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Numerical and experimental analysis of instability in high temperature packed-bed rock thermal energy storage systems

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HIGHLIGHTS

- Experiments were conducted on an air-based packed-bed thermal energy storage system.
- CFD models were developed and validated for both static and dynamic operation.
- Instability in packed-bed systems is predicted during the discharging process.
- Increase in instability is observed with the increase in standby duration.
- Comparison of reduced-scale and full-scale packed-bed systems is presented.

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ABSTRACT

Due to heat losses of preferential areas of packed-bed energy storage systems, transverse temperature variations may occur during the charging, discharging and standby processes. Furthermore, the heat losses of preferential areas of the storage tank cause a lower pressure drop in these areas resulting in an increased mass flow rate and further cooling, and thereby an enhanced transverse temperature variation, in a positive feedback loop—a phenomenon called instability. The transverse temperature variations may deteriorate the performance and thereby the economic feasibility of packed-bed energy storage systems. In this paper, numerical and experimental investigations of an air-based packed-bed rock thermal energy storage system for large-scale high temperature applications are presented. The objective of the study is to predict the instability and to analyze the effect of different standby durations and storage size on the instability of the air-based packed-bed system. Transient axisymmetric computational fluid dynamics models were developed for the standby and discharging processes of the packed-bed thermal energy storage systems. In addition, experimental investigations were carried out at a test facility located at Stiesdal Storage, Denmark, using magnetite rocks as heat storage material and air as heat transfer fluid. The results suggest that the numerical predictions are in good agreement with the test data. The instability phenomenon is found to increase with the standby duration, resulting in a maximum difference of 161 K between the maximum rock temperature and the outlet air temperature for a standby period of 10 h followed by a discharging process. Moreover, the results indicate that the maximum difference between the rock temperature and outlet temperature is 73 K and 56 K for a reduced-scale and a full-scale system (no standby period), respectively.

1. Introduction

The energy demand of the world is continuously rising because of the growing economies, increasing population and rapid urbanization. According to the International Energy Outlook 2021 published by the Energy Information Administration (EIA), the rise in global energy consumption between 2020 and 2050 is expected to be nearly 50% [1]. Currently, the share of fossil fuels to meet the world energy demand is about 80% [2]. The share of renewable energy sources needs to be increased to alleviate global warming and the depletion of fossil fuels.
[3]. However, the most promising renewable sources of energy such as solar and wind are intermittent in nature. In order to increase the availability of renewable energy in the power grid and maintain the controllable supply of the renewable energy, it is crucial to incorporate energy storage systems in the energy system.

The more mature and commercially available energy storage technologies such as pumped-hydroelectric storage, compressed air energy storage, and flow batteries have major drawbacks, including being geographically constrained (pumped-hydroelectric storage and compressed air energy storage), causing detrimental environmental impacts (pumped hydro-electricity storage and compressed air energy storage), and having high costs and short lifetimes (flow batteries) [4]. A promising novel alternative solution that resolves these issues is the pumped thermal energy storage (PTES) system. In the PTES technology, excess electricity is stored in the form of thermal energy, which can then be converted back into electricity upon demand. Among different ways of storing thermal energy, sensible energy storage is the more mature and widely used technology [5–9]. Sensible energy storage can be accomplished using a two-tank system (for example, molten salt as heat transfer fluid) or a single-tank packed-bed storage system [9,10]. Two-tank systems are widely used for thermal energy storage in concentrated solar power plant systems, consisting of two separate tanks for high temperature and low temperature molten salt [5]. However, for the scale considered in this work, the conventional two-tank molten salt based thermal energy storage system would be more costly than a single-tank packed-bed storage due to the requirements of an additional tank, antifreeze system due to the high freezing point of the molten salt, and an additional air to molten salt heat exchanger. In a single-tank packed-bed storage, the solid heat storage material is loosely filled in an insulated tank making it a porous structure. During the charging and discharging processes, the heat transfer fluid is circulated through the porous packed-bed. The heat is transferred because of direct contact between the heat transfer fluid and the heat storage material. Both the hot and cold zones are included in a single tank separated by a “thermocline zone”, being the transition zone between the hot and cold zones characterized by a thermal gradient.

Numerous numerical and experimental investigations of single-tank packed-bed thermal energy storage systems are available in the open literature. Air was reported as an advantageous heat transfer fluid compared to molten salt, steam/water, and thermal oil for industrial applications. This is because of various economical [11] and technical advantages of air, such as a capacity to be used for a large temperature range without any upper temperature limit, free availability, environmental friendliness, chemical stability, and no degradation with time [12]. The air-based single-tank packed-bed system is being used in a variety of different industrial applications such as air purification systems in industries [13,14], air preheating in glass and steel industries [13,14], gas-cooled nuclear reactors [15], catalytic reactors [16], and drying processes [17]. The reported studies on air-based packed-bed systems can be divided into large-scale energy storage setups (> 1 MWth) and small-scale (including laboratory-scale) energy storage systems. At present, air-based single-tank packed-bed systems are in the development phase for medium to large-scale energy storage applications [13].

The studies available in the open literature on air-based packed-bed thermal energy storage systems for large-scale applications are summarized next. Zunft et al. [18] performed experiments on an air-based packed-bed thermal energy storage system, which was intended to be integrated in the Jülich solar power tower. The energy storage system had a modular setup with 9 MWth of storage capacity and an air temperature between 393 K and 953 K. The authors developed a one-dimensional two-phase numerical model that agreed well with the test results. Zanganeh et al. [19] carried out an experimental investigation of a truncated conical-shaped packed-bed system with air as heat transfer fluid. The packed-bed of 6.5 MWth capacity and the maximum temperature of 923 K, was submerged into the ground for taking care of the thermo-mechanical stress generation on the bed surface. The authors developed a one-dimensional two-phase numerical model that was used to design a full-scale energy storage system for a 26 MWth round-the-clock CSP plant located in Ait-Baha, Morocco. Zavattoni et al. [20,21] carried out two-dimensional and three-dimensional computational fluid dynamics (CFD) simulations of the packed-bed thermal energy storage test facility reported in [19]. The consideration of porosity distribution and effective thermal conductivity was reported to improve the model accuracy [20]. The results indicated that radiation heat transfer at high temperatures, and based on this finding the authors suggested to consider radiation heat transfer also at low temperature in numerical models for packed-beds [21]. An experimental and numerical analysis of a 12 MWth air-based packed-bed system used in adiabatic compressed energy storage systems was presented by Geissbühler et al. [22]. Tests were performed for charging and discharging processes with the temperature ranging from ambient temperature to 823 K and a gauge pressure ranging from 0 Pa to 7 MPa. A temperature difference between numerical and experimental results up to 348 K was observed. Esence et al. [13] carried out experimental and one-dimensional numerical analysis for a structured packed-bed and granular packed-bed of 1.1 MWth and 1.4 MWth capacity, respectively, for repeated charging and discharging cycles. Ceramic plates were used to form corrugated channels in the structured packed-bed, while ballistic rocks were used in the granular packed-bed system. A mixture of air and exhaust fumes was used as heat transfer fluid in both the systems. The results suggested that the pressure drop is up to 54% higher in the granular packed-bed than in the structured packed-bed, due to channeling effects and flow heterogeneity.

In addition, various small-scale studies were reported in the literature for air-based packed-bed rock thermal energy storage systems. Anderson et al. [23] conducted an experimental and numerical analysis of air-based packed-bed systems. The authors developed a one-dimensional two-phase model and reported that the heat transfer loss coefficient should be considered in the model for accurate estimation of the storage efficiency. A comparison between experimental and CFD results of a 1.8 m height and 0.58 m diameter packed-bed system with air as heat transfer fluid was reported by Cascceta et al. [24]. The test setup was implemented at the University of Caligari, Italy. The results of the CFD model were found to be in good agreement with the test data when the temperature dependent thermal properties, inlet conditions of the heat transfer fluid, effective thermal conductivity and porosity variation in the radial direction were accounted for in the model. Lai et al. [25] carried out an experimental analysis of air-based packed-bed systems to investigate the effects of charging temperature, particle size, mass flow rate and repeated charging/discharging cycle on the performance of packed-bed systems. Sintered iron ore particles (raw material of a blast furnace) were used as the heat storage medium, while air was used as the heat transfer fluid with a maximum temperature of 573 K. Sintered iron ore particles showed good thermal performance and achieved a maximum cycle efficiency (the ratio of the energy released to the sum of the energy stored and the pump work in a charging/discharging cycle) of 66.8%. A compilation of further studies and modeling approaches of packed-bed thermal energy storage systems may be found in the following review papers: Esence et al. [9], Antwerpen et al. [26], and Calderón-Vásquez [27].

