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# High performance integrated receiver-storage system for concentrating solar power beam-down system

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**Abstract:** Concentrating solar power systems (CSP) with thermal storage units can provide dispatchable power. Here we propose a ~~new~~ modified design of a cavity receiver combined with a thermocline heat storage unit for the beam-down CSP. Instead of using a separate receiver and heat storage unit, an integrated unit consisting of an extended cylindrical cavity with a packed bed storage is proposed. The new approach was designed using validated cavity radiation and quasi-1D 2-phase numerical heat transfer models. As the concentrated irradiation can be directly absorbed in such a system, the structure used can be simplified and operation of the unit is more effective. A high solar-to-exergy conversion ratio of 0.52 was reached with an optimized design, charging and discharging efficiencies being well beyond 99% and 92% at 770°C. An important detail in the integrated receiver-storage design was the use of a circulation air flow fan, which enhanced the heat transfer inside the packed bed storage. The proposed ~~new~~ design is promising for improving the efficiency and economics of beam down CSP.

**Keywords:** concentrating solar power, heat transfer, thermal energy storage, packed bed storage, beam-down system, thermocline

## Nomenclature

*Symbols*

| | |
|---|---|
| $C$ | concentration ratio or specific area (- or $m^2/m^3$) |
| $c_p$ | specific heat (J/kgK) |
| $d$ | effective diameter of rocks (m) |
| $F$ | view factor matrix |
| $H$ | height (m) |
| $h$ | specific enthalpy (J/kg) |
| $h_{rs}$ | void to void radiative heat transfer coefficient (W/ $m^2$K) |
| $h_{rv}$ | solid surface to solid surface radiative heat transfer coefficient (W/ $m^2$K) |
| $k$ | thermal conductivity (W/mK) |
| $L$ | thickness (m) |
| $\dot{m}$ | mass flow rate (kg/s) |
| $Nu$ | Nusselt number (-) |
| $Pr$ | Prandtl number (-) |
| $p$ | pressure (Pa) |
| $Q$ | thermal energy (MWh) |
| $\dot{Q}$ | heat flux (kW) |
| $Re$ | Reynolds number (-) |
| $q$ | irradiation (W/$m^2$) |
| $\overline{q}$ | average irradiation (W/$m^2$) |
| $R$ | radius (m) |
| $t$ | time (s) |
| $T$ | temperature (K\°C) |
| $U$ | overall heat transfer coefficient (W/$m^2$K) |
| $\alpha$ | rim angle (˚) |

| | |
|---|---|
| $\beta$ | constant |
| $\gamma$ | circulation flow to output flow ratio (-) |
| $\delta$ | Dirac delta function |
| $\varepsilon$ | porosity (-) |
| $\Delta T$ | temperature difference (K\°C) |
| $\bar{T}$ | integral mean of temperature (K\°C) |
| U | emissivity (-) |
| $\eta$ | efficiency (%) |
| $\Theta$ | non-dimensional temperature (-) |
| $v$ | viscosity ($m^2/s$) |
| $\xi$ | solar to exergy conversion ratio (-) |
| $\rho$ | density ($kg/m^3$) |
| $\sigma$ | Stefan-Boltzmann constant or RMSE ($W/m^2K^4$ or m) |
| $\sigma_T$ | relative deviation of outlet temperature during discharging (%) |
| $\phi$ | constant |
| $\lvert\ \rvert$ | absolute value |

*Subscripts*

| | |
|---|---|
| *0* | initial or original point |
| *absorb* | absorbing |
| *bottom* | bottom |
| *c* | charging |
| *cav* | cavity |
| *conv* | convective |
| *cycle* | charging-discharging cycle |
| *d* | discharging |
| *eff* | effective |
| *f* | fluid |
| *fan* | fan |

| | |
|---|---|
| *inc* | incident |
| *inlet* | inlet to the discharging phase |
| *inside* | inside wall |
| *layer* | layer of storage |
| *loss* | loss |
| *max* | max |
| *net* | net |
| *outlet* | outlet to the discharging phase |
| *rad-cond* | radiative and conductive |
| *s* | solid |
| *storage* | storage |
| *surf* | cavity surface |
| *top* | top |
| *v* | volumetric |
| *w* | wall |
| $\infty$ | ambient |

*Abbreviations*

| | |
|---|---|
| CPC | compound paraboloid concentrator |
| CSP | concentrated solar power system |
| CT | computer tomography |
| EP | equilibrium point |
| HTF | heat transfer fluid |
| IRS | integrated receiver-storage system |
| LDC | low-density concrete |
| RPC | reticulate porous ceramic |
| STS | spatially thermal stratification |
| TES | thermal energy storage |
| UPC | ultra-high-performance concrete |
| VF | view factor |

## 1. Introduction

Concentrating solar thermal power (CSP) is a promising renewable energy technology, which can provide dispatchable power when connected to thermal energy storage (TES) (Kuravi et al., 2013). Therefore, developing efficient and cost-effective TES systems has high relevance for future CSP technologies (Pardo et al., 2014). Recent CSP projects seem to increasingly employ TES (Pelay et al., 2017).

The thermophysical principle of a thermal storage unit can be based on sensible heat, latent heat of fusion or vaporization, or on reversible chemical reactions (Kuravi et al., 2013). Sensible heat storage is so far the most commonly used approach because of simplicity and a wide range of low-cost materials available, though its transient heat transfer characteristics and total heat storage mass flux falls well behind the two other forms of TES (Pelay et al., 2017; Romero and Steinfeld, 2012). Typical sensible heat storage materials include e.g. rock gravel, sand or concrete (Brosseau et al., 2005; Tamme et al., 2004; Zanganeh et al., 2012), and for working fluid, molten salt, steam, or high temperature oil (Gil et al., 2010; Liu et al., 2016; Steinmann and Eck, 2006) have been used (Herrmann and Kearney, 2002; Medrano et al., 2010). Combining a packed bed of rocks as storage material and air as heat transfer fluid (HTF) has been proposed due to the inherent technical and economic advantages associated such as abundant and economical storage material, no hazardous or corrosive ingredient, and direct heat transfer mode, etc. (Zanganeh et al., 2015a). Because of the attractiveness of rock as a storage material, the effects of high-temperature (up to 1000°C) thermal cycling on rocks have received much attention in the past decade (Becattini, 2018). Good long-term stability of rock storage has been reported (Allen et al., 2014; Riaz, 1977; Tiskatine et al., 2016). Here we consider the same rock composition as used in a pilot-scale storage system (ETH), mainly containing Siliceous Limestone, Quartzite, Limestone, Calcareous Sandstone, and Gabbro (Zanganeh et al., 2012).

In this paper, we present a cavity receiver integrated with a packed-bed rock storage as a thermal energy storage for a beam-down CSP system to simplify overall design and find more effective solutions for system design. It is well-known that packed beds have been subject to extensive general methodology development and analyses of specific cases in the past (Beek, 1962; Kunii and Smith, 1960, 1961; Pfeffer, 1964; Whitaker, 1972), but the literature on integrated receiver-storage is negligible. Examples of past work on packed-bed thermal storage include e.g. determining the effective thermal conductivity of porous rocks (Kunii and Smith, 1960, 1961), heat and mass transport in fixed beds (Beek, 1962; Pfeffer, 1964), correlations for heat transfer (Whitaker, 1972), heat transfer models for high porosity and complex micro-structure cases (Kaviany, 2012), determining radiative transport properties of porous media (Ganesan and Lipiński, 2011; Petrasch et al., 2007), and different 1D-3D models for performance analysis of packed beds (Geissbühler et al., 2016; Ismail and Stuginsky Jr, 1999; Meier et al., 1991; Zanganeh et al., 2014b; Zanganeh et al., 2015a; Zanganeh et al., 2015b; Zanganeh et al., 2012; Zavattoni et al., 2015), among others. Besides, packed-bed has been widely applied in high-temperature solar receiver/reactor systems to obtain a high heat/mass transfer rate (Chueh et al., 2010; Furler and Steinfeld, 2015; Hischier et al., 2012; Keene et al., 2013). To date, the studies in terms of thermocline heat storage for solar tower and dish power systems have mainly focused on the sensitive and latent heat transfer issues (Geissbühler et al., 2016; Zanganeh et al., 2014b; Zanganeh et al., 2015a; Zanganeh et al., 2015b; Zanganeh et al., 2012). The merit of the proposed integrated receiver-storage design lies in the possibility to eliminate heat exchangers as well as complex connecting devices. Thus, a simplified structure and operation is possible, which could reduce overall costs, improve system stability, and lead to high heat transfer efficiency.

