GEOMETRIC OPTIMIZATION OF HIGH TEMPERATURE SHELL AND TUBE LATENT HEAT THERMAL ENERGY STORAGE

[1]S. SAEED MOSTAFAVI TEHRANI, [2]GONZALO DIARCE, [3]ROBERT A. TAYLOR, [4]POUYA SABERI, [5]ARDALAN SHAFIEI GHAZANI

[1,3]School of Mechanical and Manufacturing Engineering, University of New South Wales, Sydney, NSW, Australia
[2]Escuela de Ingeniería de Bilbao, University of the Basque Country UPV/EHU, Rafael Moreno Pitxitxi 2, Bilbao 48013, Spain
[4]School of Mechanical and Manufacturing Engineering, K.N. Toosi University of Technology, Tehran, Iran
[5]Mechanical Engineering Department, Sharif University of Technology, Tehran, Iran
E-mail: [1]s.mostafavitehrani@unsw.edu.au,[2]gonzalo.diarce@ehu.es,[3]robert.taylor@unsw.edu.au, [4]saberi.pya@gmail.com, [5]ardalan.shafiei@gmail.com

Abstract— A simple geometry shell and tube heat exchanger provides a straightforward design for near-term integration of latent heat thermal energy storage systems in concentrated solar thermal plants, but currently there is no literature available for this configuration in the 286-565 ˚C temperature range. Therefore, the objective of this work is to evaluate the potential of this configuration for CST-tower plants by proposing a proper design method. The work has been done by optimizing the main geometric parameters involved along with considering a market ready phase change material (H500 salt). The optimization consisted of fixing the PCM volume while varying the other geometric parameters simultaneously over a wide range. The goal was to achieve the highest amount of total stored/delivered energy in a certain amount of time with a minimum heat transfer surface area. For the selected PCM, the optimum area was found 36-63 $m^2.GJ^{-1}$ (0.12-0.22 $m^2.kWh_{th}^{-1}$). The storage charging and discharging efficiency for the selected PCM over a cycle of continuous charging and discharging were found ~99% and 85%, respectively. The results also imply that the shell and tube LHTES system is technically competitive with the conventional two-tank molten salts because of its high efficiency.

Keywords— High temperature, Phase change material, Optimization, Shell and tube tank

Nomenclatures	
C_P	specific heat at constant pressure [J/kg.K]
C_{N_p}	pipe numbers correction factor [-]
d	inner diameter of latent heat storage unit [m]
h	enthalpy [J/kg]
k	thermal conductivity [W/m.K]
L	length of tank [m]
$\dot{m}$	mass flow rate [kg/s]
N_p	number of latent heat storage modules (pipes) [-]
$\acute{N}_p$	corrected number of pipes [-]
$\dot{Q}$	amount of stored/discharged energy [W]
R	outer radius of a cylinder [m]
r_o	inner radius of pipe [m]
T	temperature [K]
T_{m1}	lower melting temperature of PCM [K]
T_{m2}	higher melting temperature of PCM [K]
$W_{CST-Tower}$	electrical power output [W]
Greek symbols	
ρ	density [kg/m^3]
λ	convective heat transfer coefficient [$W/m^2.K$]
ΔH	latent heat [J/kg]
$\forall$	volume [m^3]
Subscripts	
bottom	bottom of tank (Z/L=0)
ch	charging process
dis	discharging
f, HTF	solar salt fluid (heat transfer fluid)
Abbreviations	
CST-Tower	concentrated solar thermal Tower
HTF	heat transfer fluid
LHTESS	latent heat thermal energy storage
PCM	phase change material

I. INTRODUCTION

As reported in [1], Tower systems represent the next generation of CST plants as they can achieve higher efficiency and lower cost.By integrating a concentrated solar thermal- Tower plant with thermal energy storage (TES), excess solar energy can be stored during periods of high insolation, and then discharged later during peak demand when it provides the most value to the grid/plant operator.The literature has shown that a well-designed TES system can make power plants more cost effective – 2.5% higher net present value (NPV) and 10% reductionin thelevelized cost of electricity have been reported [2, 3].