It needs to be stressed that the aforementioned studies on large-scale and small-scale thermal energy storage systems consider only the dynamic processes (charging and discharging processes) when operating a thermal energy storage system, excess electricity is converted to heat and stored in the rocks (charging process) and recovered (discharging process) at a later point in time when there is a demand for the electricity. In practice, there may be a delay after the storage is charged until there is a need for the electricity. Therefore, the operation of thermal energy storage systems often includes a standby process, for which there is no flow of air in/out of the packed-bed and the stored thermal energy is not used. Standby process can follow either charging process or
discharging process. The time duration of the standby process can vary from a few minutes/hours to a number of days depending on the application of the thermal energy storage system. The standby process is associated with thermal degradation of the packed-bed system, hence affecting the economic feasibility of the entire thermal energy storage system. Therefore, it is important to analyze the thermal performance of the packed-bed thermal energy storage systems during such a process.

However, very few studies considering the standby process are reported in the open literature, especially for high temperature applications with air as a heat transfer medium. Most of the available studies address low temperature applications, such as space heating and solar water heating [28]. Okello et al. [29] carried out an experimental analysis of a packed-bed thermal energy storage with air as heat transfer fluid. Their results suggested that the temperature at the top of the packed-bed system decreases with time during the standby period, and the rate of decrease in temperature increases with the average temperature of the packed-bed. Mertens et al. [30] carried out transient one-dimensional simulations for quartzite rock packed-bed systems with air as heat transfer fluid. The authors also considered a standby period in their model; however, they did not provide a validation of the standby period. Yang et al. [31] developed and validated a one-dimensional two-phase transient model to investigate charging-standby-discharging and charging-standby processes of an air-based packed-bed thermal energy storage for high temperature applications. The authors reported that a 30% decay in thermal stratification (the ratio of the mean temperature gradient to the maximum mean temperature gradient along the axial distance at a given time duration) was reported after a 6 h standby period. In addition, heat loss and thermal diffusion were found to be the prominent factors for thermocline degradation.

Another concern that deteriorates the performance and thereby the economic feasibility of packed-bed energy storage systems is transverse temperature variations that may occur during the charging, discharging and standby processes. Cooling due to heat losses of preferential areas of the storage tank causes a lower pressure drop in those areas resulting in a decreased flow rate and further cooling, and thereby an enhanced transverse temperature variation, in a positive feedback loop – a phenomenon called instability [32]. A very limited number of studies that address the instability phenomenon in single-tank packed-bed systems are available in the open literature, and all but two of them [32,38] considered a liquid as the heat transfer fluid [33–37]. The physics involved in the stability of packed-beds differ for systems using a liquid or a gas as the heat transfer fluid. Liquids are generally considered to be incompressible, and their viscosity decreases with an increase in temperature. Also, the pressure drop that needs to be overcome to push the liquid through the packed-bed decreases with the increase in temperature due to a reduced viscosity of the liquid. However, for a gas, the pressure drop increases with an increase in temperature because of an increase in viscosity with an increase in temperature of the gas [32].

Davenne et al. [32] and Zavattoni et al. [38] claimed that the radial temperature variations causing the instability may be due to non-uniformities in void fraction or non-uniformities in the flow of the heat transfer fluid at the inlet. Zavattoni et al. [38] developed a two-dimensional CFD model taking into account the radial variation in porosity and validated it with the experimental results for the charging and discharging processes. It has been reported that the temperature variation in the radial direction is not significant if the tank to particle diameter ratio is >25–30 [38–40]. Furthermore, Davenne et al. [32] developed a one-dimensional model to analyze the stability of a packed-bed system during the charging and discharging processes. The authors introduced a small perturbation in their model and found the minimum perturbation size that causes an instability, the critical perturbation size.

In conclusion, results of previous works for small-scale packed-bed systems indicate that the temperature variations in the radial direction causing the instability are due to a porosity variation in the radial direction and imperfect inlet flow [32,38]. However, for large-scale packed-bed systems with a tank to particle diameter ratio >30 and large inlet manifolds, previous works suggest that the instability may be due to a porosity variation in the radial direction and imperfect inlet flow are not the main causes for the radial temperature variation/instability [38–40]. The main cause of instability in the case of large-scale packed-bed systems is the heat loss from the walls of the packed-bed tank. In addition, it has been observed that phenomena of thermal instabilities in packed-bed systems are generally less prone to arise during charging processes than during discharging processes [32]. This is because the pressure drop increases in the heated areas with the progress of the charging process, leading to lower mass flux in those areas. On the contrary, the mass flux increases in the preferential colder areas due to a lower pressure drop in these regions, maintaining the temperature uniformity in the transverse direction and making the packed-bed more stable during the charging process [32]. During the discharging process, the lower pressure drop in the preferential colder areas results in an increased mass flux and further cooling in those areas, and thereby makes the packed-bed less stable.

This paper presents numerical and experimental investigations of an air-based packed-bed rock thermal energy storage system for large-scale high temperature applications. The objective of the study is to predict the instability and to analyze the effect of different standby durations and storage sizes on the instability of an air-based packed-bed system. Since the charging process generally is more stable than the standby and discharging processes, as reported in the literature [32], and observed in our experimental results, the present study is focused on the standby and discharging processes. Transient axisymmetric CFD models were developed for the standby process and discharging process of a packed-bed thermal energy storage system. In addition, experimental investigations were carried out at a test facility located at Stiesdal Storage, Denmark, using magnetite rocks as the heat storage material and air as the heat transfer fluid. The numerical CFD models were validated based on the test results.

The main novel contributions of our work to state of the art are as follows:

- Development and validation using experimental data of two-dimensional two-phase transient CFD models for large-scale high temperature air-based packed-bed systems, considering the following aspects: i) Effective thermal conductivity for the solid (rock) and fluid (air) involving conduction, contact conduction, radiation, and mixing in axial and radial directions within the tank, ii) Heat transfer through convection within the tank, iii) Heat loss to the ambient, iv) Temperature dependent thermal properties of the solid (rocks) and fluid (air), and v) Real gas effects of air. No previous work reported such an extensive CFD model incorporating all these parameters for air-based packed-bed systems.

- Prediction of the instability during the discharging process for air-based packed systems. Only two previous studies [32,38] analyzed the instability in air-based packed-bed systems. While Davenne et al. [32] induced the instability in the packed-bed system by a small perturbation, we predict the instability arising due to the heat losses in the preferential areas of the storage tank. Moreover, while the analyses by Davenne et al. [32] are purely numerical, we predict the instability in the numerical model as well as observe the instability in the experimental work. Zavattoni et al. [38] analyzed a small-scale air-based packed-bed system with a tank to particle diameter ratio of 12.5, while we analyze a large-scale air-based packed-bed system with a tank to particle diameter ratio of 100.

- Analyses of the effect of different standby durations and storage sizes on the instability of air-based packed-bed systems. No previous work reported the effect of standby durations nor storage sizes on the instability of such storage systems.
Fig. 1. Test facility at Stiesdal Storage, Denmark.

(a) Inner and outer cylindrical tanks
(b) Side view of outer cylindrical tank
(c) Top view of packed-bed tank filled with magnetite rocks
(d) Heater assembly
(e) Packed-bed thermal energy storage system
and experimental data reported in this paper provide a solid basis for further investigations of packed-beds for different applications, benefiting both academia and industry. Overall, the paper contributes to the realization of cost-efficient large-scale storage technologies, thereby enabling the increased availability of renewable energy sources in the power grid while matching the supply and demand of energy.

Section 2 presents a description of the experimental setup and the different processes involved in the storage system. Section 3 describes the numerical modeling and methods including the governing and constitutional equations. The results and discussion are presented in section 4, while the conclusions are summarized in section 5.

