Modelling of the Annual Performance of a CAES Plant and Relative Economic Analysis

Federico Ferretto [a], Anna Stoppato[b], Nicola Destro[c] and Alberto Benato[d,e]

[a] *Department of Industrial Engineering, University of Padova, Italy, federicoferretto@libero.it*
[b] *Department of Industrial Engineering, University of Padova, Italy, anna.stoppato@unipd.it*
[c] *Department of Industrial Engineering, University of Padova, Italy, nicola.destro@dii.unipd.it, (**CA**)*
[d] *Department of Industrial Engineering, University of Padova, Italy, alberto.benato@unipd.it*
[e] *"Giorgio Levi Cases" Interdepartmental Centre for Energy Economics and Technology, University of Padova, via Marzolo 9, 35131 Padova, Italy*

Abstract:

Compressed air energy storage (CAES) technology can offer a lot of ancillary services to the electricity system such as black start capability, voltage and frequency support, integration with non-programmable renewable energy sources. CAES plants often operate on electricity spot markets by storing energy when prices are low and generating electricity when prices are high. This paper investigates the operation over a year of two different CAES plants in two different scenarios. The first scenario simulates the operation of a CAES plant as independent plant in the Italian electricity market with the aim of maximizing the earnings. The control method is based on the variation of the hourly price of energy and on the variation of the pressure of the air stored in the cavern. The second scenario simulates the operation of an integrated system built up of a CAES and a wind farm with the aim of maximizing the earnings of the integrated system and the capacity factor of the wind farm. The integration of CAES and the wind farm is considered by means of a non-direct connection. The spot markets prices are referred to the Italian market. Other ancillary services are not taken in consideration. The software used to conduct the simulations is Simulink, a dynamic package of Matlab.

Keywords:

Compressed Air Energy Storage, Economic Analysis, Energy Storage, Wind Farm, Electricity Market.

1. Introduction

One of the most relevant problems when dealing with not predictable renewable sources is adjusting their variable and intermittent availability with the users demand curve. For this reason, the rapid diffusion of energy technologies based on renewable sources, and their expected wider use in the near future, pose challenges and opportunities regarding their integration into energy supply systems [1-2].

One way to reduce this uncertainty, and to guarantee the satisfaction of the users without limiting their request, is the use of energy storage systems. In addition, conventional power plants are called to withstand large fluctuations in demand, and must be able to meet these different conditions trying to keep low costs, good efficiency and acceptable residual life reductions [3].

For this reason, even if energy storage use had a long history, both with regard to medium-large sizes for peak shaving and to micro-small sizes for isolated systems and emergency power, greatest efforts have been recently spent to achieve efficient systems for electrical energy storage able to play the important role of unifying, distributing and increasing the capacity and the performances of both renewable and fossil fuels power generation systems. The various electrical energy storage technologies differ for many aspects, such as commercial maturity, storage capacity, stored energy density, charging and discharging power, efficiency, discharge time (which is linked to the instantaneous power), costs, expected life time (in terms of both years and number of charge / discharge cycles), autonomy, reaction time, environmental impact, and reliability [4-7]. For this reason, each of them is particularly indicated for specific applications, as black start capability, voltage and frequency support, balancing, time shift, peak shaving.

According to their operational criterion they can be classified as mechanical (pumping hydro, compressed air, flywheels), electrochemical (batteries, flow batteries), electric (supercapacitors and superconducting magnetic), chemical (hydrogen and fuel cells, power to gas) or thermal (thermal storage) technologies [8-11].

In Compressed Air Energy Storage (CAES) systems, air is compressed and then stored in an underground cavern. When electricity is required, air is heated and then expands in a turbine [12]. In this way, electricity can be consumed when the prices are low and then produced when they are higher for maximizing plant earnings, or energy can be used when available and then sent to the grid when required by the users for a good exploitation of non programmable renewable energy sources.

Luo et al. [13] present an overview of different types of CAES, taking into account their technical and economical characteristics, their state of the art and the challenges for future development. Different application areas (such as power quality, load levelling, seasonal storage,...) are taken into consideration and the suitable CAES technologies identified. The potential for integration with intermittent renewable energy power generation is shown as particularly interesting.

The use of CAES for isolated systems is described in [14] as a support to hybrid plants or in CHP configuration where it supplies also thermal power to a district heating grid [15]; the economic and environmental advantage of this solution is evaluated in [16]. CAES is suitable for medium and large energy storage, but recently many studies are related to its use into small systems, also in innovative configurations integrating CAES and pumped hydro plants [17].

A roadmap for energy storage development in Europe was published by the European Association of Energy (EASE) and European Energy Research Alliance (EERA) [18]: by 2030 a cost reduction around 20% and a efficiency improvement beyond 70% are estimated for large storage plants as well as a deployment of decentralised CAES plants in the distribution grid and even at consumers.

In Lund et al. [19] the optimization of the operational strategy of a given CAES plant is presented. The goal is the maximization of the earnings when the plant operates on the electricity spot market. Since the prices fluctuations on that market are not known in advance, two practical strategies, which estimate the average price of electricity, are compared with the optimal one, showing good results.

