

Concept of the thermal integration of the compressed air energy storage system with the power plant

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Abstract

This paper presents the concept of an innovative hybrid system that integrates a compressed air energy storage system with a conventional power plant. Using simple mathematical models, the proposed hybrid system is compared with the classic adiabatic system. It also presents the results of more detailed thermodynamic analyses for the compressed air energy storage system, which is thermally integrated with a 600 MW coal-fired power plant. The first stage of integration enables the storage system to utilize the heat of compressed air (air cooling) for condensate heating, which results in partial replacement of the low-pressure regeneration of the power unit and subsequently in an increase in power. The second stage of integration heats the air during discharge of the energy storage system before the air expander, using the heat of superheated steam which is directed from the steam turbine bleeding to the high pressure regeneration exchanger. While integration organized in this way decreases power unit efficiency, it eliminates the need for (i) gaseous fuel in the energy storage system, as in the case of diabatic systems or (ii) heat storage, as in the case of adiabatic systems. Three variants of the hybrid system were analyzed. Evaluation of the hybrid system variants was made using energy storage efficiency as defined in the paper.

Keywords: Hybridization, CAES, Supercritical coal-fired power plant, Thermodynamic analysis.

1. Introduction

In countries where the energy system is based on coal, the increase in the share of intermittent energy sources, i.e. renewable sources using wind energy or solar energy [1], contributes to the adverse effects of load changes within units working in large, centralized power plants [2]. As a consequence it adversely affects the economics of the whole system. The need to reduce the power delivered to the system results here from the reduced demand for energy within national systems, which may be a consequence of periodic increases in the potential of renewable sources that have priority access to the network. The lack of ability to perform periodic, complete withdrawal of the units results in the need to maintain a technical minimum enabling a quick return to the nominal load. While at present the problem in Poland affects smaller coal units – most often 200 MW class units – it

is certain that modern power units with supercritical parameters will be forced to perform withdrawals more often [3] in response to planned shutdowns of obsolete power plant units and as the power installed in wind farms and solar power plants increases. Energy storage systems may be able to mitigate the negative effects of the larger share of renewables in the energy mix. A synergy effect can be demonstrated in the case of thermal integration of power plants with energy storage systems. Additional advantages can be found in relation to the possibility of eliminating the burning of gaseous fuel, as is the case of diabatic systems or eliminating heat storage from the system, which takes place in the case of the adiabatic systems concept.

This paper presents the idea of a system which integrates a compressed air energy storage system (CAES) [4–6] with a coal-fired energy unit and presents the results of thermodynamic analysis for selected integration cases.

2. Concept of integration

The method of integrating the compressed air storage system with the power plant is presented in Fig. 1. The essence

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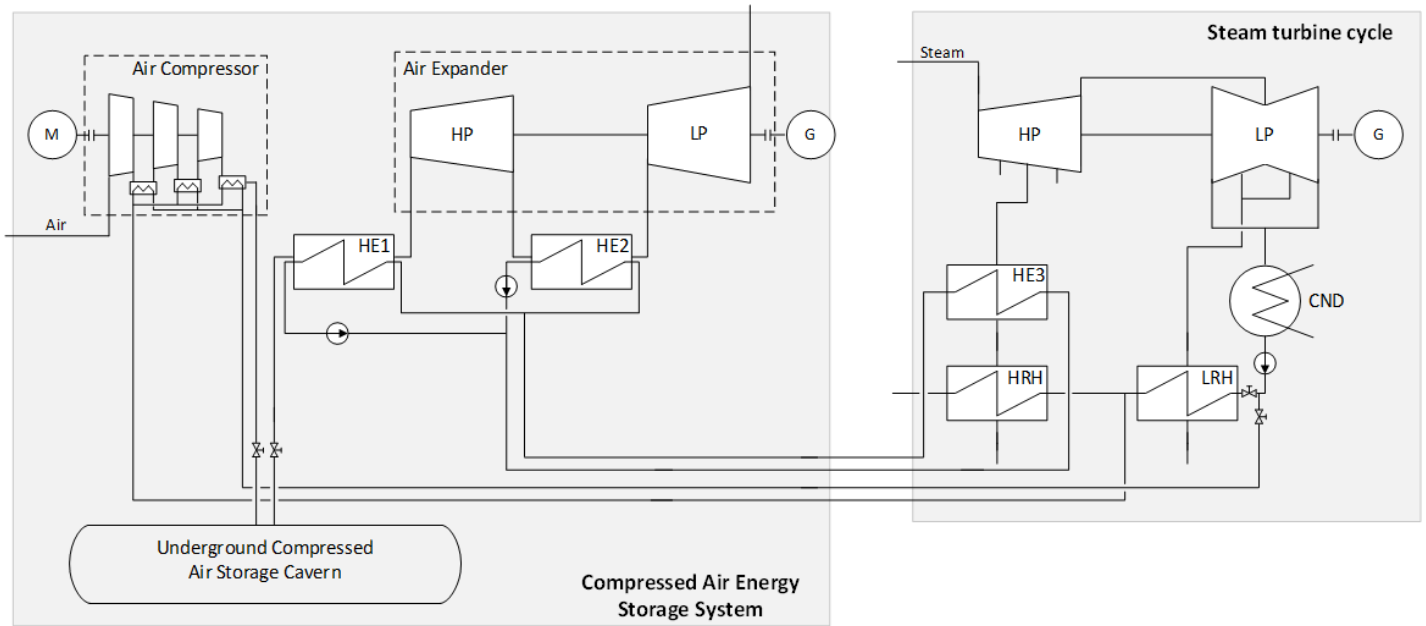


