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Compressed Air Energy Storage: Optimal Performance and Techno-Economical Indices

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Abstract

A thermodynamic and techno-economical analysis of a Compressed Air Energy Storage system subjected to an exogenous periodic electricity price function of the interconnection is presented. The fundamental parameters affecting the thermodynamic performance and the techno-economical cost-benefit indices are identified and corresponding optimisation problems are formulated. The results of the analysis permit to obtain the optimal values of the fundamental plant parameters to be used in the design process.

Key words: energy storage, CAES, compressed air, optimization, technoeconomics

1. Introduction

The largest share of the energy generated by a gas turbine is consumed by its compressor. This fact combined with the fluctuations in the demand for power and its consequent time of use pricing formed the motivation for the development of the Compressed Air Energy Storage (CAES) technology. The CAES technology consists of converting excess base load energy into stored pneumatic energy by means of a compressor for a later release through a gas turbine (turbo-expander) as premium peaking power. As the operation of the compressor is decoupled from the operation of the turbo-expander the whole amount of power produced by the turbo-expander is available at the generator terminals (except for minor electro-mechanical losses).

Although storage is a major component in CAES, this technology is not a pure storage system as fuel is added to the compressed air in a combustor prior to its expansion through the turbo-expander. An adiabatic alternative can be considered (without fuel consumption), however its viability should be assessed as the outcome of a techno-economical analysis and is therefore a design option. Therefore CAES, although an energy storage technology, is a hybrid system which includes both storage and generation from fuel consumption, unless the adiabatic alternative is adopted.

The CAES system consists of two major parts. The first is the machinery, which includes typical elements of an industrial gas turbine, with possibilities of intercooling the air during the compression process, or aftercooling, reheating and recuperating as design options. The second part of the system is the underground compressed air reservoir, which can be either of a constant pressure type, e.g. an aquifer or a depleted gas reservoir, or of a constant volume type (variable pressure), e.g. a salt dome cavern. Other types of reservoirs like excavated caverns in hard rock with or without a water compensating system to maintain an almost constant pressure, or abandoned mines have been considered. The reservoir technology, utilised over the last fifty years for seasonal natural gas storage, can be applied almost without variation to store compressed air for the CAES system.

The significant difference between peak and off-peak prices has created the motivation to develop energy storage technologies. Electric utilities often apply energy storage methods to meet daily, weekly and seasonal variations in the power load demand. Electric energy storage technologies exist for many years. The main proven technologies are pumped hydro, battery storage and flywheel energy storage.

Although all the components of a Compressed Air Energy Storage system represent proven technologies, their combination reached only very recently (with the commissioning of the CAES plant in Alabama, U.S.A.) the status of a proven technology, which has many inherent advantages. However, its implementation as a commercial one is in its beginning. A 290 MW CAES power plant has operated successfully since 1979 in Huntorf, Germany. The first unit (110 MW) of a 220 MW power plant was commissioned in 1991 in the USA by Alabama Electric using a salt-dome reservoir at the McIntosh site. A CAES 30MW pilot plant is being constructed in the island of Hokkaido, Japan. A 300MW CAES plant is being planned for construction in Mount Sedom, Israel. Another CAES plant is being considered for Taiwan. ESKOM, the South African Electric Utility considers too the option of a CAES plant for energy storage purposes.

The thermodynamic cycle parameters which affect the performance of the CAES system were identified and optimised in numerous papers e.g. Vadasz et al. (1989), Vadasz and Weiner (1988), Vadasz et al. (1987), Vadasz and Weiner (1986), although some of them consider a simplified CAES system, i.e. without reheaters or recuperator. The thermodynamic analysis is closely related to the techno-economical analysis and optimisation of the machine.

Novel CAES alternatives were proposed by Schnaid, Weiner and Brokman (1995) for combined production of power and cold and others were reviewed by Touchton (1996) suggesting a CAES cycle which is essentially a derivative of a Cascaded Humidified Advanced Turbine system (de Biasi, 1995).

The thermodynamic performance of the CAES system has a direct impact on the techno-economical viability of the plant as it affects its running cost component. The objective of the present paper is to present some results of a performance analysis and to indicate how the different thermodynamic parameters affect the efficiency of a general CAES system, i.e. including intercoolers, reheaters and a recuperator . Ultimately, a techno-economical optimisation utilises the performance analysis results to come out with the optimal design solution.

The following analysis although assumes a constant pressure type of reservoir can be easily adapted to apply for other types of reservoirs as required.

2. Thermodynamic Analysis

The CAES system consists of a compressor for charging the air into the reservoir, a combustion chamber and a turbo-expander, as presented in *Figure 1*. A motor/generator is connected through clutches to the compressor and the turbine. While in the charging mode the motor which uses off-peak power drives the compressor to inject the air into the reservoir. During peak periods the CAES system is operated in the discharging mode. Then, the compressed air is released from the reservoir and introduced into the combustion chamber where fuel is added and combusted to provide high temperature gases. The combustion gases expand through a turbine, that drives the generator to provide peaking power. A particular alternative design is possible by considering thermal storage techniques to accumulate the heat of compression for a later utilisation during the discharging process. Then the combustion chamber can be excluded, therefore providing a fuel independent system. This option is referred to as the adiabatic CAES alternative.

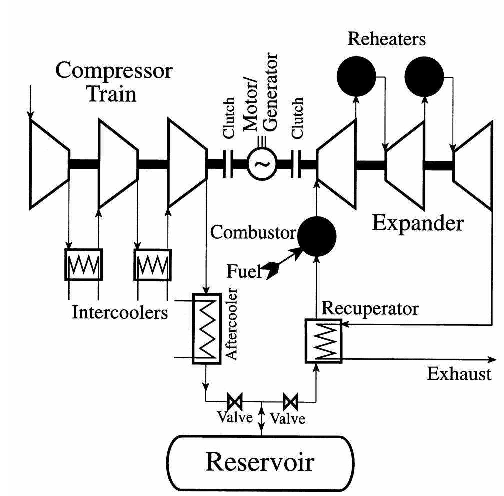

Figure 1. Schematic description of a compressed air energy storage system.