In present work, the cavity receiver integrated with a packed-bed as the thermal storage is intended for a beam-down CSP system. The concept of beam-down was initially suggested in 1970s for central receivers (Rabl, 1976), and it has been further developed theoretically and experimentally for different designs (Segal and Epstein, 1999; Segal and Epstein, 2001; Vant-Hull, 2014). Using a secondary reflector in this context enables to place the heavy components at ground level, thus enabling simpler and cheaper tower and heat transport sub-systems (Vant-Hull, 2014). Also, the concentration ratio can be improved through shortening the optical path. A few beam-down systems have been built so far (Matsubara et al., 2014; Mokhtar et al., 2014), but applications of interest include fuel production via thermochemical reactions (Furler and Steinfeld, 2015) or power generation integrated with thermal storage systems (Koepf et al., 2012; Matsubara et al., 2014; Tamaura et al., 2006). Hence, the proposed integrated receiver-storage (IRS) design with a beam-down CSP has high relevance.

To our best knowledge, the original concept of the IRS for CSP was initially proposed by Slocum et al. in 2011 (Slocum et al., 2011). In their work, a CSP system with integral storage was presented, where heliostats direct sunlight into a volumetric absorption molten salt receiver. The incident concentrated sunlight can therefore be directly absorbed when penetrating the salt. Here we employed a modified IRS design to achieve a direct absorption of solar radiation for a beam-down CSP system. Different from Slocum's design, we considered a packed bed as the storage media and air as HTF instead of molten salt aiming for simplifying the system and reducing operating complexity. Also, a recirculation device was designed to enhance the heat transfer inside the bed.

The paper is organized as follows. In Chapter 2, we describe the modified receiver-storage design, in Chapter 3 the methodology for modelling the design including the validation of the simulation model, in Chapter 4 we present the main results, and finally in Chapter 5 the conclusions.

## 2. Description of the ~~new~~ modified receiver-storage design

In this section, the technical details for the integrated receiver-storage system (IRS) are given, which are later used in the simulations and analyses. The basic CSP considered is a 450 $kW_{th}$ beam-down system located in eastern China with an average normal radiation (DNI) of 4.8 kWh/m$^2$ per day. The daily storage capacity of the IRS is designed as 3.6 MWh$_{th}$. Total optical efficiency and average concentration ratio ($C$) are set to 63% and 1000 suns, considering the state-of-the-art of the commercial heliostats and central reflectors (reflectivity 0.8 and 0.95, mirror error 1 mrad) and tracking accuracy (~2.5 mrad) (Mokhtar et al., 2014; Wei et al., 2013). Figure 1 shows a sketch of the IRS and its working principle. The operating process is as follows: at the beginning of charging of the TES, the top of the packed bed is exposed to concentrated solar irradiation and heated up from the initial temperature. Cooler air is pumped out by $M_1$ from the bottom of the storage, while another fan ($M_2$) is used to circulate the top air flow for enhancing heat exchange within the bed. Fan model AFP® produced by Daniels Fans© could be available for $M_2$ which can circulate high temperature air (up to 950°C) in a controllable volumetric flow rate (up to 50 m$^3$/s) (DanielsFans). The power of $M_1$ and $M_2$ are subject to the desired output temperature of the HTF as well as the scale of the IRS. The charging mass flow rate of $M_1$ ( $\dot{m}_{1c}$) is fixed to 0.4 kg/s while the rate of circulation air ( $\dot{m}_{2c}$) varies in the range of 0-4 kg/s. Here we define the circulation flow to output flow ratio as $\gamma = \dot{m}_{2c} / \dot{m}_{1c}$, which indicates the relation of the mass flow rates, i.e. the powers offered by $M_1$ and $M_2$. The charging time is set here to 8 hours (daytime). During the discharge, air flows through the packed bed inversely and exits from the outlets arranged on the side walls of cavity. The discharging mass flow rate of $M_1$ ( $\dot{m}_{1d}$) is fixed to 0.2 kg/s. The aperture is closed so that hot air cannot escape from the top. The discharge time is set to 16 hours (night-time). Metal grids are used for eliminating heterogeneity of flow velocity at the same cross section due to the fan effect.

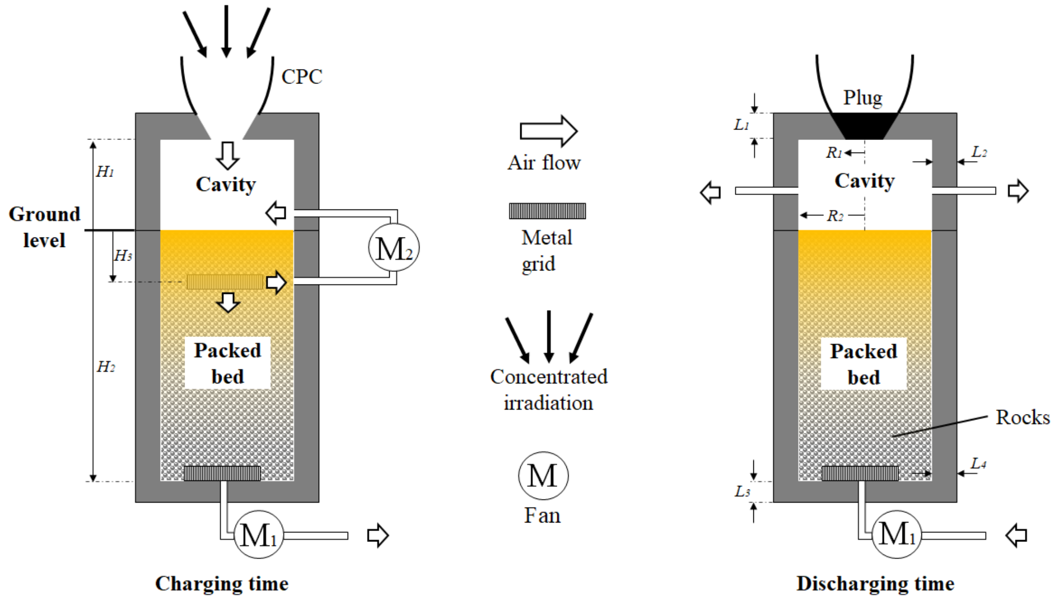


Figure 1. Scheme of the integrated receiver-storage (IRS) configuration.

The tank is immersed in the ground (except for the cavity part) for reducing the lateral load bearing caused by the expanding rock during a charge half-cycle as well as heat losses. It has a cylindrical cross section with an inner radius of $R_2$ and a total height of $H_1$+$H_2$. The insulation of the cavity contains two layers: $Al_2O_3$–$SiO_2$ (inner) and Foamglas® (outer) with a total thickness of $L_1$ for the lid and $L_2$ for the side. It is enclosed outside with a thin layer of Inconel 600. The immersed part of the tank is made of insulation (2 layers) and concrete (2 layers): Microtherm®, Foamglas®, ultra-high-performance concrete (UPC) and low-density concrete (LDC) (Martinola et al., 2010). The thicknesses of the base and side wall are $L_3$ and $L_4$, correspondingly. The packed bed is filled with rocks till the ground level. The equivalent thermal properties and geometries are selected here according to the characteristics of rocks (Somerton, 1992; Zanganeh et al., 2015a; Zanganeh et al., 2012). Heat capacities and thermal conductivities are plotted in Fig. 2 as a function of temperature. The inlets of $M_2$ ~~is~~ are inserted from the side at a height of $H_3$ beneath the ground for transporting air back to the cavity. Not shown in Fig. 1, but four symmetric inlets are designed along the lateral walls for reducing the effect of the air bypass. The aperture of cavity is $R_1$. A compound paraboloid concentrator (CPC) is coupled with the IRS configuration to improve the concentration ratio up to 1000. The details of the design are shown in Tables 1 and 2. It is worth noting that thermal ratcheting may occur when a tank filled with particulate solids is heated and cooled successively (Flueckiger et al., 2011; Kolb et al., 2011). Two conditions should be considered: 1) if the tank wall expands more than the filled particles a gap may form and then the particles may subside to fill the gap. When the tank is cooled, however, the wall contracts against the bed and may experience stresses in excess of the yield stress, resulting in plastic deformation. Cyclic operations repeats the process and the tank wall is slowly "ratcheted" outward until it fails; 2) conversely, if the bed expands more than the tank walls, it may deform the walls plastically on heat up. Ultimately over many cycles, failure of the tank could occur. Our specific design here is supposed to mitigate thermal ratcheting. In fact, the effect of the lateral earth pressure prevents the buried tank wall from expanding outward in a charge half-cycle. Moreover, the UPC can bear a high pressure while the soft structure of Microtherm® and Foamglas® as the buffered liner offers additional volume for the expanding bed. Therefore, the phenomena in improbable in our IRS system. A 3-D thermo-mechanical analysis would be necessary for an accurate evaluation, which was out of the scope of this paper.

