottom port. The height of each part is 33 mm, 312 mm and 33 mm, respectively. The perforated baffles have an overall dimension of 194 mm in length, 194 mm in width and 6 mm in thickness. Each baffle has 37 orifices with uniform or nonuniform diameters subjected to optimization, as will be described in detail in section 3 and Fig. 6. The upper and bottom baffles were supported by small cubes fixed at the lateral walls of the tank. The translucent silica gel (Loctite 5366) was used to seal up the cervices between the baffle and the tank wall so that the fluid can pass from one zone to another only through the orifices. The pipes and valves for entering the hot fluid were insulated by foam glass to reduce the heat loss to the environment.

5.2.3 PIV setup

PIV technique was used to acquire the instantaneous velocity profiles of the fluids during the charging operations. Polyamid seeding particles (DANTEC PSP-5) with an average diameter

of 5 µm and a density of 1030 kg m⁻³ were used as the seeding in our study. The proportion of seeding was adjusted to be more than 7.5 g m⁻³ during the experiments. The Stokes number in our condition was always smaller than 10^{-6} ($\ll 1$), indicating that such particles follow the flow streamlines well. The facility consists of illumination unit, image record unit, synchronous controlling unit and data processing unit. The light source used was a double cavity Nd-Yag laser (Litron Nano S65-15PIV serial LM1138). The maximum pulsed energy was 65 mJ with the minimum pulse duration at 4 ns and the wavelength of 532 nm. A cylindrical lens was used to spread the laser beam in a light sheet of about 1.5 mm in thickness, crossing perpendicularly through the walls of the tank along the center line (z=0) as shown in Fig. 3. The adaptive settings were employed, e.g., 4 Hz pulse frequency and 2 ms time interval between two frames for the charging flow rate $Q_{in}=1$ L min⁻¹ with the initial uniform OBD. The number of images during the observation duration (750 s) was kept at 3000 for each measurement.

The light scattered from the small particles was collected by a camera (DANTEC Zyla 5.5) with a 2560 \times 2160 *pixels* CMOS sensor equipped with a 105 mm optic lens .The visualization window was located at the downstream of the upper baffle so as to study the effect of flow distributor on the flow field and its time evolution. The interest area was therefore a *x*-*y* plane (*z*=0) as displayed in Fig. 3. A synchronizer was used to guarantee that the laser and the camera work synchronously.

The commercial software Dynamic studio (release 6.9) was used to control the PIV facility. The data acquisition and image analysis involved three steps.

- Image pre-processing: background image subtraction via minimum value according to the time series;
- Correlation calculation: adaptive correlation (initial pass: 64×64 pixels, final pass doubled: 16×16 pixels with a 50% overlap after several attempts considering a compromise between the measurement accuracy and the computational time).
- Spurious vector detection: validation steps (median filter, etc.) to eliminate erroneous velocity vectors.

The spatial resolution was set as 190 mm/ 2360 pixel for the initial uniform OBD case and the final number of velocity vectors in the visualization window was 17080 (= 140×122); As for the optimized OBD, the spatial resolution was set as 110 mm/2200 pixel and the final number of velocity vectors was 12500 (= 125×100).



Fig. 3. Schematic view of the cuboid thermocline storage tank, the PIV instruments and the visualization area (unit: mm).

5.2.4 CFD parameters

3D CFD simulations were performed to calculate the transient velocity & temperature characteristics inside the storage tank, providing information for the orifice size optimization of the upper baffle. The following assumptions and simplifications have been made for this study.

- Incompressible Newtonian fluid;
- Temperature-dependent thermophysical properties of fluid, as listed in Table 2;
- An eighth of 3D domain due to its symmetrical geometry (cf. Fig. 6);
- Negligible heat loss to the environment;
- Constant injecting HTF temperature (T_{in}) and flow-rate (Q_{in}) for charging;

- Homogeneous temperature profile T_c =293.15 K in the storage tank at the initial t_0 state, including solid (baffles) and liquid domains.

Material	Property	Unit	Fitting correlation T (K)
Water	Density ρ	kg m ⁻³	$739.57 + 1.9908T - 0.0038T^2$
	Specific heat Cp	J kg ⁻¹ K ⁻¹	$5438.6 - 8.07T + 0.0129T^2$
	Thermal conductivity λ	$W m^{-1}K^{-1}$	$-0.7888 + 0.0077T - 1 \times 10^{-5} T^{2}$
	Viscosity μ	$kg m^{-1} s^{-1}$	$0.0215 - 1.198 \times 10^{-4} T + 1.701 \times 10^{-7} T^2$
Polycarbonate	Density ρ	kg m ⁻³	1200
	Specific heat Cp	J kg ⁻¹ K ⁻¹	1200
	Thermal conductivity λ	$W m^{-1}K^{-1}$	0.2

Table.2. Thermo-physical properties of the fluid and solid (293.15 K<T<333.15 K) [source: engineering toolbox]

The following initial and boundary conditions were applied:

(i) The initial conditions were zero velocities and uniform temperature.

$$v_x = 0; v_y = 0; v_z = 0; T = T_c$$
 (2)

(ii) At inlet, hot HTF (T_h =333.15 K) was injected into the upper port:

$$v_y|_{y=i} = v_{in} = 0.2122 \text{ (for } Q_{in} = 1 \text{ L min}^{-1} \text{)}$$
 (3)

$$T_{y}|_{y=in} = T_{h} \tag{4}$$

(iii) Zero static pressure outlet at the bottom port:

$$P_{out} = 0 \tag{5}$$

(iv) At symmetry planes (zero normal velocity and zero normal gradients of all variables):

$$v_{normal} = 0 \tag{6}$$

(v) Conduction heat transfer in the solid domain (OBDs) [Ansys Fluent Theory Guide]:

$$q = \lambda \frac{\partial T}{\partial x}$$
(e.g., in *x*-direction); (7)

Where λ_s is thermal conductivity of polycarbonate.

(vi) Adiabatic top, bottom and lateral walls of TES tank [Mostafavi Tehrani et al., 2019]:

$$\frac{\partial T}{\partial y}|_{y=-H,0} = 0; \ v_x = 0; \ v_y = 0; \ v_z = 0$$
(8)

Fluid flow and heat transfer characteristics in the storage tank were calculated using ANSYS FLUENT code (version 19.1). The operational pressure was fixed at 101 325 Pa. Finite-volume method with collocated grid at vertex was applied for numerical discretization. Second-

order upwind differential scheme was applied for momentum and standard method for pressure. SIMPLE method was used for the pressure–velocity coupling in order to correct the pressure field oscillations.

Turbulent effect must be considered in the simulation due to the strong mixing of fluids with different velocities and densities. Therefore, most of the earlier experimentally validated numerical studies (as listed in Table 1) tended to use a turbulence model rather than the laminar model, despite the low Re_{tank} value (usually smaller than 200). The choice of turbulence model depends on the type of fluid, the working conditions, the required level of accuracy, the available computational resources and the needed time. In this study, the realizable $k-\varepsilon$ two-equation model was used. Standard wall functions were used for the near-wall region [Bahnfleth et al., 2003; Angui et al., 2018]. This model can take the buoyancy effect into account by enabling the full buoyancy option, the generation of turbulence kinetic energy due to the buoyancy G_b being considered for both the transport equations of k and ε . Another turbulent model that includes the buoyancy effect is the seven-equation Reynolds stress model (RSM), but requiring much heavier computational resources and longer calculating time. In our cases, since the values of turbulence kinetic energy at *x*-coordinate and *y*-coordiante obtained from PIV were in the same order of magnitude, one of the RANS models such as $k-\varepsilon$ model could satisfy our needs.

Simulations were performed under transient state with fixed time step of 0.001 s. Hexahedral elements, multi-zone method, sweep method were applied for meshing fluid and solid domains. A grid independence study was conducted with the increased number of total elements from 1.0 million to 1.4 million. It was found that the deviation of the mean passage time \bar{t} at 1.0, 1.1 and 1.4 million mesh was smaller than 0.01%. The grid with 1.1 million elements has finally been chosen for this study. The solution was considered to be converged when (i) normalized residuals were smaller than 8×10^{-5} for mass, momentum, *k* and ε equations and 10^{-7} for the energy equation, and (ii) the inlet static pressure become constant (less than 0.5% variation). Simulations were carried out by a computer with 2.2 GHz Intel processors (Intel Xeon CPU E5-2630 v4) and 32.0 GB RAM. The average calculation time of 18 days was required for one fully charging operation.

5.2.5 Characteristic indicators and dimensionless numbers

The Reynolds number at the inlet port and at the tank body can be calculated by Eq. 9 and Eq. 10, respectively.

$$Re_{in} = \frac{\rho v_{in} d}{\mu} \tag{9}$$

$$Re_{tank} = \frac{\rho \bar{v}_{tank} W}{\mu} \tag{10}$$

Where \bar{v}_{tank} is the mean velocity at the tank body. The physical properties of the HTF used for the calculation were based on the inlet temperature of the fluid (T_h) .

The element Péclet number (*Pe*) is introduced here to characterize the hydrodynamic status at the local element level of numerical model [Li et Tao, 2002; Dallaire, 2017; Bayón et Rojas, 2013; Patankar, 1980]. In this chapter, we introduced the modified péclet number Pe_x and Pe_y to represent simulated microscope competitive relation between convective and diffusive heat transfer with a proximate resolution of CFD model.

$$Pe_x = \frac{v_x \cdot \delta x}{\alpha} , Pe_y = \frac{v_y \cdot \delta y}{\alpha}$$
 (11)

$$\alpha = \frac{\lambda}{\rho c_p} \tag{12}$$

where δx and δy are the size of the control element and v stands for the velocity within such element. The thermal diffusivity α was calculated by the temperature dependent physical properties of each element. In this study, δx and δy are equal to 10^{-3} m.

The bulk Richardson number (Ri_{bulk}) that indicates the overall competitive effect of buoyancy force and inertial force at the scale of the entire storage tank can be calculated by Eq. 14 [Majumdar et al., 2019].

$$Ri_{bulk} = \frac{g\beta(T_{in} - T_{out})H}{v_{in}^2}$$
(14)

Where g is the gravitational acceleration. The thermal expansion coefficient β is defined by Eq. 15.

$$\beta = -\frac{\partial \rho}{\rho_c \, \partial T} \tag{15}$$

For simplification, its values are equal to 3.386×10^{-4} , 3.757×10^{-4} and 4.097×10^{-4} with the operating inlet temperature 323.15 K, 333.15 K and 343.15 K, respectively. For a typical charging process, T_{in} is maintained constant while the T_{out} increases with the charging time.

The gradient Richardson number $(Ri_{gradient})$ is also introduced in Eq. 16 [Chandra and Matuska, 2020; Li et al., 2020].

$$Ri_{gradient} = \frac{|N^2(y)|}{S^2(y)} \tag{16}$$

140

Where N^2 is the squared Brunt-Väisälä frequency (also named as buoyancy frequency) and S^2 is the squared vertical shear production, calculated by Eq. 17 and Eq. 18, respectively.

$$N^{2}(\mathbf{y}) = -\frac{g \ \partial \rho(\mathbf{y})}{\rho_{c} \ \partial \mathbf{y}}$$
(17)

$$S^{2}(\mathbf{y}) = \left(\frac{\partial v_{x}}{\partial y}\right)^{2} + \left(\frac{\partial v_{y}}{\partial y}\right)^{2} + \left(\frac{\partial v_{z}}{\partial y}\right)^{2}$$
(18)

By its definition, $N^2(y)$ is a measure of fluid stability to vertical displacements such as those caused by convection [Miyoshi et al., 2017]. For normal and stable thermocline, the value of N^2 is kept positive, meaning that the hot fluid is always located above the cold fluid. However, this value becomes negative when unexpected thermal overturning occurs, i.e., some streams of hot fluid are beneath the cold fluid. $S^2(y)$ is the sum of squared vertical shear rate tensor of the streamwise velocity in x, y and z coordinates [Panthalookaran et al., 2008]. The ratio of $N^2(y)$ and $S^2(y)$, the $Ri_{gradient}$, could be used to characterize the temeprature stratification at the local level [Li et al., 2020]. $Ri_{bulk}>1$ means that the temperature stratification is enough stable against the turbulence production shear whereas $Ri_{bulk} < 0.25$ indicates unstable temeprature stratification. In between 0.25 and 1.0 is the transition range [Abarbanel et al., 1984; Paiva et al., 2020; Gallegomarcos et al., 2019].

