Thermoelectric energy storage based on CO_2 transcritical cycles: ground heat storage modelling.

Edoardo Gino Macchi[a]*, Catherine Colin[a], Thomas Tartière[b], Denis Nguyen[c], Nicolas Tauveron[d]

[a] Institut de Mécanique des Fluides de Toulouse, Université de Toulouse, 2 Allée du Professeur Camille Soula, 31400 Toulouse, France
[b] Enertime, 1 rue du Moulin des Bruyères, 92400 Courbevoie, France
[c] BRGM Languedoc-Roussillon, 1039 rue de Pinville, 34000 Montpellier, France
[d] CEA, LITEN – DTBH/SBRT/LS2T, 17 rue des Martyrs, 38054 Grenoble, France

E-Mail:*edoardo.macchi@imft.fr

Abstract: We consider an electricity storage concept based on CO_2 transcritical cycles and geothermal heat exchangers. Developing a reliable and possibly simple model of the ground heat storage is fundamental for assessing the global performance of the system and for its optimization. Nevertheless this is not a trivial task due to the significant size of the storage and the different time-scales involved in the process. A physically-based 1D/2D model for the ground heat storage, that simulates both flow and energy transfer inside the heat exchangers and the heat transfer in the ground will be presented. The model will be then used for a preliminary optimization of process and storage parameters.

1. Introduction

As it is well-known, the massive integration of renewable energy sources generates new challenges for the supervision and regulation of electric grids. The variability and unpredictability of these sources conflict with the reliable supply of electricity required by industries and consumers: energy storage is essential to balance supply and demand. Energy storage will also play a key role in enabling to develop a low-carbon electricity system. Despite energy storage is an established technology, there is still limited storage capacity, currently mostly based on pumped-hydro storage plants. Other forms of storage are either minimal, or at a very early stage of development; today's challenge is to increase existing storage capacities and efficiencies.

Recently, thermoelectric energy storage (TEES) systems have been proposed as a new method for large-scale energy storage: electric power is stored as thermal energy using a heat pump and retrieved from the storage using a heat engine. A review of TEES system can be found in [1, 2] while the reader is referred to [3] for an overview of energy storage technologies. Among TEES systems, a thermoelectric storage based on transcritical CO_2 cycles has been recently considered as a promising method for energy storage [1, 2, 4]. A new type of large-scale thermoelectric energy storage based on transcritical CO_2 cycles and a ground heat storage has been recently proposed [2, 5]. Such a system includes a hot ground reservoir in a shallow rock massif made of a large number of vertical heat exchangers (geothermal exchangers), a cold reservoir using either ice or a phase-change material, two thermodynamic cycles (and the corresponding thermodynamic machines) for charging and discharging the reservoirs, both using carbon dioxide as a working fluid. For a description of the thermodynamic cycles please see [2, 5]. In the following sections, after describing the ground heat storage configuration, a physical-based 1D/2D model will be presented. The results of some simulations will be then discussed and a preliminary optimization of process and storage parameters will be performed.

2. Ground heat storage: description and modelling approach

The hot reservoir is made of 2160 identical geothermal exchangers organized in a serial-parallel layout and separated by 0.5 m. Each exchanger is made of rubber and is composed of a central circular pipe (11.8 cm diameter) used for fluid injection and of an annular return pipe (12.2/20 cm inner/outer diameter) in contact with the surrounding rock as showed in Figure 1a,b.

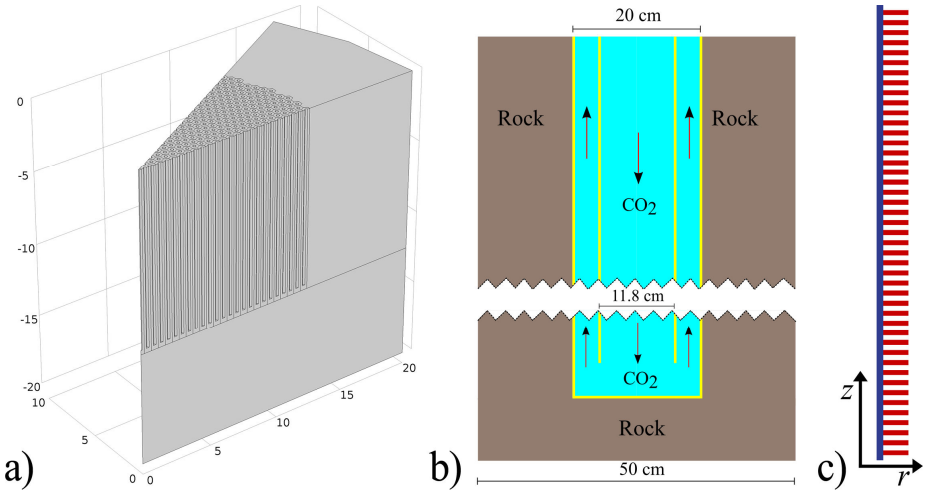

Figure 1: a) Drawing of a 30° sector of the ground heat storage, b) sketch of a single coaxial exchanger, c) 1D/2D computational model (deformed in the radial direction) - fluid domain (blue) and rock domains (red)