Table 1. Dimensions and operating conditions of the integrated receiver-storage system (IRS) design.

| **Dimensions** | | **Operating conditions** | |
|---|---|---|---|
| $H_1$ (m) | 1.5 | charging time, $t_c$ (s) | 28800 |
| $H_2$ (m) | 8 | discharging time, $t_d$ (s) | 57600 |
| $H_3$ (m) | 0.33~1.65 | HTF's outlet mass flow rate during charging, $\dot{m}_{1c}$ (kg/s) | 0.4 |
| $R_1$ (m) | 0.447 | HTF's outlet mass flow rate during discharging, $\dot{m}_{1d}$ (kg/s) | 0.2 |
| $R_2$ (m) | 2 | HTF's circulating mass flow rate during charging, $\dot{m}_{2c}$ (kg/s) | 0~4 |

| | | | |
|---|---|---|---|
| $L_1$ (m) | 0.2/0.5 | Circulation flow to output flow ratio, $\gamma$ (-) | 0~10 |
| $L_2$ (m) | 0.2/0.5 | incident radiation flux, $\dot{Q}_{inc}$ (kW) | 439.8 |
| $L_3$ (m) | 0.3/0.5/0.02/1 | initial temperature, $T_0$ (K) | 298 |
| $L_4$ (m) | 0.3/0.5/0.02/1 | ambient temperature, $T_\infty$ (K) | 293 |
| $d$ (m) | 0.003 | efficiency of fan, $\eta_{fan}$ (-) | 0.95 |
| $\varepsilon$ (-) | 0.342 | solar-to-power efficiency of commercial CSP, $\eta_{CSP}$ (-) | 0.23 |

Table 2. Main physical properties of materials used in the IRS (Furler and Steinfeld, 2015; Kelley, 1960; Somerton, 1992; ToolBox, 2005; Yang et al., 2018; Zanganeh et al., 2015a; Zanganeh et al., 2012).

| **Material** | **Conductivity & specific heat & viscosity & density & emissivity** $k$ (W/mK) & $c_p$ (J/kgK) & $\nu$ ($\times10^6$ m$^2$/s) & $\rho$ (kg/m$^3$) & Ụ (-) |
|---|---|
| Rocks (Kelley, 1960; Somerton, 1992; Zanganeh et al., 2015a; Zanganeh et al., 2012) | $k(T)=\begin{cases} k_{20}-A(T-B)(k_{20}-C)[k_{20}(DT)^{-Ek_{20}}+F]k_{20}^{-G} & k_{20}>2W/mK \\ k_{20}-A(T-B)(k_{20}-C) & k_{20}<2W/mK \end{cases}$ |
| | $c_p(T)=747.0995+0.5676\times(T-273)$ |
| | $\rho=2732.6$ |
| | Ú= 0.85 |
| Air (ToolBox, 2005; Yang et al., | $k(T)=2.35\times10^{-12}T^3-1.290\times10^{-8}T^2+4.8370613\times10^{-5}T+0.00483$ |
| | $c_p(T)=1171\times e^{-\left(\frac{T-3070}{2257}\right)^2}+691.6\times e^{-\left(\frac{T-516.2}{1673}\right)^2}+191\times e^{-\left(\frac{T+114.3}{399.4}\right)^2}$ |
| | $\nu(T)=258.7-259.4\cos(0.001214T)-88.17\sin(0.001214T)-6.35\cos(0.002428T)+49.59\sin(0.002428T)-5.995\cos(0.003642T)-0.2957\sin(0.003642T)$ |

| | |
|---|---|
| 2018) | $\rho(T) = 352.6T^{-0.9998} - 1.747\times10^{-4}$ |
| $Al_2O_3$–$SiO_2$ (Furler and Steinfeld, 2015) | $k(T) = 0.00012926T + 0.019654$ |
| | $c_p(T) = \min\{4\times10^{-7}T^3 - 1.3797\times10^{-3}T^2 + 1.5987289T + 477.6995948, 1118.44\}$ |
| | $\rho = 560.65$ |
| | Ú= 0.85 |
| Insulation (Zanganeh et al., 2012) | $k = 0.025/0.05/2.05/0.375/0.5\left(Microtherm^{®}/Foamglas^{®}/UPC/LDC/Soils\right)$ |
| | $c_p = 840(Foamglas^{®})$ |
| | $\rho = 115(Foamglas^{®})$ |
| | Ú$(T) = 0.0001982\times T + 0.5734(Inconel\ 600)$ |

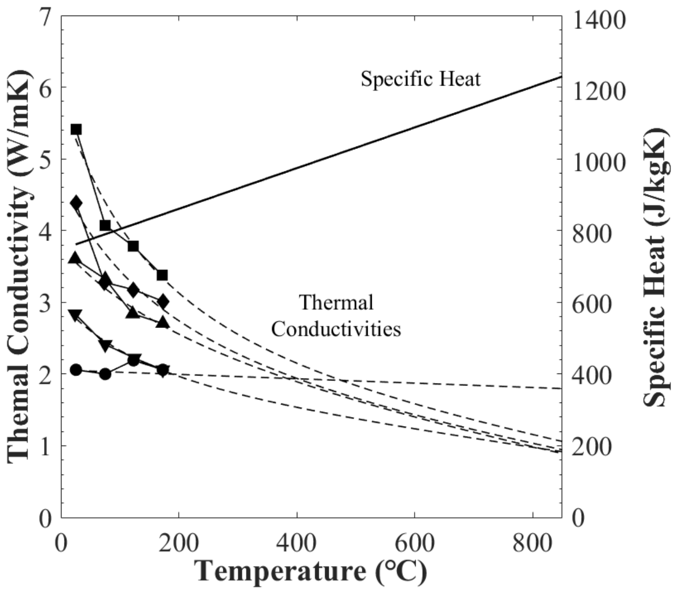


Figure 2. Thermal conductivity and specific heat of the rocks as a function of temperature. Dashed lines: Extrapolations obtained using the correlations for thermal conductivity (Somerton, 1992) and specific heat (Zanganeh et al., 2015a). Solid lines: experimental data (Zanganeh et al., 2012) corresponding to ■Quartzite, ◆Calcareous Sandstone, ▲Helvetic Siliceous Limestone, ▼Limestone, and ●Gabbro.

## 3. Modelling of the integrated receiver-storage system

To analyze and optimize the thermal performance of the integrated receiver-storage system, we developed a thermal simulation model for the cavity receiver and the packed-bed storage. The basic assumptions made in the modelling were the following:

- All materials are isotropic and the surfaces are opaque gray-diffuse;
- Ambient temperature is set at 293 K and the sky is regarded as a black-body at an 8 K lower temperature ($T_{sky}$=285 K) (Kalogirou, 2012);
- Conductive losses through insulation are 1-dimensional;
- For the discharging phase, the conductive heat losses to the cavity part (above the ground) are very small and can be ignored;
- A Gaussian distribution is used for incident solar radiation at the cavity bottom;
- Air is regarded as a non-radiative media except for the effect of void-to-void radiative heat transfer;
- A plug flow in the packed-bed so that the air mass flow rate is uniform at any cross-section perpendicular to the packed bed;
- Radial temperature differences are ignored for the packed bed;
- For the storage, the thermal inertia of the walls and the soil insulation layer is not considered.

