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Dynamic behavior of a sensible-heat based thermal energy storage

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Abstract

In this paper, a mathematical model is developed to study the behavior of thermal energy storage (TES) under operation in the particular case of Adiabatic Compressed Air Energy Storage (A CAES). The A CAES consists of storing the available extra electrical energy of the electricity network in a form of compressed air (in a cavern) to discharge it during peak periods. The TES sub-system is used to charge and discharge the corresponding heat of compression, leading to a quasi adiabatic mode and an increase in the overall electricity storage efficiency (roughly from 50 to 70%) compared to diabatic CAES. The mathematical model has been converted into a computer simulation program with all the effective parameters of heat transfer in the storage reservoir. This model used to define a geometry reservoir able of storing a given power and restore it while maintaining a required temperature level at the output of unit. The influence of the input and output parameters on the storage efficiency is studied. The results illustrate the behavior of the storage reservoir under dynamic mode.

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Keywords: energy storage, heat exchange, recycled storage material, thermal efficiency.

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1. Introduction

The need of an effective and affordable Thermal Energy Storage (TES) technology is emerging as one of the main challenges facing the world today to reach the target of the energy transition. Large-scale, thermal energy storage is an important and a key technology for the development of renewable energy and for the optimization of

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existing processes such as Concentrated Solar Power (CSP) or Compressed Air Energy Storage (CAES) [1]. Most of such large scale industrial TES are based on sensible heat storage (like the so-called two-tanks molten salt storage system for CSP) and processes to which they are dedicated require high efficiencies of storage for optimum performance [2]. Indeed the operation of the two existing industrial facilities to McIntosh (Alabama), which operate over thirty years, Huntorf (Germany), electricity storage in the form of CAES has not seen the development that was expected in the 80s. The efficiency of this form of storage is less than 50% [3]. The evolving context technique can significantly alter this situation. The new generation so-called Adiabatic CAES (A-CAES) is to retrieve the heat produced by the compression via thermal storage, thus eliminating the necessity of gas to burn and would allow consideration efficiency overall energy of the order of 70%. To date, there is no existing installation of A-CAES. Many studies describe the principal and the general working mode of storage systems by adiabatic compression of air. So, efficiencies of different configurations of adiabatic compression process were analyzed [4]. The conceptual designs of the four main components of the system (compressor, heat storage, cave and air turbine) was evaluated in the European research project, some technical risks have been identified in particular for adiabatic compression and heat storage device. The compressor outlet temperature is above 600°C and the resulting heat is stored in the TES consisting by a stack of solid material such as refractory bricks or natural stones [5]. The aim of this paper is to simulate and analyze the performances of a thermal storage reservoir integrated in the system and adapted to the working conditions of A CAES.

Nomenclature

C_{pf}: fluid thermal capacity (J/kg.K)
C_{ps}: solid thermal capacity (J/kg.K)
h: convection coefficient (W/m².K)
L: plates length (m)
L_0: plates width (m)
l: plates stickness (m)
l_e: spacing between plates (m)
t: time (s)
S_f: fluid exchange area (m²)
T_f: fluid temperature (K)
T_s: solid temperature (K)
u_f: fluid velocity (m/s)
V_f : volume of fluid (m^3)
α_f : fluid thermal diffusity (mm²/s)
α_s : solid thermal diffusity (mm²/s)
λ_f : fluid thermal conductivity (W/m.K)
λ_s : solid thermal conductivity (W/m.K)
ρ_f: Fluid density (kg/m^3)
ρ_s: Solid density (kg/m^3)

2. Diabatic compressed air energy storage (CAES)

Compressed air energy storage is a technology for electricity storage developed in recent years. This type of installation is to transfer to the peak demand, energy produced off-peak by power plants using only part of the fuel that would be consumed by a system of classical art as a gas turbine conventional [6,7]. During off-peak periods, available electricity on the network is used to compressed atmospheric air and fill underground caverns. Thereafter, during peak hours, the air flow is extract to the cave and is heated by burning natural gas in the combustion chamber.

Thus the resulting combustion gas is expanded in the turbine to generate electricity. In a conventional gas turbine, the compressor consumes two thirds of the power provided by the combustion. Unlike the CAES that uses electrical energy during off-peak hours to compress air. However the heat from air compression is lost, hence the need to burn gas to reheat the air in the process of turbine operation during peak hours.

