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Exergy analysis of a one-stage adiabatic compressed air energy storage system

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Abstract

To improve the use of renewable energy sources and increase their penetration in the renewable energy market, different types of energy storage technologies have been introduced among which compressed air energy storage (CAES) systems offer more competitive feature. In this paper, a comprehensive exergy analysis of a one-stage, adiabatic compressed air energy storage (A-CAES) system is carried out for a wide range of parameters. The governing parameters on the system performance include, but not limited to, storage pressure, pre-set vessel pressure as well as isentropic efficiencies of the major components involved in the A-CAES systems most specifically compressor and expander. By performing the exergy analysis and using the concept of exergy destruction, which is a criterion for power loss due to the irreversibility, the magnitude and source of thermodynamic inefficiencies at each part of the system is identified. The obtained results reveal that the isentropic efficiency of the machine has a significant effect on the amount of exergy destructed by the machine. In addition, the results show that generally more exergy is lost in the expander than other components of the A-CAES system during the system operation. Consequently, to improve the system performance, utilising a high-efficient expander should be a priority. It is also shown that increase in the storage pressure improves the exergy efficiency of the compressor and expander. Using the obtained results, it would be possible to find the optimum working condition for the system leading to improvement of the system performance.

Keywords: Compressed Air Energy Storage System, Exergy Analysis, performance, Second Law of Thermodynamics

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1. Introduction

Due to energy crisis, the utilization of renewable energies has attracted an increasing attention in many countries [1]. However, the main drawback of these renewable energy sources, such as wind energy, solar energy, bio-energy etc., is their intermittence, randomness and volatility nature making their development a huge challenge. As a result, energy storage technologies have been taken into account as one of the most promising methods to employ the volatile renewable energy sources [2]. Amongst different types of energy storage approaches, the compressed air energy storage (CAES) system is of great interest by many researchers due to its competitive features such as large energy storage capacity, prompt response time and long cycle lifespan [3]. In a typical CAES system, excess, cheap energy of a renewable energy resource is employed to compress the air in a storage cavern. This pressurised air is released from the cavern through an expander to generate the required power when it is on demand [4, 5]. There are currently two large-scale CAES systems in operation: the Huntorf CAES plant was built in Germany in 1978 and McIntosh CAES plant was built in Alabama, USA, in 1991. In both plants, natural gas is employed as the main heat source to pre-heat the gas before the power generation process in the expander. The cycle efficiency of the Huntorf plant and McIntosh plant have been reported to be around 42% and 53%, respectively [6-8]. Technically, these plants suffer from low efficiency, loss of heat of compression and environmental pollution due to the use of fossil fuels. To remedy, Adiabatic Compressed Air Energy Storage (A-CAES) systems have been proposed. In these systems, a thermal reservoir called Thermal Energy Storage (TES) is added to the system to absorb and store the heat of compression during charging processes. This stored thermal energy is used to pre-heat the air before its expansion process. Therefore, fossil fuels are no longer required [7, 9-11]. Over the years, many researchers have attempted to reduce the energy loss and improve the performance of the A-CAES plant using different technologies [12, 13]. Mozayeni et al. [14] performed a thermodynamic analysis to investigate the performance of a one-stage Advanced Adiabatic CAES system (AA-CAES). It was found that utilising high-efficient compressor and expander can significantly improve the performance of an AA-CAES system. As large scale CAES systems are dependent on the geophysical conditions, Kim and Favrat carried out an energy and exergy study on a single-stage Micro-CAES with man-made air vessel [15]. The results showed that best exergy efficiencies are achieved in quasi-isothermal compression and expansion processes. Yao et al. [16, 17] studied a novel combined cooling, heating and power system (CCHP) in which an A-CAES system works together with heat exchangers, gas engine, and an absorption refrigeration system. The authors reported the efficiency of around 50% for their proposed systems.

In this paper, based on the Second Law of Thermodynamics, a comprehensive exergy analysis is performed to examine the performance of different components of a single-stage A-CAES system under different operating conditions. By performing the exegry analysis of the A-CAES system concurrently with using the concept of exergy destruction, which is a measure of performance lost due to irreversibility, the magnitude and source of thermodynamic inefficiencies at major components of the system including compressor and expander are identified. The obtained results of this analysis will provide a deep understanding of operation characteristics of compressor and expander in the system. This understanding will be quite valuable in designing the most efficient A-CAES system in a specific area according to the geological, financial and market constraints of that area.