Figure 1: Concept of integration of CAES system with steam turbine cycle of coal-fired power plant

of integration consists in the fact that at the stage of charging the energy storage system, cooled compressed air is used for condensating heating directed to the selected low-pressure regeneration heat exchangers. The consequence of this procedure is the full or partial replacement of selected regenerative exchangers and, as a result, the reduction of steam intake from the steam turbine exhausts, which leads to an increase in the effective power of the turbine. The second stage of integration brings effects at the stage of discharging the energy storage system and involves the heating of air directed to the air expander (in Fig. 1 of the first and second expander section) by utilizing the heat of cooling superheated steam bled from the steam turbine and directed to selected high-pressure regeneration exchangers. The steam can be cooled down even to the saturation temperature. The consequence of the planned thermal integration is to achieve effective work of the air expander without the need to use, for example gas fuel, as is the case with the most common diabatic systems at present [7, 8]. The negative effect of integration is the reduction in effective power of the steam turbine (if there is an increase in steam flow supplying the high pressure regeneration exchangers) or the need to increase the load of the steam boiler (if the temperature of the boiler feed water lowers).

3. Evaluation of different variants of CAES system

In order to evaluate the concept of the system, a simple thermodynamic analysis was carried out for three CAES systems:

- hypothetical adiabatic system, without compressed air cooling (case A),

- adiabatic system with storage of heat from compressed air cooling (case B),
- a hybrid system integrating the CAES system with the thermal engine cycle (case C). Schemes for individual variants are presented in Fig. 2.

In case A, the system consists of: a one-section compressor, compressed air tank and expander. The system is hypothetical due to the very high air temperatures resulting from high pressure ratio due to air compression within the compressor, tank and expander. For the compressor and expander, the air temperature is limited by the properties of materials used whereas for underground tanks, depending on their type (salt caverns, rock chambers), high air temperatures may not only threaten the tank structure, but also contribute to high energy losses in the case of long term storage.

The system in case B was divided into two sections to protect the installation elements from adverse effects of high temperature. The first section implements an appropriate pressure ratio, after which the air is cooled down in an intersection cooler, where the heat received is dispersed in the environment. The second compressor section compresses the air to the target pressure level, and the compressed air before storage is cooled down in the after cooler, where the cooling medium is the heat storage medium. Stored heat is used at the stage of discharging the system to heat the air directed to the expander. At the design stage of the system, it is important to make a proper division of the total pressure ratio, which should be split between two sections, to be realized by the compressor in two sections. From a thermodynamic point of view, while maintaining the criterion of high safety of technology, the most appropriate division is the one that has the highest permissible temperature at the outlet from the

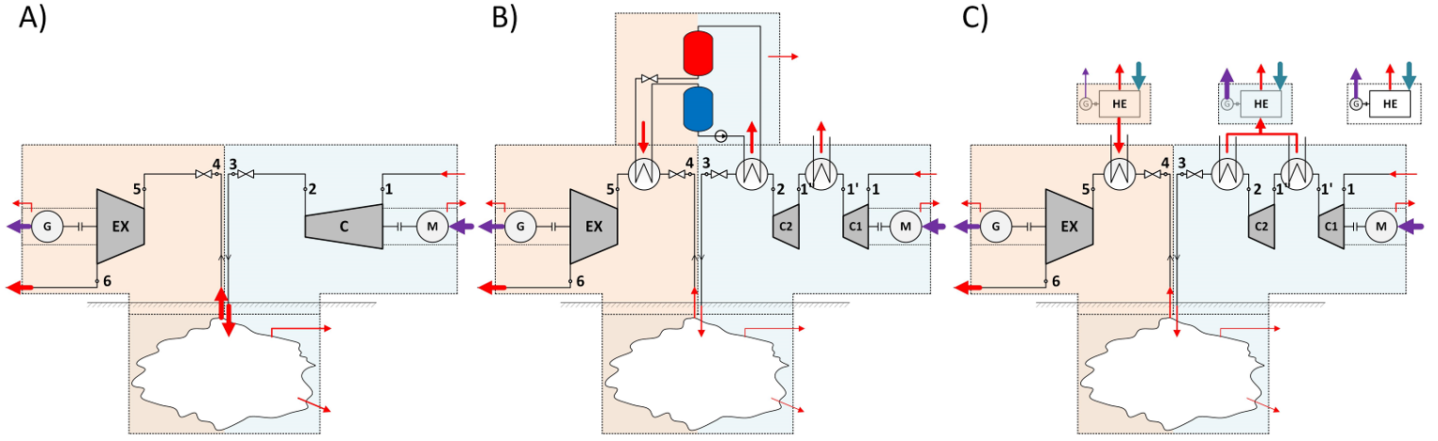


Figure 2: Schemes of variants of CAES system

second section of the compressor. With a combined pressure ratio, this will minimize the amount of heat the inter-sectional cooler dissipates to the environment. An idea similar to case B was adapted for the development of the Adele system, which is under development in Germany [9]. In this case, heat will be accumulated in a ceramic material.

Case C is the hybrid variant under consideration in this paper. It involves using the same structure of the CAES system as in case B. However, the heat received from the compressed air, in both the first and second section of the compressor, is not lost to the environment but is used in the thermal engine cycle, such as power plant. At the loading stage, this results in additional work being performed by the engine. At the unloading stage, the air before entering the expander is heated by the heat taken from the thermal engine cycle, which results in lowering its power. At the stage in which the air compressor and the CAES expander do not work, the engine works at its nominal load.