CAES is an electricity storage technology known more than 30 years ago and used in large scale. Currently, two installations operating in industrial scale exist and define the CAES. Huntorf 290 MW available 3 hours, opened in Germany since 1978 with an overall efficiency of 42%, the air is compressed and stored in cave between 50 and 70 bars in two salt caverns pressure of 150,000 m3 each to 700 meters deep. McIntosh 110 MW available 26 hours open in the United States (Alabama) since 1991, overall efficiency of 49%. The difference in performance is because the McIntosh plant recovers the turbine fatal energy to reheat the air coming out of the underground cave. The performance of CAES is defined in a particular way. In fact the air is heated in a combustion chamber with natural gas. To produce 1kWh of electricity, 0.7 to 0.8kWh of electricity is needed during off-peak hours and 1.22kWh of natural gas to heat air during peak hours [8].

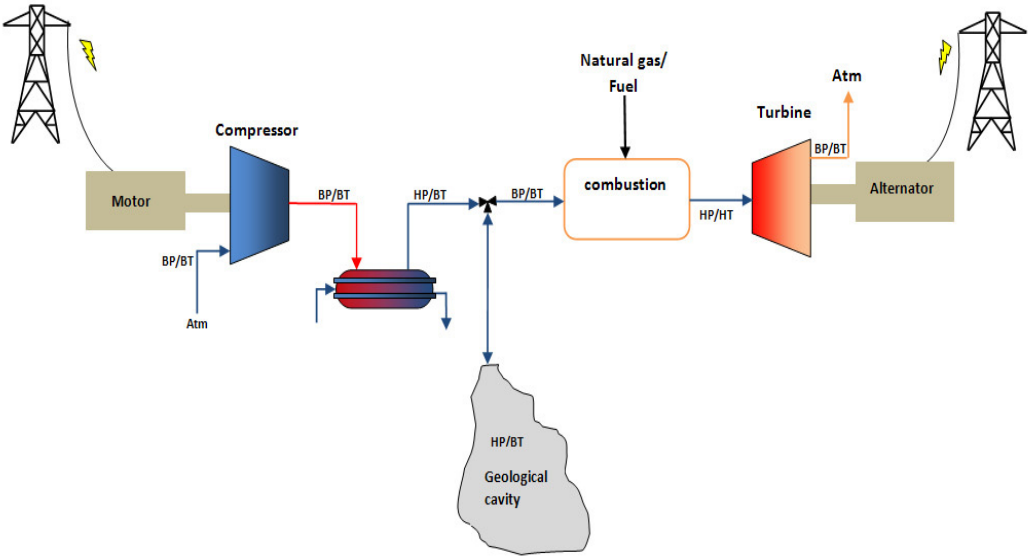

Fig.1: Scheme of diabatic compressed air energy storage (CAES)

3. Adiabatic compressed air energy storage (A CAES)

To ensure the stability and sealing of storage cave, compressed air should not be stored at high temperature. The system efficiency is reduced by air cooling before storage in the cave and the use of combustion chamber to reheat compressed air is one disadvantage of diabatic CAES. The notion of A CAES is to use heat storage unit as the central element of the system. The aim is to store compression heat via thermal energy storage and provide the required heat for expansion process.

In off-peak hours, the air flow from compressor passes through a thermal storage unit and transfer the heat. The compressed air is then stored in underground cave. When electricity is required, during peak hours compressed air passes back through the TES to be reheated. This approach avoids a combustion chamber. A CAES achieves theoretical efficiency of 70% and represents free emission electricity storage technology.