2. Description of the System Configuration

A typical A-CAES plant consists of four major components: compressor, thermal energy storage (TES), storage vessel and expender. In these systems, the air is compressed by a compressor using off-peak, cheap electricity or the excess energy of renewable energy resources, such as a wind turbine or solar energy. Before transferring to the storage vessel, the heat of compression is absorbed and stored by TES. The air compression process is continued up to the point when the air pressure in the vessel reaches to its pre-set storage pressure. When power to the grid is required, the air is released from the storage vessel and pre-heated by the stored thermal energy before the expansion process in the expander. A schematic figure of an A-CAES system is shown in Fig. 1. When performing the exergy analysis, the kinetic and potential energies along with pressure losses are negligible and the charging and discharging processes are assumed to be steady state. Moreover, TES efficiency in this study is 90% and its upper limit temperature is taken at 600 °C.

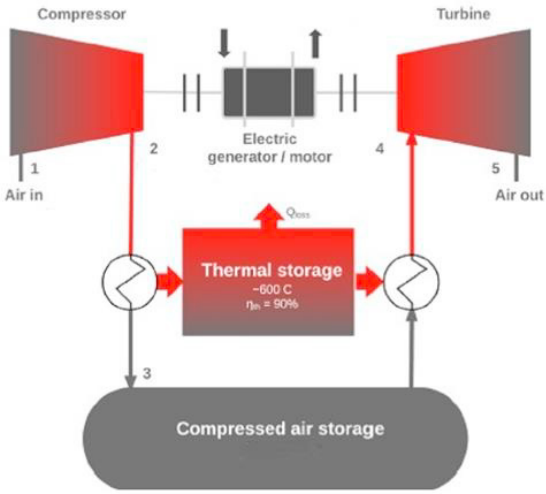


Fig.1. A schematic of an A-CAES system

3. Exergy Analysis

3.1. Compressor

For the compression process in the compressor, the exergy balance is written as follows,

$$\dot{I}_c = \dot{E}_c^+ - \dot{E}_c^- \tag{1}$$

in which, $\dot{I}_c$ is the rate of exergy destruction, $\dot{E}_c^+$ is the rate of exergy transfer to the A-CAES system by compression work and $\dot{E}_c^-$ is the rate of exergy transfer from the compressor to air flow. The two parameters $\dot{E}_c^+, \dot{E}_c^-$ are defined according to the bellow relations,

$$\dot{E}_c^+ = \dot{W}_c \tag{2}$$

$$\dot{E}_c^- = \dot{m}_a\left[\left(h_2 - h_1\right) - T_1\left(s_2 - s_1\right)\right] \tag{3}$$

where,

$$h_2 - h_1 = C_P\left(T_2 - T_1\right) \tag{4}$$

$$s_2 - s_1 = C_P \ln\frac{T_2}{T_1} - R\ln\frac{P_2}{P_1} \tag{5}$$

In these equations, subscript 2 refers to the after-compressor state and subscript 1 refers to the atmospheric state, as shown in figure 1. Also, parameters $\dot{W}_c, \dot{m}_a, h, s, T$ and P are the rate of compression work, air mass flow rate, enthalpy, entropy, temperature and pressure, respectively. Moreover, R is the specific gas constant and C_P is the air specific heat of constant pressure. In these equations, T_2 is the after-compressor air temperature, is found using the following relation:

$$T_2 = T_1 + \frac{T_{2s} - T_1}{\eta_c}, \qquad T_{2s} = T_1\left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} \tag{6}$$

In equation (6), T_{2s} is the isentropic air temperature after the compressor. The exergy transfer rate to the air flow defined in relations (3) to (5) can be split in two parts named mechanical exergy $\dot{E}_{c(M)}$ and thermal exergy $\dot{E}_{c(T)}$, as follow,

$$\dot{E}_c^- = \dot{E}_{c(M)} + \dot{E}_{c(T)} \tag{7}$$

where,

$$\dot{E}_{c(M)} = \dot{m}_a R T_1 \ln\frac{P_2}{P_1} \tag{8}$$

$$\dot{E}_{c(T)} = \dot{m}_a C_p \left(T_2 - T_1 - T_1 \ln\frac{T_2}{T_1} \right) \tag{9}$$