The most maturelarge-scale TES technology is the two-tank system, based on nitrate molten salts[3, 4]. In spite of high reliability of this kind of sensible heat storage (SHS) system, it requires a relatively high volume andcapital cost. The next most mature technology uses a combination of the same molten salts with an inexpensive granulated rock. In this type of system only one tank is used, which relies onthermocline-driven flow[5-8]. Commonly referred to as a the double media thermocline system, this technology provides a pathway towards lower volume and cost, but is limited by the thermal ratcheting phenomenon (e.g. cyclic stress/strain), which can lead to structural failure of the tank[6]. To cope with this problem, an encapsulated phase change material (PCM) can be incorporated into the design, creatinga packed bed latent heat thermal energy storage

(LHTES) system to replace the granulated rocks[9-11].While it overcomes thermal ratcheting, this configuration addssignificant pressure-drop and capital cost [12].While research on the previous configurations is ongoing, there remains a large scope for developing alternative compact storage systems that are competitive with the conventional two-tank molten salts.One alternative option that seems promising is cylindrical latent heat storage systems. Yanbing et al. [13] showed that these (at least for low temperatures) are very competitive with packed bed containers in terms of their thermal performance. Hence, the focus of this study is to determine the feasibility of using a simple cylindrical geometry latent heat storage systemdesigned for high temperature solar thermal power- Tower based plants[14].In terms of studies on simple geometry cylindrical units (shell and tube heat exchanger without external fins or heat pipes), although there are no high temperature studies for CST-Tower (286-565˚C), there are several numerical and experimental studies for other temperature ranges which can be drawn upon for validation and comparison[15-18]. Regarding the geometric parameter selection of shell and tube LHTES system, there are several studies in the literature, but all of these studies only show the trends rather than what is an optimum selection [19-22]. In fact, the literature has already shown that increasing the non-dimensional length, or decreasing the non-dimensional radius, or increasing the tube length usually enhances the LHTES unit performance. However, the literature lacks a methodology that helps with selecting the optimal geometric parameters. Therefore, this study expands the literature by proposing a method to select the optimal geometric parameters. The results are presented for 1 MW_e CST plant.

II. SHELL AND TUBE LHTES SYSTEM

As shown in**Fig. 1**, the cylindrical LHTES unit proposed for this study consists of concentric tubeswhereby the HTF and PCM are segregated. The HTF flows through the inner tubes and exchanges heat with the PCM in the surrounding region (e.g. inside the outer tube). During the charging process (melting), the hot HTF flows from top to bottom. During the discharging process (solidification), the flow is reversed.The theoretical treatment of this problem is similar to that described by Esen[23] and Visser[24].

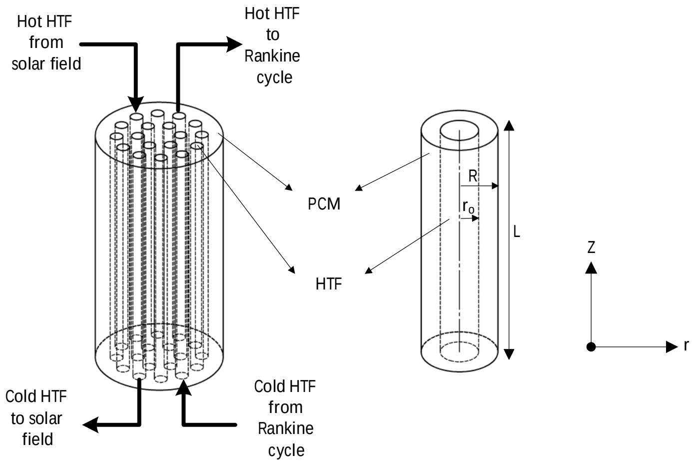


Fig. 1.Schematic of shell and tube LHTES unit.