The charging efficiency (η_{ch}) that presents the ratio of effective stored energy to the whole inlet energy can be calculated by Eq. 19.

$$\eta_{ch} = \frac{E_{in} - E_{out}}{E_{in}} = \frac{\int_0^t \dot{m} \ c_p \ (T_{in} - T_{out}) dt}{\int_0^t \dot{m} \ c_p \ (T_{in} - T_c) dt}$$
(19)

Where \dot{m} is the mass flow rate and E is the transported energy by inlet/outlet port.

The capacity ratio (σ) that indicates the ratio of real stored thermal energy to the maximum energy storage capacity of the tank is calculated by Eq. 20.

$$\sigma = \frac{E_{stored}}{E_{stored}^{max}} = \frac{E_{in} - E_{out}}{V_{tank} (Cp_{in}T_{in}\rho_{in} - Cp_c T_c \rho_c)}$$
(20)

Where V_{tank} (14.1 L) is the volume of inner fluid domain inside the tank.

5.2.6 Uncertainty analysis

An uncertainty analysis was performed following the method of Moffat [1988]. In our condition, as for Ri_{bulk} , the indepent variables in the experiments only includig inlet and outlet temperature:

$$Ri_{bulk} = f(T_{in}, T_{out}) \tag{21}$$

Therefore, the uncertainty of two energy efficiencies can be calculated using the root-sumsquure method [Kalapala and Devanuri,2020; Moffat,1988]. For example, error of Ri_{bulk} is estimated by Eq.22.

$$\delta Ri_{bulk} = \left\{ \left(\frac{\partial Ri_{bulk}}{\partial T_{in}} \delta T_{in} \right)^2 + \left(\frac{\partial Ri_{bulk}}{\partial T_{out}} \delta T_{out} \right)^2 \right\}^{\frac{1}{2}} = \left\{ \left(\frac{g\beta H}{v_{in}^2} \delta T_{in} \right)^2 + \left(\frac{g\beta H}{v_{in}^2} \delta T_{out} \right)^2 \right\}^{\frac{1}{2}}$$
(22)

Accuracy of the PT100s is $\pm (0.15 + 0.002\text{T})$ °C while flow meters were calibrated for flow rate 0.05 - 5 L min⁻¹ with the accuracy at 1%. According to typical charging operating (Q_{in} 1.0 L min⁻¹, $T_{in} = 333.15$ K), mean temperature of T_{out} is considerated to be 313.15 K after simplification. the error Ri_{bulk} is equal to 0.0113, and relative error is about 0.89% under such above operating condition.

5.2.7. Comparison between CFD and PIV results

The hydrodynamic characteristics of a typical charging process has been investigated as a reference case for the comparison between the CFD and PIV results. To do that, hot water at T_h =333.15 K and Q_{in} =1 L min⁻¹ was injected into the cuboid thermocline storage tank initially filled with cold water (T_c =293.15 K). Both the upper and bottom baffles have uniform orifices, their geometry and dimensions being given in section 3.1 and Fig. 6a.

Figure 4 shows the evolution of the velocity field in the interest area (downstream the upper baffle) at different charging times (*t*=100 s; 400 s and 700 s), obtained by PIV and CFD methods. Here, each velocity map measured by PIV is obtained by averaging 20 images (5 s) around the certain measurement moment. Due to the existence of central orifice on the upper baffle, an inflowing hot water jet together with the progressive downward penetration can be clearly seen. With the initially momentum-dominated hot fluid flowing downwards, the inertia force becomes unable to overcome the vertical shear produced by the density difference at the interface of the injecting hot fluid and the surrounding cold fluid. The vertical velocity of the jet front decreases, and in the meanwhile, a certain portion of hot fluid is turned upward away from the centerline (transition regime). The phenomenon of jet entrainment is clearly shown in Fig. 4, i.e., two branches of the buoyant plume at left and right sides of the principal jet can be observed in both CFD and PIV pictures. The buoyant plumes first rise upwards due to the buoyancy force (in the buoyancy-dominated regime) until they touch the upper baffle, and then fall downward along the walls of the tank (the edgewise dispersion regime). All the flow regimes as schematized in Fig. 1 can be captured in the obtained numerical and experimental results.

Fig. 4 also shows that calculated transient temperature fields and velocity fields are closely correlated. The inflowing thermal jet advances along with the thermocline moving downward during the charging process.



Fig.4. Characterization of the jet entrainment and buoyant plumes inside the SMT storage tank at different charging times. (a) velocity field obtained by PIV; (b) velocity field obtained by CFD; (c) temperature field obtained by CFD. Condition: $Q_{in} = 1 \text{ L min}^{-1}$, T_h =333.15 K, T_c =293.15 K, initial uniform OBD.


Fig. 5. Comparison on the length of penetration (h_j) as a function of charging time obtained by PIV and CFD methods. Condition: $Q_{in} = 1 \text{ L min}^{-1}$, T_h =333.15 K, T_c =293.15 K, initial uniform OBD.

Fig. 5 compares the jet penetration length (h_j) as a function of charging time, obtained by the CFD and PIV methods. Recall that h_j quantifies the longest extended length of the thermal jet front from the top inlet port of the storage tank to where the vertical velocity is exhausted ($v_y=0$) at the centerline.

The same trend may be observed from both the CFD simulation and the PIV measurement: the h_i value increases monotonously with the charging time t. The h_i value reaches about y=-160 mm at t = 700 s which is near the bottom border of the visualization window. The difference of the h_i values (between the CFD and PIV) ranges from 33.9 mm (31.0%) to 24.5 mm (12.1%) when the charging time t increases from 50 s to 650 s. This deviation can be derived equally from the associated uncertainties or intrinsic errors of the both methods. Firstly, despite the careful preheating step for each experiment, the small perturbation of the (initial) inlet condition (flow rate, temperature) still exit due to the needed response time of the 3-way valves, rendering a relatively larger difference of h_j at the beginning stage of the charging. Secondly, the experimental h_j value has been obtained by averaging PIV 50 images. The use of the averaged h_j value of a certain time period (12.5 s) instead of the instantaneous value may also cause the deviation from the real penetration length. Thirdly, the camera-laser-tank calibration procedure for PIV, although carefully done before each measurement, may still cause the small deviation on the visualized position. Fourthly, besides other assumptions and simplifications made for the CFD model (e.g., neglected heat loss, $k-\varepsilon$ RNG turbulent model, etc.), the use of normal to the boundary velocity profile at the inlet port rather than the "closer-to-the-reality" parabolic velocity profile will cause an underestimation of the h_i value with respect to the experimental data.

More detailed comparison on the PIV velocity profiles (velocity magnitude, x and y components) at different time and space coordinates can be found in Appendix A of this paper,

clearly showing that CFD and PIV results are in acceptable agreement. In brief, the transient velocity fields can qualitatively showcase the flow regimes of the jet entrainment and well predict the h_j value and its increasing tendency with charging time. In this regard, the CFD model and the associated parameters used in the simulation are considered to be adequate to describe the evolution of fluid flow profiles and thermocline behavior used subsequently for optimizing the size distribution of orifices on the upper baffle.

5.3. Optimization of the orifice baffle for thermocline control: the CFD results

This section presents the numerical results obtained on the optimization of the upper orifice baffle. Firstly, the optimization algorithm is briefly introduced. Then the fluid flow and temperature characteristics as well as the dynamic thermocline behaviors before and after the optimization will be compared and discussed in detail.

5.3.1. Optimization algorithm

The optimizing algorithm has been developed in a way that the size distribution of perforated orifices on the baffle is adjusted to minimize the volume of the thermocline in the storage tank. This method has been proposed and developed in 2D [Lou et al., 2020] and extended to asymmetry 2D for the design and optimization of ring-opening plate distributors in our earlier study [chapter 4], but is applied in the first time to a real 3D orifice distributor being implemented into the lab-scale thermocline storage tank. Here only the basic principles of the optimization algorithm and some key steps are introduced. More details about the algorithm itself, e.g., the optimization criterion, the numerical tested cases, the influencing parameters, etc. may refer to [Lou et al., 2020].

The basic idea of optimization is to adjust the sizes of orifices on the baffle so that the thermal front could pass through every orifice at almost the same time. Here the passage time t_i is defined as the time interval between t_0 (charging start) and the moment when the thermal front (T_{80} in this study) just traverses the i^{th} orifice on the upper baffle. In other words, it corresponds to the moment when the mean fluid temperature in the i^{th} orifice is equal to T_{80} .

Note that the temperature threshold values T_{80} and T_{20} are used in this study to define the borders of the thermocline zone, their values being calculated by Eqs. 23-24.

$$T_{80} = T_c + 80\% \times (T_h - T_c) \tag{23}$$

$$T_{20} = T_c + 20\% \times (T_h - T_c) \tag{24}$$

The temperature value is also normalized for simple interpretation and comparison:

$$T^* = \frac{T - T_c}{T_h - T_c} \tag{25}$$

145

Where T_c is the cold temperature equaling to 293.15 K.

The dimensionless thermocline volume V^* is defined as the volume proportion of the storage tank occupied by the thermocline zone [T₂₀; T₈₀].

$$V^* = \frac{V_{[T20;T80]}}{V_{tank}}$$
(26)

The mean passage time for all the orifices can be calculated Eq. 27.

$$\bar{t} = \frac{\sum_{i=1}^{N} t_i}{N} \tag{27}$$

The optimality criterion is written as:

$$t_i = \bar{t} \ (i = 1, 2, \dots N) \tag{28}$$

suggesting that the thermally piston flow could be realized when the passage times (t_i) for all orifices are identical.

An iterative program is written such that the surface area of the i^{th} orifice (s_i) is modified according to the deviation of its passage time to the mean value $(t_i - \bar{t}_i)$. Specifically for j step, the s_i will be reduced for step j + 1 if $t_{i,j}$ is found smaller than $\bar{t}_{i,j}$. Otherwise, s_i will be enlarged for step j + 1 if $t_{i,j}$ is observed larger than the mean $\bar{t}_{i,j}$ at step j.

$$\Delta s_{i,j} = s_{i,j+1} - s_{i,j} = \gamma \left(t_{i,j} - \bar{t}_j \right) \tag{29}$$

Where γ is an adjusting factor chosen to ensure a fast and stable convergence.

$$t_{manifold} = \frac{V_{manifold}}{Q_{in}} \tag{30}$$

Where $V_{manifold}$ is the volume of upper/bottom manifold (1.33L).

The dimensionless charging time t^* is defined as the ratio of the instantaneous charging time to the theoretical residence time to fill up the storage tank by an ideal plug flow at a constant flow-rate.

$$t^* = \frac{t}{\frac{V_{tank}}{Q_{in}}} \tag{31}$$

5.3.2 Optimization results

Uniform orifice configuration was used as the reference case to start the optimization (step 1), the arrangement of orifices being shown in Fig. 6. The ratio between the sum of whole opening surfaces and the inner cross-section surface area of tank, also named as the baffle porosity, is equal to 0.20. Considering such orifices arrangement for both upper and bottom baffles and the symmetry character of the storage tank geometry, one-eighth of the body has been used for simulation and optimization. Note that only the upper baffle was optimized for the charging operation while uniform orifice bottom baffle was always used in this study.

It can be observed that the radius size of the central orifice is largely reduced from 8.05 mm to 1.31 mm while other orifices are all enlarged to a similar size (8.06-8.22 mm). Based on the constraint of constant porosity of the baffle in the optimization algorithm, the passage surface area of the initial central orifice is allocated to the orifices situated on the edge of the baffle, guiding the hot fluid to occupy the whole upper manifold before crossing through the (non-uniform sized) orifices. The variation trend is in line with the 2D numerical results obtained in our earlier study [Lou et al., 2020].

