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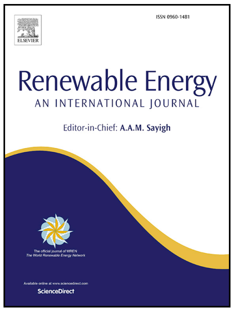

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Thermodynamic and economic evaluation of an innovative electricity storage system based on thermal energy storage

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Abstract:

One of the main issues concerning the inclusion of renewable energies as a way to produce electricity is the fluctuation of the production. This explains why there is a need for efficient solutions which will allow to store the energy when there is no need for it and release it when renewable energies are not able to sustain the production. This work concerns a Power-to-Power solution based on thermal energy storage at high temperature (around 900 °C). It relies on a simple heating loop to convert electrical energy into heat, and a thermodynamic cycle such as a gas cycle or a combined cycle to convert heat into electricity. One of the questions raised by this system is how it can be profitable from the economic point of view. Indeed a lot of studies have been conducted on energy storage, but very few propose solutions which can be relevant in terms of global costs and payback time. The aim of this study is to investigate if this system could be competitive in the European, and more specifically, French energy market. To look further into this topic, the system architecture and its components are defined. A thermodynamic model is also built to represent the behavior of the cycle. Finally, an economic discussion is performed using cost functions from the literature. The results show that the system does not accumulate enough running hours to justify high investment costs and should rely on simple technologies to insure its profitability. Furthermore, while the high capital expenditures of the whole system could be challenging, the thermal storage itself is not a big expense when compared to the cost of a natural gas fired power plant.

Keywords:

Energy, Thermodynamic Modeling, Thermal Energy Storage, Combined Cycle, Economic Analysis.

1. Introduction

At the world's scale, most of the electricity production comes from finite resources such as coal, gas or nuclear fuel. This leads to questioning the future because these resources will not last forever. Moreover, most of the electricity generation can be related to CO_2 emissions which are recognized as a main contribution to global warming. All these issues can at least be partly solved by increasing the inclusion of renewable energies as a way to produce electricity.

With the increase of the renewable energies share in the global energy mix appears the issue of the fluctuation of the production. The spreading of renewable energies at the world's scale will depend on the accessibility and efficiency of reliable storage solutions. Among these solutions, thermal energy storage is a technology that has been studied for many years [1].

1.1. Thermal energy storage as a competitive storage solution

Thermal energy storage is an energy storage solution which already had some successful commercial realizations. Sensible thermal storage is the most advanced solution, with installed capacities going up to 1010 MWh_{th} (Andasol CSP plant, Spain [2]). It relies on the variation of temperature of a storage material without phase change. In two tanks storage systems, the fluid circulates between a cold tank and a hot tank and is heated between the two, often by solar heat. In one tank or “thermocline” systems, some of the storage fluid is replaced by a solid material. This is a less mature solution which can lead to a reduction of the costs by 35% when compared to a traditional two tanks system [1]. This solution was applied once at large scale on the Solar One power plant in the USA (182 MWh_{th}) with rocks and sand as filler material and thermal oil as heat transfer fluid [2].

An emerging proposal which is adopted here is to replace the heat transfer fluid with air. Compared to traditional fluids, air has no upper or lower temperature limits, is free, non-toxic and non-inflammable [1]. Despite its weak thermal properties, it offers to store the energy at a higher temperature, enabling to increase the energy density of the storage. It is also possible to reach a higher efficiency on the heat to electricity conversion, as pointed out by the Carnot efficiency of the process. Several systems based on this type of thermal storage have emerged in the recent literature. A promising solution is “pumped heat electricity storage”, which has been studied from a thermodynamic and/or economic point of view in studies like [3]–[5]. This work will focus on another concept with simple and standard components which could insure a good industrial feasibility. As described in the next part, the system indeed mostly rely on common industrial components such as standard heat exchangers and turbomachinery, usually manufactured for conventional production cycles.

1.2. Innovative power-to-power storage system including thermal energy storage

The proposed system consists of the hybridization of storage tanks and their charging loop with a thermodynamic heat conversion cycle to produce electricity. The principle of the system is illustrated in Fig. 1 (each storage tank represents a number of storage tanks arranged in series).