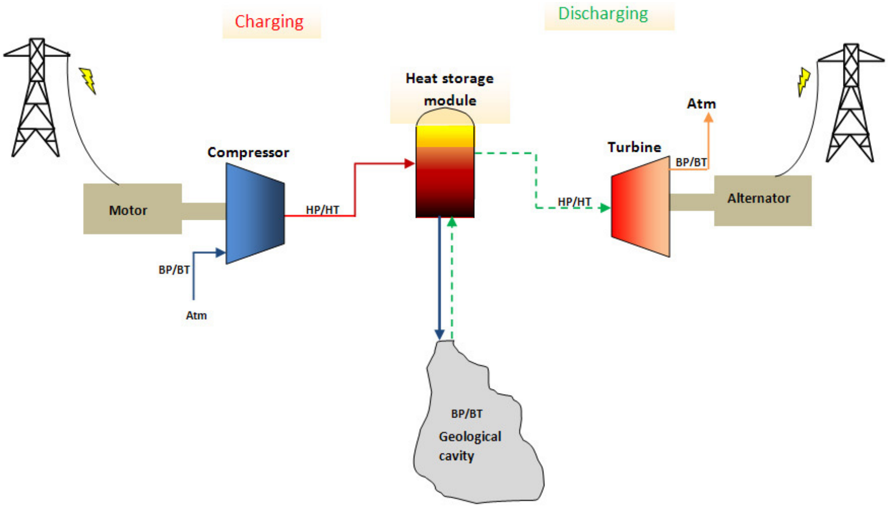

Fig.2: Scheme of adiabatic compressed air energy storage (A CAES)

4. Thermal energy storage

The thermal storage is to store and restore a quantity of energy. Several storage technology exist and are more and less suitable for a given application. In the case of A CAES, energy quantities are important; it comes to a mass storage. The European research project based on 12 to 120 MWth (thermal) for 4 to 12 hours privilege sensible heat storage media such as refractory ceramics or natural stones. Direct contact between the air pressure and the solid media has the advantage of a large heat exchange area unlike the liquid media the cost and complexity of building a heat exchanger liquid systems should be considered.

Mass storage requires large amounts of storage material [9]. Actually it is important to consider the costs of the material on the economic and environmental sides (primary energy content, CO2 content, recyclability, availability, use conflict).

On the technical side, refractory ceramics present all the qualities required for the adiabatic CAES. However, the analysis of the life cycle of these synthetic materials will be quite detrimental, mainly the cost: 4000 to 9000 Euros per tone.

Like a response to environmental issues an original approach is developed recent years to PROMES laboratory. It is the recoverability of materials at the end of life cycle from industrial waste treatment. Indeed, many materials are produced, hundreds of thousands of tons each year worldwide as by-products of industrial processes based on high temperature treatments (eg: steel slag). Others come from the thermal treatment of hazardous industrial wastes destined for the inerting (case of fly ash and asbestos waste). The thermal history of these materials generally gives them a very stable temperature behavior. From their Origins these materials have thermo-physical properties recoverable and availability with attractive prices (8 to 10 Euros per tone).

A first material from asbestos containing waste treatment (vitrification) called Cofalit was selected for thermal storage design in this article. This material at the end of life cycle, with all thermophysical qualities for thermal energy storage has been identified by the Solstock program [10].

4.1. Thermal reservoir sizing

It is about storing the quantity of heat from air compression through a solid media by varying its temperature. The amount of energy to be stored is then directly proportional to the mass of material used, its specific heat capacity and the temperature variation.

4.2. Mathematical model

Modelized storage may be represented as a stacked plate solid with interstices therebetween, crossed by air. Considering the symmetries of the systems, the elementary module representative of the storage unit corresponds to a half thickness of a plate, of length L and associated with a half-flow section for the fluid (Fig.3).

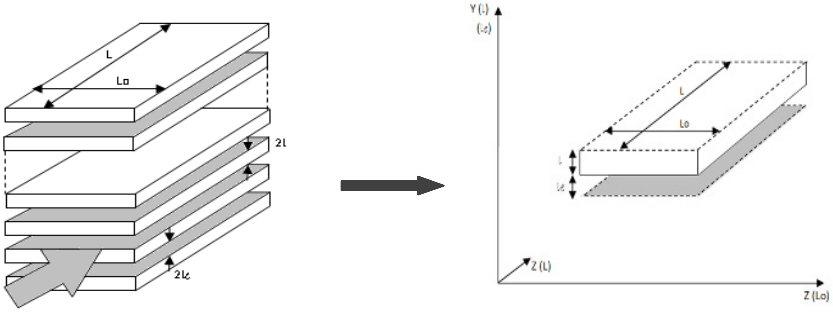

Fig.3: Architecture of the regenerator

This model leads to a two-dimensional discretization in the solid which takes account of possible anisotropy of the storage material through the definition of thermal conductivity in the two directions x and y. For the fluid, it is assumed that there is no temperature gradient in the axis perpendicular to the direction of flow. The problem is in this case one-dimensional with a gradient development in the flow direction.