In this paper, two different CAES plants are investigated in two different scenarios. The first scenario simulates the operation of a CAES plant as independent power plant in the electricity market with the aim of maximizing the earnings. The second scenario simulates the operation of the integrated system CAES and Wind Farm with the aim of maximize the earnings of the integrated system and the capacity factor of the Wind Farm.

2. Mathematical model of the CAES plants

The main components of a CAES plant are the compressor, the cavern, the combustion chamber and the turbine.

As well known, despite the large interest for this technology, at present there are only two CAES plant in operation in the world: one in Huntorf (Germany) [20] and one in Mc Intosh (Alabama, USA) [21] having 290 MW and 110 MW turbine capacities, respectively.

In this paper, the plant scheme (Fig. 1) reference is based on Mc Intosh CAES plant that consists in a compressor train (axial plus centrifugal), an underground cavern as storage reservoir, gas turbines and a heat recovery unit that heats the cold air coming from the cavern using the hot exhausted gases from the turbine. The operational parameters values are referred to Huntorf plant such as e.g. the cavern and turbine capabilities. The modelling of the off-design of the turbo-machines and equipments, such as e.g. inter and after-coolers, is based on characteristic maps founded in literature. The operation of every component is linked to the others through characteristic differential equations of first order that enable to describe the variation of the thermodynamic parameters over time.

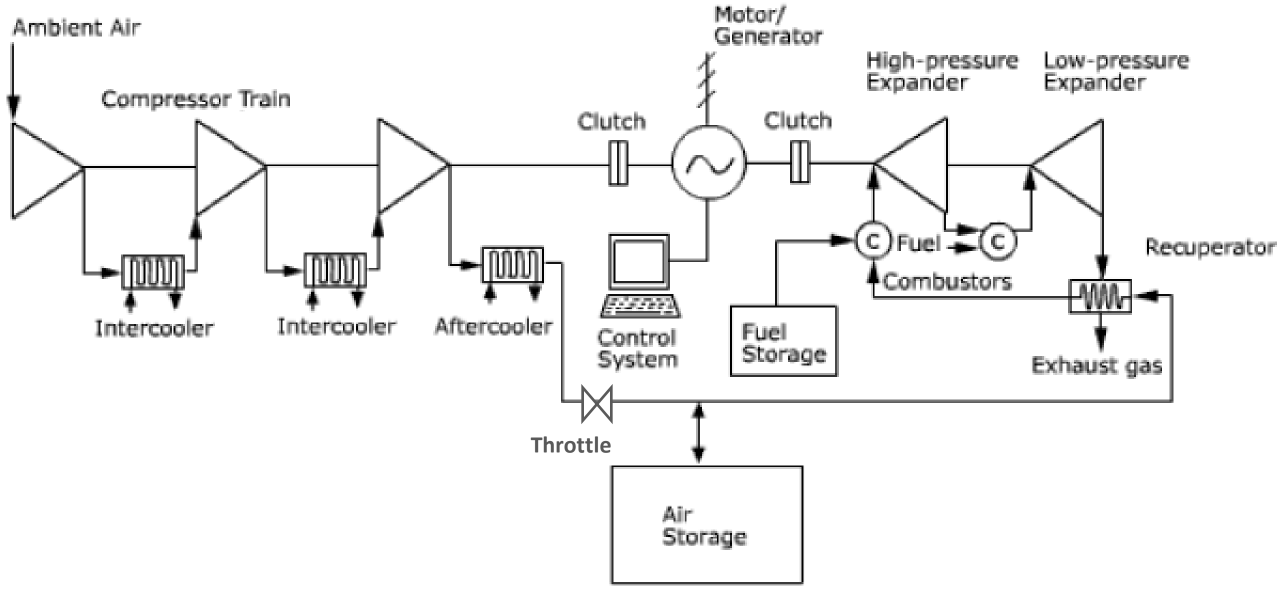

Fig. 1. Scheme of the studied plant [21].

The compressor train is modelled into three stages: the first one represents an axial compressor, while the second and third ones are centrifugal compressors. The first stage has a design flow rate of 120 kg/s, a design pressure ratio of 21 and a design isentropic efficiency of 86%. An inlet pressure of 1 bar is assumed.

The second and third stages have a design pressure ratio of 2.5 and 1.8 respectively. The model of the compressor is maintained fixed for both scenarios.

A mechanical efficiency of 90% is assumed. Table 1 and 2 summarize the characteristic maps implemented in the model for axial and centrifugal stages, respectively.

Every stage is separated from the following by an intercooler represented by tube heat exchangers air-water counter-current and modelled at partial loads according to the ε-NTU method, using the maps reported in Fig.2, where ω is the ratio between the thermal capacities of water and air.

Table 1. Reference values for the characteristic map of the axial compressor

$\dot{m}/\dot{m}_{des}$	Rc/Rc_{des}	η/η_{des}
0.667	0.574	0.953
0.708	0.620	0.965
0.750	0.667	0.966
0.792	0.722	0.977
0.833	0.778	0.978
0.875	0.816	0.983
0.917	0.853	0.983
0.958	0.929	0.989
1.000	1.000	1.000
1.042	1.062	0.991
1.083	1.122	0.965

Another air-water heat exchanger is inserted at the inlet suction of the first compressor stage, with the purpose to maintain a constant inlet temperature and unlink the performance of the axial compressor from the temperature of the external air. An after-cooler is positioned at the outlet section of the last compressor stage with the aim to maximize the volume of air that is placed in the cavern.