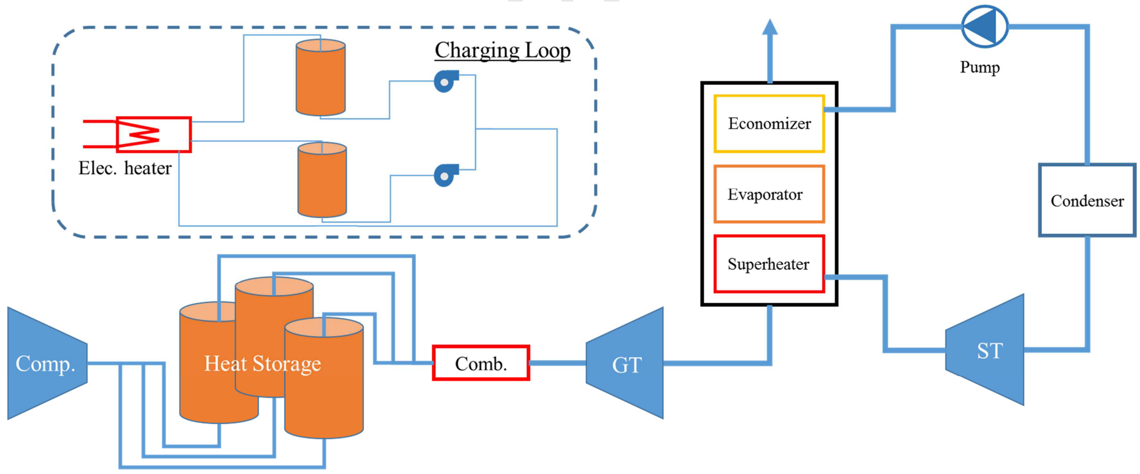

Fig. 1. Schematic representation of the power-to-power storage system (connections between the charging loop and the cycle are not shown).

The charging loop is composed of an electrical heater allowing to heat air at a temperature of 900 °C, an upper limit which was set due to technical and economic considerations on the storage material and the thermal insulation [6]. This charging loop allows to convert inlet electricity into high temperature heat which is stored in the storage tanks. These tanks are charged by the circulation of hot air in the solid filler material. Injecting the hot fluid on the upper side of the tank leads to the creation of the so-called “thermocline” zone. This thermal stratification divides the tank in a hot part and a cold part, separated by a thermal gradient. The charging loop also involves fans for air circulation. In a first approach, a dedicated fan is considered for each storage tank.

During the discharging phase, the heat stored is recovered in a gas cycle, also called Joule or Brayton cycle. A gas compressor increases the pressure and the temperature of inlet air with a given pressure ratio PR. The resulting air flows through the storage tanks and recovers heat from it (the air is injected on the lower side of the tank to avoid thermal destratification). Air can be further heated by combustion and is finally expanded in a gas turbine (GT). The remaining heat is recovered in a steam or Rankine cycle. This combination is the so-called combined cycle gas turbine (CCGT).

While the presence of the steam cycle can lead to significant improvement of the discharging cycle's thermal efficiency, it also introduces a cost that is far from negligible. The aim of this study will be to estimate the system's cost and profitability. Several design alternatives – without the steam cycle, with higher combustion temperature or without combustion – will be studied as well. In terms of storage filler material, basalt-type rocks are considered. This is a well-known cheap material suitable for high temperature energy storage [7].

The system alternates five hours of charge, when renewable electricity production is high, and three hours of discharge. This discharge time corresponds to the time length of a peak of consumption on the power grid, for instance in the morning. The system runs 300 days a year, as it should accumulate enough yearly running hours to be profitable, while it should not be able to perform more than one full charge/discharge cycle per day because of the time length of the phases.

2. Thermodynamic and economic modeling of the system

The architecture of the system is based on a combined cycle including a gas/solid sensible thermal storage at 900 °C, linked to a charging loop. The system was modeled to estimate its performance and to design the components for cost estimation. A more exhaustive thermodynamic analysis of this system, including exergy balance in each component, is carried out in [8].

2.1. Thermodynamic modeling

The model of the combined cycle is based on the first principle of thermodynamics, or energy conservation. Equation (1) gives the generic equation for energy balance in a thermodynamic process. Kinetic and potential energy $\Delta\dot{E}_c$ and $\Delta\dot{E}_p$ will be neglected. $\dot{Q}$ is the amount of heat transferred, $\dot{W}$ stands for work and $\Delta\dot{H}$ is the enthalpy variation between two states.

$$\dot{Q} + \dot{W} = \Delta\dot{H} + \Delta\dot{E}_c + \Delta\dot{E}_p. \qquad (1)$$

2.1.1. Conservation equations and design parameters

Turbomachinery and pumps are modeled as adiabatic transformations with given isentropic efficiencies. Heat exchangers were also modeled as adiabatic processes. Several design parameters allow to set a base design: pinch point temperature difference at the evaporator, approach temperature difference at the superheater, maximum steam pressure and temperature. The corresponding values shown in Table 1 are in range with typical values from the literature [9]. It should be noted that the combustion model implies complete combustion of pure methane.