Energy storage efficiency is the basic indicator of the energy efficiency of the analyzed installations [10]. By a simple mathematical model relations between the basic parameters of the installation's operation and the energy storage capacities can be presented for the three cases under consideration. The efficiency of energy storage for systems compliant with cases A and B can be calculated according to the following:

$$\eta_{CAES} = \frac{E_{el,EX}}{E_{el,C}} \quad (1)$$

where: $E_{el,EX}$ - amount of electric energy produced by the expander unit during the decompression cycle, $E_{el,C}$ - amount of electric energy consumed by the compressor during its work.

In case C, thermal integration with the engine leads to a change in engine power at the loading stage and at the unloading stage. These integration effects mean that energy storage efficiency has to be redefined for a hybrid system. Such efficiency may be calculated by using the dependence:

$$\eta_{HCAES} = \frac{\Delta E_{el,EX} - \Delta E_{el,HE(II)}}{E_{el,C} - \Delta E_{el,HE(I)}} \quad (2)$$

where: $\Delta E_{el,HE(II)}$ – loss of energy production in a thermal engine due to high temperature heat exhaust used for air heating directed to the expander, $\Delta E_{el,HE(I)}$ – the amount of energy produced additionally by the thermal engine through the use of the cooling of hot compressed air.

The amount of electric energy required by the compressor and produced by the expander unit results from energy balances:

$$E_{el,C1} = m_1 c_p (T_{C_out} - T_{C_in}) \frac{1}{\eta_{emM}} \quad (3)$$

$$E_{el,EX} = (1 - \delta) m_1 c_p (T_{EX_in} - T_{EX_out}) \eta_{emG} \quad (4)$$

In order to estimate the loss of electric energy produced in the thermal engine and the electric energy additionally produced by the heat engine as a result of thermal integration with the CAES system, the following relationships can be used:

$$\Delta E_{el,HE(II)} = Q_{II} \cdot \beta_{II} \quad (5)$$

$$\Delta E_{el,HE(I)} = Q_{I'} \cdot \eta_{I'} + Q_{I''} \cdot \eta_{I''} \quad (6)$$

where : Q_{II} – the amount of heat taken from the heat engine to heat the air directed to the expander, $Q_{I'}$ – the amount of heat taken in the intercooler of the compressor which is used in the thermal engine, $Q_{I''}$ – the amount of heat taken in the aftercooler of the compressor which is used in the thermal engine, β_{II} - electric energy loss ratio due to the use of heat from the thermal engine, $\eta_{I'}$ - efficiency of electric energy generation in the thermal engine based on the cooling heat taken from the intercooler of the compressor, $\eta_{I''}$ – efficiency of electric energy generation in the thermal engine based on the cooling heat taken from the aftercooler of the compressor.

Assuming no loss of heat in the flow path between two heat carriers of two systems, the amount of heat supplied and taken from the thermal engine cycle are determined by using the energy balances of the relevant heat exchangers from the HCAES system:

$$Q_{II} = (1 - \delta) m_1 c_p (T_4 - T_5) \quad (7)$$

$$Q_I = Q_{I'} + Q_{I''} = m_1 c_p (T_2 - T_3 + T_{1''} - T_{1'}) \quad (8)$$

Correspondingly, the electric energy loss ratio β_{II} (eq. 5) and electric energy generation efficiency $\eta_{I'}$ and $\eta_{I''}$, (eq. 6) can be estimated using the definition of efficiency of the cycle of the Carnot engine η_c :

$$\beta_{II} = \varphi \eta_{c(II)} \quad (9)$$

$$\eta_{I'} = \varphi \eta_{c(I')} \quad (10)$$

$$\eta_{I''} = \varphi \eta_{c(I'')} \quad (11)$$

where: φ – ratio of thermodynamic imperfection in relation to the Carnot engine, while the efficiency of the Carnot engine was determined by the following relations:

$$\eta_{c(II)} = 1 - \frac{2T_b}{T_4 + T_5} \quad (12)$$

$$\eta_{c(I')} = 1 - \frac{2T_b}{T_{1'} + T_{1''}} \quad (13)$$

$$\eta_{c(I'')} = 1 - \frac{2T_b}{T_2 + T_3} \quad (14)$$

Using the isentropic equation and the isentropic efficiency definition of the compressor, the temperature of the air leaving each compressor section can be determined:

- for case A,

$$T_2 = T_1 \left(1 + \frac{\beta_{C1}^{\frac{\kappa-1}{\kappa}} - 1}{\eta_{iC}} \right) \quad (15)$$

- for cases B and C:

$$T_{1'} = T_1 \left(1 + \frac{\beta_{C1}^{\frac{\kappa-1}{\kappa}} - 1}{\eta_{iC}} \right) \quad (16)$$

$$T_2 = T_{1''} \left(1 + \frac{\beta_{C2}^{\frac{\kappa-1}{\kappa}} - 1}{\eta_{iC}} \right) \quad (17)$$

Similarly, the temperature of the air leaving the expander, significant from the point of view of the conditions of increased erosion of the blades in the last stages of the expander (condensation, and even moisture crystallization in the air volume at low temperature), will be obtained using the isentropic equation and the isentropic expander efficiency definition:

$$T_6 = T_5 \left[1 - \eta_{EX} \left(1 - \beta_{EX}^{\frac{1-\kappa}{\kappa}} \right) \right] \quad (18)$$

In cases B and C, the air is cooled down before storage. The degree of cooling in the intercooler and the cooler installed in front of the pressure tank can be determined directly by the temperature to which the air is cooled ($T_{1''}$ and T_3) or by providing the value of factor γ and γ' :

$$\gamma = \frac{T_2 - T_3}{T_2 - T_{ot}} \quad (19)$$

$$\gamma' = \frac{T_{1'} - T_{1''}}{T_{1'} - T_{ot}} \quad (20)$$

For case A γ assumes a value of 0, because $T_2 = T_3$.