2. Experimental work

2.1. Experimental setup

The experimental investigations were carried out at a test facility installed at Stiesdal Storage, Denmark. Fig. 1 shows pictures, while Fig. 2 presents a schematic diagram of the test facility. The test facility consists of an inner cylindrical steel tank (liner) with a height of 5.32 m, internal diameter of 1 m, and wall thickness of 4.5 mm. The liner is insulated from the outside with rock wool material of a thickness of 0.13 m and a thermal conductivity of 0.1 W/(m·K) to minimize the heat loss to the outside. Magnetite rocks with imperfect spherical shapes and varying diameter between 8 mm to 20 mm are used as heat storage materials. The insulated liner is filled with magnetite rocks resulting in a packed-bed with a porosity of 0.36. The insulated liner tank is covered from the outside by another cylindrical tank incorporated with feeder pipes at the top and bottom of the tank. A blower (Soby, TLR model, 15 kW power) is used to get the heat transfer fluid (air) to pass through the packed-bed. In order to distribute the air uniformly in the packed-bed, flow distributors are employed at the inlet and outlet. An orifice flow meter included with differential pressure sensors installed on the suction pipe of the blower is used to measure the mass flow rate of air at the inlet during the charging and discharging processes. The mass flow rate of air is varied by varying the speed of the blower. An electric heater (10 elements of 100 kW each) is used to increase the temperature of air to the required value during the charging process. The total desired max output of the electric heater is imposed in the Programmable Logic Controller (PLC). Three-way valves (Lindab, series WH45U) are installed at different locations for flexible operation of charging and discharging processes. Similar three-way valves from Lindab were also used in the burner and flue gas applications.

The test setup was run repetitively with sequences of charging and discharging cycles. During charging, hot air enters from the top of the packed-bed through the feeder pipe, transfers heat to the rocks and then leaves from the bottom of the packed-bed at a lower temperature. During discharging, air at ambient temperature enters from the bottom of the packed-bed through the feeder pipe and is heated as it passes through the packed-bed, reducing the temperature of the packed-bed. The concept of storing and recovering heat directly in the packed-bed rock thermal energy storage system without using a heat exchanger will be used by Stiesdal Storage in their GridScale energy storage pilot plant with a charge/discharge capacity of 4 MWₑ/2 MWₑ and a storage capacity of 10 MWhₑ [41].

2.2. Measurements and uncertainty

The test facility is equipped with K-type thermocouples (405–010, TC-direct, Germany), having an outlet diameter of 1.5 mm and a 1.0 m long steel insulation to record the transient temperature at different locations of the packed-bed system; see Fig. 3. The measurement range of the thermocouples is from 233 K to 1373 K with a maximum uncertainty of ±1.5 K. Eight thermocouples (see Fig. 3) are located along the central vertical axis inside the packed-bed at heights of 0.31 m, 0.9 m, 1.4 m, 1.9 m, 2.4 m, 2.9 m, 3.4 m, and 3.9 m from the bottom of the packed-bed. Two thermocouples are placed inside the top and bottom feeder pipes to measure the inlet and outlet temperatures of the air. Thermocouples are also located on the inner and the outer surface of the insulation, and on the outer surface of the outer cylinder, all at a height of 0.31 m, 0.9 m, 1.4 m, 1.9 m, 2.4 m, 2.9 m, 3.4 m, and 3.9 m from the bottom of the packed-bed.
of 2.4 m from the bottom surface of the packed-bed. In addition, one thermocouple measures the ambient temperature. The temperature readings of the insulation surface, tank surface and ambient were used to obtain the heat loss from the packed-bed during the experiments.

Moreover, differential pressure sensors (DPS300, BDSENSORS GmbH, Germany) are placed at heights of 0.31 m, 2.175 m, and 5.120 m from the bottom of the packed-bed to measure the pressure drop along the height of the packed-bed as shown in Fig. 3. Differential pressure sensors were also used to obtain the mass flow rate of air by measuring the pressure drop across an orifice in a 100 mm pipe at the suction side of the blower, following ISO 5167-1:2003. The maximum uncertainty of the differential pressure sensors is ±1%. All the instruments were pre-calibrated within the working range of the measured parameters (temperature and pressure) before the experiments started.

3. Numerical modeling

3.1. Numerical approach

In this study, transient two-phase CFD models using the commercial software Ansys Fluent 2022 R1 [42] were developed to analyze the heat transfer performance of the packed-bed system during both the standby and discharging processes. Davenne et al. [32] showed in their three-dimensional CFD analysis that the instability arising due to non-uniform void or cold tunnel at the walls/axis grows symmetrically with time along the axis. Hence, we assumed that the instability arising due to similar losses from the tank wall will be symmetric along the axis. Therefore, to save computational time, we considered a two-dimensional axisymmetric numerical model with uniform porosity and uniform inlet flow conditions, thereby ignoring minor instabilities that may arise due to non-uniform porosity and non-uniform inlet flow conditions.

Fig. 4 shows the computational domain of the air-based packed-bed system considered in the analysis, consisting of a two-dimensional axisymmetric cylindrical tank with a height of 5.32 m and an internal diameter of 1.00 m. During the discharging process, the mass flow rate at the inlet and the pressure at the outlet were applied as boundary conditions at the inlet and outlet, respectively. A convective heat loss was considered at the walls, while the heat loss from the top and the bottom of the packed-bed was neglected during the discharging process. As for the standby process, the top and bottom of the packed-bed were considered as the walls during the simulations because of the no mass flow of air in/out of the system, while a convective heat loss was considered at the outer walls, and the top and bottom of the packed-bed system, as can be seen in Fig. 4. An axis boundary type was considered at the centerline of the computational domain for all the simulations.

The continuity, momentum, and energy equations were numerically solved with a finite volume method. The $k-\varepsilon$ model with standard wall function was employed to account for turbulence effects. The SIMPLE method [43] was adopted for the pressure-velocity coupling, and second-order schemes were used for pressure corrections and for discretizing the momentum and energy equations. The convergence criteria for the continuity and momentum, and the energy equations are absolute residuals of $10^{-4}$ and $10^{-6}$, respectively. Because of different thermal conductivities and heat capacities of the air and rocks, the local rate of change of temperature of both the media vary significantly. Therefore, consideration of the local temperature of the rocks and air is necessary in order to evaluate the heat transfer performance. In this context, the local thermal non-equilibrium (LTNE) approach was used to model the heat transfer in the packed-bed system, which uses separate energy equations for the heat storage materials and the heat transfer.
fluid. User-defined functions were used to incorporate the heat transfer coefficient, effective thermal conductivity of the rocks and air, mass flow rate at the inlet, and the pressure losses in the porous bed. A custom-field function was used to provide the initial conditions during standby and discharging processes.

3.2. Governing equations

The packed-bed system was treated as a continuous porous medium. The numerical model is governed by continuity, momentum, and energy equations with the following assumptions [24, 31]:

- Axisymmetric computational domain
- Continuous, homogeneous, and isotropic porous medium inside the packed-bed
- Spherical shape and equal average diameter of the rocks (10 mm) inside the packed-bed
- Plug flow assumption for air at the inlet of the packed-bed

With the aforementioned assumptions, the governing equations for the two-dimensional transient axisymmetric model are as described below.

**Continuity equation:**

\[
\frac{\partial (\rho \bar{V})}{\partial t} + \nabla \cdot (\rho \bar{V}) = 0
\]  

(1)

where \( \bar{V} \) is the superficial velocity vector, \( \epsilon \) is the porosity of the packed-bed, which is defined as the ratio of void volume to the total volume of the packed-bed tank, and \( \rho \) is the density of the heat transfer fluid.

**Momentum equation:**

\[
\rho \left( \frac{\partial (\bar{V})}{\partial t} + \bar{V} \cdot \nabla \bar{V} \right) = -\nabla p + \rho g + S_i
\]

(2)

where \( S_i \) is the source term and is given by

\[
S_i = -\epsilon \left( \frac{C_1}{C_1 + \frac{1}{2} C_2 |\bar{V}|^2} \right)
\]

(3)

By considering the source term in the momentum equation, the viscous and inertial momentum dissipations in the packed-bed were taken into account. The source term (Eq. 3) is comprised of two terms, the viscous loss term or Darcy term (the first term on the right-hand side of Eq. 3) and the inertial loss term (the second term on the right-hand side of Eq. 3). Here, \( p \) is the pressure, and \( \mu_f \) is the dynamic viscosity of the fluid.

\( C_1 \) and \( C_2 \) are permeability and inertial loss coefficients. The viscous loss coefficient is the inverse of the permeability coefficient (\( C_1 \)). According to Ergun [44], \( C_1 \) and \( C_2 \) are given by

\[
C_1 = \frac{\epsilon d_i^2}{150(1 - \epsilon^2)}
\]

(4)

\[
C_2 = \frac{3.5 (1 - \epsilon)}{\epsilon d_i}
\]

(5)

where \( d_i \) is the diameter of the rocks.