Table 1. Main parameters of the combined cycle thermodynamic model

Parameter	$\eta_{is,comp}$	$\eta_{is,GT}$	$\eta_{is,ST}$	$\eta_{is,pump}$	$T_{max,st}$	$P_{max,st}$	ΔT_{ap}	ΔT_{pp}
Value	0.88	0.89	0.85	0.85	550 °C	100 bar	20 °C	10 °C

As it was proposed by several authors [6], this thermodynamic evaluation relies on static models, because economic evaluation only needs design point knowledge. The only exception is the thermal storage model. Transient simulation was performed to design the tanks for constant outlet temperature (900 °C) during the discharge. This method ensures a constant temperature at the inlet of the combustor or the gas turbine, allowing to use the static model for the rest of the system.

2.1.2. Storage design

Preliminary results from the combined cycle model are used to determine the number of storage tanks and their size. This is done with a 1D Schumann-type model (2), (3) to resolve the temperature profiles in the fluid and the solid phase [1]. Effective thermal conductivities λ_{eff} are determined with Zehner & Schlünder [10] model and the heat transfer coefficient α_{eff} is from

Wakao et al [11] with the effective approach from Stuke [12]. In accordance with the literature, the packed bed void fraction ε is set to 0.37 and the rock particles diameter is 0.03 m [1].

$$\varepsilon\rho_g c_{pg}\left(\frac{\partial T_g}{\partial t}+u\frac{\partial T_g}{\partial z}\right)=\frac{\partial}{\partial z}\left(\lambda_{eff,g}\frac{\partial T_g}{\partial z}\right)+\alpha_{eff}a_s\left(T_s-T_g\right)+Ua_{vessel}(T_{ext}-T_g), \quad (2)$$

$$\left(1-\varepsilon\right)\rho_s c_{ps}\frac{\partial T_s}{\partial t}=\frac{\partial}{\partial z}\left(\lambda_{eff,s}\frac{\partial T_s}{\partial z}\right)+\alpha_{eff}a_s\left(T_g-T_s\right). \quad (3)$$

The model assumes that the porous medium is homogeneous in the radial and axial directions. It also assumes that the fluid is circulating according to a plug flow hypothesis. The only variables influencing the storage design are the gas cycle mass flow rate and the compressor PR, because the air flows directly from the compressor to the storage in the discharging phase. Sizing the storage for a stable output temperature profile during the whole discharging phase means that the thermal gradient is not totally extracted at each discharging phase. As a consequence, the storage tanks are oversized in comparison with a storage which would be completely discharged. The configuration of the storage tanks which is proposed here is a good trade-off between pressure drop (very high if all the tanks are in series) and total discharging of several tanks (not possible if all the tanks are in parallel). Pressure drop is computed with the standard Ergun equation for pressure losses through packed beds [1]. The aspect ratio of the cylindrical tanks will be set to 1 for this study, meaning that the tanks have the same length and diameter. It seems to be a good trade-off between thermal efficiency, pressure drop and mechanical stress [1]. The storage limit dimensions are set at 5.55 m x 5.55 m for 15 bar of absolute pressure, and 8 x 8 m for 10 bar of absolute pressure. The first limit is set because of industrial feasibility, the second one is a limit of the economic model.

2.1.3. Thermodynamic model outputs

In this study, the aim of the thermodynamic model is to set consistent design parameters for the system, check the thermodynamic feasibility of its functioning and estimate the corresponding flow rates. Cycle thermal efficiency was calculated with (4). Real, electric efficiency was then calculated by applying two coefficients to this thermal efficiency in (5), accounting for mechanical losses $\eta_{mechanical}$ and electric generator losses $\eta_{generator}$ (both are considered equal to 0.98).

$$\eta_{th} = \frac{\left(|\dot{W}_{GT}| - |\dot{W}_{comp}|\right) + \left(|\dot{W}_{ST}| - |\dot{W}_{pump}|\right)}{\dot{Q}_{storage} + \dot{Q}_{combustion}}, \quad (4)$$

$$\eta_{CCGT}=\eta_{th}\eta_{mechanical}\eta_{generator}, \quad (5)$$

in which $\dot{Q}_{storage}$ represents the power input from the discharge of the storage tanks:

$$\dot{Q}_{storage} = \dot{M}_g\left(h_{out,storage} - h_{in,storage}\right). \quad (6)$$

2.1.4. Thermodynamic model validation

The validation of the model of a thermodynamic cycle involving turbomachinery is often complicated because of incomplete data sets. At the same time, the thermal modeling of such a system implies well-known methodologies. The thermodynamic model was checked by comparing its results with those from [13] reproducing the behavior of a combined cycle with a GT13-E2 GT and a single pressure Heat Recovery Steam Generator (HRSG) (Table 2).