At the stage of air storage there may be a loss of mass of the stored air due to leaks and additionally heat loss through conduction through the insulation material. Weight loss is determined by the factor δ :

$$\delta = \frac{m_1 - m_4}{m_1} \quad (21)$$

while the degree of cooling of the air on the path of heat loss in the storage tank can be determined using the efficiency of heat storage:

$$\eta_{mag} = \frac{T_4 - T_{ot}}{T_3 - T_{ot}} \quad (22)$$

The amount of heat transferred to the air directed to the expander at the stage of discharging the energy storage system, in relation to the amount of heat received from the air before its storage, is determined by the α indicator:

$$\alpha = \frac{(1 - \delta)(T_5 - T_4)}{T_2 - T_3} \quad (23)$$

The α index is determined only for cases B and C; for case B, representing the adiabatic system with heat storage, the α indicator is the efficiency of heat storage in the vessel and always takes values below 1, while for case C, the value of α indicator depends on the availability and disposal of heat in the thermal cycle of the engine and can take values higher than 1. For calculation purposes in case C, the temperature of the air heated by utilizing heat taken from the integrated thermal engine can be assumed.

Using the calculation based on equations (15) - (23), an analysis of realities of plant operation parameters can be carried out, which may be crucial, e.g. due to technological barriers (thermal stability of materials used, air storage pressure). Using additional relations (1) – (14), the energy storage efficiency characteristics can be determined for the analyzed cases.

One of the basic values taken into account when planning CAES systems is the pressure to which compressed air is compressed. The higher the pressure, the greater the energy storage capacity of a given volume. The pressure level is directly determined by the pressure ratio which must be

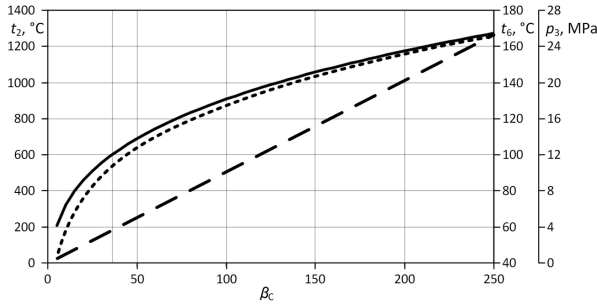


Figure 3: Temperature at the outlet of compressor, temperature at the outlet of expander, pressure at the outlet of compressor as a function of compressor pressure ratio for case A

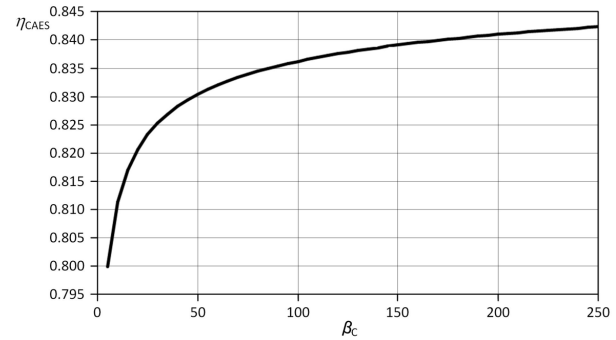


Figure 4: Efficiency of energy storage of CAES as a function of compressor pressure ratio for case A

implemented by the air compressor. For the implementation of high pressure ratios, there may be a need to use a larger number of compression sections within the machine, with planned intersectional cooling. Three cases of energy storage installations were used to analyze the impact of pressures in individual compression sections. In case A, which is the case of a hypothetical installation, we are dealing with a single-section compressor. Here, the analyses were carried out for pressure ratios in the range from 0 to 250. Cases B and C involve two-section compressors; in case B the temperature after the first section should be minimized, accompanying minimization of heat losses to the environment. Hence, a pressure ratio should also be implemented in this section of the compressor. During analysis of case B the pressure ratio for the first section was changed in the range from 1 to 15, while for the second section from 1 to 50. The analyses carried out for case C were done for the same ranges of pressure ratios for two sections of the compressor, i.e. 1 to 30.

During the analyses, the characteristics of the size that could determine the technical capabilities of the installation were determined. Among them were:

1. air temperature after the compressor or its individual sections (t_2 and t_1),
2. pressure of the air entering the pressure vessel (p_3),
3. temperature leaving the expander (t_6).

Fig. 3 presents the characteristics of the analyzed values as a function of the compressor compression ratio obtained for case A. High air temperature at the compressor outlet is a technical barrier to this solution, as higher values of pressure are needed to ensure a high value of stored air pressure (not lower than 5 MPa). Assuming a critical temperature of 600°C, the pressure ratio of the compressor should not be higher than 36. Regardless of the assumed values of the pressure ratio, the technical barrier for case A should not affect the temperature of the air leaving the expander.

Fig. 4. Efficiency of energy storage of CAES as a function of compressor pressure ratio for case A Fig. 5 presents the results of the analysis for case B. The results are presented in the form of an isoline of the values of the tested temperatures

and pressure as a function of two pressure ratios that are appropriate for the two sections of the air compressor. Fig. 6 presents in an analogous manner the value isolines for energy storage efficiency for case B, calculated in accordance with relation eq. (1). The figure on the yellow background is limited to the area of solutions that meet the following conditions:

1. air temperature leaving the second section of the compressor $t_2 \leq 600$ C,
2. air pressure leaving the second section of the compressor $4 \text{ MPa} \leq p_3 \leq 12 \text{ MPa}$,
3. air temperature leaving the expander $t_6 \geq 0^\circ\text{C}$.

