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THERMODYNAMIC ANALYSIS OF DIABATIC AND ADIABATIC COMPRESSED AIR ENERGY STORAGE SYSTEMS

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ABSTRACT: *Compressed Air Energy Storage (CAES) is a technology for storing large quantities of electrical energy in the form of high-pressure air. CAES can play a major role in meeting the challenge of making renewable energy more reliable and in the successful integration of energy generated from renewable energy into the electric grid. A thermodynamic analysis of Diabatic and Advanced Adiabatic Compressed Air Energy Storage systems under the ambient temperature, compression and expansion ratios and stages number of compression and expansion trains is conducted in this paper. This paper aims to study the impact of these parameters on the specific work as well as the efficiency of the D-CAES and AA-CAES systems. In addition, a comparison between the obtained results of D-CAES and AA-CAES systems is carried out.*
KEYWORDS- *D-CAES, AA-CAES, ambient temperature, compression and expansion ratios, round-trip efficiency.*

INTRODUCTION

Nowadays, climate changes, caused by burning fossil fuels, occur at an unprecedented rate which has a high impact on the natural resources. The production of energy will no longer be able to sustain the world without implying risks on altering the global system. The use of renewable technologies, which are considered as clean and inexhaustible resources, plays an important role in minimizing the environmental impacts, reducing global carbon emissions and the world's reliance on fossil fuels, increasing energy security and achieving sustainable development. However, the biggest disadvantages of renewable energy is its intermittent nature, as these resources do not deliver a regular supply easily adjustable to consumption

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needs. It is affected by weather conditions, time and geographic location. The use of energy storage technologies and the demand management can provide a promising solution by filtering out the variability of renewable energy [1]. Energy storage plays an essential role in integration and deployment of renewable energy systems. Currently, the development and the commercialization of energy storage technologies highly increase since they will have a significant impact on integrating renewable energy and managing power system stability. There are many types of energy storage technologies such as electrical, mechanical, thermal and chemical. Pumped hydro storage is the most widely commercial and mature energy storage technology and constitutes 97 % of the total storage capacity worldwide [2]. The corresponding facilities store energy in water form in an upper reservoir then pumped it from another reservoir. Compressed air energy storage (CAES) is also a promising energy storage technology to store electrical energy and reuse it whenever demand is needed due to its large power and energy capacity, high cycle lifespan, and fast response time. In the case of CAES, air is compressed in a period of excess energy and expands to release the energy during the period of energy shortage.

CAES has been extensively studied and developed in different locations by using different thermodynamic processes. The first CAES plant was built in 1978, in Huntorf- Germany, where compressed air is stored in a salt dome with a power of 290 MW and has been operated for more than 20 years [3, 4]. Alabama Electric Cooperative is another plant that was built in Alabama in 1991 with a power of 110 MW. The compressed air is stored in a mined salt cavern at pressure up to 75 bar. In 2001, Highview power storage slough was built in Berkshire-UK using LAES [5]. CAES General compression Gaines has been built in Texas-USA in 2012 and stored air in a cavern with 2 MW and efficiency between 70% to 75% [6]. Table 1 shows the realized and planned projects of CAES system.

Table 1. Existing and projects CAES systems

	Date, company, and location	**Technical parameters**	**Technology description**
realized	1978 Huntorf plant	290MW/ 3h	Compressed air stored in a salt dome with 310 m^3 at up to 100 bar. Combustion of natural gas during the expansion process.
	1991 Alabama Electric Cooperative McIntosh, Alabama	110MW/ 26h 65M$	The compressed air is stored in a 538 m^3 in a mined salt cavern at pressure up to 75 bar. Combustion of natural gas during the expansion process.
	2012 General compression Gaines, Texas, USA	2MW/ 250h 70-75%	Compressed air is stored in a cavern.

Planned Projects	2013 SustainX Seabrook, New Hampshire, USA	2MW/ 95% efficiency 20 years lifetime 13M$	Sprayed water CAES for heat management. Use of standard steel pipes for compressed air storage.
	2013 Alliant Techsystems Inc Promontory, Utah, USA	80MW/ 30-60min 3.6M$	Above the ground compressed air energy storage.
	2013 RWE Power Stassfurt, Germany	360MW/90MW 70% efficiency $40M for 3.5 years	Compressed air is stored in subterranean caverns
	2013 Hydrostor Toronto, Ontario, Canada	1MW/ 4MWh >25 years lifetime 65-75% efficiency	Compressed air stored underwater bags.
	2016 Apex CAES Anderson, South California, USA	317MW 350-400M$	Compressed air is stored in the subterranean salt dome
	Pacific Gas and Electric Co. Kern, California, USA	300MW 355M$	Compressed air is stored in a subterranean porous-rock depleted gas field.
	New York Power Authority New York, USA	9-10mw/ 4h30	Compressed air in steel piping. Possible combustion of natural gas during expansion
	Hydro One Toronto, Canada	3-5MW/ 1-2h 8-10m$	Molten salt thermal energy storage system at 540 to 820C. Air stored above ground at a pressure up to 110bar.
	NPPD Nebraska, USA	135MW/10h	Compressed air stored in a depleted natural gas field with a volume of 850 million m^3 to a pressure up to 60bar.

