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Numerical investigations of long-term thermal performance of a large water pit heat storage

Zichan Xie, Yutong Xiang, Dengjia Wang, Oleg Kusyy, Weiqiang Kong, Simon Furbo, Jianhua Fan

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Abstract

Water pit heat storage is an effective long-term heat storage. Experimental and theoretical investigations of a 60,000 m³ water pit heat storage in Dronninglund, Denmark was carried out with an aim to study its long-term thermal performance. Experimentally, detailed measurements were analyzed to monitor thermal behavior of the store, including flow rates, temperatures of the inlet and outlet flow, thermal stratification in the store and temperatures of the surrounding soil at four different depths. Theoretically, a simulation model of the water pit heat storage was developed based on the Type 343 model (ICEPIT) in TRNSYS. The calculated temperatures were compared to the measurements. The developed model was used to investigate the influence of soil properties and inlet arrangement on the thermal performance of the store. The result show that the simulation model predicts well the thermal behavior of the store. Over the whole year, the calculated average temperature of the store differed within ±4.7 K from the measured values. The measurements show that the annual charged and discharged heat are 12,787 MWh and 11,957 MWh respectively. The deviation of the charged and discharged energy between the simulation and the experiment did not exceed ±4%. The annual storage efficiency reached 90% in the second year of operation. The calculated ground temperatures were consistent with the measured ones at different depths. Measured heat loss of the store is 1392 MWh. As elucidated by the model, the heat loss from the cover contributed to 60% of the annual heat loss, while the heat loss from the sidewall and the bottom contributed 38% and 2% respectively. The influence of soil types on thermal performance of the store is significant. For a PTES surrounded by dry soil, the storage efficiency can reach 90%, while for a PTES surrounded by wet sandy gravel, the storage efficiency decreases by 7%. Change of the height of the middle inlet/outlet diffuser could slightly improve the storage efficiency by 1%. The restraints of the model were analyzed and the focus areas of further investigations were pointed out. The findings of the paper give good reference for researchers, designers and consultants in planning of a solar heating plant with a large water pit heat storage.

1. Introduction

In view of the current worldwide effort towards highly renewable integration, energy storage is the core technology (Lund et al., 2016; Novo et al., 2010). For the heating sector, seasonal thermal storage (STES) aroused intense concern from the 1960’s when the concept was first proposed (Mangold and Schmidt, 2009). It has been confirmed that STES can significantly increase the solar fraction of central solar heating plants from 25% to 50% (Sørensen, 2019).

Economic analysis of solar heating plants can be achieved by combining TRNSYS and optimization program like genOpt (Jensen, 2014). Tulus’s study in Mediterranean climate regions has proved that by optimizing on system level, the environmental impact of central solar heating plants with seasonal storage (CSHPSS) can be dramatically reduced with lower investment (Tulus et al., 2016). Therefore, CSHPSS is a cost-efficient solution for the near future (Tulus et al., 2016; van der Heijde et al., 2019).

There are three available technologies for STES: Sensible heat storage, latent heat storage, and chemical storage. Yet, large-scale latent heat and chemical storage are still in the experimental stage (Xu et al., 2014). Although sensible seasonal storage has been successfully utilized...
in practice such as Denmark and Germany, the permeation of this technology is still low in general. The major challenge impeding the global market for seasonal storage systems is economic feasibility (Tschopp et al., 2020). Water from natural sources is an ideal heat storage medium to reduce initial investment costs (Novo et al., 2010). Tank Thermal Energy Storages (TTES), and Water-Gravel Thermal Energy Storages (WGTES) are the most attractive techniques in the field of seasonal heat storage (Bott et al., 2019).

Many studies have stated that it is economical to expand storage volumes (Lottner et al., 2009). Also, heat losses of the heat storage decrease percentage-wise with a lower surface area/volume ratio by augmenting the storage volumes (Duffie and Beckman, 2013). Pure water takes advantage of high thermal capacity, whereas backfilling materials of the gravel-water pit can reduce the initial costs (Novo et al., 2010). Although TTES has the best performance and less dependence on location, the materials are expensive for large systems. Therefore, PTES is regarded as the most promising approach for large-scale storage (over 10,000 m³) (Ochs et al., 2020).

Large-scale thermal storage experienced rapid development in the last decade. Numerous attempts have been made in Europe (Bott et al., 2019). According to the real case studies in Denmark, water pit storages perform as expected (Jensen, 2014; Winterscheid and Schmidt, 2017; Fan et al., 2017a). Considering local geographic conditions and initial investment like excavation costs, most existing PTES were constructed as a reversed pyramid stump (Jensen, 2014; Ochs et al., 2020; Dahash et al., 2019b). Despite no severe failure with PTES on the technical level (Ochs et al., 2009), the upscaling design has not been sufficiently addressed (Dahash et al., 2019b). Current studies paid attention to optimal design for small tanks (Yang et al., 2016; Fan and Furbo, 2012), while little efforts were put on large-scale heat storages, especially for underground heat storage systems. Accordingly, finding a suitable modeling tool is significant for the planning phase of large-scale projects where the experiment is unrealistic, which plays a crucial role in implementation of this advanced technology in more countries.

The common modeling tools that have been utilized for STES can be divided into two categories: (a) computational fluid dynamics (CFD) like ANSYS Fluent and COMSOL, (b) energy system simulation such as TRNSYS and Matlab/Simulink. CFD is the most exact approach to simulate on the component level, which has been widely employed in optimal design for small tanks rather than large-scale storages (Dahash et al., 2019b). A validated CFD model of a large-scale tank developed by Panthalookaran et al. (2008) was used to investigate the influence of boundaries on thermal storage efficiency (Panthalookaran et al., 2011). Chang et al. (2017) carried out another CFD model validated by experimental tests to investigate natural convection in PTES. Besides, Fan et al. (2017b) used a CFD model to study the thermal performance of a real case in Marstal (Denmark). A model on COMSOL for large-scale water pit and tank developed by Dahash et al. (2019a) matched well with the measured data from Dronninglund (Dahash et al., 2020), Bai et al. (2020b) state that water pits take four to six years to run steadily. Nevertheless, time-consuming and complex computing is needed for each project (Dahash et al., 2019b), and typically it is only feasible for calculations of short operation periods such as the study by Fan et al. (2017b). Though Ochs (2015) compared a model based on Matlab-Simulink where only simple conditions were involved with a CARNOT model. Then, these models cannot be connected to other components which means further integration planning on system-level was impossible. Yet, integration optimization is necessary in combining thermal storage with different heat sources which is crucial to economic feasibility (Moser et al., 2018).