Fig. (7) and Fig. (8) show the results of the analysis covered in case C. The results were presented in the same way in relation to the results of case B. In case C, due to the wider range of changed values of pressure ratio implemented in the first section of the compressor air, a critical air level was exceeded in this section for air temperature, i.e. 600°C. The energy storage efficiency values possible in case C do not differ significantly from the values obtained in case B. In case B, the values for this thermodynamic efficiency index are included in the range from 0.725 to 0.778, then for case C in the range from 0.707 to 0.751

4. Discussion

This section presents the results of analyses of the case of integration of the CAES system with the thermal cycle of the coal-fired supercritical power unit. The structure of two systems and the manner of their integration are presented in Fig. 9. At the stage of loading the energy storage system, compressed air cooling energy, obtained in two intersectional coolers and a condenser installed at the compressor outlet, is used to heat condensate, which is a working medium of the coal-fired power unit. This way of using the heat makes it possible to partially replace the regeneration, thereby reducing steam bleeds from the medium and low-pressure turbine releases and increasing the power generated by the turbine set. At the stage of discharging the energy storage system, the air after leaving the tank is directed to the exchanger fed with steam taken from one of three bleeds, i.e. steam from

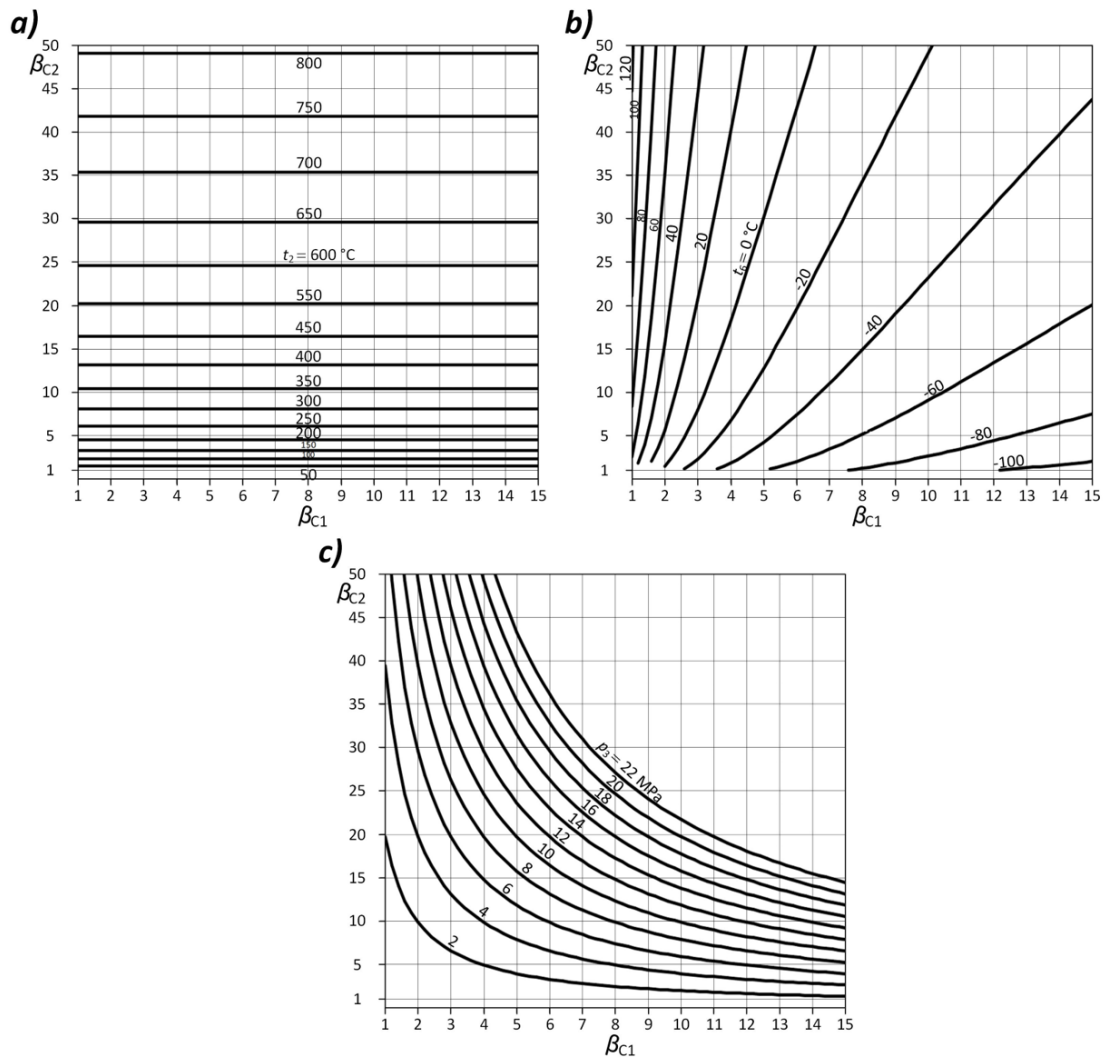


Figure 5: Temperature at the outlet of compressor (a), temperature at the outlet of expander (b), pressure at the outlet of compressor (c) as a function of compressor pressure ratio for case B