Table 2. Validation of the thermodynamic model

	η_{th}	$\dot{W}_{net,GT}$	$\dot{W}_{net,ST}$	$T_{out,GT}$	$T_{out,g}$
Reference [13]	0.53	184 MW	66 MW	506 °C	184 °C
Model	0.53	180 MW (-2 %)	71 MW (+8 %)	502 °C	184 °C

2.2. Economic modeling

On the basis of the thermodynamic model results, an economic evaluation was performed to evaluate the Levelized Cost Of Energy (LCOE).

2.2.1. Economic model

The economic modeling of an electricity supplying system cannot reach a high level of accuracy unless it is done with recent data from the sectors' manufacturers. Indeed the price of the components and the indirect costs are constantly evolving, and scale effect can be very pronounced. However, the cost and future income of such a system are among the first concerns of a decider.

The economic model is mainly based on the works of Spelling [6] and Pelster [14]. Both of these studies suggest cost functions attributed to Frangopoulos, which are abundantly used in the studies involving turbomachinery [13]. Due to the limited space, the functions involved in (7) will not be detailed here. Additional cost functions for the fans were taken from [15] and from [7], [16] for the storage (tanks, insulation, rocks). All the cost functions that are used are detailed in [17]. Table 3 shows some key parameters of this economic study.

Following the approach from Pelster [14], when the reference cost of an equipment is from a previous year, the effects of inflation are taken into account with the Marshall and Swift index. Some adjustments were made on the correction factor for the cost of the GT (reference temperature was set to 1700 K to match modern gas turbines characteristics) and the reference cost of the steam turbine was set to 500 \$/kW. Pelster's proposal to adopt a weight coefficient of 0.7 for the gas cycle equipment cost was applied, as it lead to good results when compared to the GTW Handbook [18].

$$C_{inv} = \sum C_{eqp} + C_{inst} + C_{civil} + C_{NG} + C_{eng} + C_{cont}. \qquad (7)$$

Table 3. Reference values for the storage subsystem costs

Parameter	*Value*
Reference insulation cost (mineral fiber)	310 \$/m3
Insulation thickness	0.4 m
Reference storage media cost (gravels)	0.1 \$/kg
Cost of the charging loop electrical heater (approximation)	3.5 M\$

2.2.2. Economic model outputs

For this type of study, at a low level of project definition, an accuracy of ± 20 % is believed to be acceptable [15]. The main output which will be studied is the LCOE. This value indicates the minimum electricity sale price needed to get to the break-even point at the end of the project's lifetime. The lower the LCOE, the higher the profitability of the system. This indicator was determined using the formula from [6], adding a term C_{CO_2} to account for a carbon tax:

$$LCOE = \frac{\alpha C_{inv} + \beta C_{dec} + C_{fuel} + C_{maint} + C_{lab} + C_{CO_2}}{E_{net,year}}. \qquad (8)$$

The coefficients α and β, taken from [6], translate the investment and the decommissioning costs C_{inv} and C_{dec} into annual payments, including debt interest. They take into account the operating years and the effects of the time needed for construction and decommissioning. $E_{net,year}$ is the annual amount of energy released by the system, in MWh. For practical reasons, the labor cost C_{lab} related to the salaries of the plant's employees will not be taken into consideration. It is expected that it will be in the same order of magnitude for each case. Equations (9) and (10) give α and β:

$$\alpha = \frac{(1+i)^{n_{con}} - 1}{i n_{con}} \frac{i(1+i)^{n_{op}}}{(1+i)^{n_{op}} - 1} + k_{ins}, \qquad (9)$$

$$\beta = \frac{(1+i)^{n_{dec}} - 1}{in_{dec}(1+i)^{n_{dec}-1}} \frac{i}{(1+i)^{n_{op}} - 1}. \qquad (10)$$

The decommissioning cost C_{dec} was taken as 5 percent of the cost of equipment, installation and civil engineering. Yearly maintenance cost C_{maint} is 1 percent of the civil engineering cost and 2 percent of the equipment cost, as in [6].