This paper presents a thermodynamic analysis of Diabatic-CAES (D-CAES) and Advanced Adiabatic-CAES (AA-CAES) systems under some parameters. The studied parameters are the ambient temperature, the compression and expansion ratios and the stages number of compression and expansion. This paper aims to evaluate the variation of the work consumed by the compressor, the generated work by the expander and the performance of the systems under the studied parameters. In addition, a comparison between the obtained results of D-CAES and AA-CAES has been presented in this paper.

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Compressed air energy storage (CAES)

Principle

The fundamental idea of compressed air energy storage (CAES) technology developed back to the early 1940s to perform energy storage and it was conducted in the industry in 1960s [7]. CAES is a technology of storing energy as potential energy of compressed air. It uses compressed air as a medium to store energy during off-peak hours and generate energy during peak hours. CAES uses electricity during off-peak hours to drive the air compressor and to compress the air at a higher pressure and to store it in a cavern underground (rock cavern, salt cavern and dry mining salt) or aboveground. When electricity is needed, the pressurized air stored is heated and released to drive a turbine and generate electricity. Since conventional CAES systems need an additional fossil fuel to recover the stored electricity, CAES systems are not “pure” electricity storage, but hybrid systems. The main difference between a CAES power plant and a gas power plant is that in the CAES, compressor and turbine do not operate at the same time. CAES uses electricity from the grid to compress air and to store it. However, the main components of both systems are similar. In a conventional gas turbine, roughly two thirds of the power produced is needed to pressurize the air before combustion. CAES systems generate the same amount of electricity as a conventional gas turbine power plant using less than 40 % of the fuel [8]

The CAES process can be divided into three main stages:

- Compression: the main objective in this process is to compress the air that will be stored in the air storage. During this stage, there will be mechanical conversion due to high pressure and high temperature. Compression process can be done by using multi-stages which increases the efficiency before the air reach the air storage. The compressor efficiency depends on the pressure ratio between the inlet and outlet. When the pressure ratio varies during the compression stage, it will create some difficulties while optimal efficiency should be determined during the compression stage when the pressure ratio is constant [9].
- Storage: the aim of the process is to store large quantities of air at high pressure. The storage can be accomplished in three different systems, isobaric, isochoric, or cryogenic. The storage also can be built underground or aboveground depending on the needs and requirements.
- Expansion: after the compression and the storage of the air, the energy is extracted by passing the compressed air through the expander. The energy created during the compression process is used in expansion so no heat is removed or added to the system. Providing heat is ensured by CAES plants that need to burn fuel during the expansion. Natural gas is the most common fuel used but it contributes to global warming [10]. The expander should be connected to a generator in order to extract electrical energy, and this connection can be done in different ways as per the needs and requirements [11].

Based on thermodynamic principles and whether fuel is needed or not, CAES can be classified into three concepts

- Diabatic compressed air energy storage (D-CAES)

The D-CAES system is the oldest concept of CAES. A conceptual representation of D-CAES is shown in Figure 1. The heat resulting from air compression is wasted to the environment by using intercoolers to increase the pressure ratios of the compressor and decreasing the needed power consumption. In addition, the stored compressed air in the cavern should be at low temperature in order to reduce the required cavern volume for a given storage capacity and to reduce the thermal stress in the cavern wall. Thus, the heat is neither absorbed nor given off. This happened when the temperature of the two systems is the same (thermal equilibrium). The same amount of energy is transferred in one direction as in the other. The energy of the system is:

$$\Delta U = Q - W \tag{1}$$

where U is the internal energy, Q is the heat and W is the work.

Burning fossil fuel is required to reheat the compressed air during the discharge process to prevent ice buildup in the turbine during expansion. The Huntorfs and McIntosh plants utilize D-CAES in order to reduce the power consumption.