**Energy equation:**

For the fluid phase:

\[
\frac{\partial \left( \rho c_p T_i \right)}{\partial t} + \nabla \cdot (\rho \bar{V} T_i) = \nabla \cdot (k_{eff} \nabla T_i) - h_a(T_i - T_s) + h_s a_s(T_a - T_i)
\]

(6)

For the solid phase:

\[
(1 - \epsilon) \frac{\partial \left( \rho c_p T_s \right)}{\partial t} = \nabla \cdot (k_{eff} \nabla T_s) - h_a(T_s - T_i)
\]

(7)

where \( k_{eff} \) is the effective thermal conductivity of the fluid, \( h \) is the interstitial convective heat transfer coefficient between the rocks and air, \( h_a \) is the heat transfer loss coefficient on the packed-bed tank wall surface, and \( a_s \) is the superficial area of heat storage materials (rocks) per unit bed volume, and is given by

\[
a_s = \frac{6(1 - \epsilon)}{d_i}
\]

(8)

During the standby process, there is no mass transfer in/out of the packed-bed; however, there is an internal mass transfer (circulation in the system) due to a density variation of the air inside the packed-bed system.

3.3. Constitutive equations

3.3.1. Fluid to solid heat transfer coefficients

The convective heat transfer coefficient (\( h \)) in the energy equations for fluid (Eq. 6) and solid phases (Eq. 7) is the interstitial heat transfer coefficient between the air and rock material. Various Nusselt number (\( Nu \)) correlations are available in literature to calculate the fluid to solid heat transfer coefficient in packed-bed systems [9]. In the present study, the Wakao and Kaguei correlation [45] was used to calculate the interstitial convective heat transfer coefficient (\( h \)) during the discharging process:

\[
Nu = 2.0 + 1.1 R e^{0.3} P r^{1/3}
\]

(9)

where

\[
Nu = \frac{h d_i}{k_f}
\]

(10)

\[
Re = \frac{\rho |\bar{V}| d_i}{\mu_f}
\]

(11)

\( Nu, Re, \) and \( Pr \) are the Nusselt number, Reynold number, and Prandtl number, respectively. The main difference between the charging and discharging processes and the standby process is the interstitial convective heat transfer coefficient between the rocks and air, which is natural for the standby process and forced for the charging and discharging processes. A very limited number of Nusselt correlations are available in the literature for the standby process for obtaining the convective heat transfer coefficient between the storage material and heat transfer fluid [31]; however, a previous work [46] suggests that convective flows are lower when the higher temperature region is at the top of the packed-bed, because the high temperature air will remain at the top due to lower density at high temperature. Nonetheless, the losses from the walls of the packed-bed tank may give rise to convective flows in the tank. Therefore, it is reasonable to assume that the Nusselt number for the heat transfer between the rocks and air is unity for the natural convective heat transfer. Similar conditions are encountered for the standby process in the present analysis, and therefore this assumption was made.

3.3.2. Thermo-physical properties of air and rock

In the present numerical model, air was considered as a real gas with temperature dependent properties. The Redlich-Kwong model [47] was used to predict the density of air in the model. The temperature dependent thermal conductivity of the fluid (\( k_f \)), specific heat of the fluid (\( c_p f \)), and dynamic viscosity of the fluid (\( \mu_f \)) were fitted to experimental data [31, 48] and are given as follows:

\[
k_f = 2.457 e^{-3} + 7.638 e^{-5} T_f - 2.096 e^{-8} T_f^2
\]

(12)
The thermophysical properties of the magnetite rocks such as density and specific heat capacity were measured experimentally. The average density of the magnetite rocks filled in the packed-bed was found to be 4700 kg/m$^3$. A differential scanning calorimeter (DSC) was used to measure the specific heat capacity of the rocks. The results obtained from DSC measurements were fitted in a polynomial equation and are shown in Fig. 5. Furthermore, the thermal conductivity of magnetite rocks ($k_s$) was taken from Grosu et al. [49] and fitted in a polynomial equation; see Fig. 6.

\[
c_p = 1002 + 4.754e^{-7} T_f + 4.133e^{-3} T_f^2 - 2.993e^{-7} T_f^3
\]  
\[c_p = -3.812 + 3.313 T_s - 4.44e^{-3} T_s^2 + 2.312e^{-6} T_s^3
\]  
\[
\mu_f = 1.748e^{-5} + 4.417e^{-13} T_f - 1.298e^{-16} T_f^2
\]  

3.3.3. Overall effective thermal conductivity

The overall effective thermal conductivity of the packed-bed is comprised of the sum of three contributions, namely, the effective thermal conductivity of the packed-bed with stagnant fluid ($k_{eff}^{CR}$), the

\[
k_s = 2.044 + 3.599e^{-2} T_s - 9.902e^{-5} T_s^2 + 8.919e^{-8} T_s^3 - 2.5895e^{-11} T_s^4
\]  

\[
R^2 = 0.996
\]
heat transfer of the fluid mixing and turbulence ($k_{mix}$), and the motion of the heat storage materials (rocks). The contribution of motion of the heat storage materials to the overall effective thermal conductivity was neglected in the present study. The effective thermal conductivity of the packed-bed with stagnant fluid ($k_{eff}^{0}$) is divided into four major modes of heat transfer, including conduction in the rocks and conduction in the air ($k_{eff}^{a}$), conduction through contact surfaces of neighboring rocks ($k_{eff}^{c}$), and radiation heat transfer between the rocks ($k_{eff}^{r}$). In addition, mixing of the heat transfer fluid in the axial and radial directions contributes to the overall effective thermal conductivity of the packed-bed system. Each of the contributions to the overall effective thermal conductivity is evaluated as described below.

(i) Conduction in rocks and air ($k_{eff}^{0}$)

The correlation by Zehner and Schlünder [50] was used to calculate the effective thermal conductivity with radiation heat transfer:

$$k_{eff}^{0} = \left(1 - \sqrt{1 - \varepsilon} + \sqrt{1 - \varepsilon} \frac{B + 1}{\pi - 1} \frac{1}{B} \frac{1}{1 + 1/\left(\frac{2}{\varepsilon} - 1\right)} \right)^{\frac{4}{3}}$$

where \(A = \frac{k_{a} \cdot \psi}{4 \pi \nu L}\), \(\psi = 5.67 \times 10^{-8} \text{W/(m}^2\text{K}^4)\) and the value of B is the same as described in Zehner and Schlünder [50]. \(\varepsilon\) is the emissivity of rock material, and it is considered to be one in the current analysis, because rocks at high temperature have a high emissivity (~0.98) [53,54].

The effective thermal conductivity of the heat storage material ($k_{eff}^{s}$) (rocks) was obtained by summing the thermal conductivity due to contact conduction ($k_{eff}^{c}$), radiation ($k_{eff}^{r}$), and the thermal conductivity due to conduction by rocks ($k_{eff}^{s}$):

$$k_{eff}^{s} = k_{eff}^{0} + k_{eff}^{c} + k_{eff}^{r}$$

Similarly, the effective thermal conductivity of the heat transfer fluid ($k_{eff}^{f}$) (air) was obtained by considering the contribution of mixing in axial ($k_{mix}^{a}$), radial ($k_{mix}^{r}$), and axial ($k_{mix}^{ax}$) directions:

$$k_{eff}^{f} = k_{eff}^{0} + k_{mix}^{a} + k_{mix}^{r} + k_{mix}^{ax}$$

The contribution of mixing in axial, $k_{mix}^{a}$, and radial, $k_{mix}^{r}$, directions depends on the Reynolds number (Re) and Prandtl number (Pr) and are given as follows:

$$k_{mix}^{a} = 2 \text{Re Pr}$$

$$k_{mix}^{r} = \frac{1}{10} \text{Re Pr}$$

The contribution from each mechanism to the effective thermal conductivity is shown in Fig. 7(a). It may be observed that the conduction heat transfer by rock and radiation heat transfer (for temperatures exceeding 600 K) are the major contributors to the effective thermal conductivity. In addition, it may be noted that the stress evolved in the bottom is greater than that in the top portion of the packed-bed due to the compressive load, increasing the contact conductance among the particles at the bottom of the packed-bed. An estimation using Eq. 19 suggests that the mean absolute variation in the effective thermal conductivity due to different compressive loads (0 MPa at the top, and 1.5 MPa at the bottom of packed-bed) is lower than 6%. The main reason for the small variation is that the share of contact conduction in the effective thermal conductivity is less than the share of conduction heat transfer by the rocks/air, and radiation heat transfer, see Fig. 7(a). Therefore, it is reasonable to neglect the influence of pressure on the effective thermal conductivity. We have used an effective thermal conductivity corresponding to an external pressure of 1.0 MPa in the analyses.