The system is thermally insulated; the heat exchange takes place only at the interface between fluid and solid. The thermal properties are considered constant to the average temperature of fluid and solid. Equations are:

- **Fluid**

$$\partial_f C_{pf} \frac{\partial T_f}{\partial t} = -\partial_f C_{pf} U_f \frac{\partial T_f}{\partial X} + \lambda_f \frac{\partial^2 T_f}{\partial X^2} + h \frac{S_f}{V_f}(T_f - T_s) \qquad (1)$$

- **Solid**

$$\partial_S C_{pS} \frac{\partial T_S}{\partial t} = \lambda_{SX} \frac{\partial^2 T_f}{\partial X^2} + \lambda_{SY} \frac{\partial^2 T_f}{\partial Y^2} \qquad (2)$$

The initial conditions are:

$$T_{inif} = T_f(t=0) = 300K \qquad (3)$$

$$T_{iniS} = T_S(t=0) = 900K \qquad (4)$$

The boundary conditions are:

- **Fluid**

$$CF_1 : \mathrm{X} = 0; \mathrm{T}_f (X = 0) = T_{inif} \tag{5}$$

$$CF_2 : \mathrm{X} = L; -\lambda \frac{\partial T_f}{\partial X} = 0 \tag{6}$$

- **Solid**

$$CF_1 : \mathrm{X} = 0; 0 \leq Y \leq l; \lambda \frac{\partial T_S}{\partial X} = 0 \tag{7}$$

$$CF_2 : \mathrm{X} = L; 0 \leq Y \leq l; \lambda \frac{\partial T_S}{\partial X} = 0 \tag{8}$$

$$CF_3 : Y = 0; 0 \leq X \leq L; \lambda \frac{\partial T_S}{\partial X} = h(T_f - T_S) \tag{9}$$

$$CF_4 : Y = l; 0 \leq X \leq L; \lambda \frac{\partial T_S}{\partial Y} = 0 \tag{10}$$

These equations offer the possibility to access to various sizes dimensionless reference namely:

$$\mathrm{X}^* = \frac{X}{L} \tag{11}$$

$$Y^* = \frac{Y}{L} \tag{12}$$

$$t^* = \frac{t}{t_p} \tag{13}$$

Where tp is the breakthrough time of the fluid, t_p=L/u. Introducing these dimensionless variables led to following expressions:

- **Fluid**

$$\frac{\partial T_f}{\partial t^*} = \frac{1}{Pe_f} \frac{le}{L} \frac{\partial^2 T_S}{\partial X^2} - \frac{\partial T_f}{\partial X^*} + \frac{Bi}{Pe_f} \frac{\lambda_{SY}}{\lambda_f} \frac{L}{l} (T_f - T_S) \tag{14}$$

- **Solid**

$$\frac{\partial T_S}{\partial t^*} = \frac{1}{Pe_f} \frac{le}{L} \frac{\alpha_{SX}}{\alpha_f} \frac{\partial^2 T_S}{\partial X^{*2}} + \frac{1}{Pe_f} \frac{le}{L} \frac{\alpha_{SX}}{\alpha_f} \frac{L^2}{l^2} \frac{\partial^2 T_S}{\partial Y^{*2}} \tag{15}$$

Considering the physical properties of fluid and solid are fixed, four dimensionless parameters formally define system response to namely: Pe_f, Bi, L/l , L/le.

The equations are solved numerically by the finite volume method using Comsol 3.2 simulation software. At each time step, the system is solved on the basis of the solution of the previous time. Switching from a charging period to a discharge period is fixed by a level of fluid temperature leaving the storage. The calculation is completed when the periodic steady is achieved in terms of the storage efficiency.