Previous studies, e.g. Vadasz, Pugatsch and Weiner (1989) considered a performance analysis of a CAES system without reheaters and provided a method of analysis to maximise the efficiency. One of the significant results from Vadasz et al. (1989) was the very narrow design space for the adiabatic alternative and the corresponding low efficiency associated with this adiabatic option. For this reason in the present paper the adiabatic design option is not considered, but reheating stages during the expansion process are included and expected to provide an insight on the degree of attractivity to include reheaters in the design of CAES systems.

A CAES thermodynamic cycle is presented in *Figure 2* on a T-s diagram. The processes involved are as follows:

1-2' : Compression processes, involving (n-1) intercooling stages and one aftercooling stage (the $n^{\underline{th}}$ stage) 2'-3.

2'-3 : An aftercooling process as mentioned above.

3-4 : Constant pressure (almost) preheating in the recuperator.

4-5 : Constant pressure (almost) combustion in the combustor.

5-6' : Expansion processes, involving (m-1) reheating stages.

6'-7 : Constant pressure heat transfer in the recuperator.

7-1 : Constant pressure heat transfer released through the exhaust to the environment is the process which closes the cycle.

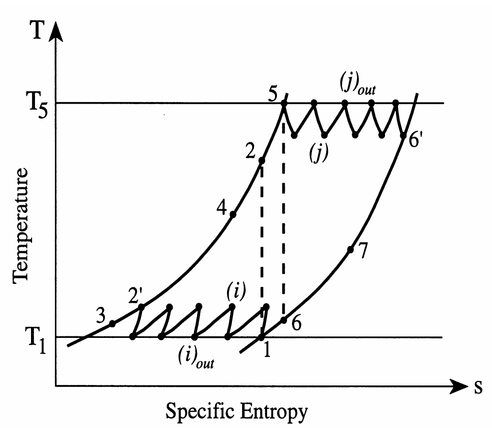

Figure 2. Temperature-entropy (T-s) diagram - CAES thermodynamic cycle.

The series of processes from state 1 to state 3 occur while the plant is operated in a charging mode, i.e. the left clutch is engaged (see *Figure 1*), the Motor/Generator operates as a Motor using grid power to drive the compressor which delivers compressed air to the reservoir. The left valve is open while the right valve is closed. This is followed by an off mode of operation when the plant is idle but the reservoir is charged. Then, during peak demand periods the plant is operated in a discharging mode, when the right clutch is engaged, the left valve is closed, the right valve is open and the Motor/Generator operates as a Generator. During this mode of operation the series of processes from state 3 to state 7 and the exhaust process 7-1 occur, producing net output power to be delivered to the grid by the Generator. This mode is followed by another off mode of operation, when the plant is idle again but the reservoir is at a low charging level (i.e. it maintains the initial necessary amount of compressed air to keep the pressure almost constant during the charging and discharging processes). It should be emphasised that during the constant pressure processes mentioned above some pressure losses exist. These pressure losses were taken into account in the analysis although not being graphically represented in *Figure 2*. When the indices representing the states appear in parentheses they refer to the intercooling or reheating stages during the compression and expansion processes respectively.

The thermal efficiency of the CAES thermodynamic cycle is defined in the form

$$\eta_{th} = \frac{w_t}{\dfrac{w_c}{\eta_{ex}} + q_f} \qquad (1)$$

where w_t and w_c stand for the specific work of expansion and compression, respectively, q_f is the specific heat from combustion in the combustor and reheaters and η_{ex} is an external thermal efficiency of a coal fired steam power plant (or any other type of base load dedicated power plant) which provides the power for the compression process. Thus the term w_c/η_{ex} is equivalent to the heat required at the coal fired steam power plant (or the alternative base load plant) to produce a compression work, w_c for the CAES plant. It can be observed that when no heat of combustion is supplied to the compressed air, i.e. q_f=0, then the efficiency of an adiabatic CAES system is obtained from Eq.(1) in the form

$$\eta_{th}^{ad} = \beta\eta_{ex} \qquad (2)$$

where β is the storage effectiveness defined by

$$\beta = \frac{w_t}{w_c} \qquad (3)$$

It should be pointed out that for comparison of the CAES system with other energy storage technologies, like pumped hydro-energy storage (PS) (which is an adiabatic system and therefore its relevant index of performance is the storage effectiveness β), the thermodynamic efficiency as given by Eq.(2) can be used for the pumped hydro-energy storage, while Eq.(1) is to be used for Compressed Air Energy Storage. It is a mistake to compare the values of β between CAES and PS.

