ANALYSIS OF GEOMETRIC PARAMETERS OF COLD PACKED BED ENERGY STORAGE FOR LIQUID AIR ENERGY STORAGE SYSTEMS

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ABSTRACT

Among large-scale energy storage systems, liquid air energy storage systems have gained a wide interest due to their advantages such as high energy density, not being geographically restricted, and being environmentally safe. An essential component that has a large influence on the performance of the system is the cold energy storage, which stores the cold energy rejected during air evaporation in the discharging cycle and transfers this cold energy to the air liquefaction stage in the charging cycle. The aim of this paper is to identify the optimal set of geometrical parameters for cold energy storage, consisting of a packed bed, maximizing the performance of the liquid air energy storage system and minimizing the cost of the cold energy storage unit. The packed bed was modelled using a transient one-dimensional model, while the rest of the components and the overall system were modelled using a steady-state zero-dimensional thermodynamic model. We analyzed the influence of the aspect ratio and diameter of the tank on the round-trip efficiency of the liquid air energy storage system and the total cost of the cold energy storage unit, while keeping the capacity of the cold energy storage constant. The results suggest that there is an optimum aspect ratio for each diameter of the tank for reaching the minimum cost for the cold energy storage unit. On the other hand, for each diameter of the tank, the value of round-trip efficiency keeps increasing by raising the aspect ratio of the tank. Nevertheless, by selecting the optimum aspect ratio concerning the minimum cost, there is a negligible penalty on the reduction of round-trip efficiency compared to its maximum value.

1 INTRODUCTION

To increase the penetration of renewable energy sources in the electricity market, it is vital to have large-scale energy storage systems (Chen *et al.*, 2009), neutralizing the fluctuating nature of renewable energy sources. Among different large-scale energy storage technologies, thermo-mechanical energy storage systems have gained increasing interest due to their large storage capacity (Vecchi and Sciacovelli, 2023). The three conventional thermo-mechanical energy storage systems are pumped thermal energy storage (PTES), compressed air energy storage (CAES), and liquid air energy storage (LAES). Recently, LAES has gained rising attention among researchers because of its advantages compared with other technologies. The LAES system is not dependent on geographic conditions, has a high energy storage density, and is environmentally safe (Morgan *et al.*, 2015). The LAES system consists of three main stages: 1) charging cycle – during off-peak hours, the ambient air is liquefied through a liquefaction process by using the excess electricity from renewable energy sources. 2) storage – the liquefied air at the end of the charging cycle is stored in liquid air tanks at ambient pressure. 3) discharging cycle – during peak hours, the liquid air is evaporated, reheated, and expanded in air turbines producing electricity. The main challenge of the LAES system is its low round-trip efficiency

compared with other mature energy storage technologies As for geometric analyses of packed beds, there are some research papers in the open literature(Antonelli *et al.*, 2017). To overcome this issue hot and cold thermal energy storage are employed to provide internal recuperation of cold and hot streams. The design of these sub-systems significantly affects the performance of the whole LAES system. Hot thermal energy storage systems have been studied in a vast range of research as they have other applications than LAES systems. However, cold thermal energy storage systems, especially cryogenic cold energy storage, are novel thermal energy storage systems and have more specific applications. (Liu *et al.*, 2023)The studies conducted on cold energy storage can be divided into three types: cold energy storage using two fluids four tanks systems that mostly work with methanol and propane as heat transfer fluids (Guizzi *et al.*, 2015), cold packed bed with phase change material as the storing medium (Tafone *et al.*, 2021), and cold packed bed with sensible heat material (Sciacovelli et al., 2017). In general, the cold packed bed is a more robust system because this system is more economically and environmentally viable, and since the working fluid is simple gases like air or nitrogen, there is not any safety concern regarding the inflammability that exists for the case with methanol and propane (Wang *et al.*, 2021). Moreover, cold packed bed energy storage systems can cover the entire temperature range of LAES cold storage, from cryogenic temperature of boiling point of air to ambient condition, in a single stage which helps to reduce the complexity of the LAES system.