Fig. 7(a) demonstrates the time evolution of the dimensionless thermocline volume V^* . It can be observed that the rising rate, the maximum value of V^* and the corresponding reaching time are different for different optimization steps. In other words, the V^* curve seems to be flattened from one optimization step to the next. Especially for the optimized configuration (step 5), the V^* curve climes to reach the shift point at $t^*=0.097$, then remains almost stable afterward. This implies that the optimized OBD helps establish a more stable thermocline between the hot and cold fluids, and manage its (almost) undisturbed advance over time.

The comparison on the V^* value clearly shows that the temperature stratification can be better maintained by optimizing the orifice configuration of the upper baffle using the proposed algorithm. Nevertheless, the effect of conjugated fluid flow and heat transfer on the thermocline evolution is still masked. A close look at the local velocity and temperature profiles is thereby necessary to provide additional insights on the jet entrainment phenomenon and buoyant plumes, which is detailed in the following sub-section.



Fig. 6. Geometry and dimension details of the upper orifice baffle tested in this study. (a) initial uniform orifice configuration; (b) optimized orifice configuration (Unit: mm). Condition: $Q_{in} = 1 \text{ L min}^{-1}$, $T_h=333.15 \text{ K}$, $T_c=293.15 \text{ K}$.



Fig.7. Time evolution of the dimensionless thermocline volume V *. Condition: $Q_{in} = 1 \text{ Lmin}^{-1}$, $T_h = 333.15 \text{ K}$, $T_c = 293.15 \text{ K}$.



5.3.3 Comparison on the velocity and temperature profiles

Fig. 8. Velocity vectors superimposed over velocity contour at half of the interested area of the SMT tank at $t^*_{manifold}$ (0.0927). (a) initial uniform orifice baffle; (b) optimized orifice baffle. Condition: $Q_{in} = 1 \text{ L min}^{-1}$, T_h =333.15 K, T_c =293.15 K,

Figure 8 shows the contours of the velocity magnitude in right half of the interest area (below the upper baffle) for the uniform orifice configuration (Fig. 8a) and for the optimized orifice configuration (Fig. 8b). Local snapshots with adaptive enlarged ratio of each have been extracted and displayed on the figure, showing the amplified velocity vectors in some typical zones. Note

that the instantaneous flow fields at $t^*_{manifold} = 0.0927$ (t = 79.5 s) are shown and compared here, corresponding to the theoretical charging moment when the upper manifold is fully occupied by an ideal plug-flow of the incoming hot water ($Q_{in}=1 \text{ Lmin}^{-1}$).

From Fig. 8(a), it can be observed that the injecting hot fluid passes easily through the (large) central orifice, pushing the cold fluid far away from the upper baffle. The momentumdominated thermal jet has large v_y component at first and then gradually exhausts. This is because the existing density difference between the hot jet and the surrounding cold fluid induces the buoyancy force, indicated by the upward velocity vectors along the vertical central line as shown in zone C. At a certain penetration length h_j , the buoyancy force, the inertia force and the gravity are balanced and offset, the v_y component being zero at this point.

The produced velocity shear yields the entrained flow that changes the advancing direction. A good part of the fluid flow has thus been brought away from the axial direction, but risen back to the upper baffle in the form of a plume. The temperature of the rising plume falls down by mixing with the surrounding cold fluid. When the plume touches the upper baffle, a small part of the fluid flow re-enters the upper manifold through the orifices (upward velocity vectors shown in zone B) while the main stream goes in horizontal (x) direction towards the lateral walls of the storage tank and bounces back (zones B and D). Vortices of different sizes can be observed: one between the inflowing jet and the rising plume branch (zone A) and another between the rising plume and the edge (zone B). The latter one stretches over a large horizontal distance, the v_x component of the fluid flow being more important.

The existence of various vortices and flow recirculating zones enhance the temperature mixing, as shown in Fig. 9a. Actually for an ideal plug-flow condition, the hot front would just cross through the upper baffle at $t^*_{manifold}$ (t = 79.5 s), i.e. at y = 0 mm in Figs. 8 and 9. But for the initial uniform OBD, the thermocline goes much deeper and reaches y = -49.5 mm. The temperature stratification is largely degraded due to the momentum-dominated inflowing jet (through the central orifice) as well as the rising plumes. Consequently, the thermocline zone has already occupied a large volume of the middle storage zone at $t^*_{manifold}$ moment.



Fig. 9. Temperature contour of the half interest area at $t^*_{manifold}$. (a) initial uniform OBD; (b) optimized OBD. Condition: $Q_{in} = 1 \text{ Lmin}^{-1}$, $T_h=333.15 \text{ K}$, $T_c=293.15 \text{ K}$.

From the velocity contours and vectors shown in Fig. 8(b), one may find that a more calm flow pattern can be achieved with the optimized upper OBD. Since the surface area of the central orifice was significantly reduced (others were enlarged), the hydraulic resistance of the central orifice facing the inlet port is thereby much higher. More hot fluid is guided toward the orifices on edge of the upper manifold (zone B). The thermal jet through the central orifice penetrates only a very small distance (zone A). The rising plumes are hardly visible and the generated vortices are much smaller (zones A and C), the hot and cold fluids being less mixed. A stable and uniform velocity profiles (upward v_y component) are formed at y < -20 mm (zone D and below) at $t^*_{manifold}$. The temperature cartography shown in Fig. 9b also indicates that the thermocline becomes more flattened and thinner compared to that of the initial uniform orifice case (Fig. 9a).

5.3.4 Convective vs. diffusive heat transfer: gradient Richardson number $Ri_{gradient}$ and Pe number

The $Ri_{gradient}$ number and the element Pe number are calculated and discussed here, so as to provide additional insights on the stratification degradation due to the effect of convective or diffusive heat transfer mechanisms. Figure 10 shows the N^2 , S^2 and $Ri_{gradient}$ curves at different depths of the interest area at $t^*_{manifold}$.



Fig. 10. Squared Brunt–Väisälä frequency N^2 , squared vertical shear production S^2 and the gradient Richardson number $Ri_{gradient}$ at $t^*_{manifold}$ for different optimization steps. Condition: $Q_{in} = 1 \text{ L min}^{-1}$, $T_h=333.15 \text{ K}$, $T_c=293.15 \text{ K}$,

Negative values of N^2 ($y = -10 \ mm$) may be found near the centerline (x between 10 and 20 mm) for steps 1 and 2, implying that the thermal overturning $\left(\frac{\partial T(y)}{\partial y} > 0\right)$ occurred in these areas. This is caused by an eddy between the downward jet from the central orifice and the rising plume, as mentioned earlier in Fig. 8(a). The value of S^2 in this eddy zone is also very high, indicating the large shear stress induced by the central inflowing jet. The thermal overturning yields the degradation of temperature stratification, characterized by $Ri_{gradient} < 0.25$ shown in Fig. 10 for steps 1 and 2. Another subcritical peak ($Ri_{gradient} < 0.25$) at $y = -10 \ mm$ could be identified at x=80 mm, corresponding to the locations where the vortices are generated as shown in Fig. 8a.

The values of N^2 become all positive after step 3, implying that all vortices that cause the thermal overturning have been eliminated. Moreover, N^2 value could be kept at 0 at the position $y = -50 \ mm$, indicating that the cold fluid is calmly maintained at $t^*_{manifold}$.

The analysis of the $Ri_{gradient}$ number (as well as N^2 and S^2) clearly shows that the temperature stratification can be better maintained from one optimization step to the next. At step 5, gradient Ri number curves are all higher than 1. Stable thermocline in the middle storage zone has thereby been achieved with help of the optimized upper OBD.

From the heat transfer point of view, it is also interesting to examine the competitive relationship between convection and diffusion, which are two mechanisms that degrades the temperature stratification. In this regard, the x and y components of the element Pe number (Pe_x and Pe_y) at $t^*_{manifold}$ for both the initial uniform OBD and the optimized OBD are calculated and compared in Fig. 11. Note that Pe_x and Pe_y values are plotted in logarithmic scale, with its upper and lower range at a certain y-coordinate and the point for the mean value.

For the initial uniform OBD (blue curve), Large Pe_x and Pe_y values can be observed in the zone (-50 mm< y < 0 mm), indicating the strong fluid mixing and the convection-dominated heat transfer. This is in line with the local velocity and temperature profiles analyzed and discussed in the above sub-section 3.3. The thermocline zone has significantly expanded in this area at $t_{manifold}^*$. A diffusion transport dominated zone may be observed at (-90 mm< y<-50 mm), indicated by the value of Pe_x and $Pe_y<1$. This is also coherent to the earlier mentioned neutral buoyancy zones C and D shown in Fig. 8a. Below y=-90, the Pe_x value becomes very small mainly because of the negligible v_x . On the other hand, the mean Pe_y values stabilizes around 1, indicating the balanced convective and diffusive heat transfer when the cold fluid progresses.



Fig. 11. Comparison on the element *Pe* numbers at $t^*_{manifold}$ for the initial uniform OBD and the optimized OBD. (a) Pe_x; (b) Pe_y. Condition: $Q_{in} = 1 \text{ Lmin}^{-1}$, $T_h=333.15 \text{ K}$, $T_c=293.15 \text{ K}$.

For the optimized OBD (orange curve), the convective transport is dominating within only a thin zone (-10 mm < y < 0 mm) just beneath the upper baffle. Since Pe_x value becomes smaller than 1 below y=-10 mm, the diffusion becomes the dominating heat transfer mechanism along the

x direction (eddy advection eliminated). Likewise, the balanced convective and diffusive transport is achieved at y <-30 mm, indicated by $Pe_y=1$ in Fig. 11b.

In brief, the optimized orifice configuration, in comparison with the initial uniform orifice configuration, could yield the diffusion-dominated transport along the x-direction and a balanced convection/diffusion transport along the y-direction at a much smaller depth of the storage tank. Both effects help alleviate the fluid mixing thus enhance the temperature stratification (thinner thermocline zone).

5.4. Influences of operating parameters on the thermal jet propagation and on the thermal performance of the SMT storage tank: experimental results

The numerical results presented in the above section has demonstrated that the optimization algorithm is really effective in maintaining the temperature stratification and manage the undisturbed thermocline advance. Based on these encouraging findings, the optimized OBD as well as the initial uniform OBD were fabricated and installed in the lab-scale storage tank. Charging experiments were then performed under various inlet flow-rate and inlet temperature conditions. Measuring the local velocity profiles by the PIV method as well as the inlet/outlet temperature evolutions of the storage tank by RTD sensors will further reveal and confirm the thermal performance improvement of the SMT storage tank by using the optimized OBD. In particular, the influences of the injecting flow rate and the temperature difference (between T_h and T_c) on the thermal jet penetration/penetration and on the thermal performances (indicated by Ri_{bulk} number, η_{ch} and σ) will be addressed, providing guideline for the design and operating of SMT storage tanks for different applications.



5.4.2 Effect of the injecting flow rate

Fig. 12. Velocity profiles of the interest area measured by PIV for different injecting flow-rate at $t^*=0.4$ and $t^*=0.8$. Condition: $T_h=333.15$ K, $T_c=293.15$ K, initial uniform OBD

The charging experiments were performed for the SMT storage tank equipped with either the initial uniform OBD or the optimized OBD, at an injecting flow-rate of Q_{in} =0.3-1.5 L min⁻¹ and a temperature difference of 40 K (T_h =333.15 K; T_c =293.15 K). Figure 12 gives a comparison on the velocity profiles of the interest area (initial uniform OBD) for different injecting flow rates, captured by PIV at t^* =0.4 and t^* =0.8, respectively. It can be observed that the higher flow rate with larger inertia force provokes the longer penetration depth (h_j) at the same t^* . For example, the front of thermal jet at $Q_{in}=0.3$ L min⁻¹ touches y=-50 mm at $t^*=0.4$ whereasat $Q_{in}=1.5$ L min⁻¹, it has already crossed over the lower border of the visualization window (y=-160 mm).