The objective of this analysis is to maximise the thermodynamic efficiency of the CAES cycle and to determine the optimal set of parameters' values which correspond to the maximum efficiency. Therefore the thermodynamic parameters which affect the efficiency are identified and introduced into Eq.(1). While more extensive analysis can be performed by considering cooling of the turbo-expander's high stages we limit the present analysis to cases when these effects are negligible. The optimal location of intercoolers (as presented by Vadasz, Pugatsch and Weiner (1988) and by Vadasz and Weiner (1992)) and reheaters was evaluated and expressed in terms of the terminal isentropic temperature ratio $R = T_2/T_1 = (p_2/p_1)^{(\kappa-1)/\kappa}$ and is inherently included in the relationships for the compression and expansion specific work and for the specific heat of combustion supplied in the combustor and reheaters. These relationships for the different processes were developed considering mass and energy balances in these processes yielding the following results

$$w_c = \frac{c_p T_1}{\eta_c \eta_{elm}} n\left(\sigma_c R^{1/n} - 1\right) \quad (4)$$

$$w_t = c_p T_1 \eta_t \eta_{elm} r_{mt} m \left(1 - \frac{\sigma_t}{R^{1/m}}\right) \quad (5)$$

$$\left\{ \frac{L}{\quad} - \varepsilon \left[+ \eta \left(\frac{\sigma}{\quad} - \right) \right] \right. \qquad - \frac{\quad}{\quad}(- \varepsilon \quad) + \left[- \frac{\sigma}{\quad} \right] \Bigg\} \quad (6)$$

where R is the terminal isentropic temperature ratio defined above, $r_{mt}=T_5/T_1$ is the maximum temperature ratio, $r_{st}=T_3/T_1$ is the storage temperature ratio representing the ability of the reservoir to store thermal energy as well. The recuperator effectiveness is defined by $\varepsilon_{RC}=(T_{6'}-T_7)/(T_{6'}-T_3)$, while η_t and η_c are the turbine and compressor efficiencies, respectively, representing the degree of deviation of the real expansion and compression processes from the corresponding isentropic processes. While polytropic efficiencies would have been more appropriate to use, it can be shown that the deviation between the results using polytropic efficiencies and the present ones using isentropic efficiencies differ by less than 3% in the estimation of the temperature ratio $\left(T_2'/T_1\right)$, and less than 6% in the estimation of the specific work, over a wide pressure ratio range ($1<r_p<280$). Therefore we use the isentropic efficiencies for compression and expansion processes which simplifies the analysis. An electromechanical efficiency η_{elm} is introduced as well to account for electrical and mechanical losses (e.g. in the motor/generator) in converting the turbine output to electrical power (or electrical power to compressor input). The other notation used in Eqs. (4) - (6) stands for the number of intercoolers (n-1), the number of reheaters (m-1), and parameters accounting for pressure losses in the intercoolers, reheaters, valves, reservoir, combustor and recuperator.. These parameters are $\sigma_t = \mu_t \mu_r^{1/m}$ which is a global pressure losses factor accounting for pressure losses in reservoir, valve, piping, combustor, reheaters and recuperator, $\sigma_h = \mu_t \mu_r$ and $\sigma_c = \mu_c^{(n-1)/n}$ which stands for pressure losses in the intercoolers. The following definitions for the μ's hold:

$$\mu_c = \left(p_{(i)}/p_{(i)out}\right)^{(\kappa-1)/\kappa}$$

$$\mu_t = \left(p_{(j)}/p_{(j)out}\right)^{(\kappa-1)/\kappa}$$

$$\mu_r = \left(p_{6'}/p_1\right)^{(\kappa-1)/\kappa}$$

The constant pressure specific heat c_p is taken constant and assumed the same value for air as for the combustion gases although the later have a value which is by 10% higher than air, a fact which penalises by 10% the amount of power generated by CAES systems. This approximation can be relaxed if a more detailed and accurate analysis is required.

The specific fuel consumption in kg fuel/kWh generated is

$$\frac{\dot{m}_f}{\dot{W}_t} = \frac{\dot{Q}_f}{H_f \dot{W}_t} = \frac{q_f}{H_f w_t} = \frac{1}{H_f}\left\{1 + \frac{\sigma_h/R}{m\left(1-\sigma_t/R^{1/m}\right)} - \frac{\varepsilon_{RC}[1+\eta_t(\sigma_h/R-1)] + r_{st}/r_{mt}(1-\varepsilon_{RC})}{m\left(1-\sigma_t/R^{1/m}\right)}\right\} \quad (7)$$

where H_f is the low heating value (kWh/kg) and the ratio between the specific work of expansion and specific work of compression is the energy storage effectiveness β, as defined in equation (3). By using the relationships for the compression and expansion specific work, i.e. equations (4) and (5) respectively, it yields the following equation for β

$$\beta = \frac{w_t}{w_c} = \frac{\eta_t \, \eta_c \, \eta_{elm} \, r_{mt} \, m\left(1-\sigma_t/R^{1/m}\right)}{n\left(\sigma_c R^{1/n} - 1\right)} \quad (8)$$

These thermodynamic relations will be useful in the techno-economical analysis.

The thermal efficiency of the CAES system is evaluated by substituting Eqs.(4)-(6) into Eq.(1) and yields

$$\eta_{th} = \frac{\eta_{telm}\eta_{cxlm} m\left[1-\frac{\sigma_t}{R^{1/m}}\right]}{\frac{1}{r_{mt}}\left\{n\left[\sigma_c R^{1/n}-1\right] - \eta_{cxlm} r_{st}(1-\varepsilon_{RC})\right\} + \eta_{cxlm}\left\{\frac{\sigma_h}{R} - \varepsilon_{RC}\left[1+\eta_t\left(\frac{\sigma_h}{R}-1\right)\right] + m\left[1-\frac{\sigma_t}{R^{1/m}}\right]\right\}} \quad (9)$$

From equation (9) it can be observed that the set of parameters which affect the efficiency consists of the following (R, r_{mt}, ε_{RC}, r_{st}, n, m). The analysis regarding the optimal value of r_{mt} is similar as in Vadasz, Pugatsch and Weiner (1989) and hence only the conclusions will be repeated here. Since r_{mt} appears only in the denominator, maximising the efficiency with respect to r_{mt} is equivalent to minimising the denominator. This results in obtaining the maximum efficiency on the boundaries of r_{mt}, i.e. either $r_{mt,max}$ or $r_{mt,min}$. The choice between the two is made according to a transition criterion on the values of R. The lower boundary of r_{mt}, i.e. $r_{mt,min}$ represents the adiabatic option which is not discussed here as mentioned earlier and therefore the optimal value of $r_{mt} = r_{mt,max}$ is set for the maximum temperature ratio. For the optimal isentropic temperature ratio the partial derivative $\partial\eta_{th}/\partial R = 0$ is set equal to zero resulting in a non-linear algebraic equation in terms of R for the extrema of η_{th}. The method is similar to that applied by Vadasz, Pugatsch and Weiner (1989). Solving for the values of R within the admissible domain yields the set of optimal values for different values of m and n. The optimal values of m and n are also evaluated, although a different process is involved in this case as the result must remain integer for both m and n.