It is clear that the hot reservoir is the central part of this storage system but it is also the most complex part to model. A complete CFD study of flow dynamics and heat transfer inside the geothermal exchangers (including the conjugated heat transfer in the rock) is viable only for a single exchanger due to the particularly high Reynolds number (between 10^5 and 10^6) and to the different time-scales of the process (from seconds, for the flow dynamics, to days, for the heat storage behaviour during multiple charge/discharge cycles). However, in order to optimize the system, we need to model at least a series of 45 exchangers (radially distributed in the ground) which is representative of the behaviour of the whole reservoir (made of 48 series). The model must be as physically-based as possible since an in-depth model validation, based on CFD simulations and on an experimental pilot, will be done only for a single exchanger. At the same time, however, we have to introduce some simplifications in order to have a viable and relatively fast model. Keeping this information in mind, in order to model a series of exchangers we decided to resort to a reduced 1D model for the thermo-fluid dynamic behaviour of CO_2 inside the exchangers, coupled with a 2D axisymmetric model for the heat transfer inside the rock. Furthermore in our model the interactions between the exchangers (i.e., heat conduction between the rock surrounding each exchanger) and the heat losses toward the rock surrounding the whole heat storage are assumed to be negligible. The former can be neglected since the temperature difference between adjacent exchangers is small, while the latter are important only during the start-up phase when the temperature of the rock that surrounds the heat storage is different from the temperature of the rock surrounding the peripheral exchangers. Concerning the fluid flow in the exchangers, please note that the flow direction is from the first (placed at the centre of the storage) to the last exchanger during charge while it is the opposite during discharge: in this way the central exchanger is always the hottest and the peripheral one is the coldest. It is important to highlight that in the one-dimensional thermo-fluid dynamic model both the injection pipe and the annular return in contact with the rock are considered: all the pipes are connected and create a single computational domain for the fluid, aligned with the positive vertical direction (see Figure 1c). Due to the intrinsic unsteadiness of the heat storage system, it is fundamental to adopt a transient model for the energy transport (both for fluid and rock). In addition, despite the temporal variation of mass flow rate and pressure is gradual, a transient model has to be used also for the flow dynamics because of the strong coupling between enthalpy, density and velocity.

3 . Mathematical and numerical model

The fluid energy conservation equation is as follows:

$$\frac{\partial(\rho h)}{\partial t}+\frac{1}{A}\frac{\partial(\rho UhA)}{\partial z}=q \qquad (1)$$

where ρ, U and h are respectively density, velocity and enthalpy of the fluid. q is a volumetric energy source/sink that accounts for the heat transfer from/to the rock surrounding each exchanger and A is the pipe cross-sectional area (that is constant in time but not in space). Please note that the heat conduction term, as well as the contributions related to the mechanical energy (rate of change of kinetic energy, rate of change of strain energy, rate of change of potential energy) have been neglected since very small compared to the other terms of the equation. The local volumetric heat source/sink q_i is computed dividing the power transferred to/from the fluid from/to the rock Q_i by the cell volume V_i:

$$q_i = \frac{Q_i}{V_i} = \frac{\Gamma_i S_i \left(T_{w_i} - T_i \right)}{V_i} \tag{2}$$

where Γ_i is the local heat transfer coefficient, S_i is the area of the face through which the heat transfer takes place, T_{w_i} is the wall temperature (equal to the rock temperature at the fluid-rock interface) and T_i is the fluid temperature (computed from enthalpy and pressure). Obviously the local heat source/sink is equal to zero in the cells belonging to the injection pipe. The heat transfer coefficient is computed using the correlations proposed by Kirillov et al. for heating and cooling (during discharge and charge) of supercritical fluids [6]; these correlations include the effect of natural convection.
The fluid mass conservation and momentum balance equation can be stated as:

$$\frac{\partial \rho}{\partial t} + \frac{1}{A}\frac{\partial \left(\rho U A \right)}{\partial z} = 0 \tag{3}$$

$$\frac{\partial \left(\rho U \right)}{\partial t} + \frac{1}{A}\frac{\partial \left(\rho U U A \right)}{\partial z} = -\frac{\partial p}{\partial z} - \frac{f \rho U \left| U \right|}{2 D_h} \tag{4}$$

where p is the fluid pressure, f is the Darcy–Weisbach friction factor and D_h is the pipe hydraulic diameter. For simplicity in Eq. 4 the gravity term has been neglected since the gravitational pressure drop is small compared to the absolute pressure. The last term on the RHS of Eq. 4 is a momentum sink due to friction: the friction factor is computed using the correlations proposed by Fang et al. for single-phase and supercritical flows [7]. Also note that, since the rubber pipe is practically smooth (pipe roughness 2 μm), the pressure change along a series of exchangers is small. Finally, to get a closed set of equations an equation of state (EOS) for the fluid must be specified. The fluid density and temperature are computed as a function of the dependent variables:

$$\rho = \rho \left(h, p \right) \quad ; \quad T = T \left(h, p \right) \tag{5a, 5b}$$

