A Comparison between Passive Regenerative and Active Fluidized Bed Thermal Energy Storage Systems

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Abstract: Active Fluidized Bed Thermal Energy Storage (sandTES) offers a promising alternative to the current state of the art thermal energy storages (TES), such as active TES based on molten salt or passive TES (Regenerators) realised as a porous packing of ceramics. The characteristic of a sandTES system applying sand in an active TES using a fluidized bed heat exchanger (HEX) is explained. The exergetic performance of a sandTES is compared to a passive Regenerator.

1. Introduction

Large scale Energy Storages are key components for the future electrical energy grid. One possible approach are Adiabatic Compressed Air Energy Storages (ACAES) [6], where air is compressed, stored at the increased pressure level in underground caverns or sub lake vessels and expanded in a turbine. In ACAES the heat resulting from the compression process is stored in an appropriate Thermal Energy Storage (TES). Electrical energy is stored in form of potential energy via pressurized air and thermal energy. For the realization of large scale TES systems the efficiency of the storage as well as the costs of the storage are important factors. In order to minimize the costs the thermal capacity of the storage material has to be maximized to guarantee that the size of the TES system is low. A comparison of the thermal capacity of different storage materials is given in figure 1.

2. Fluidized Bed Thermal Energy Storage: sandTES

The sandTES Active Fluidized Bed Thermal Energy Storage system is an active TES applying a fluidized bed HEX to transfer heat between any heat transfer fluid (HTF) and the storage medium sand. The concept consists of a stationary fluidized bed with an internal tube bundle HEX, two sand bunkers and...
auxiliary devices handling the storage medium sand (conveyors) and delivering the air enabling the fluidized bed (blowers).

In charging mode the sand is entering the fluid bed HEX at the left-side coming from the cold bunker in figure 2. The hot HTF is flowing inside the tube bundle of the HEX countercurrent to the sand. Heat is transferred from the hot HTF to the cold sand and the internal energy of the sand is increased. The heated sand is exiting the HEX on the right-side and is transported to the hot bunker, while the cooled HTF is exiting the HEX at the left-side in figure 2. The usage of fine sand (80-100 μm) keeps fluidization air mass streams low and maximizes the sand side heat transfer coefficient as shown in figure 3. The heat transmission coefficient is also influenced by the velocity of HTF inside the tubes of the HEX. The hot sand can be easily stored in huge hoppers with external and/or internal isolation. For discharging a sand TES, the flow directions of sand and HTF are altered in respect to the charge mode and heat is transferred from the sand to the HTF again with counter current heat exchanger characteristic.

Modelling the fluidized bed HEX

Crucial in designing and calculating a HEX is the temperature distribution and thus the heat transfer characteristic of the HEX. In case of a fluidized bed HEX, heat transfer is dependent on the particle size/distribution and the bed porosity. The fluidization characteristics are fitted to reach optimal values of the particle to wall heat transfer coefficient. Due to the page limitation only the core ideas can be presented here. The total fluidized bed HEX is subdivided into computational cells. A simplified sketch of one computational cell is given in figure 4. For the sand-side heat transfer the equation of Martin for the particle Nusselt Number is applied [2]:

\[
N_u = \left(1 - \psi \right) Z \left(1 - \exp(-N) \right)
\]  

In equation (1) the parameter \(Z\) is a function of fluid and particle properties, actual bed porosity \(\psi\) and the state of fluidization. The parameter \(N\) in equation (1) is dependent on the parameter \(Z\) and the maximum Nusselt Number for particle wall heat transfer. Further details about the derivation of equation (1) and a more comprehensive description of its parameters can be found in [2].

For finding bed porosity \(\psi_{\text{opt}}\) for the maximum particle to wall heat transfer coefficient for a given particle size distribution, \(\psi_{\text{mf}}\) has to be optimized [2], thus: \(\partial N_u / \partial \psi = 0\). The resulting equation for \(\psi_{\text{opt}}\) is given by equation (2).

\[
\frac{\psi_{\text{opt}}}{\psi_{\text{mf}} - \psi_{\text{opt}}} = \frac{1}{2} \left(1 - \frac{N(\psi_{\text{opt}})}{e^{N(\psi_{\text{opt}})} - 1} \right)
\]  

The subscript \(mf\) indicates minimum fluidization conditions. Having gained the optimal porosity \(\psi_{\text{opt}}\), the corresponding superficial air velocity \(u\) is estimated via the following equation taken from [7], which is valid for the flow regime near the minimum bubbling velocity:

\[
u = u_{\text{mf}} \sqrt{1 - \psi_{\text{mf}}^{1/3}} \frac{1 - \psi_{\text{mf}}}{1 - \psi_{\text{mf}}^{1/3}} \]  