Furthermore, it can be noted that the effect of mixing of air in radial and axial directions in the packed-bed provides a negligible contribution to the overall effective thermal conductivity. Fig. 7(b) depicts the actual and effective thermal conductivity of the rocks and air in the packed-bed system. The actual thermal conductivity represents the thermal conductivity of the material/fluid such as magnetite rock and air, while the effective thermal conductivity represents the thermal conductivity of the packed-bed system considering all the heat transfer mechanisms, i.e. conduction, convection and radiation. Both the rocks and air contribute to the effective thermal conductivity of the packed-bed system. It can be observed from the figure that there is a large difference between the actual and the effective thermal conductivity of the rocks and a small difference between the actual and the effective thermal conductivity of the air. This happens because of the much higher actual thermal conductivity of the rocks compared to that of the air. The effective thermal conductivity is evaluated as described below.

\begin{equation}
\begin{aligned}
k_{eff}^{s} &= \frac{k_{s}}{0.531 \left[ 3 \left( 1 - \sigma_{s}^{2} \right) \frac{S_{p}}{N_{c}} \right]}
\end{aligned}
\end{equation}

where $S_{p} = 1$ for simple cubic type packing of rocks and porosity ranging from 0.35 to 0.5, and $N_{c}$ is the number of rock particles per unit length and is given by $N_{c} = \frac{1}{L}$. The value of the Poisson’s ratio ($\sigma_{s}$) is 0.37, Young modulus ($E_{s}$) is 175 GPa, and $p_{s}$ is the external pressure on the rocks.

(ii) Conduction through contact surfaces of neighboring rocks ($K_{eff}^{c}$)

A model based on the Hertzian elastic deformation presented by Kaviany [51] was used to calculate the effective thermal conductivity due to contact conduction:

$$k_{eff}^{c} = k_{s} \left[ \frac{3 p_{s} \left( 1 - \sigma_{s}^{2} \right)}{4 E_{s}} \right] \frac{S_{p}}{N_{c}}$$

where $S_{p} = 1$ for simple cubic type packing of rocks and porosity ranging from 0.35 to 0.5, and $N_{c}$ is the number of rock particles per unit length and is given by $N_{c} = \frac{1}{L}$. The value of the Poisson’s ratio ($\sigma_{s}$) is 0.37, Young modulus ($E_{s}$) is 175 GPa, and $p_{s}$ is the external pressure on the rocks.

(iii) Radiation heat transfer between the rocks ($K_{eff}^{r}$)

A correlation proposed by Breithbath and Barthels [52] was used to calculate the effective thermal conductivity with radiation heat transfer:
Fig. 7. (a) Contribution of different mechanisms to the overall effective thermal conductivity and (b) Comparison of effective and actual thermal conductivity of air and rocks.
Fig. 8. (a) Grid dependence study and (b) time dependence study.
conductivity of the rocks and air is considered in the present study in order to analyze the porous packed-bed system filled with rocks.

3.3.4. Overall heat transfer loss coefficient through the tank wall

In the present numerical model, the wall was assumed as thermally thin, and the overall heat transfer loss coefficient \( h_w \) was considered. A separate Matlab [55] code was written for calculating the overall heat transfer loss coefficient through the walls with the rock wool insulation of 0.13 m thickness and considering the thermal conductivity variation of rock wool insulation material with temperature. The 130 mm insulation thickness is a consequence of the availability of diameters for the inner and outer pipes. The following equations were used to obtain the overall heat transfer loss coefficient through the walls:

\[
h_w = \frac{Q_{\text{loss}}}{A_{\text{ins}}(T_o - T_{\text{amb}})} \tag{25}
\]

where \( Q_{\text{loss}} \) is the heat loss, \( A_{\text{ins}} \) is the area of outer insulation surface per unit length, \( T_o \) is the temperature on the outer insulation surface, and \( T_{\text{amb}} \) is the ambient temperature.

\( Q_{\text{loss}} \) is calculated by using the following equations:

\[
Q_{n} = \frac{k(T)A}{T_{i,n} - T_{o,n}} \tag{26}
\]

\[
Q_{\text{ins}} = \sum Q_{n} \tag{27}
\]

where \( Q_{n} \) is the rate of heat transfer through the insulation at different time duration, \( A \) is the cross-sectional area per unit length, \( n \) is the number of data points at different time duration, \( k(T) \) is thermal conductivity of insulation, which is a function of temperature. A value of \( h_w \) of 0.77 W/m\(^2\)K was attained and used for the analysis.

4. Results and discussion

4.1. Grid and time dependence study

A two-dimensional axisymmetric computational domain was meshed with a quadrilateral structured grid. Three different grid element numbers, 12,500 (360 \( \times \) 35), 25,000 (500 \( \times \) 50) and 50,000 (730 \( \times \) 70) were studied with a constant time step of 0.05 s. Fig. 8(a) shows the temporal evolution of temperature at 5.3 m distance from the bottom of the packed-bed during the discharging process for different grid elements. It may be observed that the maximum percentage variation in temperature between grid size 25,000 and 50,000 is <1%.

Similarly, a time dependence analysis was performed by considering the time steps of 0.1 s, 0.05 s, and 0.03 s while keeping 25,000 grid elements. The temporal evolution of temperature at a 5.3 m axial distance from the bottom of the packed-bed during the discharging process for different time steps is presented in Fig. 8(b). It may be observed that the maximum percentage variation in temperature between a time step size of 0.05 s and 0.03 s is <1%. Therefore, 25,000 grid elements and a time step of 0.05 s are considered in the present analysis.

4.2. Model validation

The numerical model was validated by comparing its results with experimental results for both standby and discharging processes. There were no thermocouples at different locations in the radial direction inside the packed-bed. Therefore, no initial radial temperature variations in the packed-bed were considered, that is, for a given axial location, the initial temperature is considered the same along the radial direction except for the wall temperature. With the progress of the numerical simulations, the temperature variation is obtained in the radial direction.

The initial temperature profiles for the numerical simulation of the standby and the discharging processes were obtained from the experimental results and are shown in Fig. 9 along with the fitted curves. For this and subsequent figures reporting experimental data, the uncertainty range is reported for each data point. The temperature profile of the packed-bed at the end of the charging process was considered as the initial temperature profile for the standby process. Similarly, the temperature profile of the packed-bed at the end of the standby process was taken as the initial temperature profile for the discharging process. The equations obtained from fitting the test data for the initial center and wall temperatures are given by Eqs. (28–31).
Fig. 10. Comparison of the temperature in the packed-bed system during standby process at (a) the center for different axial distances from the bottom of the packed-bed and (b) the wall surface at 2.4 m.
The numerical and experimental results of the center temperature versus time during the experiments for the discharging process, as depicted in Fig. 11. The same variations in inlet mass flow and inlet temperature of the air were considered for the numerical simulations. A comparison of the numerical and experimental results of the center temperature versus time at different axial distances from the bottom of the packed-bed is shown in Fig. 12 (a). The results suggest that there is a sudden change in the rate of the temperature drop of the rocks at around 550 K. The air coming from the bottom of the packed-bed becomes hot before it reaches the point where the temperature is higher than 550 K, however, for temperatures lower than 550 K, the temperature difference between the rock and the air increases, causing the change in slope of the temperature drop and forming the thermocline region. Also, due to hot air reaching the region with temperature higher than 550 K, the assumption of no initial radial temperature variation does not affect the model results as this does not increase the heat losses from the packed-bed tank. The results obtained from the numerical model match quite well with the experimental results up to 550 K for most of the points. The average temperature deviation and the mean absolute percentage deviation between the experimental and numerical results in this region are 10 K and <2%, respectively. This suggests that the thermophysical properties and the temperature profiles of the rocks are accurately predicted.

After 550 K the cold air further cools down the region near the wall, increasing the conduction heat transfer in the radial direction and losses from the packed-bed. Due to the assumption of no initial radial temperature variation, the conduction heat transfer in the radial direction is lower in the numerical results than in the experimental results, and therefore the numerical models predict a higher center temperature than indicated by the experimental results, see Fig. 12(a). The average temperature deviation and the mean absolute percentage deviation between the experimental and numerical results in this region are 10 K and <5%, respectively.

The deviation between the numerical and experimental results can potentially be caused by a number of assumptions made in the simulations, including two-dimensional axisymmetric modeling, constant diameter and spherical shape of rocks, error in the value for the heat transfer coefficient, omission of the heat loss from the top and bottom of the packed-bed system during the discharging process and the initial temperature variation in the radial direction in the packed-bed, and the curve fitting of thermophysical properties of rocks and experimental data for the initial temperature profile. For large-scale high temperature systems, assuming a constant diameter and spherical shape of the rocks are not expected to result in any major deviation for the predictions of the center and wall temperatures [40]. The curve fitting of...
thermophysical properties of rocks, and experimental data for the initial temperature profile of the packed-bed, considered in the present study were obtained with a coefficient of determination ($R^2$) of 0.99, indicating that these factors are also not the main cause of the deviations. Therefore, it is expected that error in the value for the heat transfer coefficient, omission of the heat loss from the top and the bottom of the packed-bed system during the discharging process, and the initial temperature variation in the radial direction in the packed-bed are the main causes of the deviations observed in Fig. 12(a).