The mathematical model for the HTF is shown in Eq. 1, while the energy equation for the PCM is displayed in Eq. 2:The initial and boundary conditions for the problem are detailed from Eq. 3 to Eq. 7.

$$(\rho C_P)_f \pi r_o^2 \frac{\partial T_f}{\partial t} = -(\dot{m} C_P)_f \frac{\partial T_f}{\partial z} + \lambda(T_{r=r_o} - T_f)2\pi r_o \quad (1)$$

$$(\rho C_P)_P \frac{\partial T}{\partial t} = div\ (k_P\ grad\ T) \quad (2)$$

$$t = 0, T = 286.5^{\circ}C \quad (5\text{-}1)$$

$$t > 0, T_f(z = 0, r) = T_{f,in} \quad (5\text{-}2)$$

$$\frac{\partial T}{\partial z}(z = 0, r_o\ to\ R) = 0, \frac{\partial T}{\partial z}(z = L, r_o\ to\ R) = 0 \quad (5\text{-}3)$$

$$k\frac{\partial T}{\partial r}(z, r = r_o) = \lambda(T_{r=r_o} - T_f) \quad (5\text{-}4)$$

$$\frac{\partial T}{\partial r}(z, r = R) = 0 \quad (5\text{-}5)$$

The 1st order implicit, backward, finite difference method isapplied to discretize the HTF and PCM energy equations. Details about the numerical discretization can be found in [23, 24]. At every time step, an implicit set of simultaneous non-linear equations was solved. In this study, matrix inversion in MATLAB was usedto solve these equations, rather than the Gauss-Seidel iteration process. Using matrix inversion combined with the 'Sparse' function in MATLAB yielded a numerical code up to 20 times faster than the Gauss-Seidel method.

III. METHODOLOGY

A process for the optimal selection of geometric parameters is shown in **Fig. 2**. As can be seen, the first step is to define the operating temperature range of LHTES unit that is 286-565 ˚C and is imposed by the CST-tower plant. Hence, the inlet temperature of LHTES unit in charging is 565 ˚C, while this is 286 ˚C in the discharging process. To obtain the mass flowrate of LHTES unit, a simulation of the Rankine cycle should be performed. Considering an efficiency of ~33% for the Rankine cycle at design point, a thermal requirement of $1MW_e$ CST-tower plant is 3.33 MW_{th}[25]. Using this thermal requirement along with the aforementioned operating temperatures, the mass flowrate of the LHTES unit in discharging process can be found by Eq. (6) that is 7 $kg.s^{-1}$. The mass flowrate in charging process depends on the control strategy of the CST plant and the available thermal energy from sun. This study assumes the same mass flowrate in charging as discharging process, so the charging and discharging rate are the same. Another design constrain is the hours of storage. This parameter requires knowledge of overall performance of the integrated CST plant with LHTES unit and needs optimizations. However, according to literature, this should be between 1-15 hours [4]. This study assumes 10 hours of storage. The assumption made here does not affect the relative findings as all the design comparisons were performed based on the same design constrains. After assuming the hours of storage, the storage capacity can be calculated then by the aim of Eq. (7). Hence, the storage capacity for 1MWe CST plant for 10 hours storage is 111 GJ (30,800 kWh_{th}). Afterwards, Eq. (8) can be implemented to find the size of LHTES unit. To use Eq. (8), the PCM should be selected beforehand. There are a number of PCMs in the 286-565 ˚C temperature range[14, 26, 27], but only a few of them are cost effective. Moreover, to perform the numerical simulations, the properties of PCM both in solid and liquid phases must be known. Accordingly, these constrainsconfine the PCM selection.This study chooses a commercially available salt (H500) to perform the analysis. The physical properties of these PCM along with solar salt HTF are given in **Table 1**. The size of the LHTES unit is also shown in**Table 2**.

$$\dot{Q}_{HTF} = \dot{m}_{HTF} C_{p,HTF} (T_{HTF,top} - T_{HTF,bottom}) \tag{6}$$

$$Q_{HTF,total} = \int_0^t \dot{Q}_{HTF}\, dt \tag{7}$$

$$\forall_{LHS,totall} = \frac{Q_{HTF,total}}{\rho_s c_{ps}(T_{m1} - T_{HTF,cold}) + \rho_l h_{ls} + \rho_l c_{pl}(T_{HTF,hot} - T_{m2})} \tag{8}$$