3. Optimal Perfomance Results.

The optimal results of the analysis are presented in *Figures 3 - 4*. The results in *Figure 3a* for the optimal values of R show that R^*

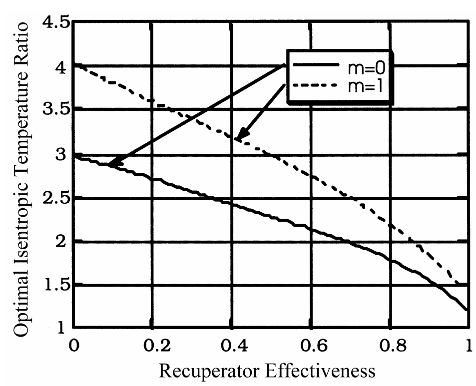

Figure 3a. Optimal isentropic temperature ratio

decreases as ε_{RC} increases. The same applies for the optimal number of intercoolers (*Figure 3b*), i.e. less intercoolers represent the optimal solution as the recuperator effectiveness increases. By relating these last two optimal results for R^* and n^* one obtains the graph presented in *Figure 4a*. Finally, *Figure 4b* presents the optimal results for the efficiency as a function of the recuperator effectiveness for a case with one reheater or without reheaters at all. As ε_{RC} gets closer to 1 the results converge to a value of 46% for the maximum efficiency.

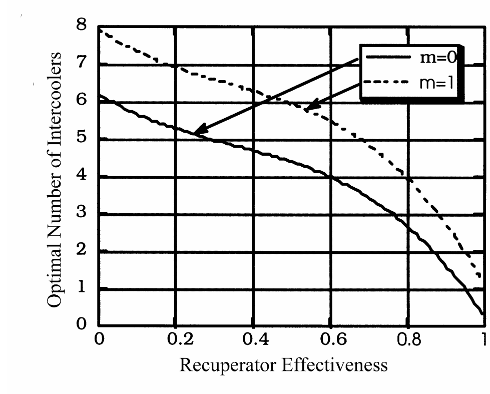

Figure 3b. Optimal number of intercoolers

The performance analysis can be concluded by stating that this example demonstrates how the performance of a CAES system can be optimised. The tendency of the optimal values obtained here will be maintained in the running cost component of the techno-economical optimisation.

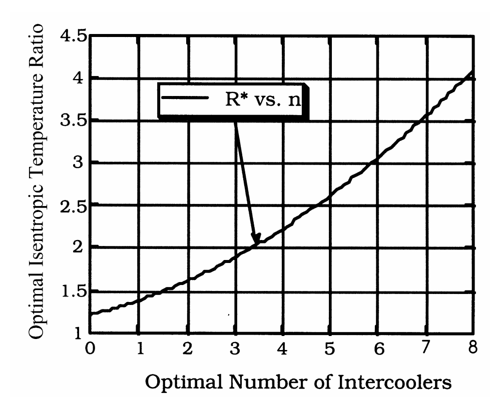

Figure 4a. Optimal isentropic temperature ratio related to the optimal number of intercoolers.

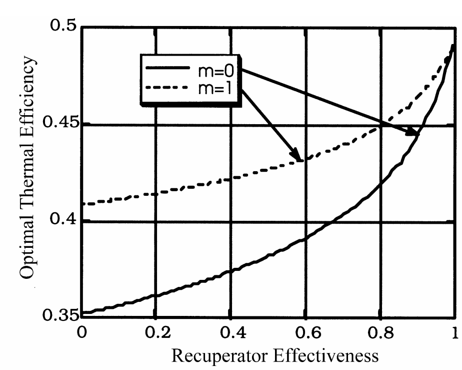

Figure 4b. Optimal thermal efficiency related to recuperator effectiveness.

4. Techno-Economical Background

There are two major approaches to provide an optimal plant design.

The first approach, which is adequate while a specific project is to be considered, selects from an inventory of existing equipment the most economical combination of technically available components to the project. This approach practically compares these combinations and comes out with the most economical one. (see Nakhamkin et al. (1991))

The second approach, "ignores" temporarily the data about availability of equipment and optimises the techno-economical indices by assuming that equipment is available for parameters values within the allowed range of variation. Then, either the available equipment having parameters which are the closest to the optimal calculated values is selected, or manufacturers are requested to develop equipment to fit the optimal specifications (see Vadasz and Weiner (1986)). Although the first approach is more practical for an ongoing project having to meet dead-lines and scheduled to be operational at a certain date, the second approach is more effective for analysing a new emerging technology and indicating what could be the economical benefit should a new generation of design and consequently equipment be developed. Therefore, this second approach is viable for long term technology development purposes.

Previous studies, e.g. Vadasz and Weiner (1986), using this approach considered a very simple CAES system, which excluded intercoolers, aftercooler, recuperator and reheaters. The present analysis is a significant extension of the simple cycle considered by Vadasz and Weiner (1986), since it includes intercoolers, recuperator and reheaters as design options into the cycle. The difficulty in incorporating these elements is to establish their normative cost functions, i.e. functions which depend only on intensive thermodynamic properties in addition to other system parameters. An example of providing such normative cost functions for compressors and turbines in CAES systems was demonstrated by Vadasz and Weiner (1987).