In the meanwhile, buoyant plumes can be observed at both sides of the central inflowing jet. The larger the flow rate, the thicker the rising branches as displayed in Fig. 12. The larger mass of hot fluid is injected per unit time at high flow rate, and proportionally a higher amount is deflected and change the direction (compared to low flow rate condition), occupying the larger volume of the storage tank in the form of rising plumes. At each flow rate, the front of thermal jet penetrates deeper with the increasing charging time. This is because the vortices and flow recirculation enhance the heat transfer so that the temperature difference becomes smaller between the inflowing jet at the central line and its surroundings. The buoyancy force is thereby reduced due to the smaller density difference.



Fig. 13. Comparison on the thermal performance of the SMT storage tank for different inlet flow rates, with initial uniform OBD or optimized OBD. (a) bulk Richardson number Ri_{bulk} versus dimensionless charging time t^* ; (b) capacity ratio σ versus t^* ; (c) Charging efficiency η_{ch} versus cut-off temperature T^*_{out} . Condition: T_h =333.15 K, T_c =293.15 K.

Figure 13a shows the bulk Richardson number Ri_{bulk} versus t^* under different inlet flow rate conditions. The Ri_{bulk} curves for different flow rates are all kept in a plateau over a certain period of time during which the thermocline doesn't influence the outlet temperature ($T_{out} = T_c$) at the bottom port of the storage tank. The Ri_{bulk} curves of the initial uniform OBD begin to fall at different t^* : $t^*=0.4$ for $Q_{in}=1.5$ L min⁻¹, 0.7 for $Q_{in}=1.0$ L min⁻¹ and about 0.8 for $Q_{in}=0.7$ and 0.3 L min⁻¹. The convection heat transfer is smaller at lower inlet flow-rate, ensuring a better temperature stratification. This can also be confirmed by the consistently higher Ri_{bulk} values of $Q_{in}=0.3 \text{ Lmin}^{-1}$ than those of the other inlet flow rates. For the optimized OBD case, very similar trend can be observed from Fig. 13a, except that the falling points of Ri_{bulk} curve are all delayed until $t^*=0.9$.

Figure 13b shows the storage capacity (σ) curves versus t^* under different inlet flow rate conditions. Recall that this parameter indicates the ratio of real stored thermal energy to the maximum energy storage capacity of the storage tank. For the initial uniform OBD case, the σ curve of a certain inlet flowrate first climes rapidly and then tends to stabilize. At $t^*=1.5$ the σ value for $Q_{in}=0.3$ L min⁻¹ reaches 0.92 while that for $Q_{in}=1.5$ L min⁻¹ approaches 0.77, still far from the fully charged status ($\sigma =1$). For the optimized OBD case in contrast, the four σ curves are almost overlapping over the charging time, reaching 0.94 at $t^*=1.5$. The departure from the ideal fully charged status is partly due to the heat loss to the environment since the SMT storage tank was not insulated during the PIV experiments which will be further discussed below.

Figure 13c shows that the charging efficiency η_{ch} decreases with the increasing dimensionless outlet temperature T_{out}^* (also called the cut-off temperature). For the initial uniform OBD case, the higher inlet flow rate results in the lower η_{ch} , except for the curve of $Q_{in}=0.3$ L min⁻¹ which falls between that of $Q_{in}=0.7$ L min⁻¹ and 1.0 L min⁻¹. Although the calm and flatten thermocline can be largely maintained for almost the whole charging time at $Q_{in}=0.3$ L min⁻¹ (as indicated by the high Ri_{bulk} number shown in Fig. 13a), the real charging time to reach the fully charged status becomes very long. As a result, the amount of heat transferred by the thermal diffusion (between the hot and cold fluids) and the heat loss to the environment (due to the lack of insulation) becomes non-negligible, resulting in the smaller η_{ch} values at a high T_{out}^* . In real-world application, the inlet flow rate and the cut-off outlet temperature should be coordinated and determined together so as to guarantee a high charging/discharging efficiency of the process. Regarding the optimized OBD case, the η_{ch} values at a certain inlet flow rate are consistently higher than its counterpart (initial uniform OBD). The three η_{ch} curves ($Q_{in}=0.3$; 0.7 and 1.0 L min⁻¹) are again almost overlapping, except for the $Q_{in}=0.3$ curve which falls rapidly after $T_{out}^* = 0.7$ due to the same reason explained above.

In conclusion, a low inlet flow rate is generally favorable to achieve a high storage capacity and charging efficiency when the inlet diffuser is less performant. This is in line with the findings of our earlier study [chapter 4]. By using the optimized OBD, the global thermal performance of the storage tank could be improved, and the impacts of the inlet flow rate (momentum-dominated thermal jet) are by and large mitigated under our tested conditions.

5.4.3 Effect of working temperature difference

The effect of the working temperature difference $(\Delta T = T_h - T_c)$ on the global thermal performance of SMT storage tank has been subjected to extensive numerical and experimental studies, but controversy still exists in the literature. Some researchers reported the positive effect of larger ΔT on the charging/discharging efficiency [e.g., Nelson et al., 1999; Dehghan et al, 2011; Chandra et al., 2019] and on the exergy efficiency [Njoku et al., 2016], while others [e.g., Shaikh et al., 2018; Wang et al., 2020] found that the temperature stratification might be degraded more easily with a larger ΔT . Therefore, investigation is still needed to reveal how this parameter impacts the local velocity (and temperature) profiles and on the global thermal performance of the SMT storage tank, especially by examining the real-time penetration characteristics of the thermal jet.



Fig. 14. Velocity profiles of the interest area for different working temperature differences at $t^*=0.4$ and $t^*=0.8$. Condition: $Q_{in} = 1 \text{ L min}^{-1}$, initial uniform OBD



Fig. 15. Comparison on the thermal performance of the SMT storage tank for different working temperature differences, with initial uniform OBD or optimized OBD. (a) bulk Richardson number Ri_{bulk} versus dimensionless charging time t^* , (b) capacity ratio σ versus t^* ; (c) Charging efficiency η_{ch} versus cut-off temperature T^*_{out} . Condition: $Q_{in} = 1 \text{ L min}^{-1}$

Additional charging experiments were performed under different working temperature difference conditions ($\Delta T = 30$; 40; 50 K; $T_c = 293.15$ K) for the SMT storage tank equipped with either the initial uniform OBD or optimized OBD. The velocity profiles of the interest area (initial uniform OBD) at $t^*=0.4$ and 0.8 recorded by PIV are displayed in Fig. 14. Although the inertia force is the same ($Q_{in}=1$ L min⁻¹), the propagation of the thermal jet clearly differ when ΔT changes. The larger buoyancy force due to the higher ΔT (cf. $T_h=343.15$ K in this study) can better balance out the inertia force of the inflowing jet, the penetration length being thereby smaller. With the charging time proceeds, the front of thermal jet penetrates deeper due to the smaller buoyancy force, i.e., smaller temperature difference between the jet and the surrounding as already explained

above. In contrast, the penetration length for $\Delta T = 30$ K ($T_h = 323.15$ K) is always the largest among the tested conditions, the upper region of storage tank being quickly occupied by the intermediate temperature fluid due to the strong convective heat transfer.

The global performance indicators of the SMT storage tank under different T_h conditions, with initial uniform OBD or optimized OBD, are presented in Fig. 15 and briefly commented below. The Ri_{bulk} curves shown in Fig. 15a indicates better temperature stratification under the higher ΔT condition, in agreement with the observation from Fig. 14. Figs. 15b and 15c show the similar trend, i.e. relatively better thermal performance, indicated by higher σ and η_{ch} values, can be achieved with a larger ΔT when the inlet flow diffuser is less performant (initial uniform OBD case).

When using the optimized OBD, the impact of ΔT on the thermal performance of the storage tank seems negligible, indicated by almost overlapping curves of σ and η_{ch} in Figs. 15b and c.



Fig. 16. Velocity profiles for different working temperature differences at $t^*=0.1$, $t^*=0.45$ and $t^*=0.8$ with optimized orifice baffle. Condition: $Q_{in} = 1$ L min⁻¹

A close look at the evolution of velocity profiles using the optimized OBD is illustrated in Fig. 16, under different ΔT conditions. Recall that the surface areas of the orifices have been reallocated by running the optimization algorithm, equalizing the passage times of the hot front for all the orifices. The flow rate passing through the orifices has also been redistributed, i.e., the central momentum-dominated jet is no longer existing and the magnitude of velocity is largely reduced. Some parameters of the high speed camera and the laser of the PIV facility have thereby been readjusted to better observe and capture this relatively small velocity magnitude (e.g., maximum velocity at 8 × 10⁻³ m s⁻¹ on Fig. 16 vs. 0.16 m s⁻¹ on Fig. 14).

Unlike the fluid structure of the initial uniform OBD case, only a tiny jet through the central orifice can be detected at $t^* = 0.1$ on Fig. 16. The front of thermal jet penetrates with the increasing ΔT value, due to reasons explained above. When the charging proceeds (e.g., $t^* = 0.45$), a second small jet appears at the location of the side orifice (the orifice at right of the central orifice shown in Fig. 6a). The interaction between the adjacent entrained buoyant plumes then takes place, illustrated by the appearance of K–H instabilities or engulfment just below the upper baffle shown on Fig. 16. Nevertheless, this intense mixing region covers a very small depth. The temperature stratification could be well maintained, corroborated by the constant Ri_{bulk} values (Fig. 15a) that not decline until $t^*>0.9$. The thermo-hydrodynamic behaviors of the rising plumes generated by multiple (neighboring) thermal jets and their mutual interferences are more complex and certainly deserve further systematic investigations, but are beyond the scope of this paper.

Note that these experiments were performed under $Q_{in}=1$ L min⁻¹, for which the convective transport is the dominant heat transfer mechanism responsible for the thermocline decay. However, at low inlet flowrate under which the real charging time is far prolonged, the influence of ΔT on the global thermal performance of the storage tank could be less significant because of the more important thermal diffusion and heat losses (as has been discussed in section 4.1). Under certain circumstances, an inverse trend might occur because smaller ΔT means smaller heat diffusion between the hot and cold fluids. Thinner thermocline zone could thereby be maintained over the long-time thermocline forwarding. More detailed discussion on this aspect has been provided in our earlier study [chapter 4]. Anyhow, the working temperature difference should be carefully determined in connection with other influencing factors (i.e., inlet flow rate; geometry of the storage tank; aimed application, etc) to achieve a compromise between the convective and diffusive dominated heat transfers.

5.5. Conclusion and prospects

This article presents a numerical and experimental work on the characterization and management of the dynamic thermocline behaviors in a SMT storage tank. The first part of the study has been focused on the CFD simulation of the charging operation and on the optimization of the orifice configuration of the upper baffle using the algorithm proposed in our earlier study [Lou et al., 2020]. Closely conjugated fluid flow and heat transfer and complex thermohydrodynamic behaviors of the thermal jet were observed. In particular, the jet entrainment phenomenon featured by the rising buoyant plumes was shown to appear, which was provoked by the interactions between the inertia force of the inflowing jet and the buoyancy force due to the density (temperature) difference. Optimizing the size distribution of orifices on the upper baffle could effectively mitigate the impact of thermal jet on the temperature stratification, indicated by the increasing values of the $Ri_{gradient}$ number from one optimization step to the next (all higher than 1 at final step). In fact, the optimized OBD could restrict the convection-dominated heat transfer ($Pe_x \ll 1$) and reach a balanced convection/diffusion transport along the y-direction($Pe_y=1$) within a very small depth downstream the upper baffle, leading to more flatten and thinner thermocline compared to that of the initial uniform OBD case.