The governing equation for the heat transfer in the rock is a simple transient heat conduction equation with constant coefficients:

$$\rho_R C_{pR} \frac{\partial T_R}{\partial t} = \lambda_R \left[\frac{1}{r}\frac{\partial}{\partial r}\left(r \frac{\partial T_R}{\partial r} \right) + \frac{\partial^2 T_R}{\partial z^2} \right] \tag{6}$$

where T_R, ρ_R, C_{pR} and λ_R are temperature, density, specific heat at constant pressure and thermal conductivity of the rock.
The mathematical model was implemented in OpenFOAM® [8] and the equations were discretised using a second-order accurate finite volume method. The set of Eqs. (1-5) is solved using the PISO algorithm [9] to compute the dependent variables (h, U and p). Fluid temperature and density are determined using the equation of state (Eqs. 5a,b). The rock temperature is computed solving Eq. 6 with appropriate boundary conditions. The thermodynamic properties (EOS) of carbon dioxide are computed using the CoolProp library [10]. The rock is assumed to be granite; its thermophysical properties are as follows: ρ_R = 2650 kg/m^3, C_{pR} = 790 J/kgK, λ_R = 3.4 W/mK.

Concerning the rock domain, a convective heat transfer boundary condition could be applied on the boundary in contact with the fluid, however, in order to ensure the energy conservation for the coupled

domain (fluid + rock), it is better to assign a Neumann boundary condition computed using the power transferred to (from) the fluid from (to) the rock Q_i:

$$-\lambda_R \frac{\partial T_R}{\partial r} = \Gamma\left(T - T_R\right) = -\frac{Q_i}{S_i} \qquad (7)$$

Homogeneous Neumann boundary conditions are applied on the other boundaries of the rock domain. Concerning the fluid, standard inflow/outflow boundary conditions are used for velocity and pressure (specifying mass flow rate/velocity at the inlet and pressure at the outlet). For Eq. 1, the fluid enthalpy is assigned at the inlet and a homogeneous Neumann boundary condition is applied at the outlet of the series of exchangers. A uniform temperature equal to 303.15 K is used to set the initial conditions for both fluid and rock domains. The fluid is assumed to be still at the beginning of each charge/discharge. A sensitivity analysis on mesh and time-step size was performed: for the vertical direction (z) a cell size of 0.5 m is chosen for both fluid and rock domains. The 45 rock domains, extending from r = 0.1 m to r = 0.25 m, are discretised in the radial direction using a non-uniform mesh refined near the rock-fluid interface (25 cells, first and last cell size in the radial direction: 1.8 mm and 14.4 mm). A finer mesh allows obtaining slightly more accurate results, however, since the results do not change sensibly, the above mentioned coarser mesh has been chosen to speed-up the simulations. An adaptive time-stepping procedure based on the maximum allowable Courant number, set to 1.0 for best accuracy, has been adopted.

4. Simulations and results

In the simulations described hereinafter some parameters have been fixed, in particular: number of exchangers in the series (n. 45), CO_2 inlet temperature and outlet pressure during charge (411.15 K and 12 MPa) and discharge (303.15 K and 12 MPa) for the series of exchangers. The initial simulation considered 5 cycles with 6h charge-6h discharge, G = 4 kg/s and 12 m long exchangers. The results show that in these conditions the effects of convection are simply too strong: the outlet temperature is too low during discharge and too high during charge (ideally during charge it should be ~30°C to avoid performance losses) and, as a result, the exergy efficiency η_{ex} is very low (47%). The latter is computed based on the total amount of CO_2 exergy extracted from/sent in the storage:

$$\eta_{ex} = \left(Ex_{out} - Ex_{in}\right)_{\text{discharge}} / \left(Ex_{in} - Ex_{out}\right)_{\text{charge}} \qquad (8)$$

This value gives an estimation of what would be the overall electrical efficiency of the storage system if we had ideal Carnot engines to produce heat (during charge) and electricity (during discharge). Several simulations were then run to evaluate the effect of each process parameter and to perform a preliminary optimization of the heat storage. In the "optimized" simulation we considered: 8h charge with G = 1.4 kg/s and 4h discharge with G = 1.9 kg/s, 30 m long exchangers (18 cycles have been modelled). Some results of this simulation are shown in Figure 2 and 3.