### *3.1 Thermal model for the cavity receiver*

Firstly, the net irradiation flux at the receiver bottom was modelled using Monte Carlo ray-tracing to obtain the matrix of the view factor (VF) for each surface element of the cavity inner walls. 1 *billion* photons were used to determine the VF matrix. The inner walls of the receiver were divided into a number of discrete meshes ($N_{surf}$). The number of nodes in radial, axial and circumferential directions were set to 50, 20, and 20 respectively. The insulation is also divided into 20 layers for solving the heat conduction. Then, the radiosity method was used to get the net irradiation flux at the cavity bottom by solving the set of equations below (Yang et al., 2018):

$$\sum_{j=1}^{N_{\text{surf}}}\left(\delta_{kj}-(1-\varepsilon_j)F_{kj}\right)\frac{q_{net,j}}{\varepsilon_j}=q_{inc,k}-\sum_{j=1}^{N_{\text{surf}}}(\delta_{kj}-F_{kj})\sigma T_j^4 \tag{1}$$

where $q_{inc,j}$ and $q_{net,j}$ represent the incident solar irradiation and the net radiative heat flux at the $j^{th}$ segment. $\varepsilon$ is the emissivity, $\sigma$ is Stefan-Boltzmann constant ($5.6704\times10^{-8}$ W/m$^2$K$^4$), $\delta$ is the Dirac delta function, and $F_{kj}$ is the VF from the $k^{th}$ to the $j^{th}$ segment.

In our case, the incident solar irradiation can just cover the cavity bottom with the special geometric design. It is assumed to obey a 2-D Gaussian distribution or 1-D Rayleigh distribution (Eq. (2)) with some constraints (Eq. (3)):

$$\frac{q_{inc}(r,\varphi)}{q_0}=e^{-\frac{r^2}{2\sigma^2}} \quad (0\le r\le\infty, 0\le\varphi\le 2\pi) \tag{2}$$

$$\frac{q_{inc}(R_2,\varphi_0)}{q_{inc}(0,\varphi_0)}=\frac{1}{e^2}; \quad and \quad \overline{q}_{inc}=\frac{\int_0^{2\pi}\int_0^{R_2}q_{inc}(r,\varphi)rdrd\varphi}{\pi R_2^{\,2}} \tag{3}$$

where $q_0$ is the peak value of $q_{inc}$ ($r$=0 m) and $\overline{q}_{inc}$ represents the average incident solar irradiation, set as $4\times10^4$ W/m$^2$ here. Thus, $\sigma=\frac{R_2}{2}$ and $q_0=\frac{R_2^{\,2}\overline{q}_{inc}}{2\sigma^2\left(1-e^{-\frac{R_2^{\,2}}{2\sigma^2}}\right)}$.

To match the scale of $\overline{q}_{inc}$, the HTF's mass flow rates of charging ($\dot{m}_{1c}$) and discharging ($\dot{m}_{1d}$) are set as 0.4 kg/s and 0.2 kg/s, respectively. The operating temperature in the interesting range of 500~800°C can be fixed by optimizing the circulating mass flow rate ($\dot{m}_{2c}$).

The concentration ratio ($C$) of 1000 *suns* was chosen based on the technical status of beam down systems (Wei et al., 2013). The rim angle of the concentrating sunlight ($\alpha$) is set as 53°. $DNI$=800 W/m$^2$. Therefore, $H_1 = \frac{R_2}{\tan\alpha}$ and $R_1 = R_2\sqrt{\frac{\bar{q}_{inc}}{C \cdot DNI}}$. $H_2$ is fixed as the minimum height ensuring the temperature of the bed's bottom close to the surroundings, with condition $\left(T_{s,f}(z=H_2)-T_\infty\right)_{\max}$<10 K. The dimensions of the IRS are shown in Table 1.

The net radiative heat flux ($q_{net}$) in Eq. (1) is calculated sequentially ~~corresponding~~ according to the temperature of the cavity bottom, i.e. the surface of the storage, varying from the initial value ($T_0$) to the maximum ($T_{max}$) ~~which it can reach~~ corresponding to the initial and steady states. Since the $q_{net}$ consists of two parts, the heat flux losses ~~through conduction, convection,~~ from the cavity walls ($q_{loss}$), and the energy absorbed by the cavity bottom ($q_{absorb}$). Therefore, the $q_{absorb}$ can be obtained from $q_{net}$ minus $q_{loss}$ and finally be fitted as a 3-order polynomial function of the temperature (Eq. (4)):

$$q_{absorb}(T) = AT^3 + BT^2 + CT + D \tag{4}$$

$A$, $B$, $C$ and $D$ are equal to -1.048×10$^{-5}$, 0.008696, -4.975 and 3.943×10$^4$ in this case.

*3.2 Heat transfer model for the packed bed*

Next, a quasi 1-D two-phase numerical transient heat transfer model for the storage is presented. Air and solid phase are separately modelled in the same 1D space based on the law of energy conservation (Eqs. 5 and 6):

Solid phase: $$\frac{\partial((1-\varepsilon)\rho_s c_{ps} T_s)}{\partial t} = \frac{\partial}{\partial z}(k_{eff}\frac{\partial T_s}{\partial z}) + h_v(T_f - T_s) + q_v\delta(z-0) \tag{5}$$

Fluid phase: $$\frac{\partial(\varepsilon\rho_f c_{pf} T_f)}{\partial t} + \frac{\partial(c_{pf}\dot{m}_f T_f)}{\partial z} = h_v(T_s - T_f) + U_w C_w (T_\infty - T_f) \tag{6}$$

Boundary conditions:

Charging :

$$\begin{aligned}
&T_f(t>0,z=0)=f^{-1}\left[\frac{h_{f,0}+\gamma h_f\left(t>0,z=H_3\right)}{1+\gamma}\right];\\
&\frac{\partial T_f(t>0,z=H_2)}{\partial z}=0;\\
&q_v=-\frac{dq_{absorb}}{dz};\\
&\frac{\partial T_s(t>0,z=H_2)}{\partial z}=0.
\end{aligned} \tag{7}$$

Discharging:

$$\begin{aligned}
&T_f(t>0,z=H_2)=T_0;\\
&\frac{\partial T_f(t>0,z=0)}{\partial z}=0;\\
&\frac{\partial T_s(t>0,z=0)}{\partial z}=0;\\
&\frac{\partial T_s(t>0,z=H_2)}{\partial z}=0.
\end{aligned} \tag{8}$$

Initial conditions: $T_f(t=0,0<z<H_2)=T_s(t=0,0<z<H_2)=T_0.$ (9)

Equation (5) and (6) are discretized with the Euler explicit method in time and with the second order central difference in space and can then be written as follows:

$$\begin{aligned}
&(1-\varepsilon)\rho_s c_{ps,n}^i\frac{T_{s,n}^{i+1}-T_{s,n}^i}{\Delta t}\\
&=\frac{k_{eff,n}^i\left(T_{s,n+1}^i-T_{s,n}^i\right)-k_{eff,n-1}^i\left(T_{s,n}^i-T_{s,n-1}^i\right)}{\Delta z^2}+h_{v,n}^i(T_{f,n}^i-T_{s,n}^i)+q_{v,n}^i
\end{aligned} \tag{10}$$

$$\varepsilon\rho_{f,n}^i c_{pf,n}^i\frac{T_{f,n}^{i+1}-T_{f,n}^i}{\Delta t}+\dot{m}_f c_{pf,n}^i\frac{T_{f,n}^i-T_{f,n-1}^i}{\Delta z}=h_{v,n}^i(T_{s,n}^i-T_{f,n}^i)+U_{w,n}^i C_w(T_\infty-T_{f,n}^i) \tag{11}$$

An optimal grid spacing of 0.066 m is chosen as this gave a good accuracy with relatively low computing time. Compared to the fine grid spacing of 0.006 m, the relative variance (Eq. (12)) is less than $10^{-3}$.

$$\frac{\sum_{Layer}\left(result\big|_{Optimal\,grid}-result\big|_{Fine\,grid}\right)^2}{\sum_{Layer}\left(result\big|_{Fine\,grid}\right)^2}<10^{-3} \tag{12}$$

where result represents the solid or fluid temperature of each layer after charging or discharging.