Charging experiments have then been performed for the lab-scale cuboid SMT storage tank equipped with initial uniform or optimized OBD. The evolution of the local velocity profiles under different injecting flow rates or temperatures were recorded and measured by the PIV method. The buoyant plumes and their dynamic evolution were again visualized. The length of jet penetration for the initial uniform OBD was found to increase with the increasing inlet flowrate and the decreasing temperature difference, the weaker buoyancy force being more difficult to balance the momentum force. For the optimized OBD case, the impacts of both factors were found far less important due to the impeded central jet and by the plume interferences between the multiple jets. The global performances of the storage tank were also calculated and compared. A low inlet flow rate and a large temperature difference are positive factors to achieve a high storage capacity and charging efficiency if the inflowing momentum-dominated jet cannot be properly handled. This can be explained by the thermo-hydrodynamics of the storage tank could always be achieved by using the optimized OBD, providing more rooms for selecting appropriate operating ranges of SMT storage systems in different aimed application fields.

As prospects, the synchronized measurement of velocity & temperature fields and their evolutions using appropriate optical-based method (e.g., LIF) is our goal. In this way, a thorough understanding of conjugated fluid flow and heat transfer characteristics may be acquired, necessary for developing some theoretical models to better predict the effect of thermal jet. A deeper examination on the fluid mechanisms and heat transfer of multiple thermal jets, as well as the interference of neighboring buoyant plumes is also one direction of our future work. Finally, the extension of the (optimized) orifice distributor method to scaled-up SMT storage tanks applied to medium/high-temperature conditions is also the aim of future studies.

Acknowledgement

This work is supported by the French ANR within the project OPTICLINE (ANR-17-CE06–0013). Technical supports from Mr. Julien Aubril and Mr. Arnaud Arrivé at LTEN are gratefully acknowledged.

Appendix 5A Supplementary measured and simulated results

In fig.4, we compare the measured and predicted absolute velocity fields at three moments during charging course. In order to get more velocity information, we draw the absolute, and the value in x/y-direction at certain fixed vertical and horizontal positions, as displayed in fig.5A.1, 5A.2, and 5A.3. At 100s, obviously, profile v_y along y-direction prove that inertia-forced thermal jet obtains the motion to go upward once it passed the neutral point whereas the difference of penetration length exists between measured and predicted results.

According to v_y along x-direction (e.g., y = -20 mm or -80 mm), measured buoyant plumes have more motion to expand and rise while measured cooled plumes at regime 4 also have more motion to go downwards. In general, the recirculation measured by PIV seems to have more motion and expand more serious at more lateral zone. As time increases, we found velocity fields at large scales increases because of thermal jet could avoid large loss of momentum at upper fluid zone near inlet port. That's the reason why we need intervene in the time evolution of inertia-forced entering thermal jet with the help of adapted flow distributors.



Fig.5A.1. Comparison of velocity profiles between CFD and PIV at t=100 s ($t^*=0.12$). Conditions: $Q_{in} = 1$ L min⁻¹ with initial uniform OBD



Fig.5A.2. Comparison of velocity profiles between CFD and PIV at t=400 s ($t^*=0.47$). Conditions: $Q_{in} = 1$ L min⁻¹ with initial uniform OBD

Because of time-averaged equations of motion for fluid flow, there is no complete flow details in the flow field, where a lot of small vortex structures are filtered out with time averaging. Moreover, the k- ε model, as one of RANS models computing the Reynolds stress, is done by the linear eddy viscosity equations, which is listed in Eq.32. In that case, the values of turbulent fluctuation u'_i in the direction of x, y and z is treated as isotropic.

TKE =
$$\frac{1}{2}\overline{u'_{l}u'_{l}} = \frac{1}{2}\left(\overline{u'_{x}}^{2} + \overline{u'_{y}}^{2} + \overline{u'_{z}}^{2}\right) = \frac{3}{2}\overline{u'_{x}}^{2}$$
 (32)



Fig.5A.3. Comparison of velocity profiles between CFD and PIV at t=700 s ($t^*=0.82$). Conditions: $Q_{in} = 1$ L min⁻¹ with initial uniform OBD

However, in actual measurement (initial momentum of thermal jet is along y direction), we resolved the TKE by Eq.33. we supposed only two turbulent fluctuations $\overline{u'_x}$, $\overline{u'_z}$ along x and z direction were isotropic. According to scale of turbulent fluctuations u'_x and u'_y , their difference couldn't be ignored. That's also one raison why simulated and measured evolution of velocity fields have the difference. Moreover,

$$\text{TKE} = \frac{1}{2} \left(\overline{u'_x}^2 + \overline{u'_y}^2 + \overline{u'_z}^2 \right) = \overline{u'_x}^2 + \frac{1}{2} \overline{u'_y}^2 \tag{33}$$

Furthermore, we added one measurement of isothermal charging $(T_h = T_c = 293.15 K)$ in order to obtain supplemental fluid information of thermal jet without the buoyancy effect.

Fig.5A.4(a) represented the velocity fields and we found that there is no time-evolution of such type of charging. In other words, once the starting period passed, the flow distribution could be regarded as steady-state whereas the time-evolution of thermal charging always exists.

Concerning to the mean velocity fields of thermal charging condition, two buoyant plumes appeared obviously at lateral zone.



Fig.5A.4. Mean velocity fields. (a)isothermal jet measured by PIV. Conditions: $Q_{in} = 1 \text{ L min}^{-1}$, $T_h = T_c = 293.15 \text{ K}$ with initial uniform OBD (b)isothermal jet simulated by CFD. Conditions: $Q_{in} = 1 \text{ L min}^{-1}$, $T_h = T_c = 293.15 \text{ K}$ with initial uniform OBD (c) thermal jet measured by PIV. Conditions: $Q_{in} = 1 \text{ L min}^{-1}$, $T_h = 333.15$, $T_c = 293.15 \text{ K}$ with initial uniform OBD (c) thermal jet measured by PIV. Conditions: $Q_{in} = 1 \text{ L min}^{-1}$, $T_h = 333.15$, $T_c = 293.15 \text{ K}$ with initial uniform OBD (c) thermal jet measured by PIV. Conditions: $Q_{in} = 1 \text{ L min}^{-1}$, $T_h = 333.15$, $T_c = 293.15 \text{ K}$ with initial uniform OBD



Fig.5A.5. Turbulent kinetic energy (TKE) profiles. (a)isothermal jet. Conditions: $Q_{in} = 1 \text{ Lmin}^{-1}$, $T_h = T_c = 293.15 \text{ K}$ with initial uniform OBD (b) thermal jet. Conditions: $Q_{in} = 1 \text{ Lmin}^{-1}$, $T_h = 333.15$, $T_c = 293.15 \text{ K}$ with initial uniform OBD

The measurement noise seems to be more serious at right side, as shown in the figures, such as fig.5A.3 or fig. 5A.5. It's because that the laser intensity decreased when the laser passes through the SMT. Limited by actual measurement condition, e.g., optic equipment, such noise isn't simple to be avoided. Concerning to the thermal charging, the TKE value isn't convergent since it is one time-dependent case. According to the resolution of camera and measured area, the used optic, the major velocity distribution could be clearly displayed whereas the magnitude of TKE on our case is too small to be dealt with.

Reference

Abdelhak O, Mhiri H, Bournot P. CFD analysis of thermal stratification in domestic hot water storage tank during dynamic mode. Build Simul 2015; 8: 421–429. https://doi.org/10.1007/s12273-015-0216-9

Abarbanel H, Holm, D.D., Marsden, J.E., Ratiu, T., 1984. Richardson number criterion for the nonlinear stability of three-dimensional stratified flow. Phys. Rev. Lett 1984;52:2352–2355. https://doi.org/10.1103/PhysRevLett.52.2352

Advaith S, Parida D, Aswathi K, Dani N, Chetia U, Chattopadhyay K, Basu S. Experimental investigation on single-medium stratified thermal energy storage system. Renew Energ 2020;164:146–155. https://doi.org/10.1016/j.renene.2020.09.092

Al-Azawii MMS, 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. https://doi.org/10.1016/j.applthermaleng.2020.115804

Al-Habaibeh A, Shakmak B, Fanshawe S. Assessment of a novel technology for a stratified hot water energy storage – The water snake. Appl Energ 2018;222:189–198. https://doi.org/10.1016/j.apenergy.2018.04.014

Altuntop N, Kilik Z, Ozceyhan V, Kincay O. Effect of water inlet velocity on thermal stratification in a mantled hot water storage tank. Int. J. Energy Res 2006;30:163–176. https://doi.org/10.1002/er.1134

Andersen E, Furbo S, Hampel M, Heidemann M, Mu["] ller-Steinhagen H. Investigations on stratification devices for hot water heat stores. Int. J. Energy Res 2008, 32, 255–263. https://doi.org/10.1002/er

Angui L, Feifei C, Wanqing Z, Bingjin S, Huang L, Effects of different thermal storage tank structures on temperature stratification and thermal efficiency during charging, Sol. Energy 2018; 173:882–892. https://doi.org/10.1016/j.solener.2018.08.025

Aristodemo F, Marrone S, Federico I. SPH modeling of plane jets into water bodies through an inflow / outflow algorithm SPH modeling of plane jets into water bodies through an in flow/out flow algorithm. Ocean Eng. 2015,105:160–175. https://doi.org/10.1016/j.oceaneng.2015.06.018

Assari M, Basirat Tabrizi H, Savadkohy 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. https://doi.org/10.1016/j.seta.2017.12.009

Bahnfleth W, Song J, Cimbala J. Measured and modeled charging of a stratified chilled water thermal storage tank with slotted pipe diffusers. HVAC & R Research 2003;9: 467–491. https://doi.org/10.1080/10789669.2003.10391081

Bai Y, Wang Z, Fan J, Yang M, Li X, Chen L, ... Yang J. Numerical and experimental study of an underground water pit for seasonal heat storage. Renew Energ 2020;150:487–508. https://doi.org/10.1016/j.renene.2019.12.080

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. https://doi.org/10.1016/j.ijheatmasstransfer.2013.01.047

Brückner S, Liu S, Miro L, Radspieler M, Cabeza LF, Lavemann E. Industrial waste heat recovery technologies: an economic analysis of heat trans- formation technologies. Appl Energy 2015;151:157e67. https://doi.org/10.1016/j.apenergy.2015.01.147

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. https://doi.org/10.1016/j.solener.2017.08.061

Chandra YP, Matuska T. Stratification analysis of domestic hot water storage tanks: A comprehensive review. Energy Build 2019;187:110–131. https://doi.org/10.1016/j.enbuild.2019.01.052

Chandra YP, Matuska T. Numerical prediction of the stratification performance in domestic hot water storage tanks. Renew Energ 2020;154:1165–1179. https://doi.org/10.1016/j.renene.2020.03.090

Calvet N, Gomez J, Faik A, Roddatis V, Meffre A, Glatzmaier G, … Py X. Compatibility of a postindustrial ceramic with nitrate molten salts for use as filler material in a thermocline storage system. Appl Energ 2013;109:387–393. https://doi.org/10.1016/j.apenergy.2012.12.078

Dallaire J. (2017). MODÉLISATION DES CHANGEMENTS DE PHASE SOLIDE-LIQUIDE : EFFET DE LA VARIATION DE LA DENSITÉ AVEC LA PHASE ET DU COUPLAGE. http://hdl.handle.net/20.500.11794/27716

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. https://doi.org/10.1016/j.solener.2017.03.008

Dehghan A, Barzegar A, Thermal performance behaviour of a domestic hot water solar storage tank during consumption operation, Energy Convers Manag 2011;52:468–476. https://doi.org/10.1016/j.enconman.2010.06.075

El-Amin MF, Al-Ghamdi A. Experiments and numerical simulation for a thermal vertical jet into a rectangular water tank. Results Phys 2018;10:680–692. https://doi.org/10.1016/j.rinp.2018.07.016

Engineering toolbox. https://www.engineeringtoolbox.com

Esen M, Yuksel T. Experimental evaluation of using various renewable energy sources for heating a greenhouse, Energy Build 2013;65:340–351. https://doi.org/10.1016/j.enbuild.2013.06.018

Flueckiger SM, Garimella SV. Latent heat augmentation of thermocline energy storage for concentrating solar power - A system-level assessment. Appl Energ 2014;116:278–287. https://doi.org/10.1016/j.apenergy.2013.11.059

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. https://doi.org/10.1016/j.solener.2017.12.062