Fig. 12(b) presents a comparison of the numerical and experimental results of the wall temperature (0.5 m from the center) at an axial distance of 2.4 m, and the outlet temperature versus time for the packed-bed.
The discharging process of the packed-bed system. The average absolute temperature deviation and the mean absolute percentage deviation between the experimental and numerical results for the packed-bed wall temperature are 36 K and <6%, respectively. The assumption of the constant heat transfer loss coefficient on the walls of the packed-bed tank, and consideration of the wall temperature based on measurement at only one point on the walls of packed-bed, are the main causes of the deviation observed in wall temperature. The outlet temperatures obtained from the numerical simulations are the mass flux average temperatures at the top of the packed-bed system. As indicated in Fig. 12 (b), the outlet temperature remains approximately the same for the first 200 min of the discharging process due to a small radial temperature variation. Afterwards, the radial temperature variation increases with the progress of the discharging process leading to an increase in the mass flux near the walls of the packed-bed, and hence a lower average outlet temperature. The average absolute temperature deviation between the experimental and numerical results for the outlet temperature is 22 K, and the mean absolute percentage deviation is <5%. The deviation is a result of an error in the prediction of the center and wall temperatures of the packed-bed, as well as the fact that the average outlet temperature is predicted at the top of the wall of the packed-bed in the simulation, while the outlet temperature is measured in the outlet pipes in the experiments.

Overall, the results indicate that the present numerical model for the packed-bed system agrees well with the experimental results, and it accurately predicts the temperature both in the axial and radial directions for different working conditions.

4.3. Instability prediction for the discharging process

Fig. 13 shows the temporal evolution of temperature contours at an interval of 60 min in the packed-bed system obtained from the numerical simulation of the discharging process. Because of the symmetry of the computational domain, only half of the packed-bed (from wall to center) is shown in the contour plots. The temperature contours at the start of the discharging process (0 min) correspond to the state at the end of the standby process, as mentioned in section 4.2. The initial boundary conditions used are the same as those used in section 4.2.

The results presented in Fig. 13 suggest that for a given axial distance, the temperature of the wall is lower than the temperature of the packed-bed. As the discharging progresses, the temperature of the packed-bed decreases, and the variation in temperature in the radial direction increases. At the start of the discharging process, the temperature at the top of the packed-bed is lower than the maximum temperature (816 K), due to the heat loss from the top (and wall and bottom) surface(s) to the surroundings during the standby process. However, with the progress of the discharging process, the temperature of the top of the packed-bed increases and becomes equal to the maximum temperature of the packed-bed. This is because the lower temperature air coming from the bottom of the packed-bed absorbs heat from the rocks on its way towards the top of the packed-bed and reaches the maximum temperature of the rocks in the packed-bed. At the top of the packed-bed, the high temperature air expels heat to the rock, and thus the rock temperature at the top of the packed-bed increases. This can also be seen from the temperature contours (Fig. 13, 0 min and 60 min), where the temperature of the top of the packed-bed is lower compared to the maximum temperature at the start of the discharging process (0 min), and the temperature is equal to the maximum temperature of the packed-bed after 60 min of the discharging process.

Two different cases, with and without initial temperature difference between the center and the walls of the packed-bed systems, are considered next. In the first case, the difference between the center temperature and the wall temperature at the start of the discharging process is 80 K throughout the packed-bed. The predicted temperature difference between the center and the wall for different time steps and axial locations are reported in Fig. 14a. As the time progresses, the temperature difference decreases in the lower part (< 3.6 m) of the packed-bed, while the temperature difference increases in the upper part (≥ 3.6 m) of the packed-bed system. The variation in the radial direction is negligible in the fully discharged region of the packed-bed, as also can be observed from temperature contour plots (Fig. 13). In the upper part of the packed-bed, the temperature difference first increases from the initial temperature difference of 80 K and then starts to decrease. The time until the temperature difference starts to decrease varies with the axial distance in the upper part of the packed-bed. Once the rocks at a given axial distance are fully discharged, the temperature difference between the wall and the center becomes negligible. The maximum difference between the wall temperature and center temperature is 121 K at an axial distance of 5.15 m after 300 min of discharging time.

In the second case, the center temperature and wall temperature are considered the same throughout the height of the packed-bed at the start of the discharging process. Fig. 14b presents the difference between the calculated center temperature and wall temperature for different time steps for this case. Unlike the first case, the temperature difference increases in both the lower and the upper part of the packed-bed system for the first 60 min of the discharging process. This happens due to higher initial heat transfer losses from the walls of the packed-bed owing to a higher temperature of the walls in the second case compared to that.
Fig. 14. Predicted temperature difference between the center and the wall for different time steps and axial locations for (a) an initial temperature difference between the center and wall of 80 K and (b) an initial temperature difference between the center and wall of 0 K.
of the first case. After 60 min of discharging, a similar behavior to that observed in the first case is observed, i.e. the temperature difference for the lower part of the packed-bed decreases, while it increases for the upper part of the packed-bed system. The maximum difference in wall temperature and center temperature is 93.4 K at an axial distance of 5.3 m after 300 min of discharging time. The maximum temperature difference between the center and the wall of the packed-bed is obtained at the top of the packed-bed because of the higher temperature at the top of the packed-bed, which increases the heat loss from the walls to the surroundings.

In order to show the effect of temperature variations, Figs. 15 and 16 show the temporal evolution of density and velocity contours, respectively, at an interval of 60 min in the packed-bed system obtained from the numerical simulation of the discharging process. The density and velocity contours indicate that the density of the air at the bottom of the packed-bed is larger than that in the upper part of the packed-bed, while the velocity of air is lower in the bottom than in the upper part of the packed-bed. This is because the temperature of air is lower at the bottom of the packed-bed, and as the air moves towards the top, the air is heated by the rocks, resulting in a higher temperature of the air and thereby lower density and higher velocity at the top of the packed-bed. However, because of the losses near the walls of the packed-bed, the rocks near the wall region get colder, decreasing the temperature and the velocity, and increasing the density of the air near the wall. The pressure loss in the packed-bed is obtained by using Ergun Eq. [44] and is proportional to viscosity, density, and superficial velocity of the air. The combined effect of the lower velocity, higher density, and lower viscosity results in a decrease in the pressure drop, thereby reducing the resistance to the flow of the air near the wall. In turn, the decrease in resistance near the wall increases the mass flux in this region.

Fig. 17a and b show the variation in mass flow rate per unit area (i.e. mass flux) in the radial direction at different axial distances and different time steps obtained from numerical simulations for the cases with an initial temperature difference of 80 K and without an initial temperature difference between the center and wall. The results suggest that the mass flux varies in the radial direction for both the cases. The variation in mass flux in the radial direction is due to the temperature gradient in the radial direction caused by the heat loss from the walls of the tank. With the decrease in temperature, the viscosity decreases, and the density of air increases resulting in a decreased resistance to the flow. Thus, the mass flux increases with the increase in radial distance. The increase in...
Fig. 17. Mass flux at various radial distance during discharging process at different time durations for (a) an initial temperature difference between the center and wall of 80 K and (b) an initial temperature difference between the center and wall of 0 K.
mass flux near the wall region increases the cooling rate further, which further increases the mass flux near the wall region creating a positive feedback mechanism, i.e. the instability phenomenon. For the first case (see Fig. 17a), the maximum percentage increase in mass flux compared with the inlet mass flux is 1.6%, 2.4%, 2.8%, 3.5%, and 5.7% after 60 min, 120 min, 180 min, 240 min and 300 min, discharging time, respectively. For the second case (see Fig. 17b), the corresponding figures are 1.0%, 1.4%, 2.0%, 2.3%, and 3.3%. Also, the maximum percentage variation in mass flux is observed at different axial distances for different discharging times. The maximum percentage variation is observed at 3.4 m for 120 min and 180 min of discharging time and 5.3 m for 240 min and 300 min of discharging time in both the cases. It is worth noting that as the discharging time increases, the region which is fully discharged does not have any noteworthy variation in mass flux in the radial direction, as may be observed in both the figures for a discharging time of 300 min and axial distance of 3.4 m.