Guo et al., (2024) did an experimental study on the dynamic performance of cold packed bed energy storage. They found that the performance of the cold packed bed gradually diminished from the start-up until reaching a stable condition after the 7th cycle. They considered two different sensible heat materials as storing material, glass, and basalt, and it was found that the packed bed with glass has smaller thermocline thickness due to higher specific heat and lower thermal conductivity of glass. Furthermore, they analyzed the effect of inlet mass flow rate and the temperature of working fluid of packed bed, and they stated that the efficiency of the system is improved by decreasing the mass flow rate and inlet temperature of the working fluid. Sciacovelli et al., (2018) analyzed a packed bed as the cold energy storage unit of the LAES system experimentally. They designed a modular configuration of a packed bed that reduced the required work of air liquefaction by 35 %. Lin et al., (2019) investigated a two-stage cascaded packed bed as the cold energy storage unit of LAES. It was reported that, by using a two-stage packed bed, the exergy losses are reduced because of a better match between temperature-duty profiles of hot and cold flows in the evaporator. Besides, they analyzed the impact of the pressure of the liquid air tank on the overall performance of LAES. They concluded that in general, the round-trip efficiency is increasing by raising the storage pressure. Further, they claimed that the round-trip efficiency of 65 % is feasible with a storage pressure of 9 bars. Fan et al., (2023) studied the effect of unstable distribution of temperature inside packed bed on the efficiency of the thermal energy storage system. They observed that heat leakage from the packed bed in addition to the non-uniform distribution of the mass flow rate of the fluid causes an uneven distribution of temperature inside the packed bed. They suggested two approaches to concur with this issue, intensify the insulation layer to minimize the heat leakage and use guide plates along the bed that help to have a uniform distribution of mass flow rate. Fan et al., (2024) designed a novel gravity-driven moving packed bed to mitigate the dynamic behavior of a common packed bed. The system was a gas-solid counter-flow heat exchanger and quartz sands were used as solid phase. They reported that the efficiency of this system improved by increasing the height and diameter of the heat exchanger as well as reducing the size of quartz sands. Wang et al., (2022) analyzed the effect of the different working fluids of cold packed bed on the performance of the LAES system. They considered four different systems, one-stage packed bed with pressurized air (10 MPa), one-stage packed bed with ambient pressure air, one-stage packed bed with pressurized propane (1 MPa), and two-stage packed bed with propane and methanol. It was observed that the best system is the one with pressurized air as the working fluid that results in a LAES round-trip efficiency of 42.8%.

Regarding the geometric analysis of packed bed, there are some research papers in the open literature, however all of them considered high-temperature packed bed system. Peng et al. (2014) investigated a packed bed system with phase change material as the storing media and molten salt as the heat transfer fluid. They analyzed the effect of geometric parameters of packed bed on the efficiency of the thermal

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energy storage system. It was stated that the performance of the system is improved by reducing the particle diameter and velocity of heat transfer fluid, and by increasing the height of the packed bed. Peng et al., (2015) studied the effect of porosity, particle diameter, and inlet pressure on the efficiency of a packed bed energy storage system. They reported that efficiency increases by reducing the particle diameter and raising the inlet pressure of the working fluid. This is due to a reduction in the thickness of the thermocline layer inside the packed bed. In addition, they claimed that the efficiency and capacity of the system decreased by increasing the porosity of the bed however, the required time to finish the charging and discharging cycles is reduced. Ahmed et al., (2019) investigated a novel design of a packed bed as a thermal energy storage system by combining sensible and latent heat (encapsulated phase change material) materials in a single bed. They found that the performance of the system intensified by reducing the porosity of the bed, velocity of the fluid, and particle diameter. Mao and Zhang, (2020) analyzed a cascaded high-temperature packed bed system with three different phase change materials. It was found that the capacity and utilization rate of the system is increased by reducing the porosity. Moreover, it was shown that the total energy stored in the system rises when the particle diameter reduces from 0.6 m to 0.1 m.

Several studies have shown that cold energy storage has a significant impact on the LAES system. For instance, Morgan *et al.,* (2015) showed that by increasing the share of recycled cold energy from 51 % to 91 % improves the round-trip efficiency of LAES systems from 8 % to approximately 50 %. Hence it is vital to investigate all the aspects of this system to achieve its best performance, which improves the performance of the LAES system. However, the investigation of the effect of geometric parameters of a cold packed bed is missing in the open literature. The aim of this paper is to identify the optimal set of geometrical parameters for cold energy storage, consisting of a packed bed, maximizing the performance of the liquid air energy storage system, and minimizing the cost of the cold energy storage unit. For this purpose, the effects of the aspect ratio and the diameter of the packed bed tank on the round-trip efficiency of the LAES system and cold energy storage unit were investigated. The results of the analysis will be useful both for industry and academia for the further development of cold energy storage units for LAES systems.

Next, section 2 discusses the methods. Afterwards, the numerical simulations are presented and discussed in section 3, while the conclusions are summarized in section 4.

2   METHODS

In this section, firstly, the layout of the system is introduced. Then, the details of the thermodynamic model of the LAES and the heat transfer model and the method to calculate the geometric parameter of the packed bed storage are presented.