Gallego-marcos I, Kudinov P, Villanueva W, Kapulla R, Paranjape S, Paladino D, ... Kotro E. Pool stratification and mixing induced by steam injection through spargers : CFD modelling of the PPOOLEX and PANDA experiments. Nucl Eng Des 2019; 347: 67–85. https://doi.org/10.1016/j.nucengdes.2019.03.011

García-Alba J, Bárcena JF, García A. Zonation of positively buoyant jets interacting with the waterfree surface quantified by physical and numerical modelling. Water 2020;12:1324 https://doi.org/10.3390/W12051324

Geissbühler L, Mathur A, Mularczyk A, Haselbacher A. An assessment of thermocline-control methods for packed-bed thermal-energy storage in CSP plants, Part 2: Assessment strategy and results. Sol Energy 2019;178: 351–364. https://doi.org/10.1016/j.solener.2018.12.016

Haller M, Cruickshank C, Streicher W, Harrison S, Andersen E, Furbo S. Methods to determine stratification efficiency of thermal energy storage processes - Review and theoretical comparison. Sol Energy 2009 ; 83 :1847–1860. https://doi.org/10.1016/j.solener.2009.06.019

He Z, Wang X, Du X, Xu C, Yang L. International Journal of Heat and Mass Transfer Cyclic characteristics of water thermocline storage tank with encapsulated PCM packed bed. Int J Heat Mass Transf 2019;139:1077–1086. https://doi.org/10.1016/j.ijheatmasstransfer.2019.05.093

Hosseinnia S, Sorin M. A systematic pinch approach to integrate stratified thermal energy storage in buildings. Energy Build 2021;232:110663. https://doi.org/10.1016/j.enbuild.2020.110663

Kalapala L, Devanuri J. Energy and exergy analyses of latent heat storage unit positioned at different orientations – An experimental study. Energy 2020; 194:116924. https://doi.org/10.1016/j.energy.2020.116924

Karbasi M, Taherian G, Minaean A. Experimental Study of Natural Convective Flow over a Hot Horizontal Rhombus Cylinder Immersed in Water via PIV Technique. J Appl Fluid Mech 2017; 10: 735–747. https://doi.org/10.18869/acadpub.jafm.73.23

Kocijel L, Mrzljak V, Gla V. Pressure drop in large volumetric heat storage tank radial plate diffuser. J Energy Storage; 29:101350 . https://doi.org/10.1016/j.est.2020.101350

Lai C, Lee J. Mixing of inclined dense jets in stationary ambient. J Hydro-Environ Res 2012; 6: 9–28. https://doi.org/10.1016/j.jher.2011.08.003

Li D, Anis A, Al F. Neap-spring variability of tidal dynamics in the Northern Arabian Gulf. Cont Shelf Res. 2020;197:104086. https://doi.org/10.1016/j.csr.2020.104086

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. https://doi.org/10.1016/j.enbuild.2013.11.086

Li Z, Tao W. A new stability-guaranteed second-order difference scheme. Numer Heat Transf B 2002;42: 349–365. https://doi.org/10.1080/10407790190053987

Majumdar R, Saha SK. Effect of varying extent of PCM capsule filling on thermal stratification performance of a storage tank. Energy 2019;178:1–20. https://doi.org/10.1016/j.energy.2019.04.101

Moffatt RJ. Describing the uncertaities in experimental results. Int J Exp Heat Transfer, Thermodyn Fluid Mech 1988; 1:3–17. https://doi.org/10.1016/0894-1777(88)90043-X

Motte F, Falcoz Q, Veron E, Py X. Compatibility tests between Solar Salt and thermal storage ceramics from inorganic industrial wastes. Appl Energ 2015;155:14–22. https://doi.org/10.1016/j.apenergy.2015.05.074

Mostafavi Tehrani S, Shoraka Y, Diarce G, Taylor RA. An improved, generalized effective thermal conductivity method for rapid design of high temperature shell-and-tube latent heat thermal

energy storage systems. Renew Energ 2019;132:694–708. https://doi.org/10.1016/j.renene.2018.08.038

Mira-Hernández C, Flueckiger S, Garimella S. Comparative analysis of single and dual media thermocline tanks for thermal energy storage in concentrating solar power plants. J. Solar Energy Eng 2015;137:031012. https://doi.org/10.1115/1.4029453

Mira-Hernández C, Flueckiger SM, Garimella SV. Numerical simulation of single- and dual-media thermocline tanks for energy storage in concentrating solar power plants. Energy Procedia 2014; 49: 916–926. https://doi.org/10.1016/j.egypro.2014.03.099

Miyoshi K, Takenaka N, Ishida T, Sugimoto K. Investigation of temperature fluctuation phenomena in a stratified steam-water two-phase flow in a simulating pressurizer spray pipe of a pressurized water reactor. Nucl Eng Des 2017;316:38–45. https://doi.org/10.1016/j.nucengdes.2017.02.006

Nelson J, Balakrishnan A, Srinivasa Murthy S. Experiments on stratified chilled-water tanks: Expériences menées avec des reservoirs d'accumulation d'eau glacée à stratification. Int J Refrigeration 1999;22:216–234. https://doi.org/10.1016/S0140-7007(98)00055-3

Njoku H, Ekechukwu O, Onyegegbu S. Normalized charging exergy performance of stratified sensible thermal stores. Sol Energy 2016;136:487-498. https://doi.org/10.1016/j.solener.2016.07.032

Nizami D, Lightstone M, Harrison S, Cruickshank C. Negative buoyant plume model for solar domestic hot water tank systems incorporating a vertical inlet. Sol Energy 2013; 87:53–63. https://doi.org/10.1016/j.solener.2012.10.001

Kuznetsov M, Yanez J, Grune J, Friedrich A, Jordan T. Hydrogen combustion in a flat semiconfined layer with respect to the Fukushima Daiichi accident. Nucl Eng Des 2015;286:36e48. https://doi.org/10.1016/j.nucengdes.2015.01.016

Oliveski De Césaro, Krenzinger A, Vielmo H. Comparison between models for the simulation of hot water storage tanks. Sol Energy 2003;75:121–134. https://doi.org/10.1016/j.solener.2003.07.009

Paiva B, Siegle E, Schettini C. Channel curvature effects on estuarine circulation in a highly stratified tropical estuary: The São Francisco river estuary. Estuar Coast Shelf Sci 2020; 238. https://doi.org/10.1016/j.ecss.2020.106723

Peltier W, Caulfield C. Mixing efficiency in stratified shear flows. Ann. Rev. Fluid Mech 2003; 35:135–167.

Shah L, Andersen E, Furbo S. Theoretical and experimental investigations of inlet stratifiers for
solar storage tanks. Appl Therm Eng 2005;25:2086–2099.https://doi.org/10.1016/j.applthermaleng.2005.01.011

Taher M, Bouchahm N, Guerira B, Bensaci C. On the thermal stratification inside a spherical water storage tank during dynamic mode. Appl Therm Eng 2019;159: 113821. https://doi.org/10.1016/j.applthermaleng.2019.113821

Shaikh W, Wadegaonkar A, Kedare SB, Bose, M. Numerical simulation of single media thermocline based storage system. Sol Energy 2018;174:207–217. https://doi.org/10.1016/j.solener.2018.08.084

Straneo F. Cenedese C. Dynamics of Greenlands glacial fjords and their role in climate. Annu Rev Mar Sci 2015;7: 89–112, https://doi:10.1146/annurev-marine-010213-135133.

Patankar SV. Numerical Heat Transfer and Fluid Flow, Washington: Hemisphere Publ. Corp, 1980.

Panthalookaran V, El-Amin W, Heidemann H. Müller-Steinhagen. Calibrated models for simulation of stratified hot water heat stores. Int J Energy Res 2008;32:661-676. https://doi.org/10.1002/er.1423

Punetha M, Choudhary A, Khandekar S. ScienceDirect Stratification and mixing dynamics of helium in an air filled confined enclosure. Int J Hydrogen Energ 2018;43: 19792–19809. https://doi.org/10.1016/j.ijhydene.2018.08.168

Roos P, Haselbacher A. Thermocline control through multi-tank thermal-energy storage systems. Appl Energ 2021;281:115971. https://doi.org/10.1016/j.apenergy.2020.115971

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. https://doi.org/10.1016/j.applthermaleng.2020.115010

Wang Z, Zhang H, Dou B, Huang H, Wu W, Wang Z. Experimental and numerical research of thermal stratification with a novel inlet in a dynamic hot water storage tank. Renew Energ 2017;111:353–371. https://doi.org/10.1016/j.renene.2017.04.007

Wang S, Davidson J. Performance of a rigid porous-tube stratification manifold in comparison to an inlet pipe. Sol Energy 2017;146:298-308. https://doi.org/10.1016/j.solener.2017.02.045

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. https://doi.org/10.1016/j.solener.2019.11.00

Chapter 6. Conclusions and future prospects

In this chapter, the main findings of this thesis have been firstly summarized. Then, the challenges and future prospects have been proposed.

6.1 Thesis summary

The **second chapter** provided a readable literature survey on thermal energy storage system, predominantly on thermocline-based storage system. Different from the conventional two-tank storage mode and thermocline-based dual-medium TES, the issue about flow misdistribution and thermocline degradation into a SMT is eager to be solved and studied. This chapter started with the TES intergradation concept, charging/discharging operating, then various methods to characterize thermal and hydrodynamic performances were summarized in detail. Concerning that the entering injection during charging and discharging processes isn't a simple inertia-forced jet, the large difference of density and velocity between the inlet thermal jet and calm stored fluid must introduce the more complex mass and heat transfer movement. Therefore, the primary impacts related to operating conditions and TES geometry were in survey in this chapter. The given TES geometry (no packing, sensible/latent packing, inlet position, flow distributor, etc) and operating condition (inlet flow rate, working temperature range, etc) communally determine the thermocline evolution and flow distribution. In particular, their interaction effect was reviewed as well as the original practical classification on flow diffusers.

The third chapter focused on the novel CFD-based optimization algorithm to the realization of the smooth thermocline evolution and quasi-thermally piston flow among perforated plate diffuser. One original optimization algorithm was developed to determine the optimal orifice size distribution on the baffle, using a novel optimality criterion of equal passage time of hot front for every orifice. The tendency of this optimality criterion along the optimization steps was correspondent to that of thermocline surface where the thermocline surface can be finally minimized. Numerical results represented that the proposed optimization algorithm could significantly improve the thermal performances, indicated by the increased values of charging/discharging /overall efficiency, and the capacity ratio, ex., the fully charging efficiency be increased by 29% by comparing the unstructured manifold geometry and the one with optimized baffles. Moreover, originally defined intermediate evaluation indicator was introduced to characterize the real-time thermal behavior, which could reduce the computational cost of the optimization problem by a factor of 6 at least. In third chapter, we built up a 2D planar CFD model and set up mean density value between 600 K-800 K to simplify the numerical model and to reduce the computational time. But a close look at the influence of the variable density values on the thermocline behaviors is in fact needed. Another important step is to extend the proposed optimization algorithm to 3D, so as to tackle the flow distribution problem in real thermocline

storage tanks with different shapes (rectangular, cylindrical or others). Therefore, these two primary drawbacks were solved in the followed fourth and fifth chapters.

The **fourth chapter** extended the SMT investigation in experiment by use of ring-opening plate equalizer. A numerical optimization procedure in series for both upper and bottom equalizers was conducted, then optimized equalizers were equipped into SMT. The different TES geometries without, with initial equalizers or with optimized equalizers were investigated to be compared. Consequently, the feasibility of the original optimization algorithm was applied into an experimentally validated 2D axisymmetric model and validated in experiment by the better thermal stratification and these improved efficiencies in first-law of thermodynamic and in second–law of thermodynamic. In particular, at a given high flow rate, our optimization algorithm could maximize the energy storage efficiency and exergy efficiency by the means of minimizing the inherent exergetic losses after addressing the irreversibility caused by the mixing. For example, Exergy efficiency η_{Ex} within S1, S2 and S5 is increased from 56.3% to 79.6 to 84.2% within the given operating condition ($T_h = 323.15 K$, $Q_{in} = 0.3 L min^{-1}$).