Table 2. Reference values for the characteristic map of the centrifugal compressor

$\dot{m}/\dot{m}_{des}$	Rc/Rc_{des}	η/η_{des}
0.705	1.00	0.928
0.725	0.967	0.964
0.788	0.945	0.988
0.815	0.927	0.994
0.833	0.909	0.998
0.870	0.858	0.999
0.906	0.818	0.996
0.928	0.782	0.976
0.951	0.727	0.964
0.960	0.673	0.934
0.978	0.636	0.904
0.987	0.600	0.880
0.993	0.545	0.843
0.997	0.509	0.798
1.000	0.473	0.693

The major limitation in this model is that it does not consider the variation of the overall heat transfer coefficient as a function of the load. The design and sizing of each heat exchanger was carried out in order to find the values for its satisfactory performance. The cooling water flow rates are determined to allow operation at a pressure of 1 bar without reaching conditions of vaporization or solidification. The power associated to the pumps is about 225 kW.

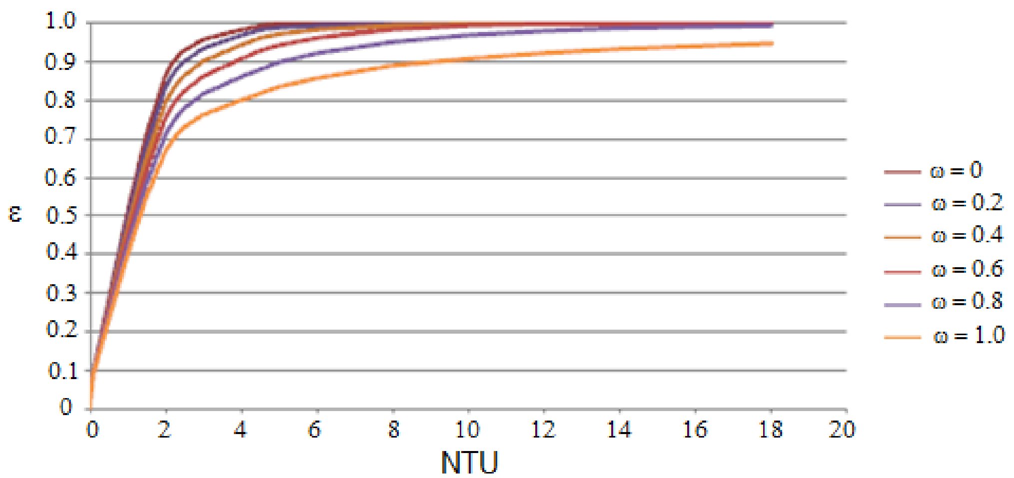

Fig. 2. Characteristic map of tube heat exchangers counter-current implemented in the software.

The system compressor-throttle-air storage has been modelled according to the performance of the compressor.

The reference model is based on the theory of gas flows through a throttling. The evolution of the mass flow through the valve depends on the pressure ratio between the upstream (p_1) and downstream (p_b) of the throttling.

$$\dot{m} = \frac{p_1 A_r}{\sqrt{R\,T_{in}}} \cdot f(k, p_b/p_1) \tag{1}$$

The coefficient f, function of k and the pressure ratio, is equal to:

$$f(k, p_b/p_1) = \sqrt{\frac{2k}{k-1}\left[\left(\frac{p_b}{p_1}\right)^{\frac{2}{k}} - \left(\frac{p_b}{p_1}\right)^{\frac{k+1}{k}}\right]}, \text{ if } \frac{\mathrm{p_b}}{\mathrm{p_1}} > \frac{p_{crit}}{p_1}, \text{ where } \frac{p_{crit}}{p_1} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}}, \quad (2)$$

or to

$$f(k, p_b/p_1) = \sqrt{\frac{2k}{k-1}\left[\left(\frac{2}{k+1}\right)^{\frac{2}{k-1}} - \left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}\right]}, \text{ if } \frac{\mathrm{p_b}}{\mathrm{p_1}} < \frac{p_{crit}}{p_1}. \quad (3)$$

For k =1.4 as for air, $p_{crit} \approx 0.528\ p_1$.

The achievement of a sonic flow regime is prevented thanks to a specific control. The optimal flow-rate signal from the throttle is connect with the compressor. The algebraic loop obtained in *Simulink* is resolved with a dynamic delay between the two signals.

The variation in time of the thermodynamic properties of the air stored is regulated by characteristic differential equations. Under the hypothesis that the cavern is rigid with an annual loss of air negligible, the equation that describes the variation of pressure p is derived from polytrophic equation:$\rho = Kp^n$, (ρ is the density, K a constant) where n depends on the type of transformation (case isothermal 1, case adiabatic 1.4). Since the air enter at 323 K, like the temperature of the surrounding terrain of the cavern, is assumed an exponent n equal to 1.

$$\dot{m}_{in} - \dot{m}_{out} = V\frac{d\rho}{dt} = \frac{\rho V}{np}\frac{dp}{dt} = \frac{V}{nRT}\frac{dp}{dt} = C\frac{dp}{dt} \quad (4)$$

where V is the volume of the tank (assumed 300000 m^3), R is the specific air constant, T is the temperature of stored air and C is the so called capacity characteristic of the storage.