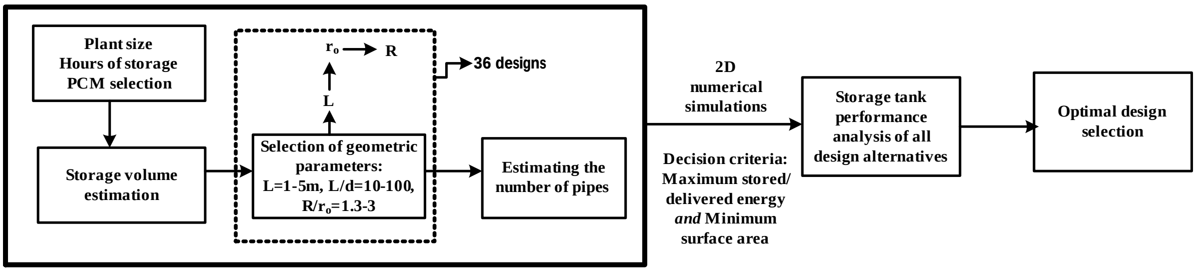


Fig. 2.A procedure to find the optimal geometric parameters.

Table 1.The physical properties of HTF and PCM.

HTF and PCMs thermal properties	Unit	HTF	PCM 2
Name		Solar Salt (40%KNO_3- 60% NO_3)	H500
Dynamic viscosity	kg/m s	0.00326	-
Melting point	˚C	-	500
Solid density	kg/m^3	-	2140
Liquid density	kg/m^3	1820	2140
Solid Specific heat capacity	J/kg.K	-	1555
Liquid Specific heat capacity	J/kg.K	1553	1555
Solid thermal conductivity	W/m.K	-	0.56
Liquid thermal conductivity	W/m.K	0.52	0.56
Latent heat of fusion	J/kg	-	140000
Decomposition temperature	˚C	600	1400
Lower melting point	˚C	-	497
Upper melting point	˚C	-	503

Table 2.The total volume of storage medium in LHTES unit.

	H500	
	PCM volume (m^3) - shell side	HTF volume (m^3) – pipe side
R/r_o=1.3	85	123
R/r_o=2	85	28
R/r_o=3	85	10

The volume obtained by Eq. (8) and shown in Table 2 can be achieved with various selections of geometric parameters. Apart from the main geometric parameters (L, R, r_o), the number of pipes is another variable when it comes to geometric design of shell and tube LHTES unit. This study suggests Eq. (9)to determine the number of pipes. Accordingly, there is a relation between the number of pipes andthe geometry of LHTES unit as well as the total volume.

Depending on what is chosen for the geometric parameters (L, R, r_o), different number of pipes is possible for a same volume of PCM.

$$N_p = \frac{\forall_{LHS,total}}{\forall_{one\ pipe} = \pi(R^2 - r_o^2)L} \tag{9}$$

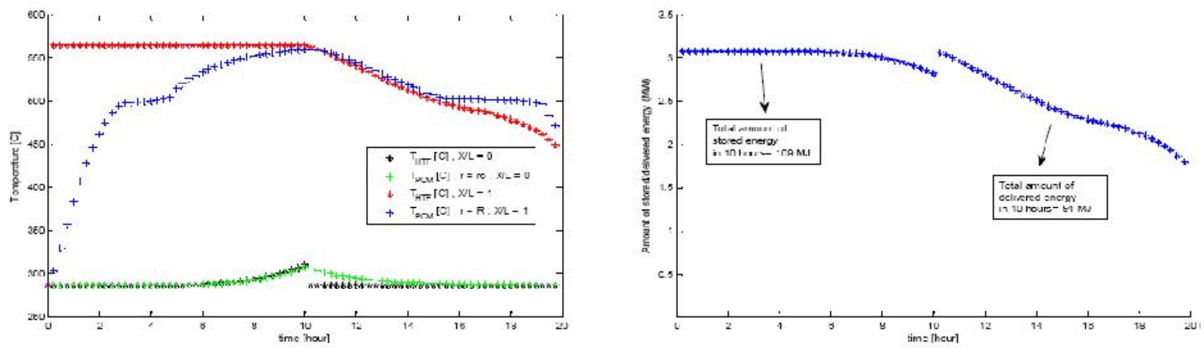
To reach the optimal design, a comprehensive set of geometric parameters was used – namely, the non-dimensional length (L/d) from 10 to 100, the non-dimensional radius $(R/r_o)$from 1.3 to 3, and finally the length $(L)$from 1 to 5 meters. Indeed, these parameters cover the entire range of feasible geometric parameters, which were considered as follows:

- For $L = 1\ m$: $0.015 \leq R \leq 0.065, 0.005 \leq r_o \leq 0.05$
- For $L = 2\ m$: $0.03 \leq R \leq 0.13, 0.01 \leq r_o \leq 0.1$
- For $L = 5\ m$: $0.075 \leq R \leq 0.325, 0.025 \leq r_o \leq 0.25$

It should be noted that each of the 36 design has a different total heat transfer surface area. To select the optimal design among 36 design alternatives, the objective function in this study is to reach the maximum amount of total stored/delivered energy with a minimum surface area. As discussed in [28], the cost of any shell and tube tank is directly related to the total heat transfer surface area. Hence, if the surface area is minimized, the storage tank cost will be minimized as well.

IV. LHTES UNIT PERFORMANCE ANALYSIS

In order to illustrate how the results for each simulated case were evaluated, and to highlight some preliminary observations, an example of the analysis is presented in this section. The timely amount of stored/delivered energy can be calculated by Eq. (6), while the accumulated amount of stored/delivered energy can be calculated via Eq. (7). In order to apply Eq.(6), historic temperatures traces of the PCM and the HTF at top and bottom of the tank should be considered. The time-wise variation of the PCM and the HTF temperatures of the cylindrical LHTESunit during whole charging and discharging periods along with the corresponding hourly and total amount of stored and delivered energy are provided in**Fig. 3** for one of the feasible designs.From the results, it can be observed that the results could be different during charging and discharging processes. This is because the initial conditions in charging and discharging process were different; the charging process started from a fully discharged state, while the discharging process started from the conditions of the tank at the end of the charging process.



$L = 1, L/d = 10, R/r_o = 1.3, N_p = 16800$

Boundary and initial conditions: $\dot{m}_{htf} = 7.14\ kg/s, T_{htf,ch} = 565\ °C, T_{htf,dis} = 286\ °C, T_{initial} = 286\ °C$

Fig. 3. The time-wise variation of HTF and PCM at top and bottom of tank (right) and the amount of stored/delivered enegy (right)