As for TRNSYS simulation environments, Ochs (2009) compared the performance of three commercial models for the simulation of TTES, Type 340, Type 342 and Type 343, and found that coupling thermal storage with ground is necessary for simulations of buried storages. An obvious misestimation of the heat loss from the bottom and sidewall appeared owing to the neglect of soil. There are several published models adopted for simulation of underground storages, including the XST-model (type 342) (Marazella, 1992), the ICEPIT-model (Type 343) (Hornberger, 2009), and the UGSTS model developed by Bai et al. (2020a). Type 342 was validated by a 2795 m³ water pit with differences in charged/discharged energy of less than 5% (Raab et al., 2005).
Type 343 has been validated by a 1050 m$^3$ gravel/water cold storage with a maximum temperature deviation from the measurement at 1.5 K in the storage (Steenhoven et al., 1996). The UGSTS model has a good agreement with the experimental data of a cylindrical underground concrete water pit with 3000 m$^3$ water. All of the studies mentioned above focus on storages with relatively small volumes (less than 10,000 m$^3$), whereas experience on modelling of thermal performance of large-scale storages is still limited.

Furthermore, although many studies mentioned that TRNSYS performs well in modeling the storage in different shapes, the focus of the available literature was more on the system level (Jonas et al., 2017; Fan et al., 2017a). TRNSYS models were widely used in sizing the thermal storage like (Urbaneck, 2004) and control strategies like (Li et al., 2019) rather than in-depth analysis of models for seasonal storage. Besides, TRNSYS models have been applied in design for CSHPSS like Gram in Denmark (Jensen, 2014), the detailed simulations for PTES were rare in available literature (Bai et al., 2020b).

The ICEPIT model (type 343) was primarily developed for calculation of water/gravel heat stores or ice reservoirs, where there was a mixture of water and solid phase (ice or gravel). This study fills the gap by validating the model for calculation of PTES with detailed measurements on a 60,000 m$^3$ PTES in a Danish solar heating plant in operation since 2014. The average storage temperature, thermal stratification in the store, utilized energy, heat loss, and other key performance indicators were investigated experimentally and theoretically on a long-term basis. The validity of the model will be elucidated. The investigation gives insights into limitations and possible improvements of the model. Furthermore, the developed model will be used to study the influence of soil properties and inlet arrangement on the thermal performance of the store. It provides reference for planning and design of water pit heat storages in different locations.

2. Methodology

2.1. Experimental method

2.1.1. The solar heating plant

The investigated water pit heat storage is part of the solar heating plant in Dronninglund, Denmark, about 30 km north from Aalborg (Fig. 1). The total collector field has 37,573 m$^2$ solar collector, which was designed to provide 40% of the local district heating network demand for 1,350 customers (Baerbel, 2013). The 60,000 m$^3$ seasonal thermal storage was put into operation May 2014. The maximum power of the collector field is 26 MW, while the maximum consumption of the district heating system in the coldest periods is 12 MW.

The heating network’s supply depends on the solar field, bio-oil boilers and natural gas boilers, illustrated in Fig. 2. The operation temperature of the district heating system is 75/40 °C in summer (non-heating season), and 80/35 °C in heating season. A bio-oil boiler driving heat pump extracts heat from the middle part of the store when water temperature is higher than 20 °C (PlanEnergi, 2011). The advantage with heat pump is not only to utilize the low-temperature heat but also to empty the storage before summer with a decrease in the heat losses and an increase in solar collector efficiency. The water pit can be heated up to 90 °C (Tschopp et al., 2020; Baerbel, 2013). This temperature was designed for a minimum lifetime of the HDPE (high density polyethylene) liner of 20 years (PlanEnergi, 2015).

2.1.2. The water pit heat storage

This water pit was built in an abandoned gravel pit, where soil consists of dry fine sand (PlanEnergi, 2011). From the geological investigation, the groundwater table is approximately 17 m under the ground in this site (PlanEnergi, 2011; Jensen et al., 2015). Thus, a hole
of 60,000 m³ with a depth of 16 m was constructed like an upside-down pyramid cone with a slope of 26.6°, shown in Fig. 1. The lengths of the top and bottom edges are 90 m and 26 m respectively (PlanEnergi, 2011), depicted as the yellow region in Fig. 5. The storage is covered by a floating insulated cover. The main part of the lid is the flexible insulation (Nomalen 28 N from NMC) with a thickness of 240 mm. There is an air gap between the floating liner and the insulation to vent away moisture. Except for the top surface, this storage is only covered with 2.5 mm HDPE liner and a layer of geotextile (Jensen, 2014).

The pipe connections were led through the bottom of the storage. Three pipes each ending with a diffusor with a thickness of 0.6 m are located at the top, the bottom and the volumetric middle of the storage respectively (Jensen, 2014), shown in Fig. 5. These pipes can be operated at the same time.

2.1.3. The experiment

The measurement includes water temperatures in the storage and pipes, ambient air temperature, and volume flowrate, which were...
recorded with a time step of 10 min. However, the data of hourly global solar irradiation was downloaded from the database Renewables.ninja using the dataset MERRA-2 (Gelaro et al., 2017). The measurement accuracy of flow meters was ±2% and that of the thermocouples was ±0.1 K (Jensen et al., 2015).