The bubbling bed behaviour is taken into account for evaluating the height \(h\) of the fluidized bed at the actual fluidization condition. This is achieved by applying the two phase theory explained in [7]:

\[
\frac{h - h_{\text{mf}}}{h_{\text{mf}}} = \frac{1}{h_{\text{mf}}} \int_0^h \left( \frac{u - u_{\text{mf}}}{u - u_{\text{mf}} + u_{\text{bubble}}} \right) \, dz
\]  

In equation (4) \(Y\) is a constant given in [7] and \(u_{\text{bubble}}\) is the rising velocity of the bubbles formed in the fluidized bed. With the obtained bed expansion the total pressure drop can be calculated via a simple force balance of the fluidized material and the pressure drop added resulting from the orifice plate of
the fluidized bed. So the required power of the blowers $P_{blower}$ providing the fluidization air can be determined via the relation for isentropic compression in conjunction with an efficiency factor of the compressor. The electrical power consumed by the conveyors $P_{conv}$ transporting the sand to the bunker is evaluated by multiplying the sand mass flow rate with the specific work required to lift the sand from the fluidized bed HEX outlet to the top of the bunker and corrected by the efficiency of the conveyor equipment.

For each cell of the HEX the fluidization characteristics are calculated considering the change of the thermodynamic properties and the boundary conditions. Since all cells connected to a single windbox are fed with the same fluidization air mass stream, their individual porosity cannot be optimized independently. Therefore the optimum porosity of all cells attached to a single windbox is determined by using the average temperature of all these cells. Based on the presented methods the thermal behaviour of each cell is described via a dimensionless HEX-model (cell method) applying the NTU-method presented in [2]. Due to the page limitation only the basic way of the proceeding is given here; a detailed description of the cell method is given in [2]. In each cell the dimensionless temperature change of both the sand and fluid mass stream is described via the following two equations:

$$\frac{1 - P_{sand}}{P_{fluid}} T^*_{sand,n+1} - T^*_{sand,n} + P_{sand} T^*_{fluid,n-1} = 0$$

$$P_{fluid} T^*_{sand,n+1} - T^*_{fluid,n} + (1 - P_{fluid}) T^*_{fluid,n-1} = 0$$

Equations (5) and (6) represent the dimensionless temperature changes of the respective stream (fluid, sand) in cell $n$. $T^*_{sand,n}$ and $T^*_{fluid,n}$ represent the dimensionless cell outlet temperatures of both the sand and fluid stream. The values of $P_{fluid}$ and $P_{sand}$ are evaluated by correlations given in [2] for a counter current heat exchanger. Formulating equation (5) and equation (6) for each computational cell leads to an algebraic system of equations for the dimensionless temperatures $T^*_{sand,n}$ and $T^*_{fluid,n}$ of all cells. Since the HEX-inlet temperatures of both streams are known, this system of equations can be solved directly. Based on the obtained dimensionless steady state temperature distribution, the real (none dimensionless) temperatures of both the fluid and sand stream can be obtained. Since the fluidization characteristics and the temperatures of the cells influence each other, the system of equations for fluidization characteristics and temperature field have to be solved iteratively. The overall algorithm for calculating the temperature distribution and fluidization characteristics inside the fluidized bed HEX is shown in figure 5.
3. Regenerator

In contrast to the sandTES Active Fluidized Bed Thermal Energy Storage system a regenerator type TES is a passive energy storage system. A regenerator system consists of a solid porous packing material through which hot and cold gas are alternately passed [2]. It can either be an accumulation of loosely stacked bricks, stones or a closely stacked accumulation of moulded bricks. During charging mode a hot gas stream enters the regenerator and flows through the channels of the packing material. The heat is transferred from the gas stream to the solid packing material. The solid material is heated up and the temperature distribution inside the solid packing is shifted to higher temperatures. By these two effects thermal energy is stored in the solid filler material. A consequence of the temperature rise of the solid material is that, when the storage is being nearly saturated, the outlet temperature of the gas stream is not constant but rising with increasing charging time. This further implies that the gas stream is not cooled efficiently towards the end of charging. In other words: the storage is filled. For discharging the gas inlet and outlet are changed. The cold gas is introduced at the former gas outlet and heat is transferred from the hot solid filler material to the gas. Hence, the storage is emptied.