The parameters related to the calculation of the LCOE are listed in table 4. The operational lifetime of 25 years is in good agreement with the usual considerations for a CCGT [6]. As this study focuses on the French market, the fuel specific cost is an estimation based on the cost of natural gas in France. The carbon tax was set to 100 €/tCO_2, a value which is expected to be reached before 2030 in this country. Both values were converted in US dollars with a current conversion rate of 1 euro for 1.18 dollars for consistency with the other reference costs of the study, which are expressed in dollars in references [6], [14]. The excess electricity used in charging phase is supposed to be free.

Table 4. Parameters for the LCOE calculation

Parameter	*Name*	*Value*
i	Debt interest rate	7 %
k_{ins}	Annual insurance rate	1 %
n_{con}	Construction time	2 years
n_{dec}	Decommissioning time	2 years
n_{op}	Operational Lifetime	25 years
c_{fuel}	Fuel specific cost (€ to $ conversion)	22.4 $/MWh
cCO_2	Carbon tax specific cost (€ to $ conversion)	118 $/tCO2
LHV_{fuel}	Lower Heating Value of the fuel	50.01 10^6 J/kg

2.2.3. Economic model validation

The total investment cost of the combined cycle part of the plant was checked by comparing the results to reference values from the Gas Turbine World Handbook 2010 [18]. The thermodynamic model was set to design a 60 MWe CCGT. The results are listed in Table 5.

Table 5. Validation of the economic model of the combined cycle

Design parameters and results	*Reference results from the GTW Handbook* [18]	*Model parameters and results*
PR	-	20
$T_{in,GT}$	-	1673 K
η_{CCGT}	-	0.58
$\dot{W}_{net}$	60 MWe	60.5 MWe
Specific equipment cost	1000 $/kWe	912 $/kWe (-9 %)
Total investment cost C_{inv}	96 to 120M$	89 M$ (-7 to -26 %)

The LCOE calculation was also verified with the results from a recent study [19] on a typical combined cycle power plant. For this estimation, the investment was not determined with the cost functions, because the net power of the plant (550 MWe) is out of the scope of the thermoeconomic model. This investment cost was determined with the values proposed in the same reference. From Tables 5 and 6, it can be seen that the cost functions tend to underestimate the investment cost and the LCOE. A source of error could be the hypothesis of complete combustion of pure methane, while real plants are fueled by natural gas with lower LHV. Furthermore, the modeled CCGT only has one level of pressure, while modern CCGTs have two or three levels of pressure at the HRSG. Still, the deviation from the reference results is small enough to validate the methodology.

Table 6. Validation of the LCOE calculation

Power plant characteristics	*Values*
$\dot{W}_{net}$	550 MWe
Total investment cost	1000-1300 $/kWe
Fuel cost	11.8 $/MWh
Construction time	3 years
Facility Life	20 years
Capacity Factor	80 – 40 %
LCOE (ref [19])	48$ - 78$
LCOE (model)	40$ - 68$

3. Case studies

To begin with, a first design called "base case" will be evaluated. Its net electrical power output is set to 60 MWe, involving a thermal storage of a few hundred megawatt-hour, depending on the thermal efficiency of the discharging subsystem. This base case does not involve combustion and the storage has an upper temperature of 900 °C. It is expected that the efficiency of the discharging combined cycle will be limited by this temperature. To see if a better efficiency could improve the profitability of the system, a second case with a combustion temperature of 1200 °C will be studied. However, a temperature of 1200 °C is still not in the range of modern gas cycles temperature. This is why a third case of combined cycle at a combustion temperature of 1400 °C will be studied. Finally, the study of modern CCGTs shows that the low power output of the system (60 MWe) is generally out of the scope for this application. To check if a cheaper, simpler system could be less thermodynamically effective but more financially competitive, the results will be compared to a final case involving a simple gas or Brayton cycle for the discharging subsystem. The design parameters for the four studied cases are shown in Table 7. PR of the gas cycle was set as an optimum for the cycle efficiency, but cannot be superior to 15 to insure the storage tank feasibility.

4. Results

The result of LCOE calculation is given in Table 7. An important design choice is that each case does not involve the same amount of stored energy. This is due to the aimed net power, which is the same for each case, while the cycle efficiency is not the same, leading to various heat input from the storage between the cases. Due to the decision to size the storage for constant outlet temperature (900 °C) during the three hours of discharge, the power from the heat storage (6) is constant.