The variation of the air temperature inside the cavern takes into account the thermal power associated with the entering and exiting mass flow rates and the thermal power exchanged by thermal conduction between the cavern and the external environment.

$$\frac{dT_{air}}{dt} = \frac{\dot{m}_{in}c_{p,air}(t_{in}-t_{air}) - \dot{m}_{out}c_{p,air}(t_{air}-t_{ground}) - \lambda_{terr}\mathrm{F}(t_{air}-t_{ground})}{\rho_{air}c_{p,air}V_{air}} \quad (5)$$

with obvious meaning of the symbols (F is the shape factor of the cavern, equal to 567 m). The variation of the air mass stored is defined by the equation $\dot{m}_{in} - \dot{m}_{out} = \frac{dm}{dt}$.

The pressure of the stored air is always maintained between about 25 bar and 63 bar, according with the mass flow-rate range that can be elaborated by the compressor. Between the tank and the inlet section of the expansion block a valve has been inserted. This throttle reduces the pressure of the air coming from the tank to 46.2 bar, when the latter is characterized by a higher pressure, while is bypassed if the pressure is lower than the control value.

The gas turbine is modelled through two expansion stages, both characterized by the same map that provides the off-design specification of mass flow rate, expansion ratio (Re) and isentropic efficiency (Fig.3).

The total design expansion ratio is equal to 46. The first stage is characterized by a design expansion ratio equal to 46/11.2, the second stage has an expansion ratio equal to 11, always insured. The gas turbine always operates in choked conditions, hence the mass flow rate elaborated is assumed constant. The two scenarios are characterized by the same expansion ratio for the gas turbine, but by two different values for the air mass flow-rate elaborated by the turbine. The mechanical efficiency is assumed equal to 92%.

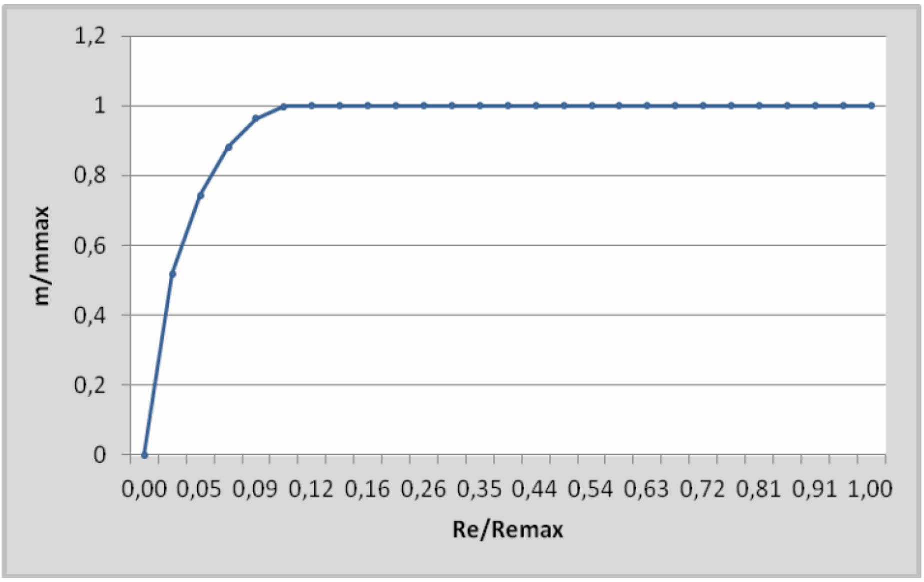

Fig. 3. Characteristic map of the gas turbine.

The two turbine stages are connected to two different combustion chambers, one operating at a pressure of 46.2 bar, the other at a pressure of 11.2 bar. The drop pressure in each chamber is assumed equal to 0.2 bar. The fuel utilized is natural gas (NG). The air / fuel ratio in the first combustion chamber is fixed at 50.

The characteristic equation qualifying the first combustion chamber refers to stationary conditions:

$$t_{gas,out} = \left[\dot{m}_{air} C_{p,air}\left(T_{air,in} - 298.15\ K\right) + \frac{\dot{m}_{NG} LHV_{NG} \eta_{CC}}{\dot{m}_{gas} c_{p,gas}}\right] + 298.15\ K \tag{6}$$

with obvious meaning of the symbols. The combustion efficiency assumed is equal to 0.98.

The second combustion chamber has a dynamic behaviour because of the variable expansion ratio of the first turbine. It is modelled in order to ensure a constant temperature of the exhaust gas at the last stage of the expansion, in order to guarantee the good operation of the heat recovery unit and the first combustion chamber. As regards the turbine start-up inertia, the power output signal has been filtered to simulate an inertia of about 10 minutes.