Model for Calculating a Regenerator

For assessing the performance of the regenerator it is modelled as a one dimensional domain containing the porous packing with assumed tubular gas channels. The unsteady conservation equations for energy of the solid material and the gas flow are spatially discretized by the finite-volume method. The resulting partial differential equations (PDE) are given by equation (7) and equation (8):

\[
(1 - \psi_g) \rho_S c_p, S \frac{dT_S(z)}{dt} = \alpha_{G,S} a_{V,S} (T_G(z) - T_S(z))
\]

\[
\psi_g \frac{d \rho_G c_p, G T_G}{dt} + \frac{d}{dz}(\rho_G h_G w) = \psi_g \frac{d}{dz}(\lambda_S \frac{dT_S}{dz}) + \alpha_{G,S} a_{V,S} (T_S - T_G)
\]

G stands for gas, S for solids, z is the coordinate in axial direction of the regenerator and \( a_V \) is the volumetric solid exchange surface in m²/m³. \( T_S \) represents the mean solid temperature which is different from the wall surface temperature of the tubular channels due to radial heat conduction in the solid filler material. The thermodynamic properties for the filler material, such as density \( \rho_S \), specific heat capacity \( c_p, S \) and thermal conductivity \( \lambda_S \) are taken constant over the temperature range considered. The density of the gas is determined from the ideal gas law by assuming constant pressure over the regenerator. The thermodynamic properties of the gas (dry air), its specific heat \( c_p, G \), and its thermal conductivity \( \lambda_G \) are evaluated by means of temperature dependent polynomials. The specific enthalpy of the gas \( h_G \) is calculated by means of

\[
h_G = \int_{T_{298.15}}^{T} c_{p,G}(T) dT
\]

The gas velocity inside the tubes \( w \) is determined by the conservation of mass. The volume fraction of the gas and the specific surface area defining the heat transfer are determined by the overall dimensions, the specific geometric arrangement of the tubular channels and the average wall thickness of the filler material. It is assumed that the tubular channels are arranged in form of an equilateral triangle, with the solid filler material occupying the space between the tubular channels. The volume fraction of gas can then be determined by the inner diameter of the tubular channels \( d_{tube} \) and the wall thickness \( s_W \) of a single channel by equation (10).

\[
\psi_G = \frac{d_{tube}^2 \pi}{2(d_{tube} + 2s_W) \sqrt{3}}
\]

With the volume fraction of gas the specific heat transfer area per m³ storage volume can be obtained according to [2] via (equation 11)

\[
a_{V,S} = \frac{4\psi_G}{d_{tube}}
\]
The heat transfer coefficient \( \alpha_{G,S} \) is calculated by using a Nusselt-correlation in conjunction with a correction for the radial heat conduction inside the walls of the tubular channels as presented in [2]. This approach is chosen because only the mean temperature of the solid material is known. The corrected heat transfer coefficient is calculated by equation (12)

\[
\alpha_{G,S} = \frac{1}{1 - \frac{d_{eq}}{d_{tube}}} \\
\text{with}
\alpha_{Nu} = \frac{Nu}{d_{tube}} \lambda_G
\]

and

\[
d_{eq} = \frac{d_{tube} - \psi_G d_{tube}}{d_{tube}}
\]

The correlation of the Nusselt number \( Nu \) has been taken from [2] for the appropriate flow regimes inside the tubular channels. The factor \( \psi \) takes into account the temperature distribution inside the walls of the tubular channel. The value of \( \kappa \) is introduced in [2] as a function of thermal diffusivity, length of charging and discharging period and the mean plate thickness \( d_{eq} \) (equation (14)) if the filler material would be made out of equivalent plates instead of tubular channels. The PDE-system formed by equation (7) and equation (8) is time integrated by means of the software package ANSYS-Fluent 13. Since the boundary conditions at the inlets and outlets are changed after each charging and discharging period as described in section 3, special care has to be given to time step sizes for the time integration of the PDE-system. The change of the boundary condition induces a rather large disturbance in the temperature distribution at the respective inlet regions. Therefore a variable time step is applied with the time step being small at the beginning of a charging and discharging period and being increased gradually towards the end of the respective period. This enables an efficient and accurate solution of the PDE-system.