2.1   System Description

The layout of the current LAES system is shown in Figure 1. This system consists of 3 stages of compression and 3 stages of expansion. The hot energy storage system consists of two tanks with Therminol-55 as the working fluid and the packed bed is used for the cold energy storage system. Referring to the charging cycle, first, the ambient air is compressed to a supercritical pressure (A7). The heat is rejected by intercoolers and aftercooler heat exchangers and is stored in the hot oil tank of the hot storage system. After the compression process, the air cools down to a cryogenic temperature (A8) by passing through the cold box. There are two sources of cold energy inside the cold box, the cold stream of air comes from the cold packed bed (C1) and the returned air after separator (A10). Afterwards, the low-temperature air is expanded to the ambient pressure in the cryogenic turbine (A9), then the liquid and gas phases of the two-phase air are separated in the separator. The gaseous part (A10) goes through the cold box rejecting its cold energy to the mainstream air and the liquid part (A_Liq) is transferred to the liquid air tank and is stored there. During discharging, firstly, the liquid air is pressurized by a cryogenic pump (A13) and then rejects its cold energy through the evaporator and leaves the evaporator as a superheated gas (A14). The cold energy is stored in the cold packed bed to

be used in the liquefication process of air in the next charging cycle. After the evaporator, the gaseous air is reheated via hot energy storage and is expanded in the air turbines producing electricity.
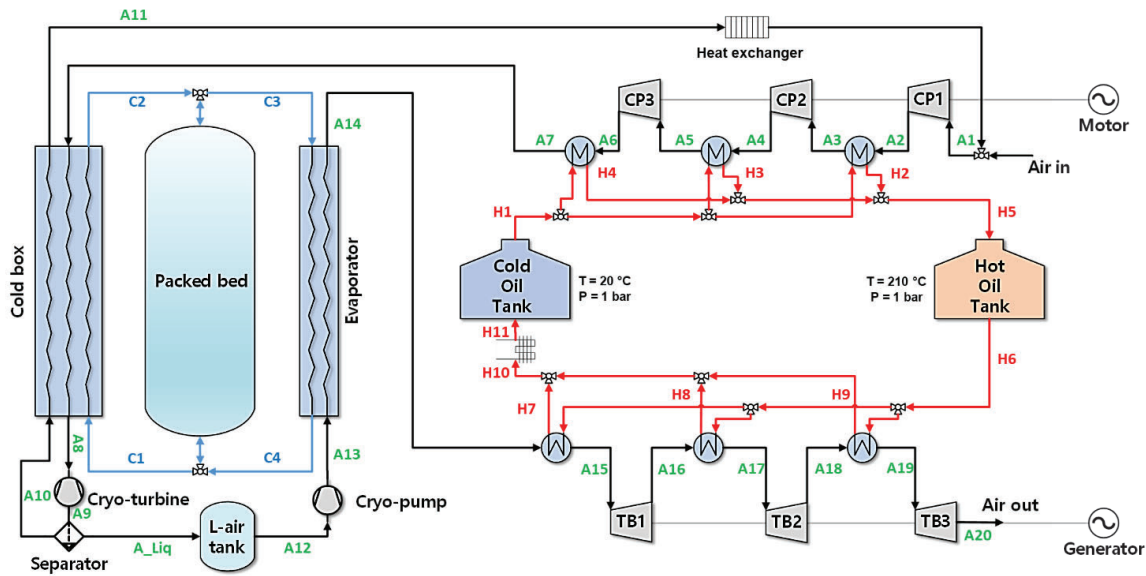

Figure 1: Schematic diagram of liquid air energy storage.

The input power to the charging cycle is fixed as 100 MW for all the cases, and the charging and discharging time of the LAES system is set to 8 hours as a large-scale LAES system. Moreover, the charging and discharging pressures for the LAES system are set to 18.5 and 10 MPa, respectively based on the investigations in our previous study (Mashayekh *et al.*, 2023). Other parameters of the system such as heat exchangers' pinch point and efficiency of turbomachinery devices are listed in Table 1.

Table 1: Input parameters of system modelling

Parameter	Value	Unit
Input power of charging cycle	100	MW
Charging/Discharging time	8	hr.
Ambient temperature	293	K
Ambient pressure	101	kPa
Charging pressure	18.5	MPa
Discharging pressure	10	MPa
Pinch points:		
Intercoolers and reheaters	5	K
Cold box	2	K
Evaporator	2	K
Isentropic efficiency of turbomachinery:		
Compressors	89	%
Cryogenic turbine	80	%
Cryogenic pump	75	%
Air turbines	90	%

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2.2 Thermodynamic Models of Components