Numerical stability was ensured by two criteria of the solid and air phases given in Eq. (13). A time step of 0.01 s (charging) and 0.02 (discharging) is used to ensure stability.

$$\begin{cases} 1-\dfrac{\left(k_{eff,n}^{i}+k_{eff,n-1}^{i}\right)\Delta t}{(1-\varepsilon)\rho_s c_{ps,n}^{i}\Delta z^2}-\dfrac{h_{v,n}^{i}\Delta t}{(1-\varepsilon)\rho_s c_{ps,n}^{i}}>0 \\ 1-\dfrac{\dot{m}_f\Delta t}{\varepsilon\rho_{f,n}^{i}\Delta z}-\dfrac{h_{v,n}^{i}\Delta t}{\varepsilon\rho_{f,n}^{i}c_{pf,n}^{i}}>0 \end{cases} \tag{13}$$

In Eq. (7), the specific enthalpy of the fluid phase is defined as $h_f=\int_{T_{ref}}^{T}c_{pf}(T)dT$, and the temperature of the mixed input air during charging is calculated from $T_f=f^{-1}(h_f)$.

3.2.1 Effective conductivity of packed bed ($k_{eff}$)

Because of the high temperature of charging and temperature gradient in the axial direction, the correlation of Kunii and Smith (Kunii and Smith, 1960; Yagi and Kunii, 1957) in Eq. (10) is applied to calculate the effective conductivity of packed bed, $k_{eff}$, which considers the thermal conductivity of both the solid and the fluid, as well as the radiative transfer, including the void to void radiative heat transfer coefficient ($h_{rv}$) and the solid surface to solid surface radiative heat transfer coefficient ( $h_{rs}$):

$$k_{eff} = k_f \left[ \varepsilon \left( 1 + \beta \frac{h_{rv} d}{k_f} \right) + \frac{\beta(1-\varepsilon)}{\dfrac{1}{\dfrac{1}{\phi} + \dfrac{h_{rs} d}{k_f}} + \dfrac{2}{3}\left(\dfrac{k_f}{k_s}\right)} \right]; \tag{10}$$

where: $\phi = \phi_2 + (\phi_1 - \phi_2)\dfrac{\varepsilon - \varepsilon_2}{\varepsilon_1 - \varepsilon_2} \quad (\varepsilon_1 = 0.476, \varepsilon_2 = 0.26)$,

$$\phi_i = \frac{1}{2} \frac{\dfrac{k_s - k_f}{k_s} \dfrac{1}{n_i}}{\ln\left( \dfrac{k_s}{k_f} - \dfrac{k_s - k_f}{k_f} \sqrt{1 - \dfrac{1}{n_i}} \right) - \dfrac{k_s - k_f}{k_s}\left( 1 - \sqrt{1 - \dfrac{1}{n_i}} \right)} - \frac{2}{3}\frac{k_f}{k_s} \quad (n_1 = 1.5, n_2 = 4\sqrt{3}),$$

$$h_{rv} = \frac{3.4424 \sigma T_f^3}{1 + \dfrac{\varepsilon}{2(1-\varepsilon)} \dfrac{1 - \acute{U}_s}{\acute{U}_s}}, \quad h_{rs} = 3.4424 \left( \frac{\acute{U}_s}{2 - \acute{U}_s} \right) \sigma T_s^3.$$

$\beta$ is constant equal to 0.9 in our case. Note that the non-uniform radial distribution or "the wall effect" can be neglected due to the large tank to rock diameter ratio (>40) (Meier et al., 1991).

3.2.2 Volumetric solid-fluid convective heat transfer coefficients ($h_v$)

Various rock-to-fluid convective heat transfer correlations have been proposed for different flow conditions (Alanis et al., 1977; Coutier and Farber, 1982; Löf and Hawley, 1948; Pfeffer, 1964). It has been found that $h_v$ may significantly affect the final results. Therefore, the correlations should be chosen with care corresponding to the operating conditions in each case. In our case the model by Alanis et al. and Coutier & Farber (Eq. (11)) was found proper:

$$h_v = l_m (\dot{m} / d)^{l_n} \tag{11}$$

where $l_m$ and $l_n$ depends on the Reynolds number (Table 3).

Table 3. Coefficients for volumetric convective heat transfer correlation.

| | $l_m$ | $l_n$ |
|---|---|---|
| Small Reynolds number (<50) (Coutier and Farber, 1982) | 700 | 0.76 |
| Large Reynolds number (50–400) (Alanis et al., 1977) | 824 | 0.92 |

3.2.3 Overall wall heat transfer coefficient ($U_w$)

For the lateral insulation of tank, the overall heat transfer coefficient is calculated as:

$$U_w = \frac{1}{\frac{1}{U_{inside}} + R_2 \sum_{j=1}^{n} \frac{1}{k_j} \ln \frac{r_{j+1}}{r_j}} \tag{12}$$

$$U_{inside} = h_{conv,w} + h_{rad-cond,w} \tag{13}$$

$$h_{conv,w} = \frac{k_f}{d}\left[3.22(\mathrm{Re}\,\mathrm{Pr})^{1/3} + 0.117\,\mathrm{Re}^{4/5}\,\mathrm{Pr}^{2/5}\right] \tag{14}$$

$$h_{rad-cond,w} = \frac{k_{eff,w}}{R_2 \ln \frac{R_2}{R_2 - d/2}} \tag{15}$$

The insulation consists of Microtherm®, Foamglas®, UPC, LDC, and soil with thicknesses given in Table 1 and conductivities in Table 2. $1/U_{inside}$ represents the heat resistance between the packed bed (including the fluid) and inner laterals wall, which mainly consists of the convective effect from the fluid phase and the radiative-conductive effect from the solid phase. $h_{conv,w}$ and $h_{rad-cond,w}$ accounts for the convection and the radiation-conduction terms calculated with Eq. (14) (Beek, 1962) and Eq. (15), correspondingly. *Re* in Eq. (14) is calculated by $\dot{m}d/(\rho\nu)_f$. $k_{eff,w}$ in Eq. (15) is obtained from correlations given by (Ofuchi and Kunii, 1965) which is similar to Eq. (10).

### *3.3. Validation of the models*

The thermal model for cavity receiver has been validated in our previous work (Yang et al., 2018). In this paper we therefore focus on demonstrating the validity of the heat transfer model for the packed bed. The case of an industrial thermal storage system for a 26 MWe CSP plant in Morocco is used here for the validation. This case differs from our design in that the storage is charged by hot air instead of solar irradiation. The dimensions and operating conditions are given in Table 4. The rest of the parameters are the same as in our case (Table 1). We simulated a 30-days period with our in-house code and compared the results to Zanganeh's numerical results which have been verified against experiments (Zanganeh et al., 2012). Figure 2 presents the non-dimensional temperature (Θ) of outlet air after each cycle and the solid phase temperature distribution vs height after charging and discharging of 1,10, 20, and 30 cycles (1 cycle=1day). The agreement of the results is good.