5. Marginal Cost and Price Functions

As the economics of a CAES system is dependent upon the instantaneous price of electricity which in turn depends on the load demand curve and is given by the marginal cost of power production at a given time it is necessary to use these data as input to the techno-economical optimisation process. The way to do so is to evaluate first the instantaneous price function using a representative load curve over a typical cycle of operation of the plant. This can be either a daily or a weekly cycle as required. The instantaneous price is then evaluated by using the principle of optimal scheduling of generation which shows that the relationship between the price and the load demand is piecewise linear and concave upward. Therefore, following Vadasz (1989), a second order Taylor expansion can be used as an approximation of this relationship in the form

$$P(t) = c_0 + c_1 N(t) + c_2 N^2(t) \qquad (10)$$

where P(t) is the instantaneous price, N(t) is the load demand at time t and c_0, c_1 and c_2 are constants. Their value will be established from the known values of the price at the minimum, maximum and average load demand. Once the instantaneous price function has been evaluated for a given load demand curve, charging and discharging price functions dependent on charging and discharging duration can be evaluated by proper integration of the instantaneous price curve. The advantage in using charging and discharging price functions is the transformation of the initial dynamic problem (i.e. time dependent) into a static problem by introducing the charging and discharging duration instead of time. Hence, the problem does not include the time explicitly and the charging and discharging duration are introduced as parameters into the system. It can be pointed out that the discharging duration is practically equivalent to the plant service factor. This process of converting the load demand data into charging and discharging price functions was presented by Vadasz (1989). It is used here as input to the optimisation process and typical charging and discharging price functions are presented in *Figure 5b.* They were evaluated using a typical daily load demand curve and its corresponding instantaneous price function as presented in *Figure5a.*

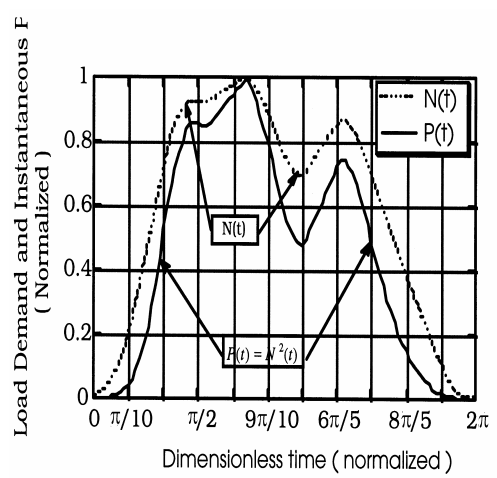

Figure 5a. Daily load demand curve and its corresponding instantaneous price.

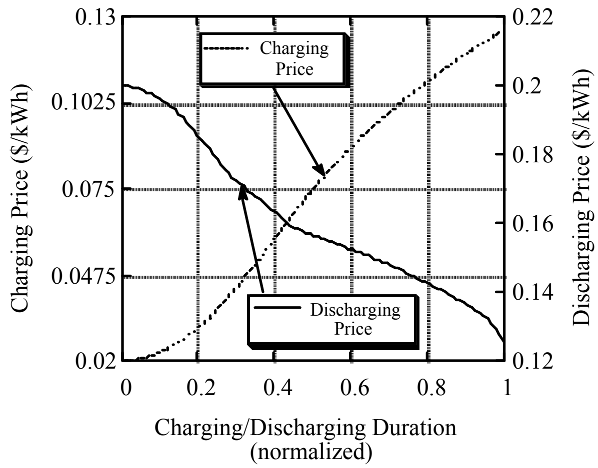

Figure 5b. Charging and discharging price functions corresponding to Figure 5a.

6. Cost-Benefit Target Functions

6.1 The cost function

The specific cost function is composed of two main cost components; namely variable and fixed charges referring to the running and capital cost respectively. These components are represented in the expression of the total annual cost ($/kW-yr) of a CAES plant, C_{tot}, as follows

$$C_{tot} = C_1 K + S h_d + C_{FOM} \quad [\$/\text{kW-yr}] \qquad (11)$$

where C_1 is the specific capital cost ($/kW installed), K is the capital recovery factor (1/yr), S is the specific variable cost ($/kWh), h_d is the plant service factor expressed in units of (operating hours/year) and C_{FOM} is the fixed operating and maintenance cost ($/kW-yr). It should be emphasised that S and all its components are expressed in terms of $ per kWh generated. The next paragraphs present the way in which the cost function depends on the CAES system parameters.

6.2 The energy cost

The variable component of the cost function, S, combines together the energy cost of charging (network electricity) and discharging (fuel) and variable operating and maintenance expenses in the form

$$S = C_{ch} + C_d + C_{vom} \quad [\$/\text{kWh}] \qquad (12)$$

where C_{ch}, C_d are the charging and discharging costs respectively, both expressed in terms of $/kWh generated and C_{vom} is the variable operating and maintenance cost ($/kWh generated) which is considered constant. Since the charging price function P_c as presented in *Figure 5b* is given in terms of $ per kWh of compression, the charging cost per kWh generated, C_{ch} may be calculated using the transformation

$$C_{ch} = P_c \frac{w_c}{w_t} = \frac{P_c}{\beta} \quad [\$/\text{kWh generated}] \qquad (13)$$

where w_c is the compression specific work, and w_t is the specific work of expansion in the turbine which is equal to the net specific work obtained at the generator terminals. We can introduce the thermodynamic relation (8) into Eq.(13) to obtain

$$C_{ch} = \frac{P_c}{\eta_t \eta_c \eta_{elm}} \frac{n\left(\sigma_c R^{1/n} - 1\right)}{r_{mt} m \left(1 - \sigma_t / R^{1/m}\right)} \quad [\$/\text{kWh gen.}] \qquad (14)$$

Equation (14) expresses the charging cost as a function of the fundamental parameters of the CAES cycle, R, r_{mt}, n, m and of the charging price function $P_c(h_c)$. The efficiencies η_t, η_c and η_{elm}, and the pressure losses factors σ_c and σ_t are considered constant.