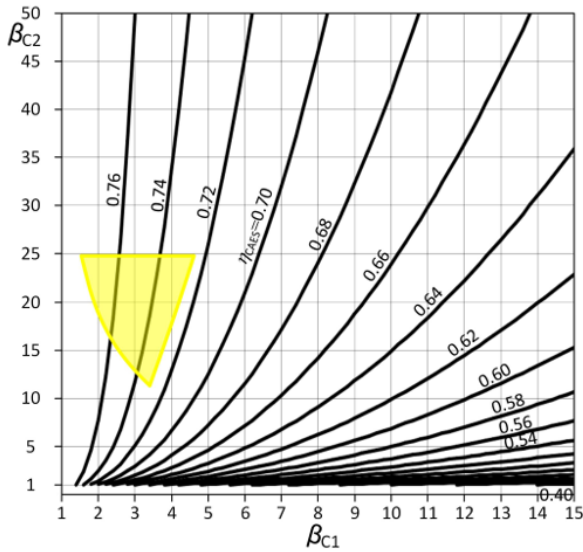


Figure 6: Efficiency of energy storage of CAES as a function of compressor pressure ratio for case B

the first turbine part?????? (Case I) or from the first turbine part???????? (Case II) or from the second part exhaust of the high pressure part (Case III). In each case, it was assumed that superheated steam directed to an additional built-in heat exchanger gives heat to the heated air to obtain the assumed enthalpy drop, determined by the following relationships:

$$\Delta \bar{h}_{bl1} = \frac{h_{11} - h_{21}}{h_{11} - h''(p_{21})} \quad (24)$$

$$\Delta \bar{h}_{bl2} = \frac{h_{12} - h_{22}}{h_{12} - h''(p_{22})} \quad (25)$$

$$\Delta \bar{h}_{bl3} = \frac{h_{13} - h_{23}}{h_{13} - h''(p_{23})} \quad (26)$$

In each of the analyzed cases, the relative enthalpy drop performed by individual heat exchangers was 0.95. The nominal assumptions for the power plant and energy storage system are presented in table (1).

Due to the assumed same times of loading and unloading the energy storage system, the efficiency of the storage system, originally defined by the dependence (2) can be presented using electric power:

$$\eta_{HCAES} = \frac{N_{el,EX} - \Delta N_{el,eq(II)}}{N_{el,C} - \Delta N_{el,eq(I)}} \quad (27)$$

where: $\Delta N_{el,eq(II)}$ - the equivalent loss of electric power of a steam turbine generator set as a result of taking more steam from turbine bleeds or increased coal consumption, $\Delta N_{el,eq(I)}$ - the equivalent gain of electric power of a steam turbine generator set as a result of taking less steam from turbine bleeds or decreased coal consumption.

Equivalent gain and power losses were determined using the dependence:

$$\Delta N_{el,eq(II)} = \dot{Q}_{II} \cdot \beta_{II} \quad (28)$$

$$\Delta N_{el,eq(I)} = \dot{Q}_I \cdot \eta_I \quad (29)$$

Heat flux \dot{Q}_{II} and \dot{Q}_I were determined by using the prepared model of the energy storage system. Parameters β_{II} and η_I were determined by using the prepared model of the coal fired supercritical power unit.

Partial results of analysis are shown in Fig. (10)–(13). Fig. (14) presents values of energy storage efficiency for the three cases analyzed.

5. Summary

The presented energy storage system concept provides an alternative to a wide range of compressed air energy storage systems. Through thermal integration with the power unit, high values of the basic energy efficiency index can be obtained, which for the studied systems is the efficiency of energy storage. Importantly, the system does not require any additional fuel supply – typically natural gas – which would significantly increase operating costs. Integration with the power unit means the way is open to harnessing the heat of compressed air cooling, which in the case of existing CAES systems requires storage for efficient use and hence high-temperature heat store equipment, which significantly increases investment costs. The analyses reported in this paper show that the scale of the CAES installation that can cooperate with a large 900 MW unit is moderate. Where single steam bleeds are used, the installed power of the compressors and expanders does not usually exceed 35 MW. 100 MW of power can be obtained, but this requires simultaneous use of steam released to a greater number of high-pressure regeneration exchangers. Locating a storage installation for compressed air in close proximity to the power unit is a central issue when planning successful integration. The team working on this technology plans to carry out analyses of abandoned hard coal mines for this purpose [11].

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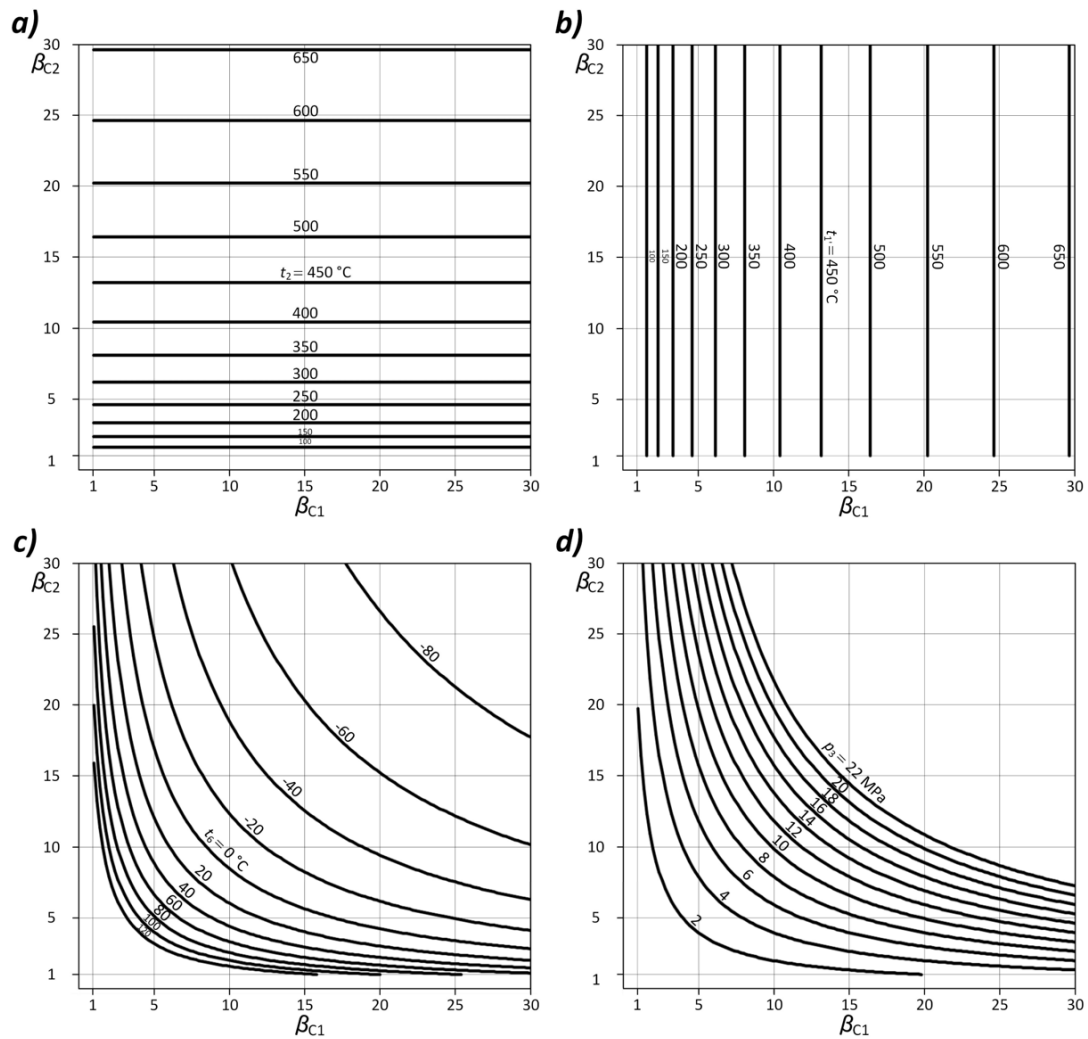