The fifth chapter extended the proposed optimization algorithm to numerical and experimental 3D TES-SMT where two perforated plate diffusers were installed. The completed time-dependent mixing regimes including: (1) Inertia dominated regime; (2) Transition regime; (3) Buoyancy dominant regime; (4) Lateral dispersion regime were observed into the main fluid storage region between the upper and bottom buffer regions by the help of using Particle Image Velocimetry (PIV) technique. Based the both predicted and measured results, transient velocity distribution was keeping changing in the charging period as well as the thermocline region evolution. Therefore, the conventional flow homogenization distribution which regarded as the optimal flow distribution seems to be too ideal to realize. That's why the main target of our original optimization algorithm is to smooth the thermocline region and stabilize thermal stratification instead of homogenization the flow distribution. Precisely, two local hydrodynamic indictors: gradient Richardson number and element Péclet number were solved, and they represented thermal stratification ability against convective perturbation and the competing relation between convective and diffusion heat transfer, respectively along the optimization steps. In addition, the timeevolution of the velocity profiles under injecting flow rates or different working temperature ranges were recorded and measured by the PIV method. Under the upper diffuser with uniform orifices, the inertia-forced jet still existed within the main storage region where the large buoyant plumes result to large mixing region. However, under the optimized upper diffuser, no more obvious inertia forced jet appeared, and penetrated multiple jets flowed out from upper diffuser whose length of jet penetration could be maintained at the equivalent distances. Charging efficiency, capacity ratio and bulk Richardson number all represented the better thermal performance and the favorable results in sensibility analysis at different operating conditions by using the original optimization algorithm.

6.2 Perspectives

In our cases, our previous measured area by PIV is the main fluid storage between the upper and bottom diffusers, thus we ignored the flow status into upper and bottom buffer regions. Besides, the flow area is invisible near the baffles because of the baffles fixing by using the translucent glue. Therefore, the interaction of multi-jet among the neighbor orifices is very interesting in our future work as well as improving the sealing issue of the squared tank.

The cyclic effect and the perturbation of operating condition such as inlet temperature have pertinent impacts on the effective heat capacity, the utilization rate, etc. We suppose that the stability against the cyclic effect and perturbation with different flow distributors must different and is worthy to be deeply explored.

Concerning the CFD model, the majority used turbulent model by using Ansys FLUENT code haven't been calibrated, in other words, these default coefficients such as buoyancy term and energy dissipation coefficient need to be adjusted by the real-world experiment results.

Multi-layer packing mode including sensible or latent materials has been largely discussed. However, recent research focus on the main issues such as how to enlarge the heat capacity (solutions: material having large phase-change enthalpy...), how to stabilize outlet temperature (solutions: improve operating conditions, last layer using latent material, packed fillers' size...), how to extend the life cycle of TES system (solution: physical stability and chemical compatibility...), etc. The interface effects between two layers are firstly proposed [Li et al.2018], where expanding / shortening phenomenon of thermocline thickness will appear while the thermocline is passing the interface of two layers. If this shortening effect could be used properly as the last two layers near outlet port, such structure may ensure the larger charging/discharging time thus improve the thermal efficiencies.

Further, the homogenous packing mode (simulated porous phase) has been deeply discussed where the packing factors are kept constant such as the filler diameter, the porosity into one layer. Anisotropic packing method may need to be deeply explored to improve the heat transfer between HTF and packing mediums.

Reference

Li, M. J., Qiu, Y., & Li, M. J. (2018). Cyclic thermal performance analysis of a traditional Single-Layered and of a novel Multi-Layered Packed-Bed molten salt Thermocline Tank. Renewable Energy, 118, 565–578. https://doi.org/10.1016/j.renene.2017.11.038

Synthèse de Thèse

Contexte général

Le principal goulot d'étranglement de technologie empêchant le déploiement efficace des ressources d'énergies renouvelables est leur nature intermittente, ainsi l'offre et la demande d'énergie ne sont pas équilibrées dans le temps [Chavan et al., 2020].Le stockage d'énergie thermique (TES) est proposé comme une technologie potentielle et une solution rentable afin de résoudre les problèmes d'inadéquation et d'améliorer la fiabilité et la dispatchabilité [Suresh et Saini, 2020; Pelay et al., 2017]. À l'exception de l'énergie solaire, l'énergie thermique provenant de nombreuses sources variables telles que l'énergie géothermique, les centrales à combustibles fossiles, les centrales nucléaires, la chaleur résiduelle industrielle et la biomasse peut également être utilisée dans le système TES [Alva et al., 2018].

Actuellement, les options disponibles du système TES comprennent principalement le stockage à deux réservoirs et le stockage à réservoir unique de technologie thermocline. Au cours de la dernière décennie, le système de stockage de type thermocline attire de plus en plus l'attention en raison de sa rentabilité (environ 35% moins cher) et de ses caractéristiques de compacité spatiale comparé au système TES conventionnel à deux réservoirs [Grirate et al., 2016].

Obstacles à la recherche

En ce qui concerne le système TES de type thermocline pendant le processus de charge / décharge, la stratification assure le fluide caloporteur chaud flottant vers la région supérieure du réservoir et le fluide froid de grande densité situant à la région inférieure. En même temps, la région de transfert de chaleur avec le gradient de température, appelée thermocline, est située à leur interface. D'une manière générale, le gradient de température de la thermocline doit être maintenu plus large possible afin d'augmenter la quantité de chaleur stockée dans la région au-dessus. En d'autres termes, limiter le volume de la thermocline est le principal défi technologique pour obtenir de meilleures performances thermiques et la stratification thermique [Li et Zheng, 2016]

Les obstacles principaux à la technologie TES de type thermocline sont les suivants :

Une distribution homogène du débit ne peut pas être maintenue portant pour le réservoir de stockage à thermocline utilisant uniquement du fluide (SMT) [Luo et al., 2020]. Le jet thermique d'entrée occupant la grande différence de vitesse et de température comparé au fluide calme initialement stocké dans le SMT doit provoquer le fort mélange des fluides chauds et froids, et donc de conduire à la dégradation de thermocline et aux performances thermiques réduites.

La technologie de stockage de l'énergie par remplissage sensible possédant l'avantage d'être bon marché et mature. Cependant, étant donné que le transfert de chaleur imparfait a lieu entre le HTF et le milieu solide, la diffusion thermique accélérera la dégradation de la thermocline. Ses inconvénients principaux aussi concernent une faible densité énergétique, température de sortie potentiellement instable. Dans la conception de la structure d'remplissage, il n'y a pas d'accord pour la solution optimale.

Le matériau à changement de phase (PCM) a attiré l'attention en raison de son importante chaleur latente de fusion. Mais, les barrières technologiques limitent sévèrement l'utilisation du PCM, y compris le coût élevé [Mostafavi Tehrani et al., 2019], la résistance thermique plus élevée fournie par sa conductivité thermique faible [Xu et al., 2015] and issue de fuite [Chandel and Agarwal, 2017].

En procédure de la commercialisation de TES, la durée de vie est également importante. Les issues existent, ex., la corrosion du sel fondu (HTF) et le cliquet thermique de la paroi du réservoir [Chang et al., 2016; Dahash et al., 2019; Pelay et al., 2020] à la condition d'une usine CSP intégrée avec TES système.

Objectif de recherche

En ce qui concerne ces barrières technologiques évoquées ci-dessus, l'optimisation du stockage de l'énergie thermique par la technologie thermocline est notre objectif principal. Nous visions à réaliser les performances thermiques et la stratification favorables à l'aide de distributeurs de fluide optimisés. Dans cette thèse, nous avons eu tendance à créer un algorithme d'optimisation original basé sur les distributeurs de fluide afin de maintenir la stratification thermique de la région thermocline. De plus, nous étions impatients d'explorer en profondeur le mécanisme de mélange dans le réservoir SMT, puis de minimiser l'effet du mélange à la dégradation de la thermocline avec les distributeurs de optimisés. Nous voulions mener l'enquête expérimentale combinée à la simulation CFD pour explorer la conception d'optimisation sur l'efficacité du système de stockage thermique.

Contenu de la thèse

Chapitre 2 : Ce travail fournit une synthèse bibliographique portant sur le stockage d'énergie thermique de type thermocline. Tout d'abord, les indicateurs les plus utilisés en caractérisant les performances thermodynamique et hydrodynamique sont résumés. Et puis, avec un accent particulier sur les impacts d'influence de la géométrie du réservoir et des conditions de fonctionnement sur les performances thermiques du réservoir de stockage SMT, en particulier l'effet de la position d'entrée, de la conception du distributeur de débit, de la plage de température de fonctionnement et du débit.

On trouve que leurs impacts sur l'thermocline ainsi que les conditions d'opération recommandées sont variés par les systèmes de stockage sans emballage, avec emballage sensible / latent. Le compromis de débit optimal dépend également des conditions de fonctionnement également. Le mode de garnissage en lit fixe, représente son meilleur caractère robuste dans l'analyse de sensibilité par rapport à l'autre structure de garnissage à haute porosité. Le mécanisme de mélange déterminé par l'advection-diffusion thermique mérite d'être approfondi pour le réservoir SMT. Les solutions conventionnelles pour atteindre une distribution de flux calme et une stratification de température stable sont également passées en revue, ex., les distributeurs de flux précédents publiés peuvent être divisés en 3 types en fonction de leur fonction principale : multi-branches, flux inversé et flux radicalaire.

Chapitre 3 : Cette étude vise à résoudre le problème de mauvaise distribution du débit dans le système de stockage à thermocline dans un seul réservoir en structurant des distributeurs appropriés au-dessus et en dessous. La solution technique est basée sur l'insertion de l'distributeur en forme de plate perforée. Un 2D CFD modèle est créé pour calculer les profils de débit et de température transitoires dans le réservoir de stockage aux opérations de chargement ou de déchargement.

Un algorithme d'optimisation original est développé pour déterminer la distribution optimale de la taille des orifices sur le distributeur perforé, en utilisant un nouveau critère d'optimalité lié aux tous les temps de passage de front chaud en traversant les orifices. L'objectif principal de création de cet algorithme d'optimisation est pour homogénéiser les temps de passage du front thermique, de manière à améliorer la stratification thermique. De plus, un nouvel indicateur d'évaluation intermédiaire est introduit pour caractériser le comportement thermique pour le travail optimisation, ce qui pourrait réduire le coût de calcul d'un facteur au moins de six. Les résultats numériques ont montré que l'algorithme d'optimisation proposé pourrait améliorer considérablement les performances thermiques, indiquées par les valeurs favorables d'efficacité de charge / décharge, le rendement de capacité. L'étude paramétrique, y compris des facteurs le rapport de hauteur/diamètre, nombre d'orifice, nombre de Reynolds, Porosité fluide du distributeur, angle de cône du réservoir, également démontre que cette méthode d'optimisation était robuste, efficace et efficiente.