Table 7. The four studied cases and their LCOE result

Design parameters	*Base Case*	*Case 2*	*Case 3*	*Case 4*
Type of cycle	Combined	Combined	Combined	Brayton
Combustion	No	Yes	Yes	Yes
$\dot{W}_{net}$	60 MWe	60 MWe	60 MWe	60 MWe
$Q_{storage}$	409 MWh_{th}	205 MWh_{th}	148 MWh_{th}	302 MWh_{th}
$T_{in,GT}$	900 °C	1200 °C	1400 °C	1200 °C
PR	10	14	15	15
LCOE	213 $/MWh	252 $/MWh	253 $/MWh	186 $/MWh

4.1. Base case: combined cycle without combustion

For a given configuration, one of the first things to evaluate is the number and size of storage tanks. Following the methodology described in part 2.1.2, this configuration needs 6 storage tanks (height: 8 m, diameter: 8 m) arranged in three series. The main thermodynamic model results are listed in Table 8. The total investment cost is about 102 M$ and the cost distribution is illustrated in Fig. 2. The storage itself represents 7 % of the total investment cost, and the equipment cost of the combined cycle alone is about 53 M$. The LCOE is 213 $/MWh for this scenario. This base case does not include any combustion, so the fuel cost C_{fuel} and carbon tax cost C_{CO_2} are equal to zero, leading to low operation costs for the system. The LCOE is mainly influenced by the investment term (eq (8)), while the share of the maintenance costs is around 10 %.

Table 8. Main thermal model results for the base case

Variable	*Value*
$\dot{M}_g$	206 kg/s
$\dot{M}_{st}$	18.3 kg/s
$Q_{combustion}$	0 MWh (for one discharge)
$Q_{storage}$	409 MWh (for one discharge)
$\dot{W}_{net,GT}/\dot{W}_{net,CCGT}$	0.7
η_{CCGT}	0.44

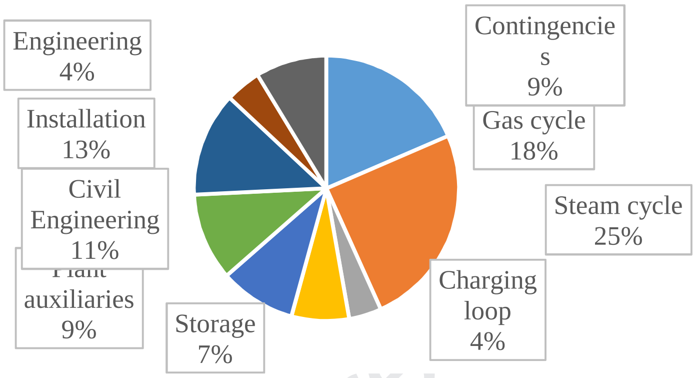

Fig. 2. Distribution of investment costs (102 M$) for the base case: CCGT without combustion.

4.2. Case 2: combined cycle with combustion at 1200 °C

This first design alternative will try to investigate if it is possible to reduce the LCOE by introducing combustion at an intermediate level of temperature. This configuration needs less storage volume because of the better thermal efficiency of the discharging cycle. However, storage tanks dimensions are constrained by the higher level of pressure (14 bar). This explains that this scenario needs as much as 12 tanks (height: 5.55 m, diameter: 5.55 m) arranged in three series. Some results are introduced in Table 9. The investment cost is about 104 M$, only 2 percent higher than the base case. The cost distribution will not be displayed again, as it is more or less the same as in Fig. 2. The LCOE is equal to 252 $/MWh for this second case. As the investment costs between this case and the base case are almost equal, the fuel cost (less than 1 M$/year) and the carbon tax (of the same order of magnitude) are likely to be responsible of the LCOE difference.

Table 9. Main thermal model results for the second case

Variable	*Value*
$\dot{M}_g$	116 kg/s
$\dot{M}_{st}$	16.9 kg/s
$Q_{combustion}$	136 MWh (for one discharge)
$Q_{storage}$	205 MWh (for one discharge)
$\dot{W}_{net,GT}/\dot{W}_{net,CCGT}$	0.67
η_{CCGT}	0.53

4.3. Case 3: combined cycle with combustion at 1400 °C

This case is the same as the previous one but it tries to establish if it would be interesting to burn more fuel to reach better thermal efficiency. The storage tanks have dimensions of 5.55 m x 5.55 m and there are 9 tanks arranged in 3 series. This scenario has the lowest stored energy of all the studied cases because the thermal efficiency of the cycle is high, which implies that less heat is needed to produce the same net power output. Main thermodynamic results are introduced in Table 10.