4. Comparison of a sandTES and Regenerator for an ACAES plant

In this work information from [6] about the ACAES project ADELE has been taken to compare the active fluidization thermal energy storage with a passive regenerator storage system. In figure 6 the sandTES plant is shown on the right-side and the regenerator explained in [6] on the left-side. The ACAES plant is supposed to operate at a cavern pressure of 70 bars, using a two stage compression with an assumed compressor outlet temperature of 600 °C. The underground cavern temperature is estimated to be about 40 °C. The passive TES (regenerator) from [6] is a 50 m high pressure vessel containing hot ceramic bricks at a pressure of 70 bar. The inner diameter of the storage material is approx. 14 m. The outer pressure vessel containing the inner repository for the hot filling material is made out of pre-constrained concrete and must be cooled by forced convection to avoid damage. The active sandTES plant consists of two sand hoppers (cold and hot), the fluidized bed HEX and auxiliary equipment such as blowers, air/air heat exchangers and mechanical transport equipment such as screws and bucket chain conveyors. For both TES systems, two major operation strategies are feasible: either the mass flow rate of the pressurized air is kept constant over a storage cycle (strategy 1) or the electrical input (275 MW) and output are kept in the same range (strategy 2). Both strategies have been investigated in case of sandTES, for the regenerator only strategy 2 has been considered. The length of the charging period is 6 h, with a massflow rate of 300 kg/s pressurized air, for strategy 2 the resulting length of the discharging period is 4 h [6].

![Figure 6: Comparison sandTES with regenerator](image)
4.1. Quantification of Results

To compare both TES approaches, the exergetic cycle efficiency $\eta_{ex,cycle}$ of the TES system is chosen (not the entire ACAES plant). As indicated in equation (15) and equation (16) below, the exergetic cycle efficiency is defined as the ratio of output to input exergy flows [3] over a complete charge/discharge cycle. In case of the sandTES also the auxiliary power (blower power and energy needed to transport the sand and to overcome the HTF side pressure drop) has to be considered.

$$\eta_{ex,cycle} = \frac{\sum \text{output exergetic flows}}{\sum \text{input exergetic flows}} = \frac{\int_0^{\tau_{cha}} m_{\text{fluid}} e_{\text{fluid}} dt + \int_0^{\tau_{dis}} m_{\text{HTF}} e_{\text{HTF}} dt}{\int_0^{\tau_{cha}} m_{\text{fluid}} e_{\text{fluid}} dt + \int_0^{\tau_{dis}} m_{\text{HTF}} e_{\text{HTF}} dt + \int_0^{\tau_{cha}} m_{\text{blower}} e_{\text{blower}} dt + \int_0^{\tau_{dis}} m_{\text{conv}} e_{\text{conv}} dt}$$  

(15)

$$e_{\text{fluid}} = h(p,T) - h(p_{env},T_{env}) - T_{env}[s(p,T) - s(p_{env},T_{env})]$$  

(16)

The subscript fluid in equation (15) and (16) represents the compressed air stream which is directed into or taken out of the storage, $\tau_{cha}$ and $\tau_{dis}$ are the charging and discharging time intervals, $m$ is the massflow rate, $e_{\text{fluid}}$ stands for the specific exergy of the pressurised air at the inlet of the storage (charging) or outlet (discharging) and $s$ is the specific entropy. In the analysis for the regenerator, the exergy loss resulting from the gas pressure drop is neglected since the velocities inside the tubular channels are rather low due to a large cross section available for fluid flow. In case of the sandTES system the exergy loss due to pressure drop is fully taken into account. The ambient temperature $T_{env}$ is taken as 15 °C, the ambient pressure $p_{env}$ is assumed to be 1 bar.

4.2. Design and Simulation of a sandTES

The challenge in designing a sandTES for an ACAES plant is to minimize in parallel the temperature spread between the sand mass stream and the pressurized air mass stream and the pressure loss of the pressurized air through the tube bundle of the sandTES fluidized bed HEX. Since both criteria are contradictory (the more turbulent the inner pipe flow, the better the heat transmission value, but the larger the pressure drop), an optimum has to be found. This challenge becomes significant at low storage pressure ACAES (<25 bar) due to low pressurized air density, but can be handled accordingly at a storage pressure of 70 bar applied in the ADELE project.