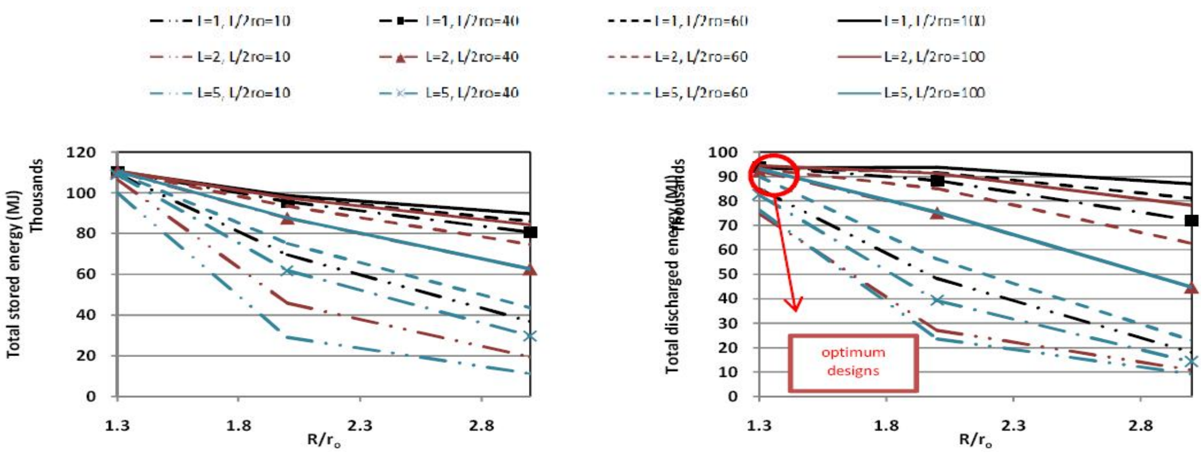


Fig. 4. The charging (left) and discharging (right) performance of the LHTES unit with various set of geometric parameters.

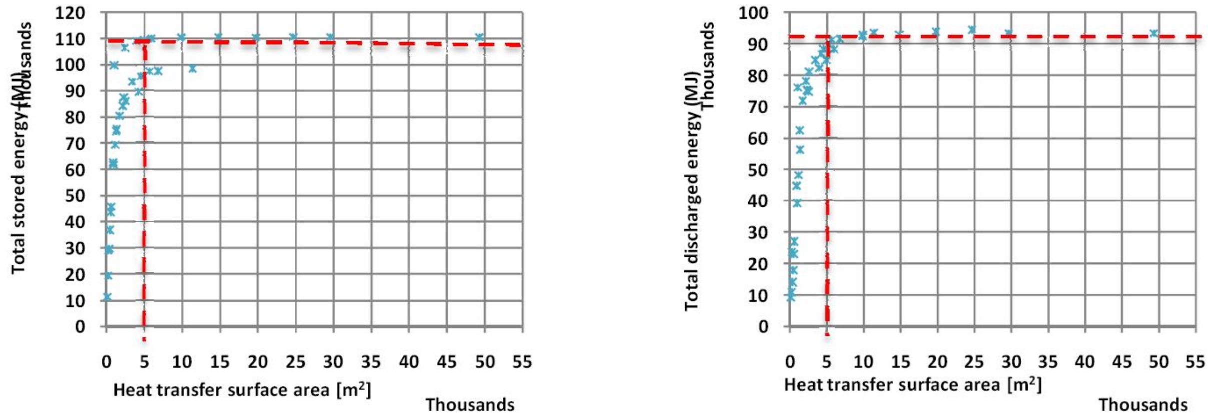


Fig. 5. The charging (left) and discharging (right) performance of the LHTES unit with various set of geometric parameters as a function of total heat transfer surface area.

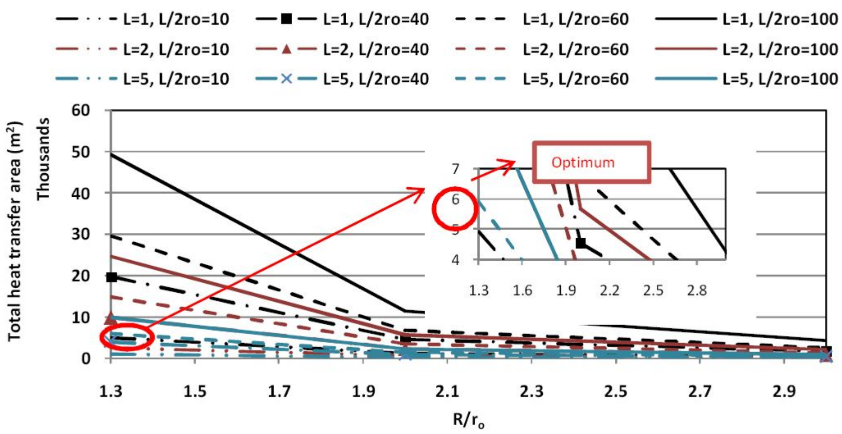


Fig. 6. The values of the total heat transfer surface area for various set of geometric parameters.