As the compressed air flow passes through the TES, its thermal exergy is absorbed by the thermal storage. This stored exergy will be later used for preheating the compressed air before expansion process. The remaining part of the flow exergy, which is the mechanical exergy, is then transferred to the storage vessel.
Also, the exergy efficiency of the compressor is defined by,

$$\eta_{ex} = \frac{E_c^-}{E_c^+} \tag{10}$$

where, E_c^+ is the total exergy transfer to the system by compression work and E_c^- is the total exergy transfer from the compressor to air flow. These parameters are expressed as,

$$E_c^+ = W_c = \frac{W_{c,s}}{\eta_c} \tag{11}$$

$$E_c^- = M_a \left[R T_1 \ln\frac{P_2}{P_1} + C_p \left(T_2 - T_1 - T_1 \ln\frac{T_2}{T_1} \right) \right] \tag{12}$$

In these relations, W_c is the total compressor work and M_a is the total air mass transferred to the vessel. Besides, η_c is the compressor isentropic efficiency and $W_{c,s}$ is the total isentropic work consumed by an ideal compressor. This ideal work is found using the following equation:

$$W_{c,s} = M_a \left(\frac{k}{k-1} R T_1 \left[\left(\frac{P_2}{P_1} \right)^{\frac{k-1}{k}} - 1 \right] \right) \tag{13}$$

It should be mentioned that for an isentropic compressor, entropy remains constant during the compression process and, therefore, the exergy efficiency will be equal to 1.

3.2. Thermal Energy Storage

After absorbing the heat of compression by the Thermal Energy Storage (TES), the compressed air temperature is degraded to the ambient temperature T_1. The rate of exergy transfer from air to the TES due to transfer of heat of compression is expressed as,

$$\dot{E}_q^- = \dot{m}_a C_p \left(T_2 - T_1 - T_1 \ln\frac{T_2}{T_1} \right) \tag{14}$$

Moreover, the rate of exergy transfer from the TES to air before the expansion process in the expander is presented by,

$$\dot{E}_q^+ = \dot{m}_a C_p \left(T_4 - T_1 - T_1 \ln\frac{T_4}{T_1} \right) \tag{15}$$

In the above relation, T_4 is the compressed air temperature before expansion process which is obtained using the following relation,

$$T_4 = T_1 + \eta_{th} \left(T_2 - T_1 \right) \tag{16}$$

In this relation, η_{th} is the thermal efficiency of the TES and is defined as the ratio of heat energy transferred from the TES to air flow before expansion to the heat energy absorbed from the air flow by TES after compression. In this study, η_{th} is considered to be 0.9.

3.3. Storage Vessel

The rate of exergy transfer to the storage tank by the pressurized air is given by,

$$\dot{E}_v^+ = \dot{m}_a R T_1 \ln \frac{P_2}{P_1} \tag{17}$$

P_2 is the after-compressor pressure which is equal to the maximum pressure of the vessel or storage pressure $(P_{\max})$. By integrating the above relation, the total exergy transfer to the storage vessel during air compression is given by,

$$E_v^+ = \int \dot{E}_v^+ dt = M_a R T_0 \ln \frac{P_2}{P_0} \tag{18}$$

Furthermore, during the power generation process, the rate of exergy leaving the storage vessel can be obtained by using,

$$\dot{E}_v^- = \dot{m}_a R T_0 \ln \frac{P}{P_0} \tag{19}$$

In this relation, P is the vessel pressure gradually decreasing during the discharging process from $P_{\max}$ to P_{set}. P_{set} is the initial vessel pressure at the beginning of air charging process. The total exergy leaving the vessel when generating power is also given by,

$$E_v^- = \int \dot{E}_v^- dt = \int \dot{m}_a R T_1 \ln \frac{p}{P_1} dt = V P_{\max} \left[\ln \frac{P_{\max}}{P_1} - \frac{P_{set}}{P_{\max}} \ln \frac{P_{set}}{P_1} + \frac{P_{set}}{P_{\max}} - 1 \right] \tag{20}$$