4.3. Breakthrough time and breakthrough temperature

The heat front must imperatively be developed in fluid and solid. As in separation processes, the quality of the forehead can be qualitatively highlighted by the shape of the breakthrough curve, which in this case corresponds to the fall of the fluid output temperature. Still by analogy to separation process, it is possible to define a breakthrough time of the thermal front which corresponds, during discharging phase to fluid outlet temperature below a limit value. For this example the breakthrough temperature is fixed to 840K. It represents an increase of the temperature at least equal to 90% of the maximum increase. Destocking efficiency is directly defined by the percentage of energy used in breakthrough time on the total energy available in the regenerator.

5. Results and Discussion

The efficiency of heat exchangers is commonly presented as abacuses according to the number of transfer unit NTU. Dimensionless equations (14) and (15), it follows four numbers whose depends the response of storage: Pe_f ; Bi ; L/l ; L/le . The number of transfer units can be expressed by a combination of three of these dimensionless numbers:

$$NTU = \frac{hS_f}{\dot{m}C_{pf}} = \frac{hl}{\lambda_{SY}}\frac{\alpha_f}{leU_f}\frac{\lambda_{SY}}{\lambda_f}\frac{L}{l} = \frac{Bi}{Pe_f}\frac{\lambda_{SY}}{\lambda_f}\frac{L}{l} \qquad (16)$$

Commonly used for the characterization of heat exchangers, it was preferred to dimensionless number L/l for results presentation in the fig.4; these graphs show the evolution of the performance of thermal storage depending of the NTU and for a range of value of three other dimensionless numbers.

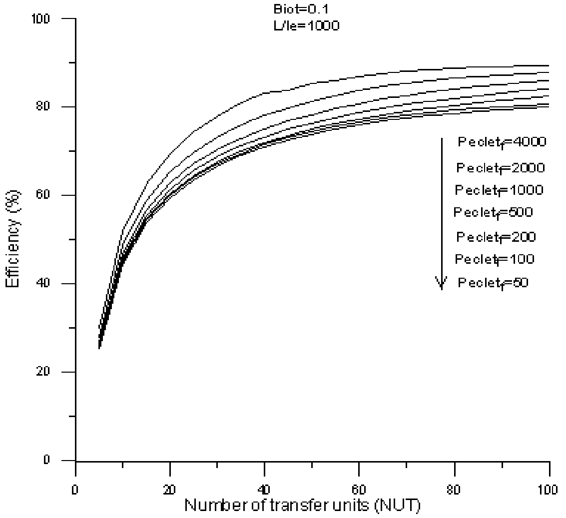

Fig.4 (a): Storage efficiency evolution according to NTU and Pe_f

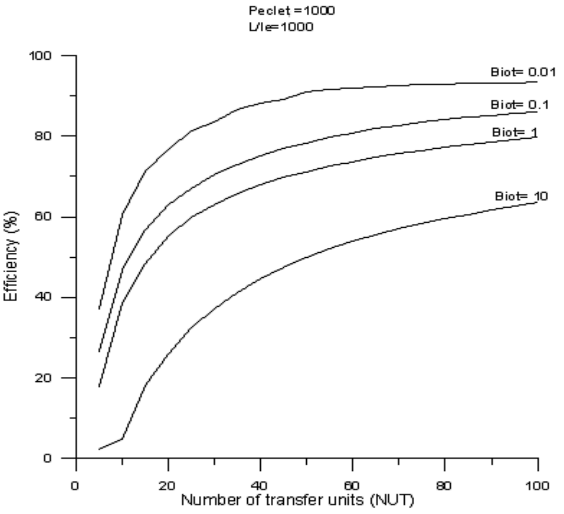

Fig.4 (b): Storage efficiency evolution according to NTU and Bi

In Fig.4 (a) the influence of the Peclet number is presented if the value of the Peclet number is large, the transfer is more effective. This can be explained by a more turbulent fluid flow and heat transfer resistance is then reduced.

Increasing of Peclet number is expressed by increasing of fluid velocity or by largest interstices between plates. This last parameter determines the quantity of fluid passing through the plates and dedicated to absorb or transfer energy. Fluid velocity is an important parameter in the heat exchange, when it increases the thickness of the hydrodynamic boundary layer decreases and therefore the heat exchange coefficient is increased. In contrast, a low fluid speed is perturbed by the thickness of the boundary layer and a low heat exchange.