The mathematical correlations of thermodynamic models for different components of the system are implemented in the MATLAB environment. All the thermo-physical properties of air are calculated by REFPROP library (Ewl *et al.*, 2018). For turbomachinery, the definition of isentropic efficiency is used to calculate the power of the component. The power consumption of compressors and output power of expanders are calculated following Equations (1) and (2).

$$\dot{W}_{Com} = \dot{m}(h_{out} - h_{in}) = \dot{m}(h_{out.is} - h_{in})/\eta_{is,Com} \tag{1}$$

$$\dot{W}_{Tur} = \dot{m}(h_{in} - h_{out}) = \eta_{is,Tur}\,\dot{m}(h_{in} - h_{out.is}) \tag{2}$$

In these equations, $\dot{W}$ is the turbomachinery power, $\dot{m}$ is the mass flow rate of air, h is specific enthalpy, and η is isentropic efficiency. The subscripts Com, Tur, out and in refer to compressors, turbines, inlet and outlet flows, respectively. is refers to an isentropic process.

For modelling the heat exchangers, pinch point temperature difference is used with a known pinch point temperature for each heat exchanger. It is assumed that all heat exchangers are counter-flow. The equations of heat balance and pinch point temperature are given as Equations (3), (4), and (5).

$$\dot{Q}_H = \dot{m}_H(h_{H_{in}} - h_{H_{out}}) \tag{3}$$

$$\dot{Q}_C = \dot{m}_C(h_{C_{out}} - h_{C_{in}}) \tag{4}$$

$$Pinch\ point = \min(T_H - T_C) \tag{5}$$

In the above equations, $\dot{Q}$ is the heat transfer rate, and the subscription of H and C represents hot and cold streams, respectively. For the cold box where there are three streams, it is presumed that there is no heat transfer between the two cold streams and $\dot{Q}_C$ is the summation of the heat transfer rate for the two cold streams.

The overall performance of the LAES system is evaluated in terms of the round-trip efficiency, RTE, of the system:

$$RTE = \frac{W_{discharging}}{W_{charging}} = \frac{\int_0^{t_{dis}} \dot{W}_{Tur} - \dot{W}_{CRP}\ dt}{\int_0^{t_{ch}} \dot{W}_{Com} - \dot{W}_{CTB}\ dt} \tag{6}$$

The subscript CRP refers to cryogenic pump and the subscript CTB refers to cryogenic turbine.

2.3 Packed Bed Modelling

For calculating the heat transfer performance of packed bed storage, the energy balance equation is used. The energy equation for fluid and solid particles is presented in Eq. (10) and Eq. (11), respectively. The packed bed storage is considered a 1D cylindrical tank and radial distribution of temperature based on Eq. (12) is considered inside the particles.

The following assumptions are made:

1. The distribution of velocity in the entire packed bed tank is constant and uniform. This assumption is valid for systems with $D/d_p > 30$ (Cohen and Metzner, 1981).
2. The heat transfer mechanism between the heat transfer fluid and particles are convection and conduction. Radiation is neglected.
3. The porosity is uniform along the packed bed tank.
4. The particles are of the same size and spherical shape.
5. The heat loss of the tank to the ambient is considered.

$$\varepsilon\rho_f c_{p.f}\left(\frac{\partial T_f}{\partial t} + u_f\frac{\partial T_f}{\partial z}\right) = \varepsilon k_f\frac{\partial^2 T_f}{\partial z^2} + \frac{6(1-\varepsilon)}{d_p}h_{fp}(T_s - T_f) + h_w(T_w - T_f) \tag{10}$$

$$(1 - \varepsilon)\rho_s c_{p.s} \frac{\partial T_s}{\partial t} = (1 - \varepsilon)k_s \frac{\partial^2 T_s}{\partial z^2} + \frac{6(1 - \varepsilon)}{d_p} h_{fp}(T_f - T_s) \tag{11}$$

$$\rho_p c_{p.p} \frac{\partial T_p}{\partial t} = k_p \left(\frac{\partial^2 T_p}{\partial r^2} + \frac{1}{r} \frac{\partial T_p}{\partial r} \right) \tag{12}$$

where ε is the porosity of the packed bed, ρ is density, c_p is the specific heat of fluid, k is thermal conductivity, d_p is the particle diameter, h_{fp} is the heat transfer coefficient of fluid to solid, h_w is the heat transfer coefficient of heat loss to the ambient, and T is temperature. Subscripts f, p, and s are fluid, particle, and solid, respectively.