The discharging cost, C_d per kWh generated is related to the fuel price, P_f ($/kg fuel) and to the specific fuel consumption, $\dot{m}_f / \dot{W}_t$ (kg fuel/kWh generated) as presented in Eq.(7). Therefore, one can present the discharging cost in the form

$$C_d = P_f \frac{\dot{m}_f}{\dot{W}_t} = \frac{P_f}{H_f} \frac{q_f}{w_t} = P_{Hf} \left\{ 1 + \frac{\sigma_h / R - \varepsilon_{RC}[1 + \eta_t(\sigma_h / R - 1)] - (1 - \varepsilon_{RC}) r_{st} / r_{mt}}{m\left(1 - \sigma_t / R^{1/m}\right)} \right\} \qquad (15)$$

where the heat price P_{Hf} is defined as the ratio between the fuel price and the low heating value, i.e.

$$P_{Hf} = \frac{P_f}{H_f} \quad (\text{\$/kWh heat of combustion}) \qquad (16)$$

Finally equations (14) and (15) can be introduced into Eq.(12) to obtain a relationship between the running cost component and the CAES plant parameters.

6.3 The capital cost

The capital cost, C_I is expressed in terms of its components in the form

$$C_I = r_w C_c + C_t + r_g C_g + r_w C_{in} + C_{re} + C_{RC} + C_r + C_s \qquad [\$/kW_{inst.}] \qquad (17)$$

where C_c, C_t, C_g and C_{in} represent the cost per kW installed of compressor, turbine, generator and intercoolers, respectively. The coefficients r_w and r_g are used to convert the cost of all components and express it in terms of \$ per kW installed of turbine, as some components' cost are established per kW installed of compressor or motor. Hence

$$r_w = \frac{\dot{W}_c}{\dot{W}_t} \quad \text{and} \quad r_g = \frac{\dot{W}_g}{\dot{W}_t} \qquad (18)$$

are the compressor to turbine capacity ratio and the generator to turbine capacity ratio, respectively. Since the generator capacity is over-rated to meet the maximum between the turbine and compressor capacity, it is given by

$$\dot{W}_g = \max\left[\dot{W}_c, \dot{W}_t\right] \qquad (19)$$

and therefore the value of r_g is found by introducing Eq.(19) into Eq.(18)

$$r_g = \max\left[r_w, 1\right] \qquad (20)$$

The other terms in Eq.(17) are the cost of reheaters (including the cost of the combustor) C_{re}, the cost of the recuperator C_{RC}, the cost of the reservoir C_r and supplementary costs including pipe lines, clutches, valves and any other costs not explicitly included in the other terms.

Some useful relationships between the plant parameters can be established by using the mass conservation equation for the air flowrate over a complete CAES cycle as follows

$$\frac{\dot{m}_c h_c}{M} = \frac{\dot{m}_t h_d}{M} \qquad (21)$$

where $\dot{m}_c$, $\dot{m}_t$ are the mass flowrate in the compressor and turbine respectively, h_c, h_d are the charging and discharging duration, respectively (expressed in hours per year) and M is the number of cycles per year. By introducing the definition of the compression and expansion specific work as

$$w_c = \frac{\dot{W}_c}{\dot{m}_c} \qquad w_t = \frac{\dot{W}_t}{\dot{m}_t} \qquad (22)$$

into the mass balance equation (21) we obtain

$$r_h = \frac{h_d}{h_c} = \frac{\dot{W}_c}{\dot{W}_t}\frac{w_t}{w_c} \qquad (23)$$

where the new parameter r_h represents the discharging-charging duration ratio. Therefore by using (23), (18) and (8) we get the following equation which relates the discharging-charging duration ratio to the compressor-to-turbine capacity ratio

$$r_h = r_w \beta \qquad (24)$$

where the dependence of β on thermodynamic parameters is given by Eq.(8). Thus the value of r_w is constrained by Eq.(24) due to mass balance considerations.

The major difficulty in defining the capital cost is to establish normative cost functions which depend on intensive thermodynamic properties only and other (non-property) system parameters. An example of providing such normative cost functions for compressors and turbines in CAES systems was demonstrated by Vadasz and Weiner (1987). Accordingly, their specific cost (\$/kW installed) is related to the known cost of a reference type of compressor and turbine through the thermodynamic system parameters in the form

$$C_c = \alpha_{co} \frac{\left[R^{\kappa/(\kappa-1)} - \eta_c (R-1) - 1\right]}{R^{\kappa/(\kappa-1)}(R-1)} \qquad (25)$$

$$C_t = \alpha_{to} \frac{\left[1 + \eta_t \left(\frac{\mu_t}{R} - 1\right) - \left(\frac{\mu_t}{R}\right)^{\kappa/(\kappa-1)}\right]}{\left(1 - \frac{\mu_t}{R}\right)} \qquad (26)$$

where α_{co} and α_{to} are coefficients which depend on the reference data only

$$\alpha_{co} = \frac{C_{co} R_o^{\kappa/(\kappa-1)} (R_o - 1)}{\left[R_o^{\kappa/(\kappa-1)} - \eta_c (R_o - 1) - 1\right]} \qquad (27)$$

$$\alpha_{to} = \frac{C_{to}\left(1 - \frac{\mu_t}{R_o}\right)}{\left[1 + \eta_t \left(\frac{\mu_t}{R_o} - 1\right) - \left(\frac{\mu_t}{R_o}\right)^{\kappa/(\kappa-1)}\right]} \qquad (28)$$