Table 10. Main thermal model results for the third case

Variable	*Value*
$\dot{M}_g$	86 kg/s
$\dot{M}_{st}$	17.5 kg/s
$Q_{combustion}$	172 MWh (for one discharge)
$Q_{storage}$	148 MWh (for one discharge)
$\dot{W}_{net,GT}/\dot{W}_{net,CCGT}$	0.65
η_{CCGT}	0.56

The total investment cost is 100 M$, 2 % lower than the base case. This can be explained by the lower number of storage tanks, and the better thermal efficiency which allows to reduce the mass flow rates of the turbomachinery. The LCOE for this case is 253 $/MWh. Even if this cycle reaches a better thermal efficiency than case 2, the increase of fuel cost and carbon tax (around 1.2 M$/year each) leads to equal LCOE for these two cases.

4.4. Case 4: gas cycle with combustion at 1200 °C

This last case will check if cutting the investment costs by removal of the steam cycle can offer a better LCOE, whereas thermal efficiency will decrease. For this case, 20 storage tanks are arranged in 4 series (height: 5.4 m, diameter: 5.4 m). In terms of stored energy, this scenario is just below the base case. Main thermodynamic results are summed up in Table 11. The total investment cost is only 64 M$. The cost distribution is illustrated in Fig. 3 and the LCOE is equal to 186 $/MWh. Fuel cost is the highest of the four studied cases (around 1.4 M$/year), but the low investment cost leads to the lowest LCOE. The storage (tanks, insulation and filler material) costs about 12 M$ and represents 19 % of the total cost of the plant.

Table 11. Main thermal model results for the fourth case

Variable	*Value*
$\dot{M}_g$	175 kg/s
$\dot{M}_{st}$	0 kg/s
$Q_{combustion}$	204 MWh (for one discharge)
$Q_{storage}$	302 MWh (for one discharge)
$\dot{W}_{net,GT}/\dot{W}_{net,CCGT}$	1
η_{CCGT}	0.36

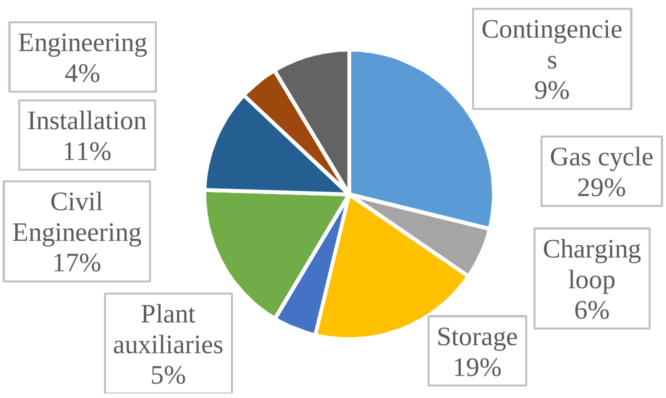

Fig. 3. Distribution of investment costs (64 M$) for the 4th case: gas cycle at 1200 °C.

5. Discussion

In the previous part, the results of four case studies were shared. All the cases lead to the same electrical power output (60 MWe). This could be discussed because it leads to various size and heat input from the storage, varying from 148 to 409 MWh for 3h of discharge. At the same time, the four systems share identical gross revenue, which is interesting for comparison.

The lowest LCOE is the one from the simple gas (Joule-Brayton) cycle. With low investment costs, it can offer an interesting profitability during the lifetime of the project. The base case (combined cycle without combustion) offers a 15 % higher LCOE. However, an interesting aspect of this configuration is its insensitivity to CO_2 tax variations. As these taxes are expected to grow up in a near future, the choice of a system without CO_2 emissions such as this one could be a safe decision. Still, the carbon tax needs to be raised to as much as 250 \$/tCO_2 for the base case LCOE to outperform the simple gas cycle (case 4) LCOE.

The main conclusion of this study is that for this type of functioning, the combined cycle does not accumulate enough running hours to counterbalance its high investment cost. On the considered level of net power, a combined cycle is indeed around three times more expensive than a gas cycle [18]. Its main advantage is that it leads to an increase of thermal efficiency between 10 and 20 points, but the yearly gross income is not sufficient enough to justify the investment. A simple way to verify this is to put the capacity factor (ratio of running hours per year) to 0.4 in case 2, by allowing the system to perform two charge/discharge cycles a day and additional combustion for the remaining hours. The LCOE drops to 118 \$/MWh (- 53 %). High investment cost could also be a brake for potential commercial applications. A solution could be to insert the gas cycle, the storage and its charging loop in a project of repowering of an old steam plant.