Figure 7: Temperature at the outlet of compressor (a), temperature at the outlet of first stage of compressor (b), temperature at the outlet of expander (c), pressure at the outlet of compressor (d) as a function of compressor pressure ratio for variant C

Table 1: Assumptions for analysis

Quantity	Value	Unit
Supercritical coal-fired power unit subsystem		
Net power output	900	MW
Live steam parameters	680/29.7	°C/MPa
Reheated steam parameters	693/5.9	°C/MPa
Pressure in condenser	0.005	MPa
Feed water temperature	305	°C
Gross efficiency	49.44	%
Net efficiency	45.64	%
Compressed Air Energy Storage Subsystem		
Charging time	8	h
Discharging time	8	h
Internal efficiency of air compressor stages	88	%
Internal efficiency of air expander stages	90	%
Electricity generator efficiency	98.5	%
Air compressor pressure ratio	70	-
Relative pressure drops in the flow of factors through heat exchangers	2	%
Relative heat losses in the heat exchangers	1	%

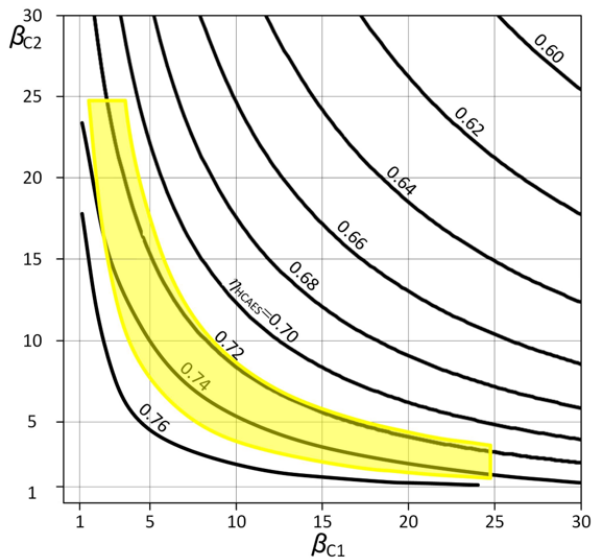


Figure 8: Efficiency of energy storage of CAES as a function of compressor pressure ratio for case C