As a consequence of instability in the packed-bed system, the outlet temperature of the air is lower than the maximum temperature of the rocks in the packed-bed during the discharging process. Due to this, the stored energy in the packed-bed is not fully utilized, resulting in a reduced exergetic efficiency of the packed-bed system and a reduced round-trip efficiency of the energy storage plant including the packed-bed system. The difference in the outlet temperature and the maximum temperature of the rocks in the packed-bed tank obtained from numerical simulations is shown in Fig. 18 for the cases with an initial temperature difference of 80 K (case 1) and without an initial temperature difference between the center and the wall (case 2). Initially, the difference between the maximum temperature of the rocks and the outlet temperature of air is negligible due to the small variation in mass flux variation in the radial direction for both the cases. The difference between the maximum temperature of the rocks and the outlet temperature of air is negligible due to the small variation in mass flux in the radial direction for both the cases. The difference between the maximum temperature of the rocks and the outlet temperature of air increases after 240 min of discharging time, reaching a maximum of 45 K and 30 K at 310 min of discharging time for case 1 and case 2, respectively. That there is a temperature difference in both cases, indicates that the instability occurs no matter if there is an initial temperature difference between the center of the tank and the wall or not. After 310 min of discharging time, there is a decrease in the difference between the maximum temperature and the outlet temperature. This is because the packed-bed is close to completely discharged, and the average and maximum temperatures of the entire packed-bed are small, resulting in a drop in the difference between the maximum temperature and the outlet temperature.

4.4. Effect of standby duration on the instability

The standby periods 1 h, 3 h, 5 h, and 10 h followed by a discharging process are analyzed. In addition, as a reference, a case without a standby period is presented. The initial and boundary conditions used for the analysis of standby periods are the same as those used for the validation of the numerical model of the standby periods reported in section 4.2. Fig. 19 depicts the temperature contours after different standby durations, and the average temperature ($T_{avg}$) of the packed-bed for different standby periods. The results suggest that the average temperature of the packed-bed decreases as the standby duration increases. The drop in average temperature is 2 K for a standby period of 1 h compared with the reference case (0 h standby period) and 26 K for a standby period of 10 h compared with the reference case. The drop in temperature is largest near the wall, top and bottom of the packed-bed due to the heat loss to the ambient during the standby period. Moreover, it may be observed that the radial temperature gradient increases with the duration of the standby process, due to the conduction between the rocks and natural convection in the air during the standby process. As a result, the maximum temperature of the rocks drops from 816 K (0 h standby period) to 809 K after a standby period of 10 h.

The temperature difference in the radial direction compared with the center temperature (center temperature minus the temperature in radial direction) at a 5.3 m axial distance for different standby periods followed by 300 min of discharging is shown in Fig. 20. It can be observed from the figure that the difference in temperature in the radial direction compared with the center temperature increases with the standby time duration. Also, the temperature difference decreases from the wall towards the center reaching an almost negligible difference at a radial distance of 0.1 m, which is in agreement with the results reported in section 4.2. The maximum temperature difference occurs after a standby...
The results suggest that initially the difference between the maximum standby periods followed by a discharging process is shown in Fig. 21. The standby duration. Thus, the instability phenomenon increases with the standby duration causing it to be lower for the 10 h standby duration than that of the 1 h, 3 h and 5 h standby durations. Moreover, the results indicate that the temperature difference increases after 240 min of discharging time for all the cases. The maximum temperature difference occurs after 310 min of discharging time for all cases, and is found to be 48 K, 90 K, 119 K, and 161 K for the standby periods 1 h, 3 h, 5 h, and 10 h, respectively. After 310 min of discharging, there is a dip in the temperature difference between the maximum temperature and the outlet temperature, as was observed also for the case of discharging with no-standby duration (see section 4.3). This occurs because after 310 min of discharge of the packed-bed, the packed-bed is close to completely discharged, and the average and maximum temperatures of the entire packed-bed are small.

As a result of instability in a packed-bed, the cumulative discharged energy of the packed-bed is affected. Fig. 22 shows a comparison of the cumulative discharged energy for ideal discharging (without instability) and discharging after different standby durations (with instability). The ideal discharging case refers to the discharging of the packed-bed with no heat losses, no radial temperature variation at the start of discharging and no standby duration, while heat losses from the walls of packed-bed and an 80 K temperature difference between the center and the wall of the packed-bed are considered for the other cases. The results are presented for a time duration of 300 min, because after 300 min, there is a rapid decrease in the outlet temperature of the air, which is limited by the minimum allowable temperature at the inlet of the turbine in thermal energy storage systems.

The results shown in Fig. 22 suggest that the cumulative discharged energy is highest for the ideal discharging process and decreases with the increase in standby duration. By 300 min discharging time, the decrease in the cumulative discharged energy with respect to that of the ideal discharging case is 2.1%, 2.7%, 4.3%, 5.8%, and 10.7% for 0 h, 1 h, 3 h, 5 h and 10 h standby durations, respectively. At the same point in time, the outlet temperature of the air of the packed-bed is 580.3 K, 573.9 K, 570.4 K, 560.2 K, 549.7 K and 499.2 K for the ideal discharging case, and 0 h, 1 h, 3 h, 5 h and 10 h standby duration cases, respectively. Moreover, the results shown in the figure indicate that the cumulative discharged energy increases linearly with time for all the cases, however, there is a decrease in the discharged energy for each time step towards the end of the considered time duration, flattening the curves. This is because, towards the end of the discharging process, the thermocline has reached the top of the packed-bed and hence the average temperature of the packed-bed is decreased, which results in a smaller amount of energy discharged for each time duration.

4.5. Influence of geometry of the packed-bed system on the instability

In this section, the influence of size of the packed-bed system on the instability phenomenon is analyzed by comparing a reduced-scale and full-scale packed-bed system with no standby duration, a constant mass flow rate, and a constant Biot number ($hL_c/k_{liner}$) of 0.136 on the outer walls of the packed-bed tank. The figure for the Biot number was obtained by assuming a heat transfer loss coefficient ($h$) from wall of the packed-bed of 0.77 W/m²K, a characteristic length ($L_c$) of 5.32, and a thermal conductivity of liner material ($k_{liner}$) of 30.12 W/mK. However, because of the large height of the full-scale packed-bed, the assumption of a constant heat transfer loss coefficient over the length of packed-bed may lead to an underprediction of the significance of the instability phenomenon. The height and diameter of the reduced-scale system are 5.32 m and 1.00 m, respectively, while the height and diameter of the full-scale system are 12.00 m and 5.00 m, respectively. The dimensions of the reduced-scale and full-scale systems selected in this study are based on the laboratory-scale prototype and large-scale pumped thermal energy storage systems of Stiesdal Storage. Both the systems were designed in such a way that their charging and discharging times are the same. The mass flow rate at the inlet of the packed-bed system is 0.37 kg/s and 22.2 kg/s for reduced-scale and full-scale systems, respectively.

The outlet temperature of the air and maximum temperature of rock versus time is presented for both the systems in Fig. 23. The results
suggest that the difference between the maximum temperature of the rocks and the outlet temperature of air is lower in a full-scale system than that of a reduced-scale system, being 73 K and 56 K for the reduced-scale and full-scale systems, respectively. The surface area of walls per unit storage volume is much lower for the full-scale system compared to that of the reduced-scale system. This results in a smaller temperature variation in the radial direction leading to a lower temperature difference in a full-scale system compared to that of a reduced-scale system. In the case of a full-scale system, the temperature difference occurs only at the end of the discharging process, while the temperature difference occurs throughout the discharging process for the reduced-scale system. At the end of the discharging process, the stored energy is concentrated to the topmost part of the packed-bed, and the radial temperature variation at the top becomes increasingly significant as

![Fig. 20. Temperature difference in radial direction compared with the center temperature at a 5.3 m axial distance for different standby periods followed by 300 min of discharging.](image1)

![Fig. 21. Difference between the maximum temperature of the rocks and the outlet temperature of the air of the packed-bed system for different standby periods followed by a discharging process.](image2)
time passes, and therefore, the temperature difference is observed only at the end of the discharging process for the full-scale system. It needs to be stressed that the maximum temperature difference reported in this section is much smaller than those presented in previous sections, namely sections 4.2–4.4, because of no standby duration and a constant inlet mass flow rate. Overall, the results indicate that the size of the packed-bed system affects the radial temperature variation, and thereby the instability phenomenon.

5. Discussion

The analysis presented in this paper is limited to a packed-bed system with a relatively large height to diameter ratio of 5.32 compared to the height to diameter of 1 to 2.5 typically considered in papers in the open literature. In addition, different parameters such as inlet mass flow rate, inlet temperature of the heat transfer fluid, heat transfer loss coefficient of the wall of the packed-bed tank, type of rocks, diameter of the rocks, and porosity of the packed-bed were not varied. As the heat transfer
mechanisms inside the packed-bed and losses from the packed-bed are affected by the geometry and operating parameters of the packed-bed, the aforementioned parameters are expected to affect the significance of the instability phenomenon.