The heat recovery unit that heats the cold air coming from the tank through the hot exhausted gases coming from the low-pressure turbine stage allows a saving of fuel in the first combustion chamber of about 22%. The recovery unit has been modelled through a shell and tube heat exchanger, operating in co-current. The characteristic equations are defined by two differential equations that describe the variation of the output temperatures of gas and air in time.

$$C_{gas} \frac{dT_{gas,out}}{dt} = \dot{m}_{gas} C_{p,gas}\left(T_{gas,in} - T_{gas,out}\right) - \frac{T_{gas,out} - T_{air,out}}{R_{b-a}} \tag{7}$$

$$C_{air} \frac{dT_{air,out}}{dt} = \dot{m}_{air} C_{p,air}\left(T_{air,in} - T_{air,out}\right) + \frac{T_{gas,out} - T_{air,out}}{R_{b-a}} \tag{8}$$

where R_{b-a} is the thermal resistance that takes into account the heat exchange both by convection and conduction through the metal wall of the tube. The convection coefficient is variable with the scenario assumed and calculated through the correlation of Dittus-Boelter (for the exhausted gases in tube) and the method of Kern (for the air in the shell).

Exhausted gases can’t reach an outlet temperature below 473 K.

3. Operation strategy methodologies

The control method is based on the variation of the hourly price of energy and the variation of the pressure of the air stored in the tank.

The reference market price of energy is the Italian Single National Price (PUN) for the year 2013, whose trend is shown in Fig.4. The control strategy, conditioned by the hourly price of energy, is based on the annual average hourly price P_{avg}. This strategy is called *peak shaving strategy*, which plans to buy energy in the hours where the price is lower than the price P_{avg}-ΔP and sell energy when the price is higher than P_{avg}+ΔP, where$\Delta P = P_{avg} \cdot \frac{(1-k)}{(1+k)}$, k is the product of the efficiencies of compression and expansion. From the analysis, the average annual price is equal to 62.99 €/MWh while the symmetric bands are identified at 68.92 €/MWh and 57.06 €/MWh.

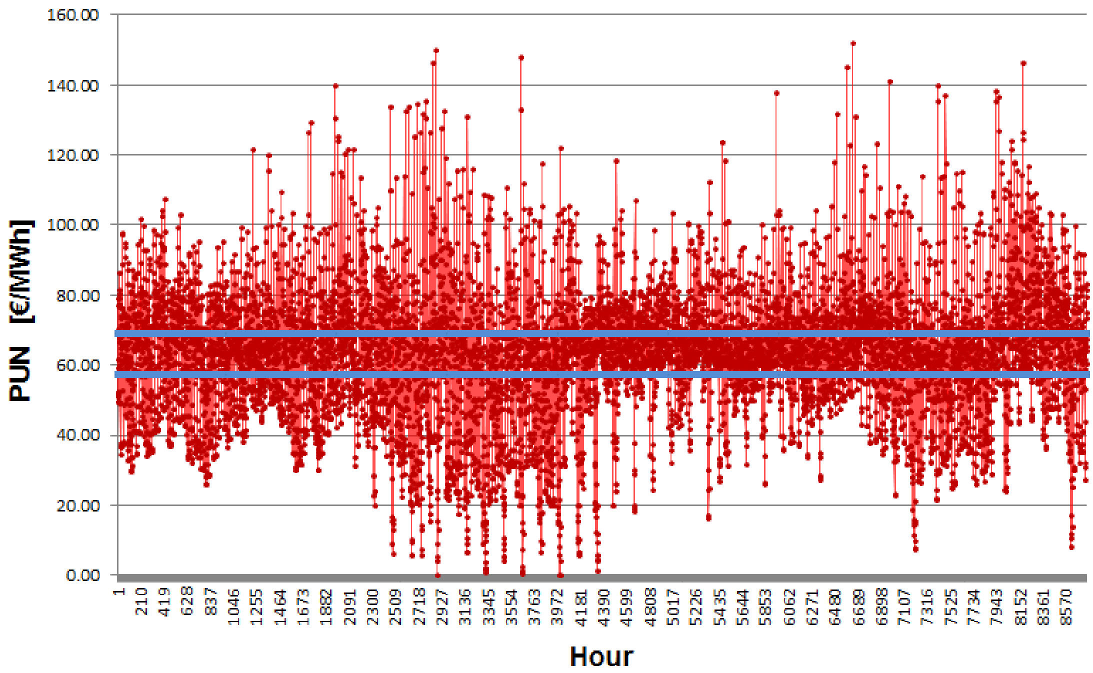

Fig. 4. Evolution of the annual hourly price of energy and relative band.

In practice, the strategy identifies a symmetric band with respect to the average annual energy price and amplitude ΔP, by storing air in the tank when the hourly price is below the band and by generating energy when the hourly price is above the band.

Consequently, the plant model permits the simultaneous operation of the compressor and the turbine.

In the second scenario, the turbine may works when the Wind Farm power production is below a certain limit.

Other parallel strategy control is based on the variation of the pressure of the air stored in the cavern. For both scenarios, the turbine can be switched on when the air pressure is greater than 25 bar. The control operation of the compressor is different depending on the considered scenario. In the first scenario the compressor can be switched on when the air pressure in the tank is below 46 bar. The lack of an algorithm that considers also the variable costs of the fuel and the machines is compensated with a strict control on the thermodynamic conditions of the air stored. A lower value of pressure control means lower variable costs of turbine and compressor but also loss of earnings. The strategy is well suited with the uncertainty fluctuations of the spot markets prices.