The locations of the heat flux sensor in the cover and temperature sensors in the storage and soil are demonstrated in Fig. 3. The temperature sensors were arranged along two lines, evenly distributed from the bottom to the top of the storage. Each line has 16 thermocouples, from 0.5 to 15.5 m and from 1 to 16 m respectively. There are four temperature sensors placed underground in the position C. The depths of the temperature sensors are 10, 15, 20 and 25 m below the top surface. Accordingly, the shortest distances from the sensors to the inclined sidewall are 9, 13, 18 and 22 m respectively. The temperature of the groundwater was in the range of 8.1 to 8.8 °C in 2014 (Jensen et al., 2015).

Besides, temperature sensors are placed in the inlet/outlet pipes in the technical room in order to obtain heat flow of the PTES. Response time of the temperature measurement can be as long as 20 min under low flow rate conditions caused by thermal inertia of remaining water volume in the pipes between the PTES and the technical room. Temperature measurement in the storage at the level of 0.5 m, 2 m, 5 m, 6 m, 7 m, 8 m and 16 m was invalid, thus partly disregarded in the analysis. Additionally, since the flow meters might give a noise signal even though there is no flow in the pipe, the lower threshold value of the flow rate measurement was set to 1.5 m³/h. Any flow rates lower than 1.5 m³/h will be disregarded. Although the mean water temperatures in pipes were recorded for 10 min, the data was invalid when the water was stagnant. Therefore, the data of water temperatures in pipes was removed while flowrates were zero.
2.2. Numerical analysis

The original Type 343 model was used to calculate thermal performance of the underground heat storage with only one pair of direct inlet/outlet. A modified model was developed by the authors to enable calculations of up to 8 pairs, whose positions can be set by users in order to address three pairs of inlet/outlet in Dronninglund water pit heat storage. Since the model is developed based on an assumption of an axisymmetric boundary condition, it can only calculate heat storages in a shape of a cylinder or a reversed truncated cone.

The Type 343 was based on the following assumptions (Steenhoven et al., 1996):

1. Constant thermal conductivity and constant heat capacity for each region are used in the calculation. Both thermal conductivity and heat capacity of the soil around the PTES might vary slightly over the year, the averaged values are representative.
2. Constant heat transfer coefficients were set for surfaces of the storage. Generally, the free convection heat transfer coefficient has a minor influence on heat loss of the store. Thus, this assumption will not sacrifice prediction accuracy of the model.
3. The thickness of the liner and geotextile around the side and the bottom wall is neglected. Thermal resistance was included in the total heat transfer coefficients. Since thermal resistance of the liner and the geotextile is slight, this simplification is considered acceptable.
4. The embankments around the upper part of the storage is disregarded, which means that the whole storage is underground.
5. Temperature inversion is not allowed in the storage. The adjacent boundary temperatures refer to the adjacent soil temperature and heat transfer between the soil and the ground are taken into consideration when the storage temperatures are calculated. The interior temperature of the storage is calculated by Eq. (2), based on a modified Euler-method, as described in TRNNSYS manual (Klein, 1988). The coefficients, \( f_1 \) and \( f_2 \) represent the exported heat coefficient from the element (W/K) and imported heat into the element (W), given in Eqs. (3) & (4) respectively.

\[
\frac{\partial T}{\partial t} = f_1 \times T_j + f_2
\]

where

\[
f_1 = -K_j - K_{j-1} \cdot A_j U_{side}
\]

2.2.2. Heat transfer in the storage region

Vertical thermal conduction, forced convection during charging and discharging via the direct water flow and heat transfer between the storage and the ground are taken into consideration when the storage temperatures are calculated. The interior temperature of the storage is calculated by Eq. (2), based on a modified Euler-method, as described in TRNNSYS manual (Klein, 1988). The coefficients, \( f_1 \) and \( f_2 \) represent the exported heat coefficient from the element (W/K) and imported heat into the element (W), given in Eqs. (3) & (4) respectively.

\[
V_j(C_p\rho)_j \frac{\partial T_j}{\partial t} = f_1 \times T_j + f_2
\]

where

\[
f_1 = -K_j - K_{j-1} \cdot A_j U_{side}
\]

2.2.1. The grid scheme

Type 343 is a two-dimensional model, which simplifies the storage region into one dimension, presented in Fig. 4. The storage geometry can be determined by the heights of water layers and diameters of the top and the bottom surfaces. Cells created in the water region depend on the input heights, while other domains’ cells will be generated automatically.

For the regions above the storage, grids along vertical direction rely on the thickness of insulation and the thickness of soil above the storage, where the widths of cells are based on soil region (a)(b). As the thin components of the PTES, including HDPE liner, geonet and extruder welding (Jensen, 2014), have negligible effects on the thermal performance, only one-layer thermal insulation with the thickness of 0.24 m above storage was considered in this research.

Eight horizontal layers are formed below the storage, whose height increase by a factor of 1.5 from 0.3 m adjacent to the storage to 14.8 m close to the boundary. Similar treatment of gridlines is also utilized for the soil(d) region, with seven columns expanding to 16.1 m from the top edge of the store.

As shown in Fig. 4, denser gridlines will be generated to the place closer to the corner for the region soil(b). Regarding the region soil(a), Fig. 6 illustrates that every two layers of storage correspond to one soil layer of the region beside the storage, which is divided into n/2 columns evenly in the horizontal direction. It means odd layer numbers of the storage region are not allowed in Type 343.

In order to make the soil boundary match with the storage boundary, the storage was divided into 20 layers with the same heights of 0.8 m. Three diffusers were arranged properly in layers 1, 7 and 20 shown in Fig. 6. Meanwhile, to best represent the realistic thermal stratification in the upside-down pyramid stump heat storage in Dronninglund, the original storage volume, depth and slope were used in the model. It means that the surface areas of the storage in the model will be different from the original shape, shown in Fig. 5. The difference in surface area can be compensated by modifying the heat transfer coefficient of the surfaces proportionally, given in Eq. (1), because a product of the surface area and the heat transfer coefficient gives the heat loss coefficient. Therefore, correct heat loss from the PTES can be calculated.