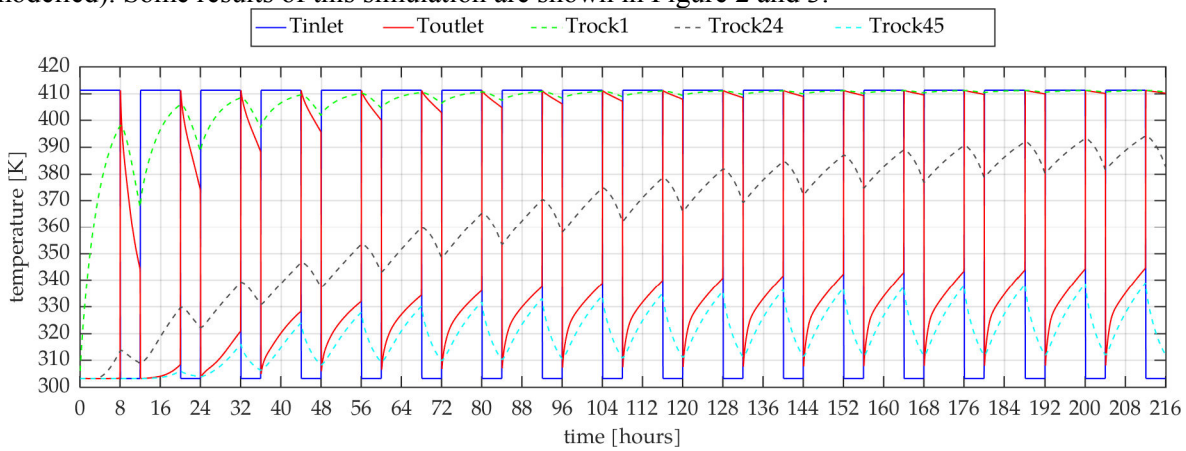

Figure 2: Temporal evolution of inlet and outlet temperature for a series of exchangers (continuous lines) and volume-averaged rock temperature for 1st, 24th and 45th exchanger (dashed lines)

Increasing the exchangers' length, reducing the mass flow rate and using different flow rates during charge and discharge allowed us to increase the exergy efficiency up to 73%. The amount of energy recovered from the storage during a discharge is about 60 MWh for the whole storage (48 lines). In these simulations a daily cycle of utilization of the storage (with relatively long charge and discharge, as in an arbitrage application) was examined. However, keeping in mind that the patterns of access to energy storage are changing, it will be interesting to consider also shorter cycles. The coupling of heat storage model and thermodynamic cycles' models is ongoing; such a tool will allow optimizing the whole energy storage system for different operating conditions.

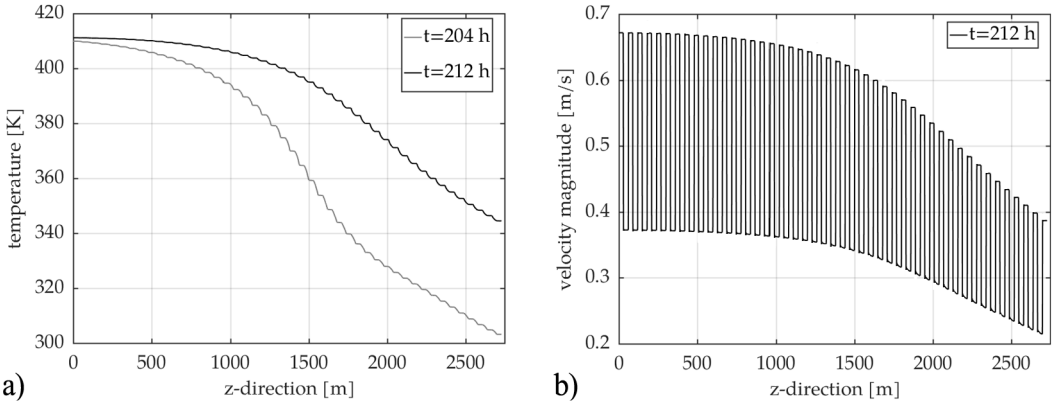

Figure 3: Spatial distribution of CO_2 temperature (a) and velocity (b) along a series of exchangers during charge (212 h, end of last charge) and discharge (204 h, end of second-last discharge)

5. Conclusion

The ground heat storage modelling approach described in this work resorts to geometrical reduction, while introducing some reasonable approximations, in order to make the dynamic simulation of several charge/discharge cycles for the whole heat storage viable. A 1D model is adopted for the thermo-fluid dynamic behaviour of CO_2 inside the exchangers while a 2D axisymmetric model is used for the heat transfer inside the rock. Preliminary simulations show that a suitable optimization of both process and storage parameters is required to increase the exergy efficiency of the proposed ground heat storage to satisfactory values.

Acknowledgements

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