The second scenario considers an annual wind distribution found in literature and referred to Irish wind conditions for 2009 (Ballintlea South, Cork). The Wind Farm is represented by 60 VESTAS turbines with a rated power of 3 MW for each unit. The annual capacity factor of the Wind Farm is

29.3% with a peak power production of about 136 MW.

The connection between the wind farm and the CAES plant has been chosen as not direct. A direct connection means absorption of energy by CAES plant during peak Wind Farm production and the generation of energy by CAES plant during periods of low wind intensity. For the Wind Farm and CAES plant defined in this paper, the energy storage system can't offer a durable and significant support to the Wind Farm [12, 22].

The non-direct connection means that the compressor is switched on when hourly price of energy is convenient *or* the pressure of the air stored in the tank is below a certain critic limit, imposed at 55 bar. The gas turbine is activated when the hourly price of energy is convenient or the Wind Farm power production is below 35 MW *and* the pressure of the air stored in the tank is higher than a certain limit (25 bar).

3.1 Sensitivity analysis

The economic analysis ensures a high degree of reliability, since the calculation of costs and revenues from the annual simulation is based on accurate fluctuations of energy prices, but is highly sensitive to economic parameters such as the cost of investment and the discount rate. From these considerations, it has been carried out a sensitivity analysis on these fixed parameters that greatly affect the result. The financial-economic indexes taken in consideration are the Net Present Value (NPV), the discounted pay-back time and the index of profit, defined as the quotient between NPV and initial investment cost [13, 21- 24].

The economic and financial parameters common to both scenarios are as follows:

- Discount rate = 6%;
- Cost of natural gas = € 0.34/m^3;
- Annual price of natural gas = +0.7%/year;
- Annual electricity price = +2.2%/year;
- Investment CAES cost, including cave, components and equipment necessary = 0.625 mil €/MW;
- Cost CAES annual O&M = 0.0095 mil€/MW/year;
- Life of the investment CAES = 30 years.

4. Results

For both the scenarios, an annual simulation has been performed. Fig.5 reports the evolution of the pressure in the cavern and the mass flow rate in the first week of January for the first scenario. Note that the compressor is activated when the price is convenient or the pressure of the air stored is below a critic value.

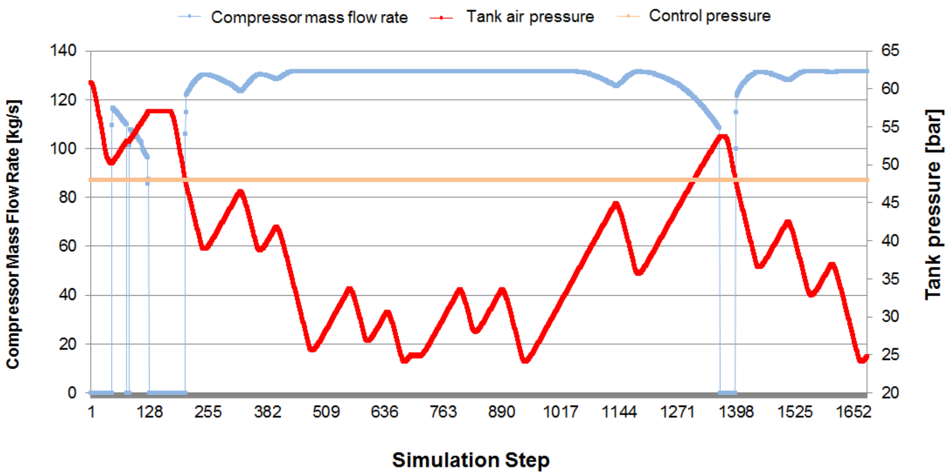

Fig. 5. Evolution of the pressure of the air stored (red line) and the mass flow-rate elaborated from the compressor (blue line) in the first week of the year. The results are referred to the first scenario.

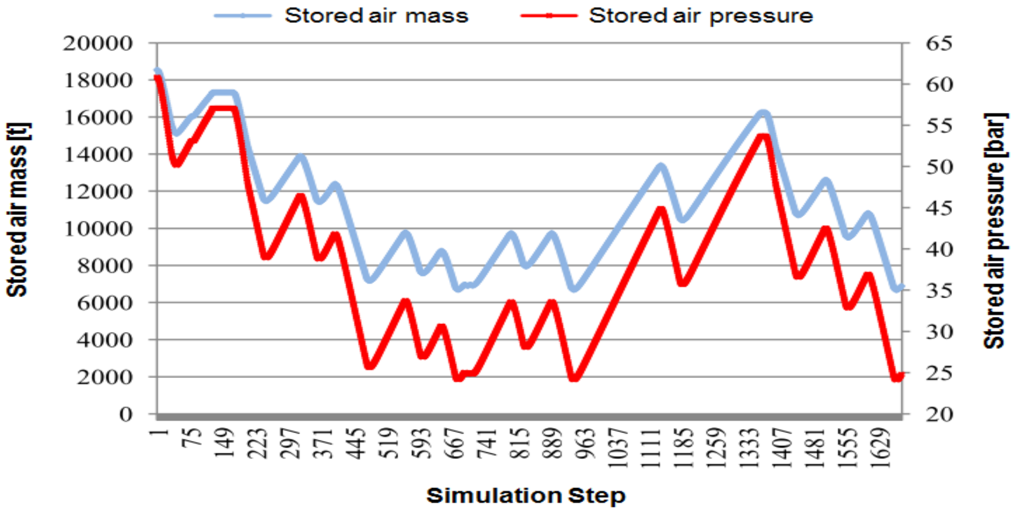

Fig. 6. Evolution of the air mass stored (blue line) and the relative pressure (red line) in the first week of the year. The trends are referred to the first scenario.