\[
U_{simulation} = U_{original} \times \frac{A_{original}}{A_{simulation}}
\]
flow, where \( m \) is the mass flow of water. Since heat transfer caused by flow also depends on the flow direction, \( M_i \) and \( M_o \) of the storage layer between inlet and outlet are defined separately as following when the inlet entrance or outlet entrance is at a higher position. \( M_i \) and \( M_o \) are equal to zero in the top layer and the bottom layer respectively because there is no water flow from the adjacent storage layer.

\[
\begin{align*}
M_i &= mC_p; M_o = 0, \quad \text{Inlet entrance at higher position} \\
M_i &= 0; M_o = -mC_p, \quad \text{Inlet entrance at lower position}
\end{align*}
\]  

(8)

2.2.3. Heat transfer in the soil/insulation region

The transient energy balance equation for the calculation in the 2D soil/insulation region can be summarized as Eq. (9). The coefficients, \( f_1 \) and \( f_2 \) represent similarly as that in 1D region.

\[
f_1 = -K_i\Delta_z - K_i\Delta_z, \quad f_2 = K_i\Delta_z + K_i\Delta_z + K_i\Delta_z + K_i\Delta_z
\]  

(10)

Due to the assumption of first-type boundary condition, the boundary conditions of soil/insulation; regions are set as Eq. (12). \( \alpha \) is the ground surface absorptivity.

\[
\begin{align*}
K_{i,k} = 0, K_{i,k+1} = 0; \quad K_{i,k} = 0, K_{i,k+1} = 0 \\
T_{k+1} = T_{k+1} + 2T_{k}A_{gr} + \frac{2T_{k}A_{gr}}{\Delta_w} T_{k+1}, \quad T_{k+1} = T_{k+1} = 8.3 \degree C
\end{align*}
\]  

(12)

2.2.4. Energy calculation in the model

Charged and discharged energy can be obtained directly from the model as Eq. (21), where the positive value represents charging. Since the calculation is based on mass balance, the inlet mass flow is equal to the outlet mass flow. Variable heat capacity of water is neglected in this model, referring to 4187 J/(kg-K).

\[
Q_{\text{ch/disch simulation}} = m_o \times (T_{o} - T_{i}) \times C_{p,o} \times \Delta t
\]  

(21)

Eq. (22) shows that the total heat loss from the storage consists of three parts (\( Q_{\text{loss}} > 0 \) means heat gain): heat loss from the top surface, the sidewall and the bottom surface. The heat losses are calculated by multiplying the time step \( \Delta t \), surface area \( A \), heat transfer coefficient \( U \), and temperature difference between the storage and the adjacent soil or insulation \( \Delta T_{i,\text{soil/insulation}} \).

\[
Q_{\text{loss}} = Q_{\text{loss,top}} + Q_{\text{loss,side}} + Q_{\text{loss,bot}}
\]  

(22)

Internal energy content of the storage is given in Eq. (23) with a reference temperature of 0 \degree C. The density of water in each storage layer is calculated as Eq. (24). Internal energy change \( \Delta Q_i \) is defined as the internal energy content in the end minus that in the start of the period.
\[ Q_u = \sum_{j=1}^{20} (T_j - T_{bas}) \times C_p \times V_j \times \rho_j \] (23)

### 2.2.5. Parameters

The initial conditions of the PTES and the surrounding soil have an impact on thermal performance of the PTES. Since only uniform temperatures are allowed in the model for the PTES and for the surrounding soil in the start of the simulation, special treatment is necessary to generate the best possible initial conditions. One year pre-simulation with a heat balance of the PTES was carried out to generate thermal stratification in the PTES and temperature gradient in the soil region. The result of the pre-simulation was used as the initial conditions of the second year.

The initial homogeneous soil temperature and infinite ground temperature were set as 8.3 °C based on the measured data in 2014. According to the annual average storage temperature at 50 °C in 2015, volumetric specific heat of water at 4137 kJ/(m³·K) and thermal conductivity of water \( \lambda_w \) at 0.63 W/(m·K) were fixed in Type 343. The influence of temperature on water density was considered as the following equation (Hornberger, 2009):

\[ \rho_w = 0.778 + 0.0719543 T - 8.387377 T^2 \times 10^{-3} + 5.424757 T^3 \times 10^{-5} - 1.9017 T^4 \times 10^{-7} \] (24)

From the geological investigation (PlanEnergi, 2011) and reported properties of sand (Hamdhan and Clarke, 2010a; Łukawska et al., 2020; Pagola et al., 2017), the thermal conductivity was set to 1.5 W/(m·K) and the constant volumetric thermal capacity was set to 1800 kJ/(m³·K). Yet, the water content of soil is controversial (Dahash et al., 2020), which should be investigated further. Ground surface absorptivity was assumed to be 0.8 (Tajfar et al., 2020).

For the PE/PEX insulation with a thickness of 240 mm, the thermal conductivity and the volumetric thermal capacity were assumed to be 0.06 W/(m·K) and 80 kJ/(m³·K) respectively (Almanza et al., 2004). Considering a 30%-50% degradation of the thermal conductivity caused by high temperature and moisture (Ochs et al., 2008), it was higher than the standard value of 0.04 W/(m·K) for new materials, as published by the company (NMC Termonova oy, n.d.). A revisionary heat transfer coefficient of 26.6 W/(m²·K) was chosen for the top surface of the PTES, referring to the thermal resistance for outdoor surfaces of 25 W/(m²·K) (Ochs, 2009).