Thus, whenever the compressor and turbine costs are known for a specific thermodynamic set of parameters (the reference set) their corresponding costs for a different set of parameters can be evaluated using Eqs.(25) and (26). These equations express the common fact, stating that (for the same flowrate) the higher the pressure ratio, the lower the additional incremental cost ($/kW installed) of a com-pressor or a turbine. Similarly, normative cost approximations are being developed for the intercoolers, recuperator and reheaters (including the combustor). As such the cost of intercoolers and recuperator were developed by assuming that it is proportional to the heat transfer area of these heat exchangers.

6.4 The net benefit and yield functions

The economical benefit indices can be presented either as the net annual benefit function B, or the yield function defined as the net annual benefit normalised with respect to the annual cost, i.e. $Y = B/C_{tot}$. Both B and Y can be used as target functions in the optimisation process. The net annual benefit function is obtained by subtracting the total cost function (11) from the revenues. The later are given by the product of the discharging price and the discharging duration. Therefore

$$B = (P_d - S)h_d - C_I K - C_{FOM} \quad [\$/\text{kW-yr}] \quad (29)$$

where the discharging price $P_d(h_d)$ is a function of the discharging duration, h_d (which is equivalent to the plant service factor) and is presented in *Figure 5b*.

7. Method of Optimisation

As indicated in Eq.(29) the net benefit function B, would be affected by seven independent variables represented by the vector $x = (r_{mt}, R, r_h, h_d, \varepsilon_{RC}, m, n)^T$. Since these variables are constrained due to thermodynamic, technological and economical reasons, they are mathematically represented by inequality constraints which are often called regional constraints because they restraint complete regions in the design space. The admissible design space is presented as a hyper-rectangle in the form

$$r_{st} \le r_{mt} \le 4.91 \quad (30)$$

$$\sigma_h \le R \le 3.5 \quad (31)$$

$$0 \le r_h \quad (32)$$

$$0 \le h_d \le \gamma \quad (33)$$

$$0 \le \varepsilon_{RC} \le 1 \quad (34)$$

$$1 \le m \le m_{max} \quad (35)$$

$$1 \le n \le n_{max} \quad (36)$$

where γ represents an additional constraint resulting from the requirement that the CAES plant cannot operate in the charging and discharging mode at the same time, thus

$$h_d\left(1 + \frac{1}{r_h}\right) \le 8760 \quad (37)$$

Since the objective function to be maximised (29) is non-linear, a mathematical programming algorithm is to be applied for the optimisation. The BCONF subroutine of the IMSL Library (IMSL Inc. (1991)) was used for this purpose. The value of γ was established through an iterative procedure which guaranties that the additional constraint (37) is satisfied.

8. Optimal Techno-Economical Results

An example of using the proposed procedure is presented for demonstration purposes. Typical cost data were taken from Vadasz and Weiner (1986). The heat price P_{hf} was varied between 0.01 and 0.1 $/kWh of heat from combustion and the optimisation procedure was repeated for each incremental value of P_{hf}. The resulting optimal values of the isentropic temperature ratio and of the recuperator effectiveness are presented in *Figure 6* together with the corresponding maximum values of the net annual benefit. The results show that the optimal recuperator effectiveness increases with increasing the heat price reaching a value of 0.51 for the maximum heat price considered. The optimal isentropic temperature ratio also increases with increasing the heat price as long as the heat price is not excessively high (i.e. less than 0.078 $/kWh). Over and above a critical heat price value ($P_{hf,cr} = 0.078$ $/kWh) the optimal isentropic temperature ratio reaches its maximum value ($R^* = 3.5$) on its upper boundary and it remains unchanged as the heat price increases. The net annual benefit decreases smoothly when passing through this critical heat price point. It should be emphasised that at a critical value of $P_{Hf,cr}$ Vadasz and Weiner (1986) identified a transition to the adiabatic CAES option (see *Fig. 7*) from their simple model (i.e. excluding recuperator, reheaters and intercoolers). It is observed here (*Fig. 6*) that incorporating a recuperator introduces an additional degree of freedom in the optimal design and the results suggest utilising this option as a better alternative to an adiabatic system.

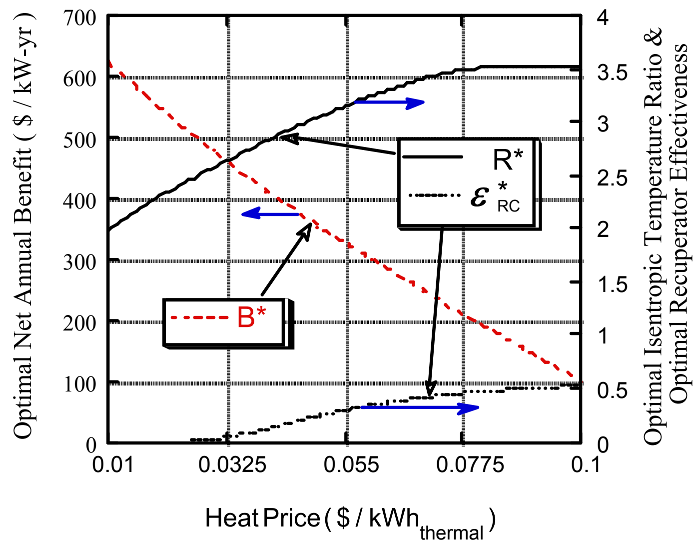

Figure 6. Optimal results from an example demonstrating the procedure.