In order to apply heat transfer between the fluid and solid, an empirical correlation is applied. For the Nusselt number, the following correlation is used (Wakao *et al,*. 1979):

$$Nu = \frac{h_{fp} d_p}{k_f} = 2 + 1.1 Pr^{\frac{1}{3}} Re_p^{0.6} \tag{19}$$

The heat loss to the ambient is calculated based on the overall heat transfer coefficient from the inner to the outer wall, see Eq. (21). The inner convection heat transfer coefficient is defined according to Beek (Beek), see Eq. (22).

$$\frac{1}{U_w} = \frac{1}{h_i} + \frac{D}{2} \sum_{j=1}^{m} \frac{1}{k_j} ln \left(\frac{d_{j+1}}{d_j} \right) \tag{21}$$

$$h_i = \frac{k_f}{d_p} \left[\left(0.203 Pr^{\frac{1}{3}} Re_p^{\frac{1}{3}} \right) + \left(0.220 Pr^{0.4} Re_p^{0.8} \right) \right] \tag{22}$$

where h_i is the inner convection heat transfer coefficient, Re is Reynolds number, and Pr is the Prandtl number.

For the boundary conditions, adiabatic boundary conditions are assumed for the outlet surfaces and fixed inlet temperatures are assumed as the boundary conditions of inlet surfaces.

Figure 2 shows the schematic view of the packed bed system with the geometric parameters.
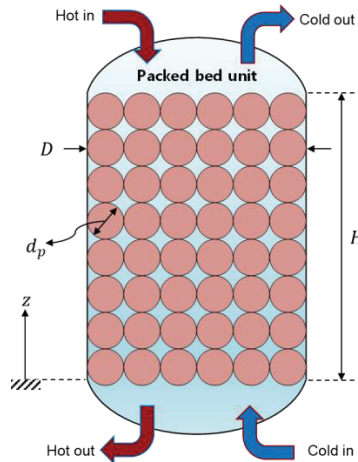

Figure 2: Schematic view of packed bed.

To see the effect of different sizing of cold packed bed on the LAES performance the following approach is used. We considered a constant capacity for the cold energy storage system and calculated the total required volume of the cold energy storage unit based on Equation (7) (Niedermeier *et al.*, 2018). Afterwards, by knowing the diameter and the height of the packed bed tank for each case the volume of a single tank is calculated using Equation (8), and the total number of required packed bed tanks is calculated as given in Equation (9).

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$$V_{tot} = \frac{E}{[(\rho_f . C_{p_f} . \varepsilon) + (\rho_s . C_{p_s} . (1 - \varepsilon))](T_{top} - T_{bot})} \tag{7}$$

In this equation E is the capacity of the cold energy storage system that is assumed $240 \ MWh_{th}$ in this study based on the available cold energy during evaporation of liquid air in our previous study (Mashayekh *et al.*, 2023). ρ and C_p are density and specific heat capacity, and ε is the porosity of the bed. The subscripts f and s refer to fluid and solid particles, respectively. T_{top} and T_{bot} are the top and bottom temperatures of the packed bed tank. Based on the given diameter and height, the volume of a single tank in each case was calculated as follows:

$$V_{1-tank} = \frac{\pi}{4} D^2 H \quad , \quad H = D.AR \tag{8}$$

where, $D, \ H, \ AR$ are diameter, height, and aspect ratio of the tank. The total number of required packed bed tanks is calculated as given in Equation (9).

$$N = \frac{V_{tot}}{V_{1-tank}} \tag{9}$$

For estimating the cost of the packed bed tank, the following correlation in Equation (10) is used (Turton, 2012). This equation calculates the purchase cost of a packed bed tank and installation in millions of Euros (updated for 2022 by the CEPCI index). It is assumed that the material is carbon steel, the working pressure is equal to ambient pressure, and the tank stands in vertical orientation.

$$Cost = \left(\frac{1.94}{10^6}\right) 10^{\left[3.49 + 0.44Log(V) + 0.11\left(Log(V)\right)^2\right]} \tag{10}$$

In this correlation, V is the volume of packed bed tank in m^3.
As the amount of the storing material is constant for all the cases, the cost of particles is not considered in the evaluation of the cold energy storage unit cost.
Quartz is used as the storing particles in the packed bed; its thermophysical properties are listed in Table 2 (Morgan *et al.*, 2015).