Since there was no insulation between the water pit and the ground around the side and the bottom wall, the heat transfer coefficient was determined based on water-side convection and the thermal conduction in the liners. Compared with the impact of natural convection, the forced convection can be neglected in large-scale storage (Bai et al., 2020a). The heat transfer coefficient of both the side and the bottom were given by Eq. (25), where \( f_{area} \) is the ratio of the real surface area to the surface area in the simplified model. Results were presented in Table 1.

\[ U = f_{area} \times \frac{1}{h + \frac{\delta_{bas}}{\lambda_{bas}}} \] (25)

The natural convection coefficient along the inclined wall and Grashof number \( Gr_{side,v} \) can be calculated by Eqs. (26)–(27) (Bergman et al., 2011). The temperature difference between the inner cell and the outer cell was assumed to be 10 K (Forkel and Daniels, 1995). The inclined angle of the sidewall \( \theta \) is 26.6° in this case (Jensen, 2014). The properties of water were used to calculate \( h_{bas} \) based on the annual average temperature of the storage at 50 °C.

\[ h_{side,v} = \frac{\lambda_{side,v} \sin^2(\theta)}{H} \left[ 0.825 + \frac{0.387 (Gr_{side,v} Pr_{side,v})^{1/3}}{1 + \left( \frac{Pr_{side,v}}{Pr_{water}} \right)^{2/3}} \right]^2 \] (26)

\[ Gr_{side,v} = \frac{cos(\theta) \beta \cdot (T_{side} - T_w) D^3}{v^2_{side,v} \sin(\theta)^3} \] (27)

When \( Gr_{P} \) is larger than \( 10^7 \), the heat transfer coefficient along the lower surface of a cold plate is given as Eqs. (28)–(29) (Bergman et al., 2011). In terms of the annual average bottom temperature, the properties of water at 32 °C were used to calculate \( h_{bas} \). In the above equation was the equivalent diameter of the lower surface.

\[ h_{bas} = \frac{6.6 \rho_{bas} T_{bas} Pr_{bas}}{L_{bas}^2} \] (28)

\[ Gr_{bas} = \frac{g \beta (T_w - T_{bas}) D^3}{64 v_{bas}^2} \] (29)

### 2.2.6. Experimental energy calculation

The measured charged and discharged energy were calculated by Eq. (30), with the property of water, \( \rho \) and \( C_p \) based on the inlet and outlet temperature according to Eq. (24) & (31). Thus, the difference between the measurement and the simulation depends on water temperature in the outlet pipes. Experimental total heat loss was calculated indirectly by the energy balance as Eq. (32), where the change of internal energy content of the PTES was calculated by Eq. (33). Each measured temperatures represent the temperature of the layer 0.25 m below and 0.25 m above the sensor as shown in Fig. 3.

\[ Q_{ch/disch.measured} = ((C_p \rho) v_{in} \times T_w - (C_p \rho) v_{out} \times T_{in}) \times V_{in} \times \Delta \tau \] (30)

\[ C_p = 4432.6 - 1.819 \times (T + 273.15) + 0.0033 \times (T + 273.15)^2 \] (31)

\[ Q_{loss.measured} = \sum Q_{ch/disch.measured} + \Delta Q_{dis.measured} \] (32)

### Table 1

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<th>Bottom</th>
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### Table 2

| Key values for the PTES in Dronninglund in 2015. |
|-----------------|-----------------|-----------------|
| Items | Experimental result | Simulation result |
| Storage heat capacity | 5213 MWh | 5198 MWh |
| Maximum temperature | 89.2 °C | 89.0 °C |
| Minimum temperature | 12.7 °C | 12.7 °C |
| Storage efficiency | 89% | 90% |
| Charged energy | 12,787 MWh | 12,236 MWh |
| Discharged energy | 11,957 MWh | 11,585 MWh |
| Internal energy change | −562 MWh | −594 MWh |
| Thermal losses | 1392 MWh | 1245 MWh |
ΔQ_{st,measured} = \sum_{j=1}^{32} \int_{T_{j,\text{start}}}^{T_{j,\text{end}}} (C_p \times \rho) dT \times V_j \quad (33)

3. Experimental validation

The key figures for the PTES in Dronninglund in 2015 are listed in Table 2. Comparing the simulation and the experiment, there is a good agreement on the discharged energy. The reasons for deviation in charged energy and heat loss will be discussed in this section.

To easily compare different storage designs, it is worth analyzing the accuracy of critical indicators. There were two common thermal performance indicators: The storage efficiency \( \eta_{st} \) and the heat capacity \( Q_{\text{capacity}} \), see Eq. (34) & (35). The results proved that the calculated thermal performance of the PTES by the simulation is reliable. Since Type 343 exactly predicts the operating temperature range of the storage, it can be used for optimization in operation under the safe operating temperature to make sure the stability of the expensive floating cover (Jensen, 2014).

\[
\eta_{st} = \frac{Q_{d} + \Delta Q_{st}}{Q_{ch}} \quad (34)
\]

\[
Q_{\text{capacity}} = (C_p \rho)_{w} V_{w} (T_{st,max} - T_{st,min}) \quad (35)
\]

3.1. The storage region

3.1.1. Temperature

The average temperature of the PTES determines the energy content of the storage. The experimental mass weighted average temperature obtained by mass weighted average of 32 temperature sensors was compared to the calculated average temperature, shown in Fig. 7. The result shows a good agreement between the measured and the calculated temperatures throughout the whole year, with a difference below 0.5 K at the end of the year. The maximum difference was 4.7 K in August. The relatively notable error could result from measurements and simulation. Firstly, the cause of the error could be inaccurate measurement of flow rates in the pipes. Although considerable error over 300 m\(^3\)/h appeared in few short periods, the periods with a difference between the outlet volume flow and the inlet volume flow larger than 5 m\(^3\)/h accounted for over 3% of the whole year. The errors are significant during charging with a frequent switch of the operation modes or with high flow rates in the pipes. This model’s restriction also contributes to the disappointing result in the summer, which will be discussed in detail in 3.1.4.

The simulated charged and discharged energy were directly determined by the outlet water temperature since the flow rate and inlet water temperature were given. Fig. 8 displays the outlet temperatures of the three inlet/outlet pipes. It can be seen that the simulation result matches well the experiment. The fluctuating temperature of measurement could be explained by the temperature measurement delay in the pipes when operating mode switches. Although the flowmeters can respond immediately, it might take twenty minutes (2 time steps) for the temperature sensors installed in the technical room to react to the changes when the flow rate was low. There are connection pipes between the PTES and the temperature sensors in the technical room. Thermal inertia of the remaining water volume in the pipes caused the delay. When the supply water temperature is not high enough for the top inlet, the middle pipe will be utilized as an inlet to charge the PTES. During discharge, the middle pipe could be used to extract heat from the PTES if water temperature in the middle of the PTES is appropriate for the district heating. Thus, the fluctuation was severest in spring when the operating mode of the middle pipe changed frequently.