6. Conclusions

This paper presents numerical and experimental investigations of an air-based packed-bed rock thermal energy storage system for large-scale high temperature applications. The objective of the study is to predict the instability and to analyze the effect of different standby durations and storage sizes on the instability of the air-based packed-bed system. Transient axisymmetric computational fluid dynamics models were developed for the standby process and discharging processes. In addition, experimental investigations were carried out, and the experimental results were used to validate numerical models. The main conclusions of the work are summarized as follows:

1. The results of the numerical models for standby and discharging processes of packed-bed systems agree well with the experimental results in both the axial and radial directions for different working conditions (<6% in mean absolute deviation).

2. The maximum difference between the maximum temperature of the rocks and the outlet temperature of the air is found to be 45 K and 30 K, assuming an initial temperature difference of 80 K between center and wall, and no initial temperature difference between center and wall, respectively. These results suggest that the instabilities will be present in the packed-bed system even if there are no radial temperature variations at the end of charging process.

3. The instability phenomenon is found to increase with the standby duration, resulting in a maximum difference of 161 K between the maximum rock temperature and outlet air temperature for a standby period of 10 h followed by the discharging process.

4. The instability is found to be affected by the size of the packed-bed system. The maximum difference between the rock temperature and outlet temperature of air is 73 K and 56 K for a reduced-scale and a full-scale system, respectively.

The results presented and the models developed in this paper can serve as a basis for further analyses on the instability in high temperature air-based packed-bed systems, including the estimation of the effects of different geometrical and operating parameters on the instability, with the aim of reducing the instability and maximizing the performances of such systems.

Nomenclature

Latin characters

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>a_s</td>
<td>Superficial area of heat storage materials (rocks) per unit bed volume ( \text{m}^2/\text{m}^3 )</td>
</tr>
<tr>
<td>a_w</td>
<td>Area of packed-bed walls ( \text{m}^2 )</td>
</tr>
<tr>
<td>B</td>
<td>Deformation parameter</td>
</tr>
<tr>
<td>C</td>
<td>Shape parameter</td>
</tr>
<tr>
<td>C_1</td>
<td>Permeability coefficient</td>
</tr>
<tr>
<td>C_2</td>
<td>Inertial Coefficient</td>
</tr>
<tr>
<td>C_p</td>
<td>Specific heat capacity ( \text{J/(kg.K)} )</td>
</tr>
<tr>
<td>D</td>
<td>Empirical coefficient</td>
</tr>
<tr>
<td>d</td>
<td>Diameter of solid particles (rocks) ( \text{m} )</td>
</tr>
<tr>
<td>E_s</td>
<td>Young’s modulus of rocks ( \text{Pa} )</td>
</tr>
<tr>
<td>F</td>
<td>Tortuosity</td>
</tr>
<tr>
<td>g</td>
<td>Acceleration due to gravity ( \text{m/s}^2 )</td>
</tr>
<tr>
<td>h</td>
<td>Convective heat transfer coefficient ( \text{W/(m}^2.K) )</td>
</tr>
<tr>
<td>k</td>
<td>Thermal conductivity ( \text{W/(m.K)} )</td>
</tr>
<tr>
<td>L_c</td>
<td>Characteristic length of cylinder ( \text{m} )</td>
</tr>
<tr>
<td>Nu</td>
<td>Nusselt number</td>
</tr>
<tr>
<td>N_i</td>
<td>Number of particles per unit length</td>
</tr>
<tr>
<td>p</td>
<td>Pressure ( \text{Pa} )</td>
</tr>
<tr>
<td>Pr</td>
<td>Prandtl number</td>
</tr>
<tr>
<td>p_s</td>
<td>External pressure exerted on the rocks ( \text{Pa} )</td>
</tr>
<tr>
<td>Q</td>
<td>Rate of heat transfer ( \text{W} )</td>
</tr>
<tr>
<td>q</td>
<td>Heat flux ( \text{W/m}^2 )</td>
</tr>
<tr>
<td>Re</td>
<td>Reynold number</td>
</tr>
<tr>
<td>S_f</td>
<td>Packing variable</td>
</tr>
<tr>
<td>S_i</td>
<td>Source term</td>
</tr>
<tr>
<td>T</td>
<td>Temperature ( \text{K} )</td>
</tr>
<tr>
<td>t</td>
<td>Time ( \text{s} )</td>
</tr>
<tr>
<td>v</td>
<td>Superficial velocity ( \text{m/s} )</td>
</tr>
<tr>
<td>x</td>
<td>Axial coordinate ( \text{m} )</td>
</tr>
<tr>
<td>Z</td>
<td>Axial Distance ( \text{m} )</td>
</tr>
<tr>
<td>r</td>
<td>Radial coordinate ( \text{m} )</td>
</tr>
<tr>
<td>r_s</td>
<td>Radius of rocks ( \text{m} )</td>
</tr>
</tbody>
</table>

Greek characters

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>γ</td>
<td>Ratio of thermal conductivity of fluid and solid ( \text{s} )</td>
</tr>
<tr>
<td>e</td>
<td>Void fraction of the bed, i.e. volume fraction of fluid</td>
</tr>
<tr>
<td>e_s</td>
<td>Emissivity of heat storage materials (rocks)</td>
</tr>
<tr>
<td>µ_f</td>
<td>Viscosity of fluid ( \text{Pa.s} )</td>
</tr>
<tr>
<td>ρ</td>
<td>Density ( \text{kg/m}^3 )</td>
</tr>
<tr>
<td>σ_s</td>
<td>Poisson’s ratio</td>
</tr>
<tr>
<td>ψ</td>
<td>Stefan-Boltzmann constant ( 5.67 \times 10^{-8} ) ( \text{W/(m}^2.K^4) )</td>
</tr>
</tbody>
</table>

Subscripts

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>amb</td>
<td>Ambient</td>
</tr>
<tr>
<td>eff</td>
<td>Effective</td>
</tr>
<tr>
<td>f</td>
<td>Fluid (air)</td>
</tr>
<tr>
<td>i</td>
<td>Inner insulation surface</td>
</tr>
<tr>
<td>ins</td>
<td>Insulation surface</td>
</tr>
<tr>
<td>liner</td>
<td>Liner material</td>
</tr>
<tr>
<td>mix</td>
<td>Mixing of fluid due to turbulence</td>
</tr>
<tr>
<td>n</td>
<td>Different time durations</td>
</tr>
<tr>
<td>o</td>
<td>Outer insulation surface</td>
</tr>
<tr>
<td>s</td>
<td>Solid (rock)</td>
</tr>
<tr>
<td>w</td>
<td>Outer surface of the inner cylinder wall (liner)</td>
</tr>
<tr>
<td>0</td>
<td>Conduction in rock and air</td>
</tr>
<tr>
<td>C</td>
<td>Conduction through contact surface of neighboring rocks</td>
</tr>
<tr>
<td>R</td>
<td>Radiation heat transfer by rocks</td>
</tr>
</tbody>
</table>

Abbreviations

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>CFD</td>
<td>Computational Fluid Dynamics</td>
</tr>
<tr>
<td>EIA</td>
<td>Energy Information Administration</td>
</tr>
<tr>
<td>EUDP</td>
<td>Energy Technology Development and Demonstration Program</td>
</tr>
<tr>
<td>IEA</td>
<td>International Energy Agency</td>
</tr>
<tr>
<td>PTES</td>
<td>Pumped thermal energy storage</td>
</tr>
<tr>
<td>LTNE</td>
<td>Local thermal non-equilibrium</td>
</tr>
</tbody>
</table>

CRediT authorship contribution statement

Rohit Kothari: Writing – review & editing, Writing – original draft, Visualization, Validation, Software, Methodology, Formal analysis, Data curation, Conceptualization. Casper Schytte Hemmingsen: Writing – review & editing, Software, Methodology. Niels Vinther Voigt: Writing – review & editing, Investigation, Conceptualization. Angelo La Seta: Writing – review & editing, Conceptualization. Kenny Krogh Nielsen: Writing – review & editing.
Writing – review & editing, Investigation, Conceptualization. Nishith B. Desai: Writing – review & editing, Conceptualization. Akhil Vijayan: Writing – review & editing, Conceptualization. Fredrik Haglund: Writing – review & editing, Supervision, Project administration, Funding acquisition, Conceptualization.

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.

Data availability

The data that support the findings of this study are available from the corresponding author, Rohit Kothari, upon request.

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