Chapitre 4 : Ce chapitre présente une nouvelle procédure d'optimisation en série sur la base du distributeur de fluide avec ouverture en forme d'anneau (ROPD) qui pourrais être pratiquement utilisé dans le système TES réel pour minimiser le mouvement de mélange et maintenir la stratification thermique. Dans un premier temps, un modèle CFD validé a suivi une telle procédure pour concevoir les deux distributeurs au-dessus et en-dessous avec les conditions de fonctionnement fixées. Évolution temporelle du volume de thermocline surveillé s'avère le système TES avec les deux distributeurs restreint efficacement la propagation de thermocline. Le champ de
la vitesse montre que ROPD empêche la pénétration de jet thermique dominé par l'inertie dans la région de stockage intermédiaire. Dans cette étude, des mesures expérimentales ont été réalisées pour un réservoir de stockage SMT cylindrique soumis aux opérations de chargement ou de déchargement. Des ROPDs avec ou sans configurations optimisées ont été installés dans la cuve et leur capacité à atténuer les perturbations induites par le jet thermique a été testée. Les principales conclusions sont résumées comme suit:

- L'algorithme d'optimisation proposé dans [Lou et al., 2020] a été validé expérimentalement et s'est avéré efficace, efficient, robuste et simple à mettre en œuvre.
- Une plus grande différence de température de fonctionnement entre les HTFs favorise le niveau de stratification de température pour le processus de charge. Pour la décharge, l'influence de la différence de température de travail est moins évidente dans nos conditions testées.
- Dans la condition de TES avec deux ROPDs sans optimisation, l'efficacité de charge ou de décharge diminue fortement lorsque le débit d'entrée est supérieur à 1 L min⁻¹ (*Re_{tank}*>131). Lorsque des ROPDs optimisés sont utilisés, l'influence du débit d'entrée peut être limitée dans une certaine mesure.
- Une série d'études paramétriques réalisées démontre que la combinaison des ROPD optimisés pourrait atténuer efficacement les impacts du jet thermique et ainsi améliorer significativement les performances globales du réservoir de stockage pour tous les cas testés. Par exemple, comparé à TES sans distributeurs, le rendement exégétique (η_{Ex}) pourrait être amélioré de 56,3% à 84,2% (condition : T_h = 323,15 K, T_c = 293,15 K, Q_{in} = 0,3 L min⁻¹, décharge).

Chapitre 5 : Afin d'explorer en profondeur le mécanisme de mélange et l'évolution temporelle de la vitesse issue du jet thermique, un travail numérique et expérimental sur la caractérisation et la gestion des comportements dynamiques de thermocline dans un réservoir carré à l'aide de technique d'visualisation (PIV). La première partie de l'étude s'est focalisée sur l'optimisation de la configuration du distributeur à orifice (OBD) à l'aide de l'algorithme proposé dans notre étude précédente [Lou et al., 2020]. Selon la simulation CFD de l'opération de charge, des profils de vitesse et de température étroitement couplés et des comportements thermo-hydrodynamiques complexes du flux d'injection ont été observés.

• L'optimisation de l'OBD au-dessus pourrait effectivement améliorer la stratification de la température, indiquée par les valeurs croissantes du nombre $Ri_{gradient}$ d'une étape d'optimisation à la suivante (toutes $Ri_{gradient} > 1$ à l'étape finale).

• L'OBD optimisé pourrait limiter le transfert de chaleur dominé par la convection ($Pe_x \ll 1$) et atteindre un transport de convection / diffusion équilibré le long de la direction y ($Pe_y=1$) sur une très courte distance en aval l'OBD au-dessus, conduisant à une zone de thermocline plus aplatie et plus mince comparé à celle avec l'OBD initial à orifice uniforme.

Des expériences de charge ont ensuite été effectuées pour le réservoir carré de stockage à l'échelle du laboratoire équipé d'un OBD à orifice uniforme ou optimisé. L'évolution des profils de vitesse locale sous différents débits ou températures d'injection a été enregistrée et mesurée par la méthode PIV. Les performances globales du réservoir de stockage (caractérisées par le nombre de Richardson, le rendement de capacité et l'efficacité de charge) ont également été calculées et comparées.

- Les panaches flottants et leur évolution dynamique ont été visualisés. On a constaté que la longueur de pénétration du jet pour l'OBD à orifice uniforme augmentait avec l'augmentation du débit d'entrée et la diminution de la différence de température, la force de flottabilité plus faible étant plus difficile à équilibrer la force d'inertie.
- Pour le cas OBD optimisé, les impacts ont été jugés beaucoup moins importants par le jet central obstrué et par les interférences du panache entre les multiples jets.
- Un faible débit d'entrée et une grande différence de température sont des facteurs positifs pour atteindre un rendement de stockage et une efficacité de charge élevée si le jet dominé par la force d'inertie entrant ne peut pas être correctement géré sans distributeur propre.
- De meilleures performances thermiques du réservoir de stockage pourraient toujours être obtenues en utilisant l'OBD optimisé, offrant plus d'espace pour sélectionner les plages de fonctionnement appropriées des systèmes de stockage SMT dans différents domaines d'application visés.

Reference

Alva G, Lin Y, Fang G. An overview of thermal energy storage systems. Energy 2018;144:341–378. https://doi.org/10.1016/j.energy.2017.12.037

Chandel S, Agarwal T. Review of current state of research on energy storage, toxicity, health hazards and commercialization of phase changing materials. Renew Sustain Energy Rev 2017; 67: 581–596. https://doi.org/10.1016/j.rser.2016.09.070

Chang Z, Li X, Xu C, Chang C, Wang Z, Zhang Q, ... Li Q. The effect of the physical boundary conditions on the thermal performance of molten salt thermocline tank. Renew Energ 2016;96: 190–202. https://doi.org/10.1016/j.renene.2016.04.043

Chavan S, Gumtapure V. Numerical and experimental analysis on thermal energy storage of polyethylene/functionalized graphene composite phase change materials. J Energy Storage 2020; 27:101045. https://doi.org/10.1016/j. est.2019.101045.

Dahash A, Ochs F, Janetti MB, 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 Energ 2019;239:296–315. https://doi.org/10.1016/j.apenergy.2019.01.189

Grirate H, Agalit H, Zari N, Elmchaouri A, Molina S, Couturier R, Experimental and numerical investigation of potential filler materials for thermal oil thermocline storage, Sol. Energy 2016;131: 260–274, https://doi.org/10.1016/j.solener.2016. 02.035.

Li G, Zheng X. Thermal energy storage system integration forms for a sustainable future. Renew Sustain Energy Rev 2016;62:736–757. https://doi.org/10.1016/j.rser.2016.04.076

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. https://doi.org/10.1016/j.ces.2014.11.051

Mostafavi Tehrani S, Shoraka Y, Nithyanandam K, Taylor RA. Shell-and-tube or packed bed thermal energy storage systems integrated with a concentrated solar power: A techno-economic comparison of sensible and latent heat systems. Appl Energ 2019;238:887–910. https://doi.org/10.1016/J.APENERGY.2019.01.119

Pelay U, Luo L, Fan Y, Stitou D, Rood M. Thermal energy storage systems for concentrated solarpowerplants.RenewSustainEnergyRev2017;79:82–100.https://doi.org/10.1016/j.rser.2017.03.139

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. https://doi.org/10.1002/ep.13388

Suresh C, Saini R. Experimental study on combined sensible-latent heat storage system for different volume fractions of PCM. Sol Energy 2020;212:282–296. https://doi.org/10.1016/j.solener.2020.11.013

Xu B, Li P, Chan C, Tumilowicz E. General volume sizing strategy for thermal storage system using phase change material for concentrated solar thermal power plant. Appl Energ 2015;140:256–268. https://doi.org/10.1016/j.apenergy.2014.11.046

Acknowledgements

Throughout my dissertation around three years, undertaking this PhD is a truly lifechanging experience for me and it takes a long journey to complete. It would not have been possible to do without the support and assistance that I received from many people. I would like to first thank the French National Research Agency (ANR) for the funding opportunity to undertake my thesis work within the framework of OPTICLINE project (ANR-17-CE06–0013).

From the bottom of my heart, thank you for my supervisor Mme. LUO Lingai, for her guidance and support that have considerably helped me at all crucial points throughout the time of my thesis.

I would like to express my sincere gratitude to my co-director Mr.FAN Yilin, for his great availability, the unwavering support and belief in me. His extensive knowledge and plentiful experience in solving scientific and practical problems has always inspired me to improve the quality of the research work.

Many thanks to Mr. BAUDIN Nicolas and Mr. Roux Stéphane who help to establish the optic measurement system. The advancement related to optic measurement can't be realized without the immense help from Mr. BAUDIN Nicolas.

I'd like to thank the members of LTeN, starting with the director of our laboratory Mme Castelain Cathy for her continuous help.

My thanks also go out to the support from the study and manufacture department SEF, in particular the support from Arnaud and Julien who are so generous with their expertise and time in fabricating the experimental setups and establishing the control and monitor system. Owing to their assistance, it accelerated and facilitated me accomplish the operational feasibility of experimental test-rig and fluid loops.

I would like to say a special thank, the good people of administrative departments, Amandine and Michelle-Anne, for their assistance, understanding and encouragement. Many thanks also to Nathalie of IT department for her continuous support on hardware and software equipment.

In addition, I would like to thank all members of the project OPTICLINE, Mr. FIAMANT Gilles, Mr. OLIVES Régis, Mr. NEVEU Pierre, Mr. Falcoz Quentin, Mr. REBOUILLAT Baptiste, Mme. VANNEREM Segolene, Mme. GERLAND Sarah, and Mr. DEJEAN Guilhem. I thank them for their various comments, questions, and discussion during our regular internal exchange meetings which allow me to think more critically in searching for explanations.

I'd like to thank all the former and current doctoral and post-doctoral students. It is their kind help that has made my study and life in LTeN a convivial moment. The special thanks are expressed to Baoshan, Eliane and Li for their support and encouragement.

Finally, I would like to express my gratitude to my friends in France, Marlaine, Phuimai and Peng. Without their tremendous accompanies and encouragement in the past six years, it would be impossible for me to complete my engineering and PhD study.

Wanruo LOU Feb,2021,

Nantes





Titre : Optimisation du stockage de chaleur par la technologie thermocline

Mots clés : Stockage d'énergie thermique (TES); Seul-médium réservoir (SMT); Stratification thermique; Algorithme d'optimisation; Jet thermique; Distribution de vitesse

Résumé : Le stockage d'énergie thermique par la technologie thermocline est interessant par son faible coût. Cependant, l'entrée / sortie inappropriés peuvent provoquer un fort mélange de fluides chauds et froids et perturber la stratification de la température, entraînant des réductions de performances. L'objectif de la thèse est d'optimiser la conception de stockage thermocline avec un haut rendement énergétique et une bonne qualité de stratification thermique par diffuseurs de fluide optimisés.

Un algorithme d'optimisation original basé sur CFD est développé pour déterminer la distribution optimale de vitesse et la manière de propagation de la thermocline dans un réservoir sans garnissage (SMT). Une methode pratique d'homogénéisation les temps de séjour du front thermique pour aplatir la zone thermoclineest a été proposé, basé sur l'insertion d'un plaque perforé optimisé.

Les résultats obtenus par l'algorithme d'optimisation sont ensuite validés expérimentalement avec un ballon de stockage SMT à l'échelle du laboratoire, en mesurant les évolutions locales de température du fluide lors des opérations de charge et de décharge. Les résultats montrent que notre distributeur à ouverture annulaire optimisé pourrait considérablement améliorer l'efficacités énergétique et exergétique dans une large plage de conditions d'opération.

Après cela, le mécanisme de mélange dynamique et le phénomène d'entraînement du jet sont particulièrement étudiés par mesures des profils d'écoulement à l'intérieur d'un SMT rectangulaire en utilisant la méthode de vélocimétrie d'image de particules (PIV). La relation concurrente locale entre les mécanismes de transfert de chaleur par convection et de diffusion sur la dégradation de la stratification de température est particulièrement explorée.

Title: Optimization of thermal energy storage by the thermocline technology

Keywords: Thermal energy storage (TES); Single medium thermocline (SMT); Temperature stratification; Optimization algorithm; Thermal jet; Flow distribution

Abstract: It has received an increasing attention because integrated TES systems can largely enhance the reliability and the dispatchability. Low-cost single storage tank based on the thermocline technology becomes an alternative to commonly used two-tank system. However, the improper inlet/outlet manifolds may cause the strong mixing of hot and cold fluids and disturb the temperature stratification, resulting in reduced thermal performances of the storage tank. An original CFD-based optimization algorithm is developed 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 is proposed, based on the insertion of a geometrically optimized perforated

The feasibility of optimization algorithm is then 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. Results show that optimized Ringopening plate distributors can significantly imporve the energy and exergy efficiencies under large range of operating conditions.

After that, the mechanism of dynamic mixing and the jet entrainment phenomenon are particularly addressed by measuring the flow profiles inside a rectangular SMT tank using Particle Image Velocimetry (PIV) method. The local competing relation between the convection and diffusion heat transfer mechanisms on the degradation of temperature stratification is particularly explored.