Additionally, the results show that even if the initial average temperature was the same, assumed data could not achieve the same thermal stratification at the very beginning. It shows a 5 K difference in the top pipe (Fig. 8 (a)).

<table>
<thead>
<tr>
<th>Conditions</th>
<th>Top/Middle</th>
<th>Top/Bottom</th>
<th>Middle/Bottom</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>Charging</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Measurement (MWh)</td>
<td>879</td>
<td>10,329</td>
<td>1551</td>
<td>12,787</td>
</tr>
<tr>
<td>Simulation (MWh)</td>
<td>826</td>
<td>10,005</td>
<td>1240</td>
<td>12,236</td>
</tr>
<tr>
<td>Deviation (%)</td>
<td>–6</td>
<td>–3</td>
<td>–20</td>
<td>–4</td>
</tr>
<tr>
<td>Discharging</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Measurement (MWh)</td>
<td>857</td>
<td>6866</td>
<td>4205</td>
<td>11,957</td>
</tr>
<tr>
<td>Simulation (MWh)</td>
<td>725</td>
<td>6534</td>
<td>4160</td>
<td>11,585</td>
</tr>
<tr>
<td>Deviation (%)</td>
<td>–15</td>
<td>–5</td>
<td>–1</td>
<td>–3</td>
</tr>
</tbody>
</table>
3.1.2. The charged and discharged energy

Table 3 shows the annual charged and discharged energy for different operation conditions. The difference between the calculated and the measured total energy was under ±4%, for both charge and discharge. As mentioned before, operating mode changed frequently in spring when the middle pipe was widely used. Therefore, a significant relative error can be seen under the charging condition from the middle pipe (20%). As for the discharging condition from the top pipe widely used on summer nights, the huge difference of 15% was caused by the underestimated energy content during summer. However, the absolute errors for both conditions were less than 320 MWh (2.5% of the total charged or discharged energy).

3.1.3. Heat loss

The flux sensor on the top cover of the PTES was located in the insulation layer, as depicted in Fig. 3. Fig. 9 shows that the calculated heat flux of the cover is consistent with the measured data at the average level. The heat flux from the cover is positively correlated to the water temperature of the top part (Fig. 8 (a)), but the relationship is non-linear. It is also influenced by the ambient air temperature. Moreover, the simulated yearly heat loss of the top cover is 5.6% less than the measured result of 786 MWh, where the measured heat loss from the top cover is calculated by the production of the measured heat flux and the top area at 8100 m². Therefore, predicted heat loss by the model is considered reasonable.

Nevertheless, the constant thermal conductivity of the insulation used in this model does not consider the influences of wind speed, ponding, moisture and material degradation. Additionally, the measurement of a single heat flux sensor is uncertain. Hence, this simulation result cannot match the measured heat flow in terms of fluctuation. The main reason for the fluctuation might be moisture in the insulation, but further studies should be conducted to prove it.

Fig. 10 compares the monthly heat loss from the model with heat loss calculated by energy balance based on the measured charged energy, discharged energy and internal energy changes (Eq. (32)). The variation of heat loss is less evident than that around the sidewall owing to good insulation on the top. The severest heat loss occurred in August when the storage temperature was highest.

Due to cooling down of the PTES by the heat pump, the water temperature in the top of the storage can be lowered to about 20 °C in early March (Fig. 15 (b)). Since the soil temperature around the storage was warmer than the storage, the heat stored in the soil transmitted back to the storage through the side/bottom surfaces, resulting in a heat gain in January - February and in December. These heat gains are shown as negative heat losses in Fig. 10.

The heat loss is presented in Table 4. The yearly heat loss was underestimated by only 2% in 2015, referring to 1275 MWh in the report (Schmidt and Sørensen, 2018). On basis of an average ambient air temperature of 9.5 °C, the calculated heat loss coefficients were listed as well. The total heat loss coefficient was 0.21 W/(m²⋅K). A part of the heat loss was recovered when the heat pump was operating in winter, leading to smaller coefficients of the sidewall and the bottom wall. Therefore, the calculated heat loss coefficients for the bottom and sides of the PTES account for the insulating and heat-storing effects of the surrounding soil.

It should be noted that although the bottom and side surfaces of the PTES were not insulated by insulation material, the soil around the PTES serves as a good thermal insulation. Therefore, the heat loss coefficient of the top lid from water to the ambient air (0.19 W/(m²⋅K)) is slightly larger than that of the side surface (0.16 W/(m²⋅K)). Overall, the accumulated heat loss from the cover accounts for the most remarkable part.

Fig. 9. Heat loss from the cover (time step:10 min).

Fig. 10. Heat loss from the envelope.

Fig. 11. Contribution to the heat loss.

Table 4

<table>
<thead>
<tr>
<th>Items</th>
<th>Top</th>
<th>Side</th>
<th>Bottom</th>
<th>Total</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat loss, MWh</td>
<td>742</td>
<td>476</td>
<td>27</td>
<td>1245</td>
</tr>
<tr>
<td>Surface area, m²</td>
<td>8100</td>
<td>8300</td>
<td>676</td>
<td>17,076</td>
</tr>
<tr>
<td>Average temperature, °C</td>
<td>65.9</td>
<td>50.0</td>
<td>31.4</td>
<td>50.0</td>
</tr>
<tr>
<td>Heat loss coefficient, W/(m²⋅K)</td>
<td>0.19</td>
<td>0.16</td>
<td>0.21</td>
<td>0.21</td>
</tr>
</tbody>
</table>
of 60% (Fig. 11). Heat gain from the sidewall in winter can partly compensate for the considerable heat loss in summer.