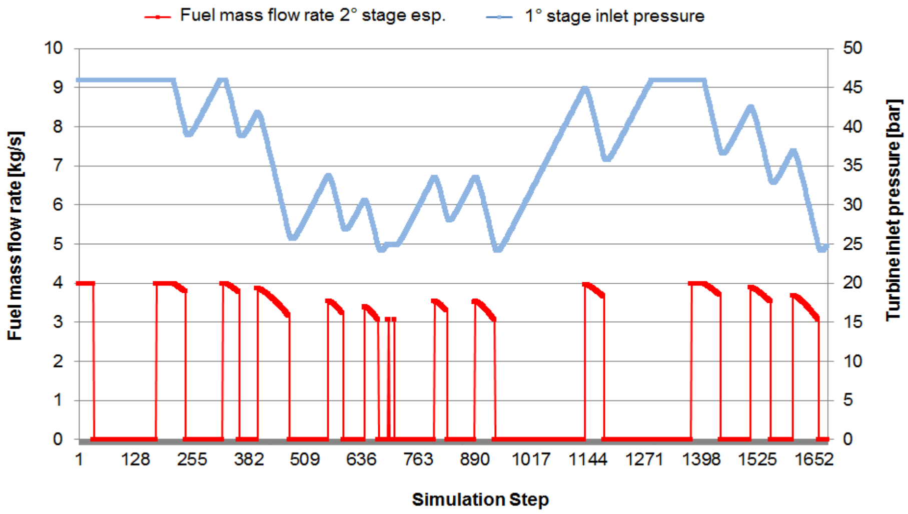

Fig. 7. Fuel mass flow (red line) in the second combustion chamber as a function of inlet air pressure to the first stage of expansion (blue line) in the first week of the year for the first scenario.

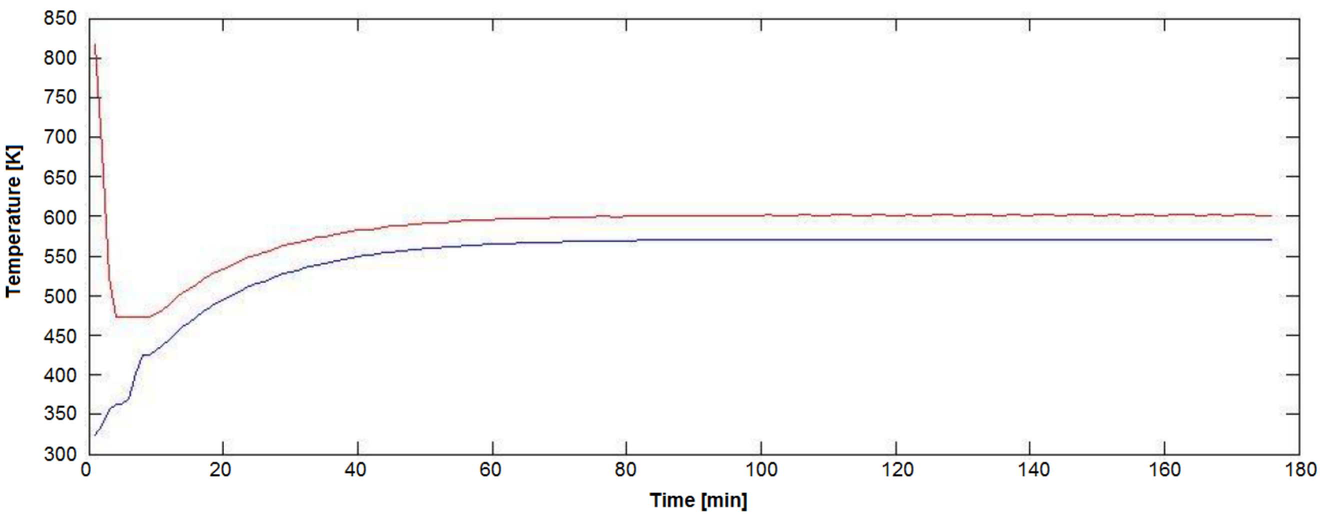

Fig. 8. Output temperature of the exhaust gas and of the air entering the first combustion chamber.

Fig. 6 shows the evolution of the mass and its pressure in the cavern. Fig. 7 reports the trends of fuel mass flow rate and air pressure in the first week of January for the first scenario. Also the turbine is activated when the price is convenient.

The annual simulation of the two scenarios shows the results reported in Table 3.