Table 2: Thermophysical properties of quartz

Properties	Value	Unit
Density, ρ	2630	kg/m^3
Specific heat capacity, C_p	710	$J/kg.K$
Thermal conductivity, k	1.83	$W/m.K$

3 RESULTS AND DISCUSSION

3.1 Validation of Numerical Model

The thermodynamic model of the LAES is validated by comparing the results of our current model with experimental data from the discharging cycle of the LAES pilot plant presented by (Sciacovelli et al., 2017), see our previous paper (Mashayekh *et al.*, 2023).
Furthermore, the developed model for the packed bed energy storage with sensible heat particles is validated by comparing its results with the experimental results of (Meier et al., 1991) for a packed bed with a height of 1.2 m and a diameter of 0.15 m during charging. The diameter of the quartzite rock particles is 0.02 m. Figure 3 presents the temperature of the air as the working fluid along the bed axial locations at different charging times. The results suggest that the maximum mean absolute percentage error is 2.8 %, indicating that the model for sensible heat particles provides accurate results.
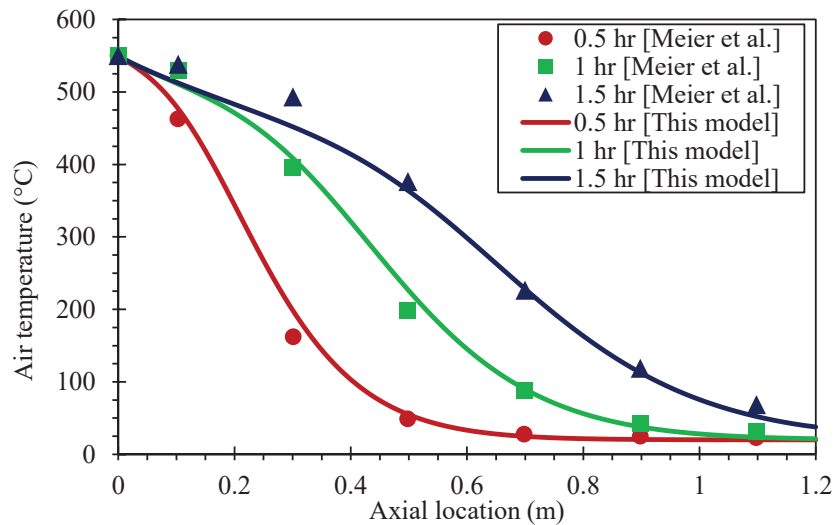
Figure 3: Validation of the packed bed model for sensible heat particles.

3.2 Analysis of Packed Bed Geometry

In this section, the effects of the diameter and the aspect ratio of the cold packed bed tank are investigated on the round-trip efficiency of the LAES system and the cost of the cold energy storage unit. For this purpose, the diameter of the tanks varies from 3 to 10 m with steps of 0.5 m, and for each of the diameters of the tank, 6 different aspect ratios are considered, from 0.5 to 5. In total, these analyses end up with 90 different cases. The diameter of particles is set to 0.02 m for all cases. The results are presented in Figure 4. The results suggest that for each aspect ratio, the round-trip efficiency of LAES keeps rising by increasing the diameter of the tank. On the other hand, the minimum cost of the cold energy storage unit is achieved at specific value for the diameter of the tank. According to the results, the increase rate in cost is greater than the increase rate in LAES round-trip efficiency for the diameter of the tank higher than its value regarding the minimum cost. In other words, the cost of the cold energy storage unit can be minimized without a significant drop in the round-trip efficiency of the LAES system. For example, for the case with an aspect ratio of 5, by increasing the diameter of the tank from the economic optimum value to the maximum value of 10 m, the cost of the cold energy storage unit reaches 7.8 M€ from 5.9 M€ (around 32% increase), while the round-trip efficiency of LAES increases from 44.1 % to 44.3 % (less than 0.5 % increase).

Further, the results presented in Figure 4 suggest that for a fixed value of aspect ratio, by raising the diameter of the tank, the height and the volume of each tank is increasing and therefore, the required number of tanks is reduced, as the total required volume is constant.

The total cost of the cold energy storage unit is calculated as the product of a single tank's cost and the number of the required tanks. The cost of a single tank is proportional to the volume of the tank, while the total number of required tanks is inversely proportional to the volume of a tank as described above. These considerations explain the trend of the cost curves presented in Figure 4. Therefore, for each aspect ratio, for the diameter of the tank lower than its economic optimum value, the reduction of the number of tanks is dominant which causes a reduction in the cost. While for the diameter of tanks higher than this value, the growth of the single tank cost is dominant which causes an increment in the cost of cold energy storage unit.