### 3.1.4. MIX number

Apart from the overall indicators mentioned above, thermal stratification in the store is also significant for PTES design in pipes arrangement and optimization in control strategies. MIX number, a dimensionless indicator in the range of 0 to 1, can be used to evaluate the stratification quality of PTES. It is defined by the moment of energy $M$ as Eq. (37).

$$MIX = \frac{M_{\text{stratified}} - M_{\text{exp}}}{M_{\text{stratified}} - M_{\text{fully-mixed}}}$$  (36)

$M_{\text{exp}}$ is the energy-momentum in the pit. $M_{\text{stratified}}$ and $M_{\text{fully-mixed}}$ are the energy-momentum of hypothetical conditions with the same energy content as the experimental condition, on behalf of perfectly stratified condition and fully mixed condition, respectively. MIX number equals to 1 means that the water in the pit is fully mixed with a uniform temperature $T_{\text{average}}$, while that equals to 0 means the pit is divided into two separate layers at distinguished temperatures, $T_{\text{hot}}$ and $T_{\text{cold}}$. $T_{\text{hot}}$ and $T_{\text{cold}}$ refer to the maximum and minimum temperature in the storage over the year, which present the potentially best thermal stratification.

In the calculation, the water pit was divided into 20 layers with the same volumes, where the moment of energy $M$ is calculated as follows. As the boundary line between hot water and cold water is likely to be in the middle of one layer, a factor based on the share of hot and cold volume in the middle layer was introduced to correct the separated volume when the stratified moment of energy was calculated.

$$M = \sum_{i=1}^{20} (\rho_i c_p V_i T_i)$$  (37)

Fig. 12 exhibits the analysis of MIX number. It is clear that the largest error appears in the summer when the simulated internal energy content is lower, caused by an overestimated mixing. Fig. 13 shows the temperature profile from mid-June to August. The drop in water temperature adjacent to the cover induced by intensive natural convection (Ochs et al., 2020) is not seen in the simulation because temperature inversion is not considered in the model. There is only one uniform temperature for the 0.8 m top layer, conducting a lower outlet water temperature in the top pipe. In general, this simulation result with milder trends cannot match well in a thin thermocline layer due to limited layers. A similar reason can explain the difference in MIX number in other periods.

Additionally, Fig. 13 reveals the reason for the noticeable deviation of the simulation in summer. During charge of the PTES in summer, water is taken from the bottom of the PTES, heated by the solar collector field before entering into the PTES through the middle and/or the top opening. A higher modeled water temperature at the bottom of the PTES means a smaller simulated charged energy, as shown in Eq. (21), resulting in an underestimated total energy content of the storage. Although it can be partly compensated by a smaller discharged energy due to a lower water temperature at the top, the discharged energy is typically much smaller than the charged energy in summer. Hence, the simulated internal energy content is smaller than the measured one.

Overall, the measured and simulated water temperature matched well throughout the whole year. Nevertheless, in order to precisely simulate the large-scale storage, more layers in the storage should be used, especially for the upper part. Otherwise, the thermal performance will be underestimated by a wide thermocline (Fernández-Seara et al., 2007).

### 3.2. The soil around the PTES

Hourly ground temperatures, including the ambient temperature, insulation temperature and ground temperature are generated by the model in a 2-D array. Due to the simplification of the pyramidal-shaped PTES with a truncated cone, the range of the top surface in the model is different from that in reality. The horizontal distance from the symmetry axis to the edge of the top surface was 4 m further in the simulation. Thus, the simulation result was presented as a ‘belt’, whose upper boundary presents the soil temperature at the horizontal distance of 45.5 m away from the centerline of the storage while the lower boundary presents that of 49.4 m away from the centerline, shown in
Fig. 14 shows hourly soil temperatures at different depths. The initial soil temperature of 2015 is obtained by a one-year pre-simulation. Although Type 343 can be used to predict the range of soil temperature, it overestimates the temperature increase at four depths. From measurements, the soil temperature increased by only 3 °C at 10 m below the ground in 2015. The soil temperature under 15 m below the ground kept stable after July in 2015, while the temperature rises continuously in simulation. This is because uniform soil condition is assumed in the simplified model, but in reality, there is a layer of relatively dry soil in depth from 0 m to 17 m, while in depth below 17 m, the soil is saturated with groundwater (PlanEnergi, 2011; Jensen et al., 2015). The impact of groundwater was not taken into account. Another explanation is that the limited soil region is not big enough where the adiabatic boundary of the model domain affects the temperature distribution. Moreover, increasing temperatures over the year at four depths indicate a dynamic response of the soil temperature. Even after two years of operation, thermal conditions of the ground has not yet been stabilized.

Fig. 15 shows temperature profiles of the PTES and the surrounding soil for four typical days, including the beginning of 2015, the day with coldest PTES, the day with the hottest PTES and the final day of 2015. The soil close to the top surface was significantly influenced by the ambient environment and the PTES. Therefore, the impact area in the superficial layer by the PTES is small. Soil temperature on the vertical direction is highly related to the thermal stratification in the storage. However, temperature growth in the soil boundary can be observed on both directions when the boundary temperature was set to 8.3 °C. The distortion of temperature distribution is resulted from adiabatic boundaries which cannot be avoided due to a small domain size, shown in Fig. 15 (c) and (d). A larger domain size is recommended for further study.

From January to March, the energy content of the storage decreased rapidly when the heat pump was extracting heat from the top and middle diffusers, making it possible for heat gain from the surrounding soil. From March to August, the storage was charged gradually. The top part along the sidewall can be warmed to 70 °C in August, whereas the ground close to the bottom wall was around 50 °C. From August to December, the storage was cooling down gradually. After a one-year operation, the impacted range of soil was widened.
4. Analysis

4.1. Site

Geotechnical conditions and soil properties will influence site selection, design and thermal performance of the PTES. Two key soil properties, namely thermal conductivity and thermal capacity, were investigated in this study (Hamdhan and Clarke, 2010b). Properties of three typical soils are listed in Table 5. The thermal conductivity for dry coarse sand is typically 0.25 W/(m·K) while it is as large as 4.44 W/(m·K) for grey silty gravel. The thermal capacity also varies from 1440 kJ/(m³·K) for dry soil to 3205 kJ/(m³·K) for wet grey sandy gravel. A reference soil was introduced to analyze the impact of these two parameters. Thermal properties of the soil around the investigated PTES is included in Table 5 as soil type 5.