Table 3. Results of the thermodynamic parameters for the first scenario

Parameter		*First scenario*	*Second scenario*
Compressor operating hours	h/y	4487	4458
Rated power turbine	MW	290	128
Nominal flow turbine	kg/s	310	140
Hours turbine operation	h/y	3641	5343
Natural gas design flow rate	kg/s	10.2	4.5
Annual fuel flow rate	mil m^3	7.57	7.04
Total energy produced	TWh	645	647
Total energy consumed	TWh	525	499
Simple cycle efficiency	-	0.575	0.392
Round-trip efficiency [1]	-	0.948	0.624
Heat-rate[2]	-	0.631	
Energy-ratio[3]	-	3.314	

[1] defined as $E_{turb}/\left(E_{NG} * 0.476 + E_{compr}\right)$

[2] defined as E_{turb}/E_{NG}

[3] defined as E_{turb}/E_{compr}

For the first one the air mass at the initial time is 18,550 tons. The air mass stored at the end of year is 10,400 tons. The total net earnings, not discounted, from the first year of operation are equal to 5,283,837 €.

For the second scenario, the initial air mass is of 18,550 tons. The air mass stored at the end of year is 16,440 tons. The total net earnings, not discounted, from the first year of operation are equal to 6,168,976 €. The increase of the capacity factor of the Wind Farm is significant, approximately of 140%, from 29.3% up to 70.3% for the integrated system. Currently, the ancillary service offered by the storage system does not provide economic returns in the electricity market. In reference to the Italian imbalanced market for wind energy, the calculations of the charges or premiums are evaluated by an algorithm based on direct measurements on wind turbines taken in consideration and on the previous predication conducted by the operator of the electricity market. Currently, in Italy, an average value of this fee is 7.7 c € per MWh. The premiums are evaluated for CAES plant.

For the first scenario, at the thirtieth year the NPV calculated is negative. The capital cost of the investment at year zero is prohibitive, equal to 181,250,000 €, and the annual revenues can't cover the cash outlay.

Table 4. NPV for the first scenario related to the first and thirtieth year

Year	Earnings	Discounted earnings	NPV
1	€ 5,283,837	€ 4,984,752	-€ 176,265,248
30	€ 11,619,043	€ 2,022,993	-€ 81,768,315

For the second scenario, at the thirtieth year the NPV calculated is positive. The capital cost of the investment at year zero is lower than first scenario, equal to 80,000,000 €.

This has a direct influence on the economic results.

Table 5. NPV for the second scenario related to the first and thirtieth year

Year	Earnings	Discounted earnings	NPV
1	€ 6,168,976	€ 5,819,789	-€ 74,180,211
30	€ 14,082,526	€ 2,451,910	€ 38,440,140

The discounted pay-back is between the 17th and 18th year, the index of profit is about 54%. The calculation of the NPV for the Wind Farm is not taken in consideration.

As mentioned earlier, a sensitivity analysis is proposed which aims to highlight the change in the NPV of the CAES plant valued at thirtieth year. The most critical indices from a financial standpoint are the discount rate and investment cost per MW of the CAES plant.

Table 6. Sensitivity analysis on NPV valuated at the thirtieth year for the first scenario.

[m€/MW]	Capital cost CAES	5%	6%	7%	8%
0.438	-30%	-13,759,357	-27,393,315	-38,721,761	-48,198,612
0.469	-25%	-22,821,857	-36,455,815	-47,784,261	-57,261,112
0.500	-20%	-31,884,357	-45,518,315	-56,846,761	-66,323,612
0.563	-10%	-50,154,357	-63,788,315	-75,116,761	-84,593,612
0.625	0%	-68,134,357	-81,768,315	-93,096,761	-102,573,612
0,688	10%	-86,404,357	-100,038,315	-111,366,761	-120,843,612
0.750	20%	-104,384,357	-118,018,315	-129,346,761	-138,823,612

Table 7. Sensitivity analysis on NPV valuated at the thirtieth year for the second scenario.

[m€/MW]	Capital cost CAES	5%	6%	7%	8%
0.,438	-20%	70,790,953	54,440,140	40,862,342	29,510,713
0.469	-10%	62,726,953	46,376,140	32,798,342	21,446,713
0.500	0%	54,790,953	38,440,140	24,862,342	13,510,713
0.,563	10%	46,726,953	30,376,140	16,798,342	5,446,713
0.625	20%	38,790,953	22,440,140	8,862,342	-2,489,287

5. Conclusions

The simulations have shown that the most critical issues related to the operating performance of the CAES system are dependent on the strategy control and the maintenance of optimum conditions of the air stored. The CAES system is highly flexible, especially when compared with other conventional thermal plants and the ability to manage the phases of compression and expansion over time, makes it competitive in the electricity market. The economic analysis has, however, shown that the investment cost is still not competitive. The analysis also highlighted how the size of the system can greatly affect the profitability of the plant: although the earnings can be substantial, large sizes, up to 250 MW, result in important investments that present a difficult economic return. Smaller sizes, below 150 MW, present a lower risk, a greater attraction to the investment and can find a significant penetration in the electricity market. A management system based only on the variability of the energy prices is feasible but highly risky, given also that energy prices are expected to undergo rapid changes due also to the introduction of renewable energy sources. The direct connection of the CAES storage system to renewable energy sources not programmable, such as e.g. wind power systems, may not be significant and durable. The intermittence of wind power, especially in countries with limited wind intensity, is too strong for a considerable enhancement of the integrated system. Thus, a non-direct connection can offer a more significant support to the Wind Farm with the aim to level the Wind Farm power output and its annual capacity factor.

In this case, incentive systems need to be present for a significant economic support for the penetration of CAES systems into the electricity market.

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