3.4. Expander

In the discharging process, the exergy balance for the expander is expressed as follows,

$$\dot{I}_e = \dot{E}_e^+ - \dot{E}_e^- \tag{21}$$

In which, $\dot{I}_e$ is the rate of exergy destruction in the expander, $\dot{E}_e^+$ is the rate of exergy transfer from heated compressed air flow to the expander and $\dot{E}_e^-$ is the resulting exergy transfer from the expander to the power generator. These two parameters are defined according to the following relations by assuming that the expander outflow pressure gets the atmospheric pressure,

$$\dot{E}_e^- = \dot{W}_e \tag{22}$$

$$\dot{E}_e^+ = \dot{m}_a \left[(h_4 - h_5) - T_1 (s_4 - s_5) \right] \tag{23}$$

where,

$$h_4 - h_5 = C_P (T_4 - T_5) \tag{24}$$

$$s_4 - s_5 = C_P \ln \frac{T_4}{T_5} - R \ln \frac{P_4}{P_1} \tag{25}$$

In which, subscripts 4 and 5 refer to the states before and after expander, respectively, as shown in figure 1. It should be noted that P_4 is the air pressure before expansion process being equal to the vessel pressure.

The exergy efficiency of the expander is also given by,

$$\eta_{ex} = \frac{E_e^-}{E_e^+} \tag{26}$$

where, E_e^+ is the total exergy transfer from the air flow to the expander and E_e^- is the total exergy transfer from the expander to the power generator, as defined bellow,

$$E_e^+ = \int \dot{E}_e^+ dt \tag{27}$$

$$E_e^- = W_e = W_{e,s} \times \eta_e \tag{28}$$

where, η_e is the expander isentropic efficiency and $W_e, W_{e,s}$ are total real power and ideal isentropic power, respectively, generated by the expander. The total isentropic power is calculated using the following equation:

$$W_{e,s} = \int \left(\frac{k}{k-1} RT_4 \left[1 - \left(\frac{P_1}{P} \right)^{\frac{k-1}{k}} \right] \right) dm \tag{29}$$

4. Presentation of Results

In this section, the obtained results from exergy analysis of the A-CAES system are presented. In figure 2, variations of the exergy transferred to the A-CAES by compression work and the exergy transferred to the power generator by expansion work are depicted versus the storage pressure ranged from 1 MPa to 10 MPa. According to this figure, when increasing the storage pressure, both input and output exergy terms increase. This increase is sharper at higher values of the storage pressure. Also, the effect of isentropic efficiency of the compressor and expander on exergy distributions during compression and expansion processes is well noticed. For the air compression in which the air pressure is increased from the atmospheric pressure to the storage pressure, utilizing a low-efficient compressor causes the input exergy to the system to be raised. For instance at the storage pressure of 10 Ma, if the compressor efficiency is improved from 0.65 to 0.95, the input required exergy to the system is reduced from 137.35 MJ/m^3 to 93.976 MJ/m^3 being equivalent to a reduction of 31.5 %. Similarly during the air expansion, if an expander operating with a low efficiency is employed, a lower portion of the air flow exergy is transferred to the generator for the purpose of power generation.