To check the accuracy of the methodology, a sensitivity analysis on the financial parameters of the LCOE calculation was carried on the best case (case 4). The results (Table 12) show that the LCOE calculation is not overly sensitive to its entry data. Moreover, the low dependence of the LCOE to the fuel specific cost and carbon tax cost can be explained by the reduced fuel consumption thanks to the heat brought from the thermal storage. As it is difficult to have a good visibility for the future of combustion systems, it can be seen as an interesting feature.

Table 12. Sensitivity analysis on the financial parameters of the LCOE calculation on case 4

Parameter	*Value*	*Variation of the LCOE*
i (debt interest rate)	0.05	- 10 %
(base case: 0.07)	0.09	+ 11 %
Operational Lifetime	20 years	+ 6 %
(base case: 25 years)	30 years	- 4 %
Fuel cost	17.9 \$/MWh	- 3 %
(base case: 22.4 \$/MWh)	27 \$/MWh	+ 3 %
Insurance rate	0.005	- 3 %
(base case: 0.01)	0.015	+ 3 %
Carbon tax specific cost	94 \$/t_{CO2}	- 3 %
(base case: 118 \$/t_{CO2})	142 \$/t_{CO2}	+ 3 %

Finally, this study allows to compare this system with other storage solutions. The order of magnitude of the stored energy is about a few hundred megawatt-hour. In the field of energy storage, its main competitors are pumped hydroelectric energy storage, compressed air energy storage or battery farms. According to a 2016 study by Lazard, compressed air and pumped hydroelectric can both offer more interesting or equivalent LCOE (respectively 116-140 and 152-198 \$/MWh [20]). Despite this advantage, it has to be kept in mind that these technologies can only be built at very specific locations. The proposed system does not share these restrictions and can be plugged in any possible location. Batteries also offer this possibility, while their cost stands around 200-400 \$/MWh to store a similar amount of energy [21]. The LCOE of the proposed system is of the same order of magnitude. A perspective could be to compare the system with batteries from a larger point of view, including life cycle analysis, ageing of the storage capacities and decommissioning.

6. Conclusion and perspectives

Because of the progression of renewable energies in the energy mix, electricity storage is expected to be a discussed topic in the near future. In this study, an innovative electricity storage solution based on thermal energy storage was introduced. The inlet electricity, converted into heat in the charging phase, is restored with a thermodynamic cycle in the discharging phase. The results showed that improving the thermal efficiency does not necessarily lead to cost improvement for such a system which is designed to run a few hours a day. An illustration of this assessment is that the most profitable case is the one which only implies a gas cycle for the discharging phase (LCOE = 186 \$/MWh). Finally, the order of magnitude of the levelized cost of energy seems to be competitive with more established solutions like batteries. An interesting perspective would be to add exergy criteria and optimize the system with regards to exergy destruction and investment cost.

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Nomenclature

Letters

a_s	total surface area of solid to bed volume, m^2/m^3
a_{vessel}	external surface area of packed bed to bed volume, m^2/m^3
C	cost, \$
c_p	specific heat capacity, J/(kg K)
$\dot{E}$	energy, W
h	specific enthalpy, J/kg
$\dot{H}$	enthalpy, W
i	debt interest rate
k_{ins}	annual insurance rate
$\dot{M}$	mass flow rate, kg/s
n	number, -
PR	pressure ratio, -
$\dot{Q}$	heat, W
T	temperature, °C
u	interstitial fluid velocity, m/s
U	overall heat transfer coefficient, W/(m.K)
$\dot{W}$	work, W
z	axial coordinate, m

Greek symbols

α	heat transfer coefficient, $W/(m^2.K)$
ε	packed bed porosity
λ	thermal conductivity, W/(m.K)

η efficiency
ρ density, kg/m^3

Subscripts and superscripts

ap approach
civil civil engineering
comp compressor
con construction
cont contingencies
dec decomissioning
eff effective
eqp equipment
eng engineering
g gas
inst installation
is isentropic
maint maintenance
NG natural gas substation
op operating years
pp pinch point
s solid
st steam
th thermal

Acronyms

CCGT combined cycle gas turbine
GT gas turbine
HRSG heat recovery steam generator
LCOE levelized cost of energy
LHV lower heating value
ST steam turbine

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Thermodynamic and economic evaluation of an innovative electricity storage system based on thermal energy storage

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Highlights

- An innovative solution based on thermal energy storage is introduced
- A thermoeconomic analysis is performed
- Improving the thermal efficiency does not necessarily lead to cost improvement
- The most profitable case is the one which only implies a gas cycle for the discharging phase (LCOE = 149 $/MWh)
- The levelized cost of energy seems to be competitive with more established solutions