In Fig.4 (b) shows the influence of the Biot number. More Biot is low more the storage is efficient. For Peclet and L fixed. Biot number of smaller than 0.1 allows to reach considerable performance, this thermal phenomenon is reflected in thermally thin body that imply the heat conduction inside the body is much faster than the heat convection away from its surface, and temperature gradients are negligible inside of it.

The three previous graphs showing the evolution of the performance of storage are used as abacuses. Indeed, each point on the graph corresponds to storage efficiency and is associated with four dimensionless numbers. With these four numbers and with reference to the basic equations, it is possible to identify geometry of plates and corresponding interstices. Table 1 lists the parameters associated with two points of Fig.4 (a).

Table 1. Examples

	Example A	Example B
Biot number (Bi)	0.1	10
Number of transfer Units (NTU)	60	30
Peclet number (Pe)	100	1000
The geometric ratio (L/le)	1000	100

The evolution of the heat front at different times is shown in Fig.5 (a) and Fig.5 (b) for two examples. For the first case, the displacement of heat forehead is marked; this stiffness forehead leads to engenders heat transfer quality.

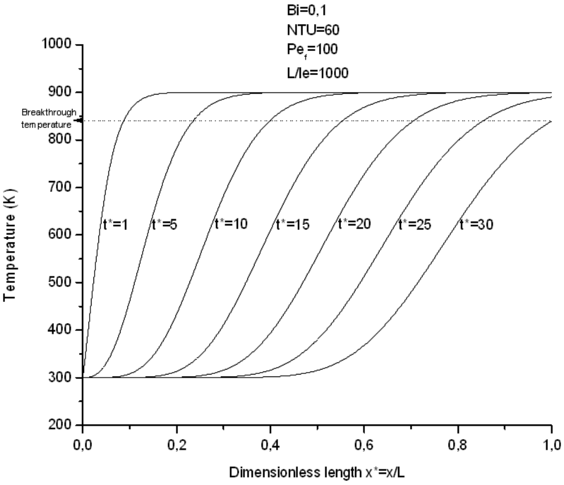

Fig.5 (a) Fluid temperature profile at outlet regenerator (Example A)

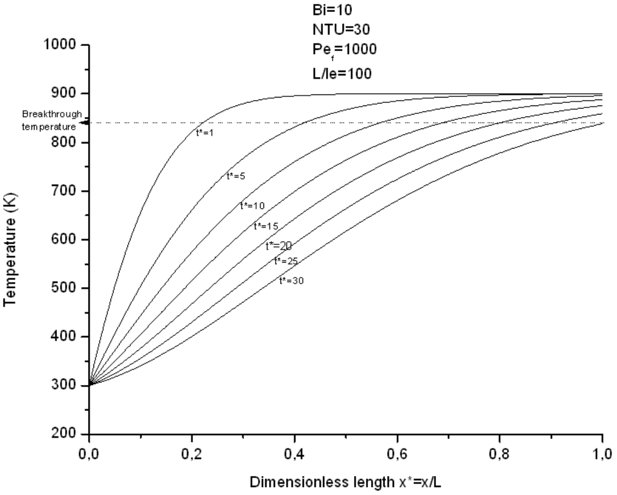

Fig.5 (b) Fluid temperature profile at outlet regenerator (Example B)

Breakthrough curves were also determined it is the evolution of fluid outlet temperature in function of time (Fig.6). The breakthrough time that has been defined for this study as the time after which the outlet temperature is equal to 90% of the maximum possible difference between the temperatures of the fluid and solid, derives directly from this curve. The quality of the forehead in example A engenders an efficiency of 52% at the breakthrough time against 17% for example B.

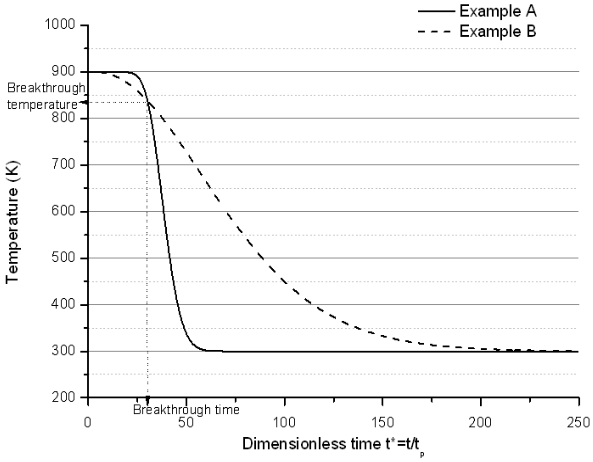

Fig.6 Breakthrough curves

The evolution of the quantity of heat transferred by the solid follows the same trend as the heat forehead. Over time the material is discharging slowly to reach the breakthrough time. For example A, the temperature distribution in the solid evolves uniformly unlike the example B.