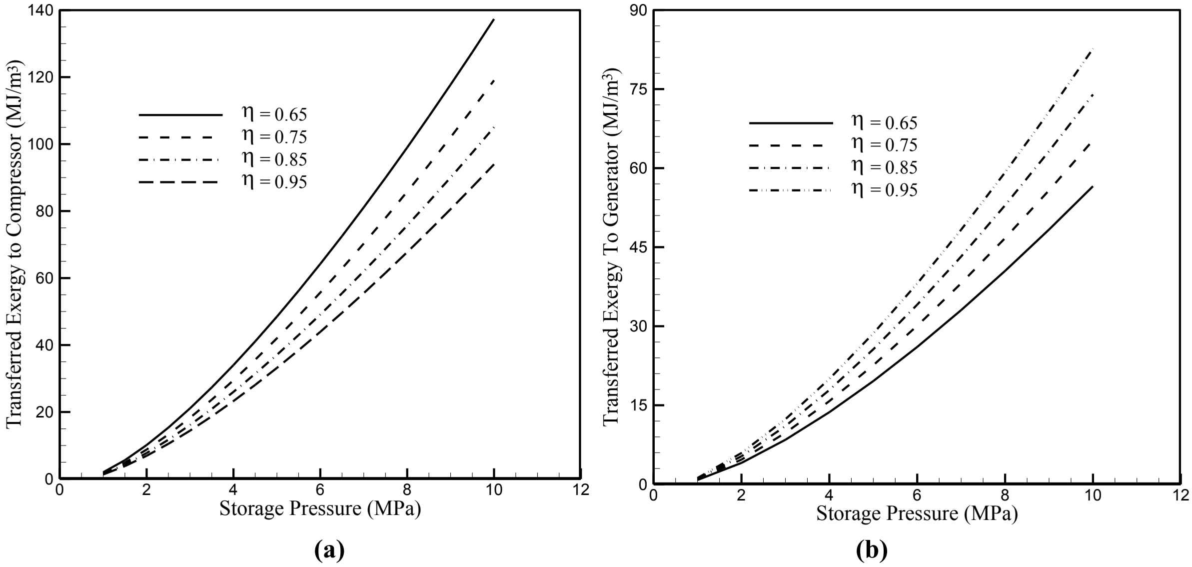
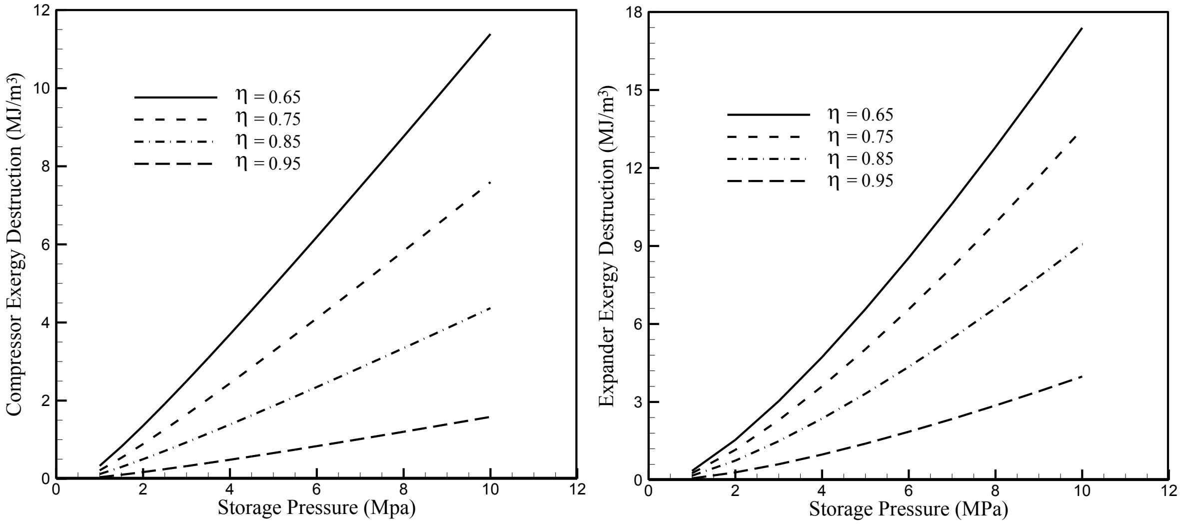