Table 4. Main parameters of the case for the validation of the model.

| **Dimensions** | | **Operating conditions** | |
|---|---|---|---|
| $R_{top}$ (m) | 20 | $\dot{m}_c$ (kg/s) | 132 |
| $R_{bottom}$ (m) | 16 | $\dot{m}_d$ (kg/s) | 66 |
| $H$ (m) | 25 | $T_c$ (°C) | 650 |
| $d$ (m) | 0.03 | $T_d$ (°C) | 150 |

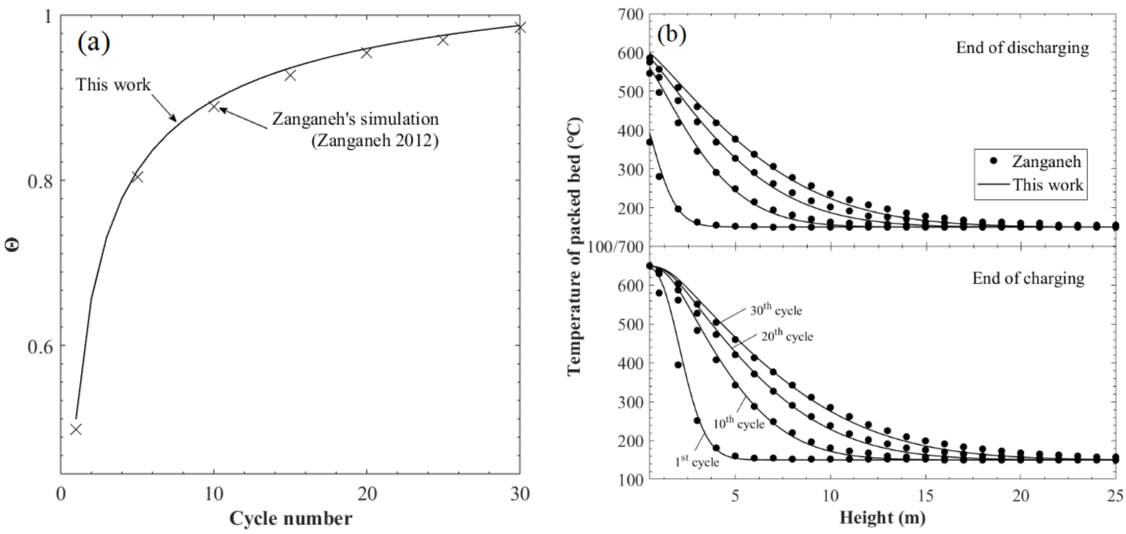


Figure 2. Validation of the in-house model to Zanganeh [6]. (a) Nondimensional temperature ( $\Theta = \dfrac{T - T_d}{T_c - T_d}$ ) of outlet air after each cycle. (b) The solid phase temperature distribution vs height after charging and discharging of 1, 10, 20, and 30 cycles.

## 4. Results and discussion

The performance of the ~~new~~ IRS system is assessed in the following aspects ~~a range of criteria~~: spatial thermal stratification (STS), transient outlet air temperature during discharging ($T_{outlet}$), cavity absorbing heat efficiency ($\eta_{absorb}$), charging and discharging efficiencies ($\eta_{charging}$, $\eta_{discharging}$), and the total solar-to-exergy conversion ratio ($\xi_{cycle}$). The results are presented in detail in the next.

### *4.1 Thermal characteristics of the IRS*

In the proposed design, the incident beam directly strikes on the first layer of the packed bed. Air is forced to sweep through the bed from the top towards the down carrying heat to the lower layers. The thermocline of the IRS after the charging and discharging phases is illustrated in Fig 3. It corresponds to the 15th cycle with $\gamma$=6 and $H_3$=0.33 m.

The temperature distribution of the packed bed after discharging is similar to that of the fluid phase fitting the general Gaussian equation (Fig. 3a). The temperature decays in axial direction ($z$) as $e^{-z^2}$. In the solid phase, the highest temperature is reached at the top due to the heating effect of direct solar irradiation. The temperature drops when moving downwards and approaches an inflection point at $z<H_3$. The rest of the thermocline follows the Gaussian distribution. For the fluid phase, the temperature in the first layer is lower and increases to the peak then gradually decreases with z. The distribution is similar to that of the packed bed when $z>H_3$. The ~~critical~~ point (marked with a dotted circle in Fig. 3b) is ~~also~~ called the equilibrium point (EP), where the temperature of two phases are identical. Above the EP, the air is at relatively lower temperature than the packed-bed and it thus absorbs heat from the packed bed. Inversely, the heat is released from air and stored in the bed below the EP.

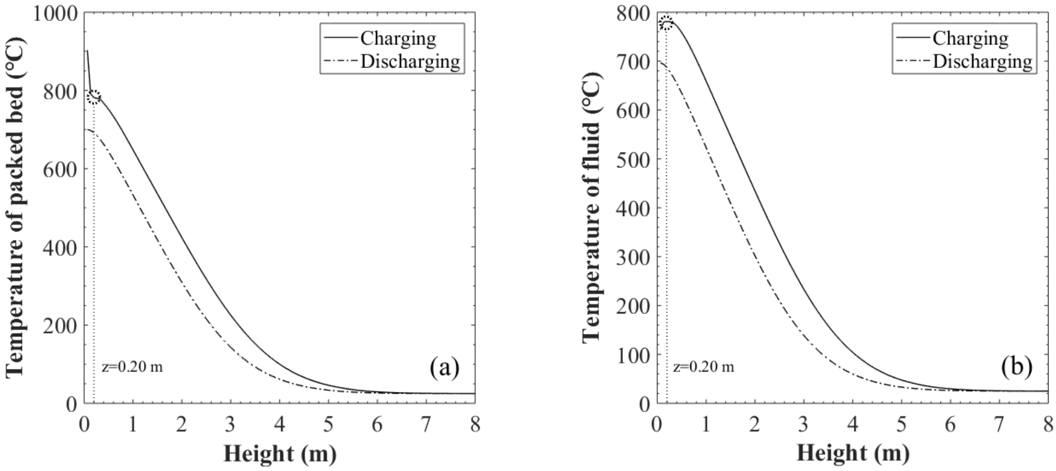


Figure 3. Thermocline distribution of (a) packed bed, (b) fluid, as the function of temperature vs height after charging\discharging. The equilibrium point (EP) is at $z$=0.20 m and $T$=781°C is marked with a dotted circle in the figure.

The temperature evolution of 1st layer of the packed bed ($T_{layer1}$) is also presented for demonstrating the charging-discharging performance during multi-cycle performance. Figure 4 depicts the curve of $T_{layer1}$ against time within 15 cycles. The mean of $T_{layer1}$ is also shown. The triangle and circle markers correspond to the charging and discharging phases, respectively. The oscillation of temperature is rather intense at the start-up stage but then decreases rapidly during the first ~ 5 cycles. After 30 cycles, the mean temperature of the charging and discharging periods approach steady-state values of 884°C and 766°C in this case.

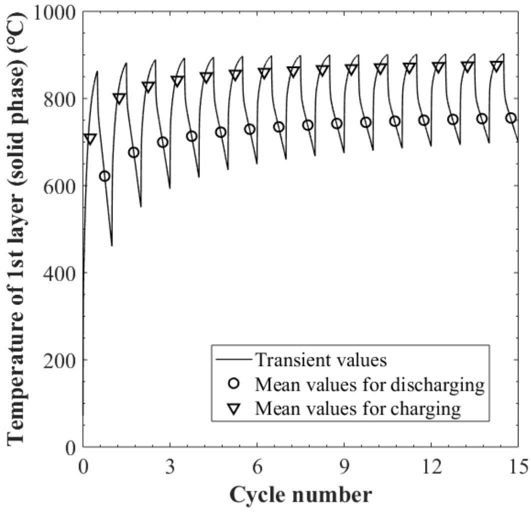


Figure 4. The temperature evolution of the 1st layer of the packed bed during a multi-cycle simulation.

*4.2 Reheat effect*

In the present design, air fan $M_2$ plays an important role on the performance of the thermal storage due to the reheat effect (Fig. 1) as a certain amount of air flow can be reheated through the reflow. Next, we mainly discuss the impacts of the circulation flow ratio ($\gamma$) and the circulation flow length ($H_3$). Table 5 shows the effects of these parameters; a more detailed discussion in given in section 4.2.1 and 4.2.2.