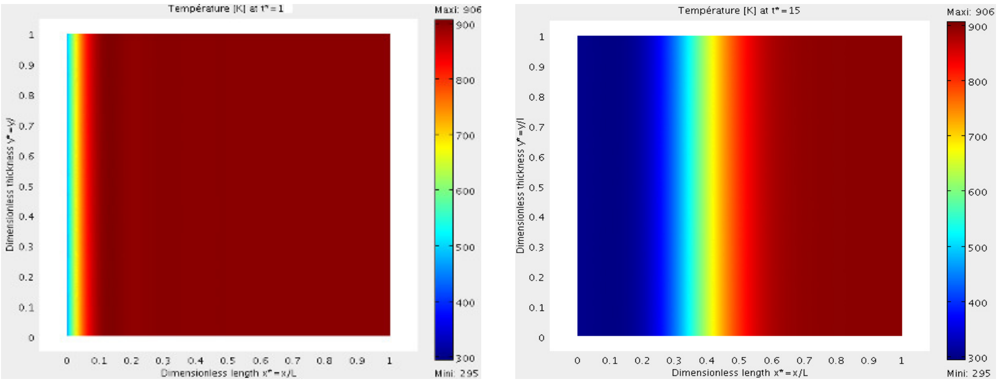

Fig.7 (a) Solid temperature profile evolution (Example A)

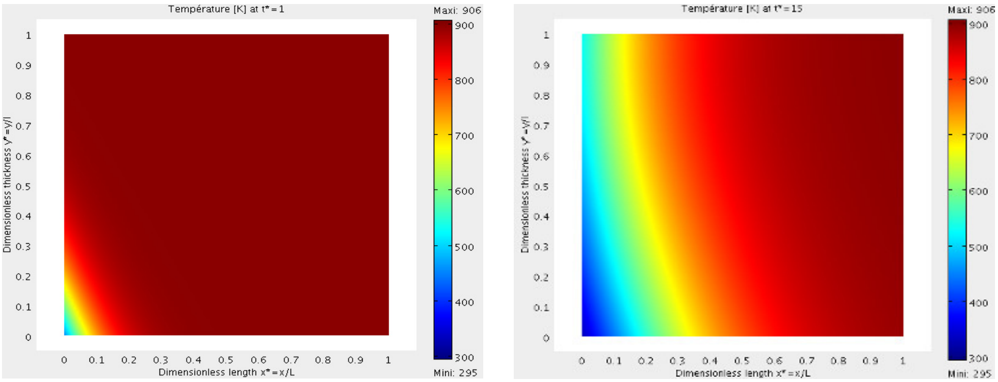

Fig.7 (b) Solid temperature profile evolution (Example B)

6. Conclusion

In this study the performances of a heat storage constituted of solid flat plates are determined. To describe the behavior of the system, the dimensionless energy balances equations are used. The modeling is based on the energy balances performed on an elementary volume of solid plate coupled to an elementary fluid flow section. The numerical simulation of storage tank is performed with Comsol 3.2. Several geometric configurations were simulated. System performances are presented as abacuses of functions in dimensionless numbers which characterize the system. From these abacuses, it is possible to define geometry of the thermal reservoir then determine the corresponding temperature profiles. The dimensioning of thermal reservoir is intended to process A CAES. The purpose is to store a quantity of energy of the compressed air upstream and to discharge it subsequently to reheat the air while maintaining a minimum level of output temperature. When the discharge lasts in the time while respecting the requirements temperature (breakthrough time) it expresses the availability of a quantity of electricity provided by the turbine power generator downstream of thermal reservoir. The value of the breakthrough time associated to the temperature profile of the fluid outlet temperature would allow to finalize the dimensioning of storage and thus to determine the duration of turbine operation as well as the process efficiency

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