Table 1: Design parameter sandTES

| HEX dimensions | Tube bundle configuration | Turbo-machinery | Massflows (Charge) | Overall parameter |
|----------------|--------------------------|-----------------|-------------------|------------------|
| Width: 17.3 m | $d_{\text{piping}}$: 57 mm | $\eta_{\text{blower}}$: 0.7 | $m_{\text{press,air}}$: 300 kg/s | $P_{\text{el,in}}$: 275 MW |
| Depth: 22.2 m | $d_{\text{piping,hor}}$: 9 | $\eta_{\text{compressor}}$: 0.75 | $m_{\text{sand}}$: 382.5 kg/s | $k$: 450 W/m²K |
| Height: 3.9 m | $d_{\text{piping,vert}}$: 26 | $\eta_{\text{turbine}}$: 0.9 | $m_{\text{air,fluid}}$: 5.1 kg/s | $A_{\text{net}}$: 10746 m² |

The simulations have been done with the Matlab program developed in [4, 5]. The design parameters are shown in table 1. For both operational strategies/simulations the same sandTES plant is used. In figure 7 the temperature vs. normalized surface curves of pressurized air and sand mass stream are shown. In strategy 2 the objective is to keep the power constant. Therefore the losses of the entire storage process must be compensated, what can only be achieved by increasing the mass streams during discharge accordingly. The electric power output, the exergetic efficiencies and mass flows are shown for both strategies in figure 8.

![Figure 7: Temperature course of strategy 2](image-url)
From the illustration of the composition of all losses on the left-hand side of figure 8 can be seen that the biggest losses are caused by imperfect heat transmission. This very loss can be minimized by increasing the net HEX area.

4.3. Simulation of a Passive Regenerator

The overall dimensions and design parameters of the regenerator type TES as considered in this work are summarized in table 2.

| Parameter                          | Variable | Unit | Value | Ref. |
|------------------------------------|----------|------|-------|------|
| Inventory Height                   |          | m    | 50    | [7]  |
| Inner Diameter                     |          | m    | 14    | [7]  |
| Diameter of single flow channel    | \(d_{\text{tube}}\) | m    | 0.01  | -    |
| Thickness of flow channel wall     | \(s_W\) | m    | 0.005 | -    |
| Specific surface area for heat transfer | \(a_{\text{VS}}\) | m²/m³ | 100.0 | -    |

For the regenerator, two different filler materials have been investigated: Silica Bricks and castable ceramics. Thermodynamic data (such as density, specific heat and thermal conductivity) of the investigated storage media have been taken from [1]. Since the exact temperature distribution inside the regenerator is not known neither for the end of a charging nor a discharging cycle, the time integration of equation (7) and equation (8) has been performed until a steady state oscillation of the cold end (charging) and hot end (discharging) temperatures was reached. The resulting gas temperature distributions inside the regenerator during a charging and discharging period for the application of ceramic bricks as filler material are depicted in figure 9. In figure 9 the gas temperature at the end of a charging period at the outlet of the regenerator (height=0 m, inlet into the cavern) is higher than the cavern temperature. In this work it is assumed that the cavern temperature is uniform and constant, thus the energy corresponding to the temperature difference between colder cavern and hotter pressurized air (at charging) is lost.

4.4. Comparison sandTES to passive Regenerator

Figure 10 provides a comparison of the exergetic efficiency over a charging and discharging cycle of the sandTES-system and the regenerator-system. The exergetic efficiency of a sandTES-system in case of immediate charging and discharging is slightly lower than the efficiency of the regenerator system. The main explanations are that sandTES requires additional energy for the transportation of the storage medium sand and that the temperature difference between fluid and storage is slightly higher.
According to [6] the outer wall of the concrete pressure vessel needs to be cooled which leads to heat losses of approximately 3% per day. Due to the temperature distribution of the filler material it is not straightforward to transform a heat loss into an exergy loss. For simplification it is assumed that the temperature distribution in the bulk is not modified during heat loss. This assumption allows to set the exergy loss proportional to the heat loss. The main conclusion is that the longer the storage time is the more advantageous a sandTES becomes from an exergetic point of view. From an industrial point of view many other criteria have to be considered, such as Capex, Opex, security, durability etc.

5. Conclusions
Active Fluidization Thermal Energy Storage (sandTES) offers a promising alternative to the currently available (state of the art) and investigated TES. A sandTES can be applied over a wide temperature range using a storage medium which is abundantly available, cheap and easy to handle. The natural material “sand” can be easily stored in large quantities over a long period of time with limited losses. A consequence of using an active TES is that auxiliary devices and power are needed for the transportation of sand. This results in associated additional investment and maintenance costs, especially for all rotating equipment (screws, bucket chain conveyers, blowers).

The comparison of sandTES to a passive Regenerator [6] showed that both approaches deliver comparable net exergetic performance values; for storage time lower than 24 hours, the passive Regenerator is slightly better whereas the sandTES is advantageous for longer storage periods.

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