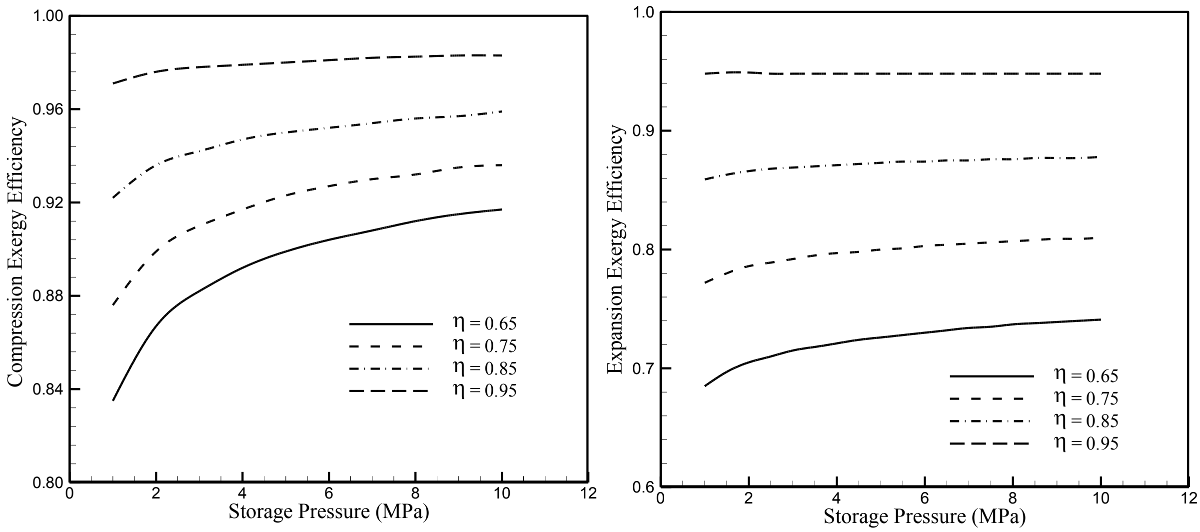
Fig. 2. Effect of the storage pressure on (a) transferred exergy to the compressor (b) transferred exergy to generator at different compressor and expander efficiencies

Due to the irreversibility existing in steady-flow devices, entropy is generated and exergy is destroyed. Clearly, the less exergy destruction in a machine, the more efficiency of that machine will be. Distributions of the exergy destruction in a compressor and expander operating at different isentropic efficiencies are presented in figure 3 in the selected range of storage pressure. As it is revealed in this figure, more exergy is destroyed in the expander than compressor at the same operating conditions. For instance, at the storage pressure of 10 MPa, exergy destruction in the expander working with the isentropic efficiency of 0.85 is 9.069 MJ/m^3 which is 2.076 times larger than that in the compressor operating at the same efficiency. In fact, during the discharging process, the vessel pressure gradually crosses the line from the maximum pressure to the pre-set pressure. Therefore, a higher portion of the air flow exergy is destroyed in the expander due to the variable input pressure of the air, compared to the compressor functioning in a fixed pressure ratio.



(a) **(b)**

Fig. 3. Distributions of exergy destruction versus storage pressure in (a) compressor (b) expander



(a) **(b)**

Fig. 4. Exergy efficiency of (a) compressor (b) expander versus storage pressure

Figure 4 examines distributions of the exergy efficiency of the compressor and expander for wide ranges of storage pressure and selected isentropic efficiencies. As shown in this figure, the increase of the storage pressure generally causes the exergy efficiency of both compressor and expander to improve. This improvement is more noticeable for a compressor or expander operating at a low isentropic efficiency, i.e. $\eta = 0.65$. For instance, by taking into account a compressor working with the efficiency of 0.65, if the storage pressure increases from 1 MPa to 10 MPa, the compression exergy efficiency is improved from 0.835 to 0.917. This improvement for an expander working with the same efficiency is from 0.685 to 0.741. This figure illustrates that generally exergy efficiency of a compressor is higher than that of an expander as more exergy is destructed in the expander when generating the power.

5. Conclusion

In this study, an exergy analysis was performed to study the performance of major components of a one-stage Adiabatic Compressed Air Energy Storage (A-CAES) system in different operating conditions. In this study, the parameter pre-set pressure is defined as the air initial pressure in the storage vessel. Using this definition, the effect of storage pressure, which is the maximum pressure in the vessel at the end of compression, on the exergy performance of the compressor and expander is investigated at different isentropic efficiencies of these machines. The analysis demonstrated that the isentropic efficiency of the compressor or expander has a major effect on the amount of input exergy to the system during compression or output exergy from the system during expansion. In addition, it was shown that generally the expander is more exergy destructive than the compressor. For instance at the storage pressure of 10 MPa, when both compressor and expander work with the efficiency of 0.85, the exergy destruction in the expander is 2.076 larger than that of the compressor. The analysis also revealed that increase in the storage pressure brings about the improvement of the exergy efficiency of both compressor and expander. This improvement is more noticeable for a compressor functioning at a low isentropic efficiency.

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