Table 5. Amount of solar ~~heat~~ energy absorbed and outlet temperature for the different values of $\gamma$ and $H_3$.

| $\gamma$ ($H_3$=0.33 m) | 0 | 2 | 4 | 6 | 8 | 10 |
|---|---|---|---|---|---|---|
| $Q_{absorb}$ (MWh) | 2.35 | 2.77 | 2.87 | 2.92 | 2.95 | 2.97 |

| | | | | | | |
|---|---|---|---|---|---|---|
| $\overline{T}_{outlet}$ (°C) | 652 | 735 | 754 | 764 | 770 | 774 |
| $\Delta T_{outlet}$ (°C) | 277 | 160 | 131 | 116 | 107 | 100 |

| $H_3$ (m) ($\gamma$=4) | 0.33 | 0.66 | 0.99 | 1.32 | 1.65 |
|---|---|---|---|---|---|
| $Q_{absorb}$ (MWh) | 2.87 | 2.89 | 2.89 | 2.90 | 2.90 |
| $\overline{T}_{outlet}$ (°C) | 754 | 759 | 760 | 758 | 757 |
| $\Delta T_{outlet}$ (°C) | 131 | 102 | 84 | 77 | 74 |

### 4.2.1 Effect of circulation flow ratio ( $\gamma$)

The amount of solar ~~heat~~ thermal energy absorbed by the storage ($Q_{absorb}$), the outlet flow temperature ($T_{outlet,min\backslash max}$, $\Delta T_{outlet}$ and $\overline{T}_{outlet}$) and the relative temperature deviation ( $\sigma_T = \frac{T_{outlet} - \overline{T}_{outlet}}{\overline{T}_{outlet}} \times 100\%$) during discharging are considered here for assessing the impact from $\gamma$. Here $T_{outlet,min\backslash max}$, $\Delta T_{outlet}$ and $\overline{T}_{outlet}$ represent the extremes, the difference, and the integral mean value of the outlet flow temperature, respectively. $H_3$ is set as 0.33 m and $\gamma$ varies from 0 to 10 as shown in Table 5. Fig. 5a shows the simulated results at the end of the 30$^{th}$ cycle. $Q_{absorb}$ and $\overline{T}_{outlet}$are relatively low (2.35 MWh and 653°C) when $\gamma$=0. A clear improvement is found at $\gamma$ ~6. The temperature gap ($\Delta T_{outlet}$) decreases as well. Beyond $\gamma$ = 6 the values stabilize explained by the heat transfer enhancement between the packed bed and air when using the reflowing air fan $M_2$. Thus, a lower $\Delta T_{outlet}$ and a higher $\overline{T}_{outlet}$ can be reached. Meanwhile, for the reason, $|\sigma_T|$ can be also reduced at the later stage of discharging. Fig. 6a presents $|\sigma_T|$ drops down to 5.2% from 10.2% at the end of the 30$^{th}$ cycle when increasing $\gamma$ from 0 to 8. Due to the limitations in the geometry and materials, the thermal storage approaches this optimal stage after increasing $\gamma$.

### 4.2.2 Effect of circulation flow length ($H_3$)

Next we consider the effects of the circulation flow length ($H_3$) by fixing $\gamma$=4 and varying $H_3$ from 0.33 to 1.65 m. The simulation results of the 30$^{th}$ cycle are given in Fig. 5b. The effect of $H_3$ on the amount of absorbed heat is very limited. $Q_{absorb}$ increases by 1.1% only when quintupling $H_3$. However, $\Delta T_{outlet}$ is furthered decreased from 131 to 74°C as the circulation length is broadened from 0.33 m to 1.65 m under the same flow ratio value. $\overline{T}_{outlet}$ barely changes with $H_3$. The peak value 760°C is obtained with $H_3$=1.32 m. Note that $T_{outlet,min}$ gets closer to $\overline{T}_{outlet}$ when increasing $H_3$, because attenuation of $T_{outlet}$ at later stage of discharging improves. The $|\sigma_T|$ is <2.5% when $H_3$>0.99 m (Fig. 6b), which is crucial to the operation of the power block. The effect is positively correlated to $H_3$ and stabilizes after $H_3$=$H_2$/8.

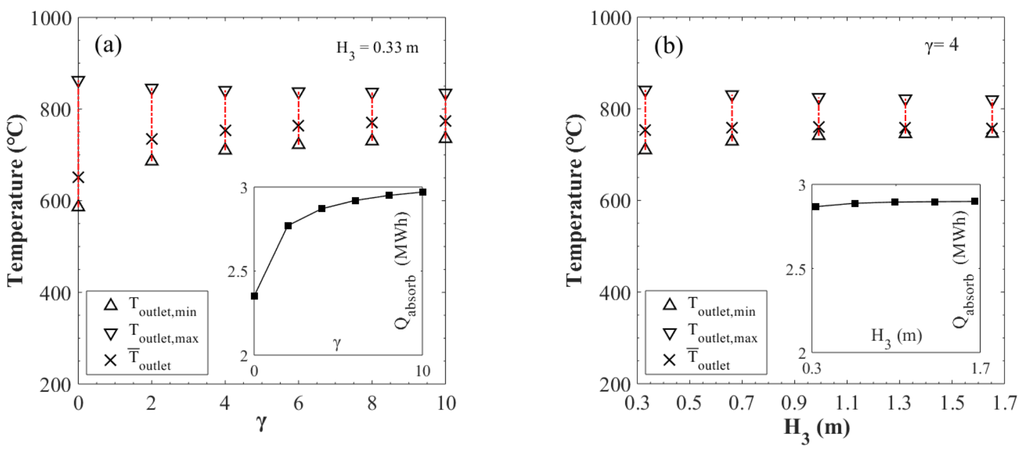


Figure 5. Effects on outlet temperature ($T_{outlet,min}$, $T_{outlet,max}$, $\overline{T}_{outlet}$) and the absorbed amount of solar heat ($Q_{absorb}$) during the discharging phase of the 30$^{th}$ cycle by varying (a) circulation flow ratio, $\gamma$; (b) circulation flow length, $H_3$;. The dashed red line depicts the gap between the maximum and the minimum outlet temperature ($\Delta T_{outlet}$).

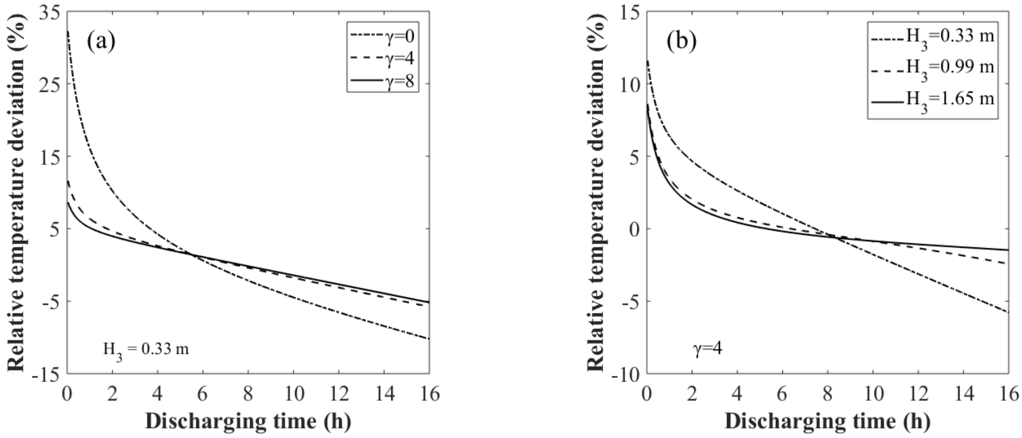


Figure 6. The relative temperature deviation ( $\sigma_T$) during discharging of the 30th cycle under the fixed conditions: (a) $H_3$=0.33 m, (b) $\gamma$=4.

Summarizing, the performance of the thermal storage can be improved either through increasing the circulation flow rate or the circulation flow range. But, meanwhile, the room for improvement is also limited by the other factors such as the geometry and materials. Eventually, the limit is supposed to be approached when the values of $\gamma$ and $H_3$ are large enough, corresponding to 6 and 1 m in this case. It will be further discussed in terms of the local efficiencies and total conversion ratio in the last section.