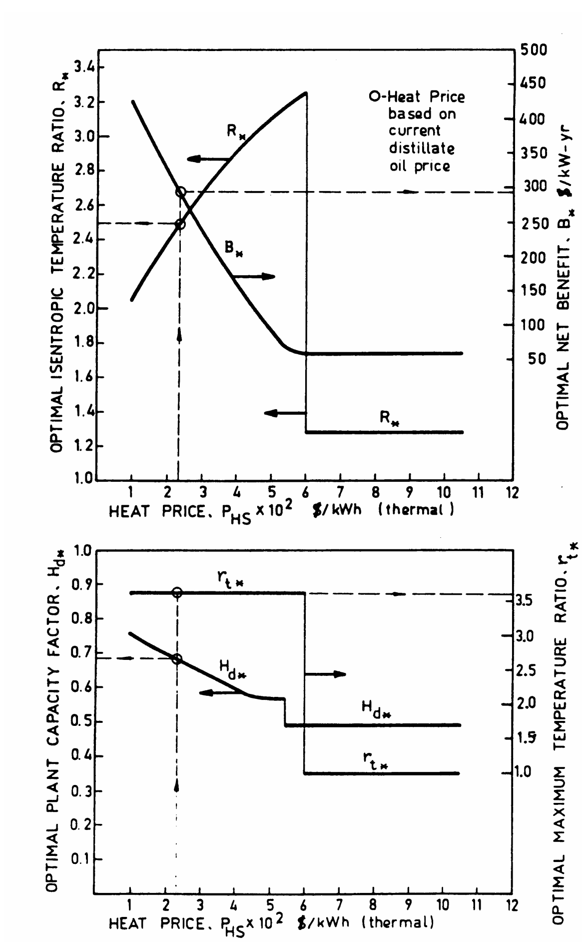

Figure 7. Optimal results using a simple model, Vadasz, Weiner (1986).

9. Conclusions

A demonstration of applying an optimal design procedure by using techno-economical analytical tools was presented for a Compressed Air Energy Storage system. The parameters affecting the cost-benefit balance of the CAES plant were identified and possible optimisation criteria were suggested. The results of the presented example show the advantage of using this procedure to analyse different design options and for sensitivity analysis of the project benefit to future unpredictable variations in fuel price.

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Nomenclature

B	Net annual benefit ($/kW-yr)
c_p	Constant pressure specific heat (J/kg^{0}C)
C_{tot}	Total annual cost ($/kW-yr)
C_I	Specific capital cost ($/kW installed)
C_{FOM}	Fixed operating and maintenance cost ($/kW-yr)
C_{ch}	Charging cost ($/kWh generated)
C_d	Discharging cost ($/kWh generated)
C_{VOM}	Variable operating and maintenance cost ($/kWh generated)
C_c	Compressor specific cost ($/kW installed of compression)
C_t	Turbine specific cost ($/kW installed turbo-expander)
C_g	Generator specific cost ($/kW installed of generation)
C_{in}	Intercoolers specific cost ($/kW installed of compression)
C_{re}	Reheaters specific cost ($/kW installed of turbo-expander)
C_{RC}	Recuperator specific cost ($/kW installed of expander)
C_r	Reservoir specific cost ($/kW installed of turbo-expander)
C_s	Supplementary specific cost ($/kW installed of expander)
h_c	Charging duration (h/yr) or dimensionless as specified
h_d	Discharging duration (h/yr) or dimensionless as specified
H_f	Low heating value (kWh/kg)
K	Capital recovery factor (1/yr)
m_c	Mass flowrate in the compressor (kg/h)
m_t	Mass flowrate in the turbo-expander (kg/h)
m_f	Fuel consumption (kg/h)
m	Number of reheaters (including the combustor)
n	Number of intercoolers (including the aftercooler)
p	Pressure
P	Instantenous price of electricity
P_c	Charging price ($/kWh)
P_d	Discharging price ($/kWh)
P_f	Fuel price ($/kWh)
P_{Hf}	Heat price ($/kWh)
q_f	Specific heat from combustion (kJ/kg)
Q_f	Rate of heat from combustion (kW)
r_h	Discharging-charging duration ratio
r_w	Compressor to turbine capacity ratio
r_g	Generator to turbine capacity ratio
r_{mt}	Maximum temperature ratio
r_{st}	Storage temperature ratio
R	Terminal isentropic temperature ratio
T	Temperature
w_c	Specific work of compression (kJ/kg)
w_t	Specific work of expansion (kJ/kg)
W_c	Compressor's installed capacity (kW)
W_g	Generator's installed capacity (kW)
W_t	Turbo-expander's installed capacity (kW)
Y	Yield function

Greek Symbols

α_{co}	Reference cost coefficient for compresor
α_{to}	Reference cost coefficient for turbo-expander
β	Energy storage effectiveness
ε_{RC}	Recuperator effectiveness, defined in the text following Eq. 6
κ	Polytropic constant
μ	Pressure losses coefficient
σ	Pressure losses factor
η_c	Compressore efficiency, i.e. the degree of variation from the isentropic compression

$\eta_{cxlm} = \eta_c\, \eta_{ex}\, \eta_{elm}$

η_{elm}	Electro-mechanical efficiency
η_{ex}	External thermal efficiency
η_t	Turbo-expander efficiency, i.e. the degree of variation from the isentropic expansion

$\eta_{telm} = \eta_t\, \eta_{elm}$

Subscripts

(i)	Refers to an intermediate intercooling stage
(j)	-Refers to an intermediate reheating stage
o	Reference value

Superscripts

*	refers to optimal values.

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