V. THE OPTIMAL SELECTION OF GEOMETRIC PARAMETERS

The results of parametric study on the geometric parameters as a function of total stored/delivered energy are shown in **Fig. 4**. This figure shows that the maximum total stored/delivered energy is achieved when the non-dimensional radius is 1.3. As this non-dimensional parameter increases, the amount of stored/delivered energy decreases.**Fig.5** represents the results as a function of total heat transfer surface area. The first conclusion that can be inferred from the results is that a certain design with the same total exchange surface but obtained from different geometric parameters (i.e.: any two points that lie in the same vertical line) can show a different charging/discharging performance. This fact implies that if the geometric parameters are not chosen carefully, the maximum potential of the LHTES unit will not be achieved. The second conclusion that can be reached from the results is that for any PCM, the total amount of stored/delivered energy increases as the total surface area increases; however, there is always a maximum surface area beyond which the energy stored/delivered no longer increases. For the PCM studied in this paper, the optimum heat transfer surface area beyond which increasing the surface area has no effect on the amount of total stored/delivered energy is between 4000 and 7000 m^2. The maximum stored energy is ~110 GJ, while the maximum delivered energy is ~93 GJ. Therefore, a charging efficiency of 99% and discharging efficiency of 85% can be achieved with an optimized system. Considering the storage capacity of 111 GJ, the optimal shell and tube LHTES unit surface area was calculated, and lied between 36-63 m^2.GJ-1 (0.12-0.22 $m^2.kWh_{th}^{-1}$). This range is of great importance because it not only helps to find the optimum value of geometric parameters, but also it aims to have an estimation on the cost of the optimal shell and tube LHTES unit for CST-tower plants.With regard to the optimal range of the surface area and by considering the maximum amount of total stored/delivered energy depicted in **Fig. 4**, the designs that meet these two requirements are highlighted in **Fig. 6**. As the goal is to reach the maximum total stored/energy with minimum surface area, the optimum value of length and the non-dimensional radius are 5 meters and 1.3, respectively. The optimum non-dimensional length is also60. The results here are valid for the constrains that described before in section 4. The results here are valid for the constrains that described before in section 3. Choosing the non-dimensional ratios greater than 1.3 or the non-dimensional lengths below 60 leads to the fewer amounts of total stored/delivered energy, but it also means the lower surface area (e.g. lower storage cost). The decision regarding the absolute optimum geometric parameters requires the results of the integrated operation of the LHTES unit with other CST-tower components. This final decision can be made based on the economic criteria such as the internal rate of return or NPV [29].

CONCLUSIONS

In the present paper, we have focused on the optimum design and technical feasibility of a shell and tube latent heat storage unit that is suitable for use with concentrated solar power tower plants and operates between 286-565˚C. To determine the best design alternative, 36 numerical experiments were performed. The performance of the charging and discharging process was evaluated based on the total amount of stored or discharged thermal energy over 10 hours of charging or discharging. All design alternatives were analyzed under the same boundary conditions, imposed by standard, steady operating conditions of a CST-tower plant. A commercially available salt (H500) was. The following conclusions can be drawn from this study:

(1) It was observed that for each of the selected PCM, there is an optimum heat transfer surface area beyond which increasing the surface area has no effect on the amount of the total stored/delivered energy. The optimum surface area found to range between 36-63 $m^2.GJ^{-1}$ (0.12-0.22 $m^2.kWh_{th}^{-1}$). This surface area can be achieved by different combinations of the geometric parameters ($L, L/d, R/r_o$), but only a few of them will guarantee the highest amount of total stored/delivered energy.
(2) For the PCM studied here, the optimum range of L/d and R/r_o were found to be ~40-60 and ~1.3-1.8, respectively. It should be noticed that the higher ratios of L/d and the smaller values of R/r_o increase the total number of pipes which is not desirable in terms of the total cost of the LHTES unit.
(3) The charging and discharging efficiency of the LHTES unit with the selected PCM over a cycle of continuous charging and discharging found to be around 99% and 85%, respectively.

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