Simulations were carried out to investigate the influence of soil properties on thermal performance of the PTES. The same operation and boundary conditions were used for the six years simulations. Fig. 16 shows monthly heat loss of the PTES in six years for different soil properties. As shown in Fig. 16, dry soil has better insulation, which significantly reduces the heat loss from the PTES over the years. However, during the winter period, the amount of energy gain of the storage from the surround soil is also the lowest for the soil type 1.

Fig. 17 shows annual storage efficiency in terms of different soil types. The thermal performances of the storages are increased year by year due to gradual thermal balance between the PTES and the soil.

There is a large increase of storage efficiency from the first to the second year, while it almost does not change from the fourth year to the sixth year. A thermal balance is reached after 4 years for all cases. Dry soil with the lowest thermal conductivity achieves the highest efficiency at 90%, while the storage efficiency of wet sandy gravel (Soil 3) is only 83%.

Additionally, the effect of thermal conductivity is evident in comparison with the thermal capacity. By comparison of the results of the soil type 3 and the reference soil (Fig. 17), the difference caused by the thermal capacity is almost eliminated in the third year with a difference less than 1%. Then, the thermal quality (highest temperature in the storage) is hardly impacted by the soil conditions, especially during the charging period.

4.2. Inlet arrangement

Typically, three inlet/outlet openings at different heights of the PTES are utilized to enhance flexible operation of the storage. The positions of the inlet/outlet diffusers are designed by rule of thumb in guidelines, because of lacking solid documentation related to the influence of the diffuser positions on PTES’s thermal performance. Referring to the original design of the storage in Dronninglund (scenario 3), four scenarios with different positions of the middle diffuser were investigated in this study. The positions of the top and the bottom diffusers are kept unchanged. The inlet water temperatures and flow rates of the three diffusers were also assumed the same as those used in Section 3.

Only the simulation result of the fourth year was presented in this paper. As listed in Table 6, under the assumed conditions, the yearly discharged energy in the fourth year increases with the increase of the position of the middle diffuser. But the energy taken from the top diffuser decreases correspondingly (Fig. 18) because of the impact on thermal stratification (temperature profile) in the PTES. In comparison with the original design, the heat recovery rate can increase by approximately 1% by this improvement.

<table>
<thead>
<tr>
<th>Scenarios</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
</tr>
</thead>
<tbody>
<tr>
<td>The layer of the middle diffuser (count from the top)</td>
<td>3</td>
<td>5</td>
<td>7</td>
<td>9</td>
</tr>
<tr>
<td>The height of the middle diffuser from the bottom, m</td>
<td>14.0</td>
<td>12.4</td>
<td>10.8</td>
<td>9.2</td>
</tr>
<tr>
<td>Storage efficiency, %</td>
<td>88.7</td>
<td>88.3</td>
<td>87.9</td>
<td>87.5</td>
</tr>
<tr>
<td>Charged energy, MWh</td>
<td>11,327</td>
<td>11,257</td>
<td>11,150</td>
<td>11,032</td>
</tr>
<tr>
<td>Discharged energy, MWh</td>
<td>10,050</td>
<td>9939</td>
<td>9800</td>
<td>9655</td>
</tr>
</tbody>
</table>

![Fig. 16. Thermal losses from the pit during six years.](image1)

![Fig. 17. Annual storage efficiency during six years.](image2)

![Fig. 18. Yearly discharged energy from the diffusers in the third year for the different inlet arrangements.](image3)
Since the heat taken from the middle diffuser is mainly used by the heat pump as low temperature heat source, a higher supply temperature for the heat pump leads to a higher COP and eventually a saving of fuel for the heat pump. However, because the heat extracted from the top diffuser is intended for direct use, a decrease of energy taken from the top opening leads to a result that the auxiliary boiler operated frequently. Thus, a system simulation integrating the PTES with the CHP generator, the solar collector field, the heat pump etc. is necessary in order to optimize the inlet/outlet diffusers.

5. Conclusion

Thermal performance of a large water pit heat storage in Dronninglund, Denmark is investigated experimentally and theoretically. By comparing the experimental results with the simulation results, the validity of a modified Type 343 for PTES is analyzed. The model is used for parametric investigations on the influence of soil properties and diffuser positions. The results show that:

(1) The annual measured charged and discharged energy of the store are 12,787 MWh and 11,957 MWh respectively. Compared to the measured data, the model predicts the charged and discharged energy with a deviation of ~4% and ~3% respectively. The accuracy of the model is acceptable, however, the limitation of the layer number causes errors in predicting the thermal stratification, especially for a PTES as large as 60,000 m³. The analysis of MIX number shows that the model underestimated thermal stratification in summer. More layers in the storage domain is necessary in order to improve accuracy of the model.

(2) For the PTES without insulation around the sidewall and bottom, the primary heat loss is from the cover (60%). Although heat loss from the sidewall is considerable in summer, heat gain in the winter can partly make up the loss, resulting in 38% of the total heat loss of the store.

(3) Thermal condition of the shallow soil around the PTES is significantly influenced by both the storage and the ambient temperatures. The soil adjacent to the storage is mainly affected by water temperatures in the storage. A higher temperature gradient is observed in the soil at a depth of 5 m due to large heat gain from the store and large heat loss to the ambient air.

(4) It takes four years for the soil to reach a thermal balance under the investigations conditions. The PTES surrounded by dry soil has an annual storage efficiency of 90%, which is 7% higher than PTES surrounded by wet soil.

(5) For the storage with three inlet/outlet diffusers, it is advisable to move the middle diffuser higher. Extra 250 MWh can be discharged from the storage when the middle diffuser is 3.2 m higher than the actual height. But further analysis on system level should be carried out to confirm this finding.

(6) Several restrictions of the model have significant influences on the prediction accuracy: limitation of layer number (maximum 20), numerical diffusion caused by large/variable layer volume, non-specific initial temperatures to each layer, overestimation of soil temperature due to small domain extent and the inability to take into account multiple soil types and eventually groundwater flow. Further improvement of the model is necessary in order to increase the precision of the model.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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References