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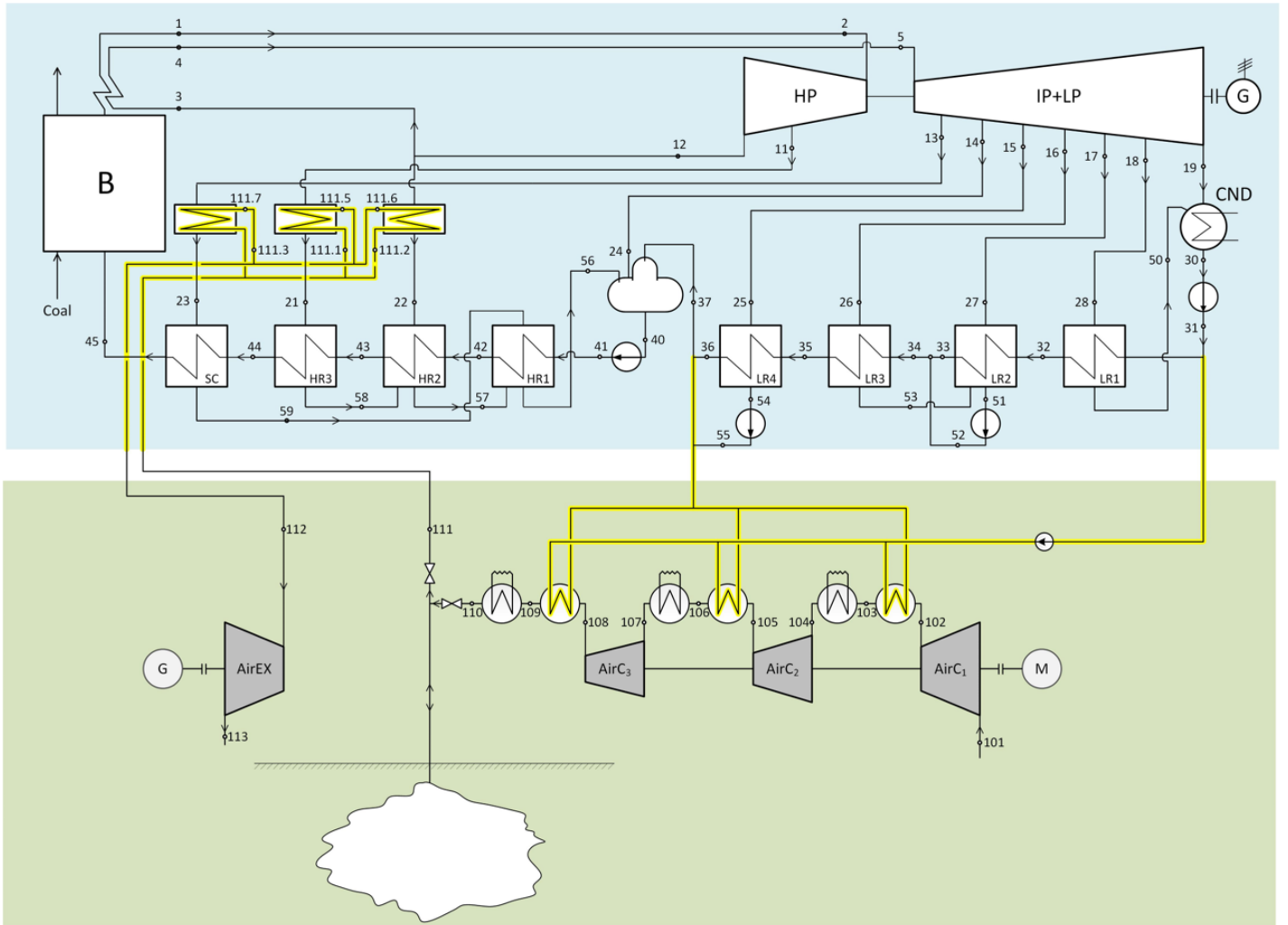


Figure 9: Scheme of supercritical coal-fired unit integrated with CAES subsystem

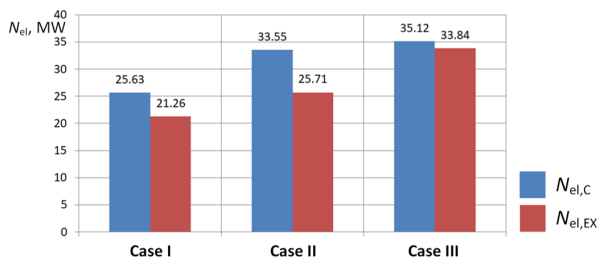


Figure 10: Compressor power and expander power for three cases of Hybrid CAES

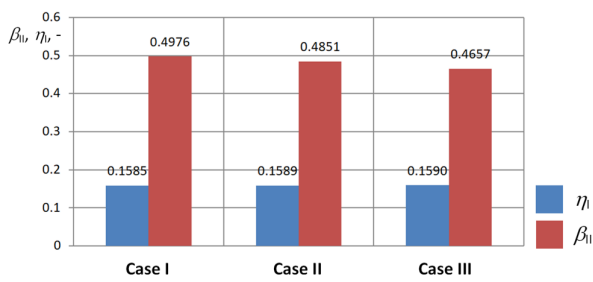


Figure 11: Electric energy loss ratio due to the use of heat from thermal engine and electric energy generation in a power unit based on the cooling heat taken from the intercooler of the compressor for three cases of Hybrid CAES

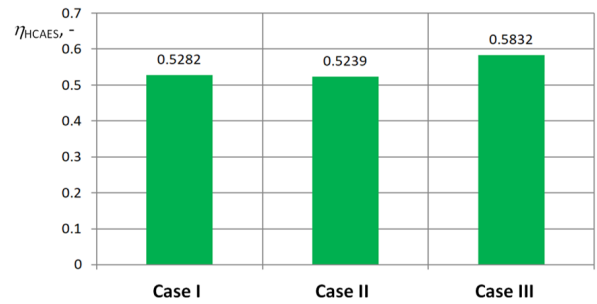


Figure 14: Efficiency of energy storage for three cases of Hybrid CAES

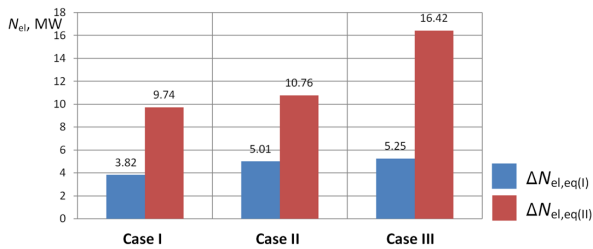


Figure 12: Equivalent loss and gain of electric power of a steam turbine generator for three cases of Hybrid CAES

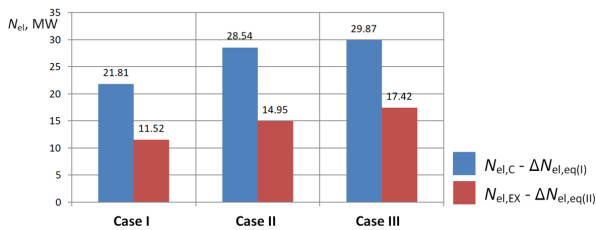


Figure 13: Differences of the analyzed electric powers for three cases of Hybrid CAES