In conclusion, it can be stated that the taller packed beds have better performance with respect to the LAES round-trip efficiency. However, from the economic point of view, there is an optimum for the size of the tank and beyond that value, the cost rises.
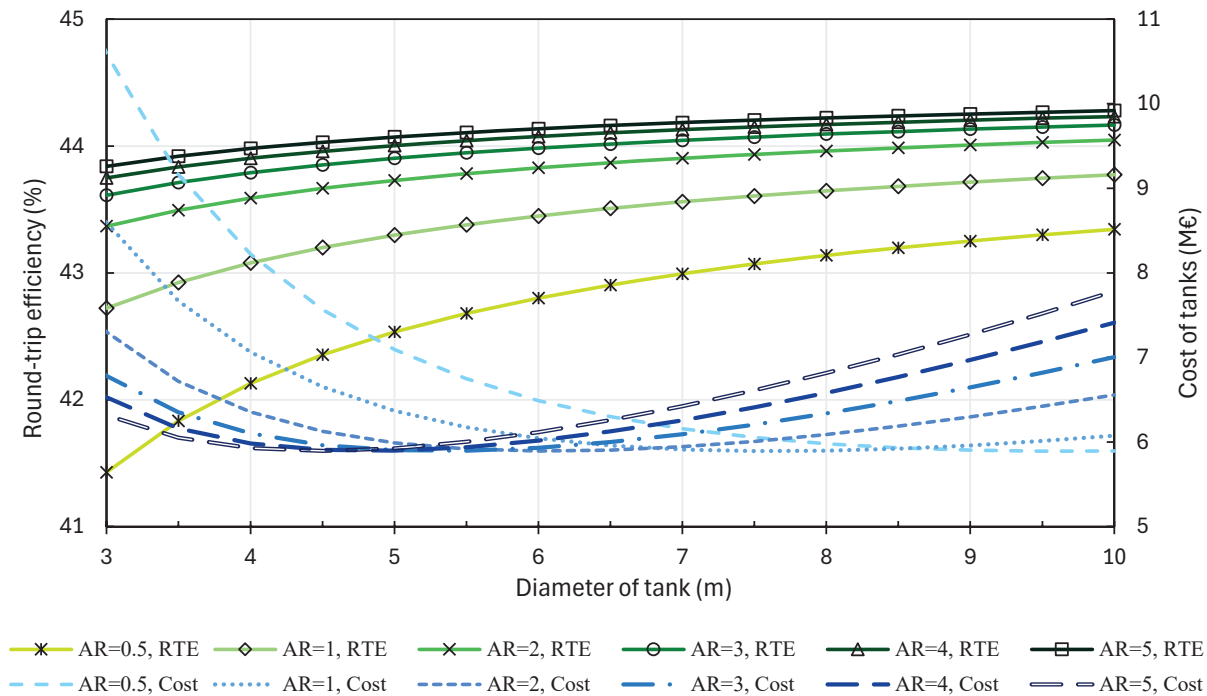
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Figure 4: Round-trip efficiency of LAES system and cost of cold energy storage unit as a function of packed bed tank's diameter for different aspect ratios.

Figure 5 depicts the optimum aspect ratio of the tank for each value of the diameter of the tank to reach the minimum cost of the cold energy storage unit. It is shown that the aspect ratio is decreasing by increasing the diameter of the tank. The results indicate that for each diameter of the tank, a suitable aspect ratio can be selected to achieve the minimum cost of cold energy storage unit. It is worth mentioning that all the reported cases in Figures 5 and 6 correspond to the minimum cost of the cold energy storage unit, which has an equal value of 5.9 M€ for all the various cases.