*4.3 Efficiency of the IRS*

The amount of heat obtained during absorbing, charging, and discharging can be calculated from the following equations:

$$Q_{absorb} = \int_0^{t_c} \pi R_2^2 q_{absorb} dt \tag{16}$$

$$Q_{charging} = \int_0^{t_c} \int_0^{H_2} \pi R_2^2 (1-\varepsilon) \rho_s c_{ps} T_s dz dt \tag{17}$$

$$Q_{discharging} = \int_0^{t_d} \int_{T_{inlet}}^{T_{outlet}} \dot{m}_d c_{pf} dT dt \tag{18}$$

Then, the thermal efficiencies and the solar-to-exergy conversion ratio can be defined by Eqs. (19)- (22):

$$\eta_{absorb} = \frac{Q_{absorb}}{\dot{Q}_{inc} \cdot t_c} \tag{19}$$

$$\eta_{charging} = \frac{Q_{charging}}{Q_{absorb}} \tag{20}$$

$$\eta_{discharging} = \frac{Q_{discharging}}{Q_{charging}} \tag{21}$$

$$\xi_{cycle} = \frac{\int_0^{t_d} \left( \int_{T_{inlet}}^{T_{outlet}} \dot{m}_d c_{pf} dT \right) \left( 1 - \frac{T_\infty}{T_{outlet}} \right) dt}{\dot{Q}_{inc} \cdot t_c} \tag{22}$$

Figure 7 illustrates the these parameters during a 30-days' operation. For the start-up phase, $\eta_{discharging}$ is quite low, but after 10 cycles it rapidly climbs from 46.8% to 86.6%. As a result, the exergic conversion ratio improves from 0.25 to 0.43. However, $\eta_{absorb}$ and $\eta_{charging}$ still drop during this interval, which is mainly because the heat loss through the aperture and the insulation is increasing as thermal energy is gradually accumulating and stored in the packed bed. Eventually, the storage approaches a steady cyclic behavior after 30 cycles and the $\xi_{cycle}$ is >0.52. The $\eta_{absorb}$, $\eta_{charging}$ and $\eta_{discharging}$ are then equal to 79.6%, 99.2%, and 92.6%, respectively.

The energy needed for the air fans ($Q_{fans}$) is calculated in Eq. (23). The derivative of the pressure versus height in the packed bed ( $\frac{dp}{dz}$) refers to the Ergun equation (Ergun and Orning, 1949) modified with a buoyancy term (Andersen, 2003). The results indicate that $Q_{fans}$ is an order of magnitude lower than the thermal heat loss though the walls.

Hence, it can be neglected in the analysis.

$$Q_{fans} = \frac{\int_0^{t_c+t_d} \int_0^{H_2} \left| \frac{dp}{dz} \right| \frac{\dot{m}}{\rho_f} dzdt}{\eta_{fan} \cdot \eta_{CSP}} \tag{23}$$

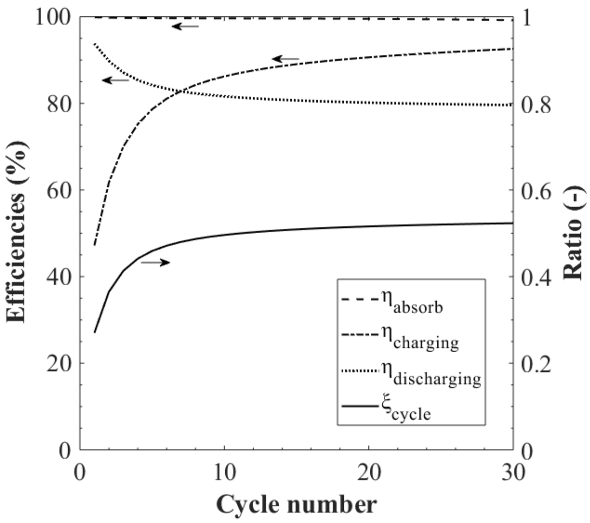


Figure 7. Thermal efficiencies and solar-to-exergy conversion ratio during a month of operation with $\gamma$=6, $H_3$=1 m.

*4.4 Comparison of receiver and storage to existing CSP plants*

Finally, we compare our IRS to existing CSP plants: the Solar One in California (Kolb et al., 1991), the 100 MWh$_{th}$ TES system in Ait Baha (Zanganeh et al., 2014c), and the CSPonD in Masdar (Gil et al., 2017), based on our results. The comparison is done against $\eta_{absorb}$, $\eta_{charging}$, $\eta_{discharging}$, and $\xi_{cycle}$ shown in Table 6. Similar to our case, these three plants employed a thermocline single-tank for the storage design. We found that the IRS performs well and has actually the highest absorbing efficiency and solar-to-exergy conversion ratio as well as a good storage efficiency ( $\eta_{storage} = \eta_{charging} \cdot \eta_{discharging}$).

The advantages of the modified IRS receiver and storage compared to existing CSP systems can be summarized as follows:

- Direct absorption of the solar irradiation is more efficient;
- The structure of IRS is simplified by eliminating the conventional receiver, storage system and heat exchanger;
- During charging, the descending airflow formed in the cavity can prevent the convective heat loss through the aperture;
- A higher temperature (>700°C) is achieved enabling the use of more efficient thermal engines, e.g. the Brayton or Stirling cycles;
- Circulation air flow can improve the heat transfer rate, uniform the temperature distribution in the upper part thus improving the attenuation of outlet temperature during discharging.

Table 6. Comparison of IRS to three existing CSP systems

<table>
<tr><th rowspan="2"></th><th rowspan="2">$\bar{T}_{outlet}$</th><th rowspan="2">$\eta_{absorb}$</th><th>$\eta_{charging}$</th><th>$\eta_{discharging}$</th><th rowspan="2">$\xi_{cycle}$</th></tr>
<tr><th colspan="2">$\eta_{storage}$</th></tr>
<tr><td rowspan="2">IRS (this study)</td><td rowspan="2">760°C</td><td rowspan="2">80%</td><td>99%</td><td>93%</td><td rowspan="2">0.52</td></tr>
<tr><td colspan="2">92%</td></tr>
<tr><td rowspan="2">Solar One (DeLaquil et al., 1991; Kolb et al., 1991)</td><td rowspan="2">566°C</td><td rowspan="2">69%</td><td>N/A</td><td>N/A</td><td rowspan="2">0.42</td></tr>
<tr><td colspan="2">90%</td></tr>
<tr><td rowspan="2">Ait Baha (Zanganeh et al., 2014a; Zanganeh et al., 2014c)</td><td rowspan="2">560°C</td><td rowspan="2">77%</td><td>98%</td><td>91%</td><td rowspan="2">0.46</td></tr>
<tr><td colspan="2">89%</td></tr>
<tr><td rowspan="2">CSPonD (Gil et al., 2017; Slocum et al., 2011)</td><td rowspan="2">550°C</td><td rowspan="2">75%</td><td>N/A</td><td>N/A</td><td rowspan="2">0.48</td></tr>
<tr><td colspan="2">95.5%</td></tr>
</table>

**Conclusions**

In this paper, we proposed a modified integrated receiver-storage (IRS) system for a beam-down CSP plant. The structure is based on a cylindrical packed rock-bed storage.

Unlike ~~current~~ conventional designs, the IRS can directly store solar irradiation without a complicated heat exchange mechanism which leads to a much simplified structure. We developed a combined thermal model for cavity radiation and storage charging-discharging processes which was used in the thermal analysis indicating that the performance of IRS was very satisfying.

The reheating element coupled to the IRS proved to enhance heat transfer in the storage improving the total thermal efficiency by 11% . The outlet temperature gap could be narrowed by a factor of 3 after adding a circulation air fan to the storage, without any major increase in the parasitic losses. The circulation flow ratio and length affect the reheat effect of the IRS. Optimal parameter values for our $1MW_{th}$ design were $\gamma$=6 and $H_3$=1 m.

For the optimal case, a mean temperature of 770℃ was reached and the output temperature gap was within 68℃. No temperature attenuation phenomenon was observed during the later period of discharging. The solar-to-exergy conversion ratio of the IRS (0.52) can be considered good.

The modified IRS shown in this paper is a promising design for future CSP systems. Further work could include assessing the thermo-mechanical stability, improving the absorbing efficiency of the cavity more and analyzing the feasibility of IRS for high temperature thermochemical reaction systems.

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