In Figure 6 the optimum value of the diameter of the tank for the minimum cost is reported for different aspect ratios (left ordinate). Moreover, on the right ordinate, the round-trip efficiency of the proposed cases is presented. From Figure 6, it may be observed that having a higher aspect ratio results in smaller diameters for reaching the minimum cost and at the same time, the round-trip efficiency is increasing. For example, for the two extreme cases, the aspect ratio of 0.5 and 5, the values of the diameter of the tank concerning the minimum cost are 9.5 and 4.5 m, respectively, and the round-trip efficiency of the LAES are 43.3 % and 44.1 %, respectively. These results agree with the results obtained by (Cascetta *et al.*, 2014) and (Ortega-Fernández *et al.*, 2017). They analyzed the effect of different aspect ratios on the performance of a hot packed bed energy storage system. They reported that a higher aspect ratio of the tank results in higher efficiency of the packed bed energy storage system. However, due to the constraints in the manufacturing and structural design of the packed bed tank, having a narrow and tall (low D and high H) vertical structure is not practical. Based on literature and manufacturers (Cascetta *et al.*, 2014; Ortega-Fernández *et al.*, 2017), to avoid installation issues an aspect ratio of two is suggested. Based on the results of the current study, the economic optimum value of the diameter of the tank for the aspect ratio of 2 is 6 m (Mashayekh *et al.*, 2023), resulting in a round-trip efficiency of 43.8 %. While the round-trip efficiency can be increased 0.2 percent points by having the diameter of tanks of 10 m, the cost of cold energy storage unit will increased by 11.3 %.
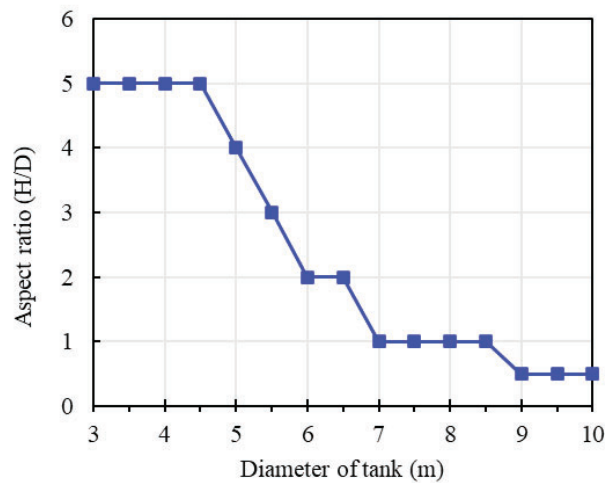
Figure 5: Tank aspect ratio for each diameter of tank to reach minimum cost.
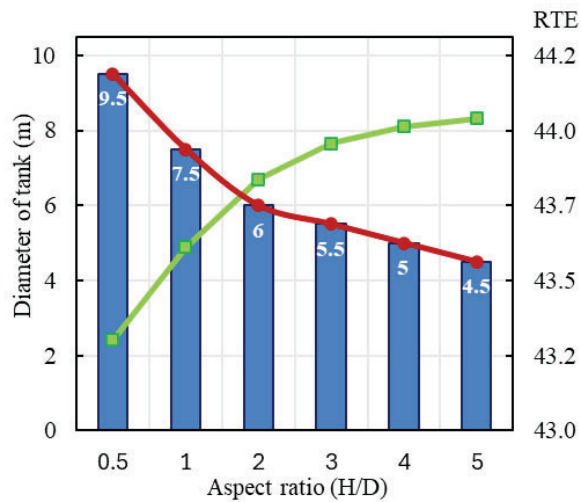

Figure 6: Tank diameter for each aspect ratio to reach minimum cost (left ordinate) and the round-trip efficiency of the respective case (right ordinate).

4   CONCLUSIONS

This paper presented a parametric analysis of geometric parameters of cold packed bed energy storage for liquid air energy storage systems. The parameters investigated are the diameter and the aspect ratio of the cold packed bed tank. For this purpose, the diameter of the tanks varied from 3 to 10 m with steps of 0.5 m, and for each of the diameters of the tank, 6 different aspect ratios were considered, from 0.5 to 5. In total, these analyses ended up with 90 different cases. For each of these cases, the round-trip efficiency of LAES and the cost of the cold energy storage unit were calculated. The developed numerical thermodynamic model of LAES and 1-dimensional model of packed bed heat transfer were used to obtain the results. The results suggest that for each specific value of aspect ratio, the round-trip efficiency of the liquid air energy storage system rises by increasing the diameter of the tank. On the other hand, for each value of the aspect ratio of the tank, there is an optimum value for the diameter of the tank to reach the minimum cost. Moreover, the results indicate that the value of the aspect ratio or diameter of the tank that corresponds to the minimum cost could be considered the best case due to the not significant drop in the round-trip efficiency compared to its maximum value. Based on the current results, the minimum cost of the cold energy storage unit is 5.9 M€ and this minimum value is equal for all the different aspect ratios. The maximum round-trip efficiency among the cases with minimum cost is 44.1 % for the case with a diameter of tank of 4.5 m with an aspect ratio of 5. However, because of the manufacturing and installation constraints, an aspect ratio of two is recommended in the literature. According to the results, for the aspect ratio of 2, the corresponding diameter of the tank to reach minimum cost and the round-trip efficiency are 6 m and 43.8 %.

NOMENCLATURE

C_p	Specific heat capacity, $J/kg.K$	$\dot{m}$	Mass flow rate, kg/s
CRP	Cryogenic pump	p	Pressure, Pa
CTB	Cryogenic turbine	$\dot{Q}$	Heat transfer rate, W
D	Packed bed diameter, m	RTE	Round-trip efficiency
E	Capacity of cold energy storage, J	t	Time, s

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H	Packed bed height, m		T	Temperature, K
h	Specific enthalpy, J/kg		V	Volume, m^3
k	Thermal conductivity, $W/m.K$		$\dot{W}$	Turbomachinery power, W

Greek symbols

ε	Porosity of packed bed		ρ	Density, kg/m^3
η_{is}	Isentropic efficiency of turbomachinery			

Subscript

amb	Ambient		s	Solid particles
Com	Compressor		Tur	Turbine

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