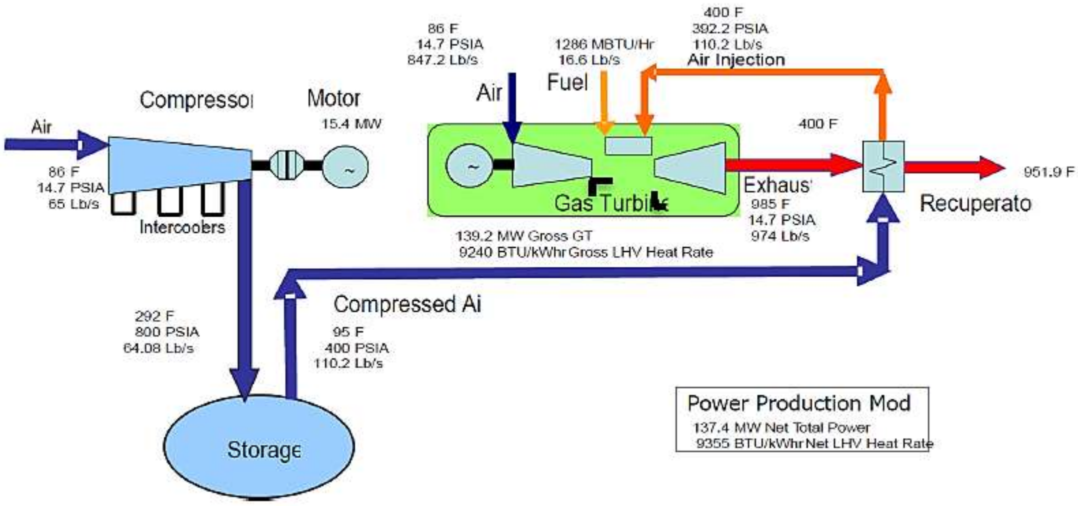

Figure 1. Conceptual representation of D-CAES system [12]

- Advanced Adiabatic compressed air storage

The Advanced-Adiabatic compressed air energy storage (AA-CAES) was proposed and gained attention in recent years in order to improve the cooling process of the compressed air and to increase the efficiency. AA-CAES systems retain the heat produced by the gas compression within a heat storage by using a thermal storage and return it to the gas when the gas is expanded through an air turbine to generate power (see Figure 2). The materials of thermal storage can be fluid such as molten salt solution or solid such as concrete and stone. Therefore, the difference between D-CAES and AA-CAES is the use of an additional fossil fuel to reheat the compressed air before expanding it through an air turbine. In AA-CAES systems, the compressed air is reheated by restoring the stored heat in the charging process. Storing the heat during the charging process and reuse it in the discharging process increase the overall efficiency of the system.

There is no heat lost or gained in AA-CAES, therefore all the change in the energy is interpreted by the performed work. According to the first law of thermodynamics, the internal energy of the system is:

$$Q = 0 \tag{2}$$

The internal energy decreases by an amount equal to the work done by the system when there is no heat.

$$PV^{\gamma} = constant = \mathrm{m} \tag{3}$$

Where $\gamma = C_p/C_v$ is the molar heat capacities.

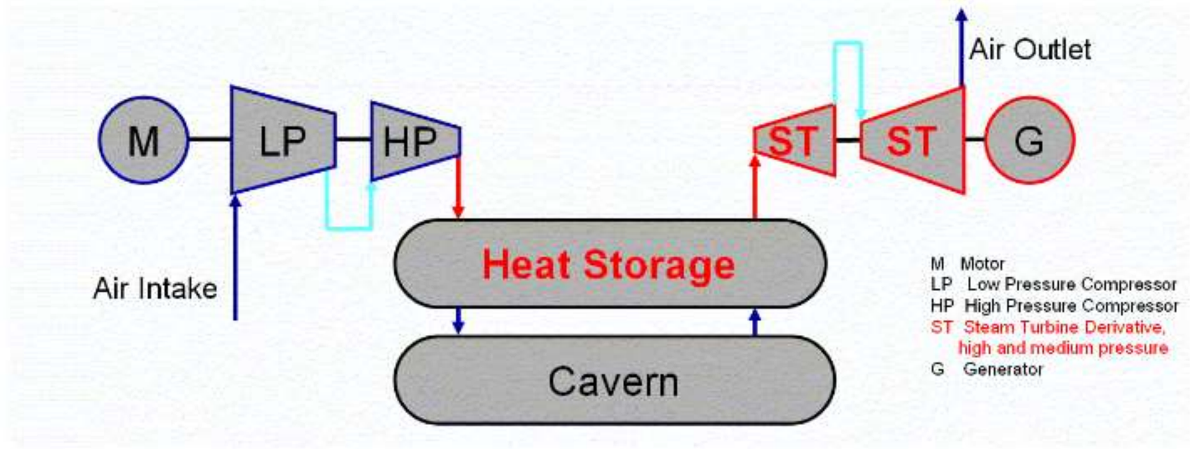

Figure 2. Conceptual representation of AA-CAES system [13]

- Isothermal compressed air storage (I-CAES)

The isothermal compressed air storage was proposed in order to overcome some of the limitations of the D-CAES and AA-CAES such as using numerous stages to compress, cool, heat and expand the air. The concept of I-CAES is presented in Figure 3. The advantages of I-CAES system are achieving true isothermal compression and expansion, improving round-trip efficiency and reducing capital costs. The principle of this technology is to compress and expand the air at near constant and close to ambient temperature to decrease the work of the compressor. In addition, the use of thermal storage is not needed in this technology. Two companies developing technology for ICAES are SustainX and Light Scale. The round-trip efficiency of an ICAES can reach 80 % with an estimated capital cost of 1000-1500 €/MW, and an expected life time around 30 years, with 15000 cycles [14].

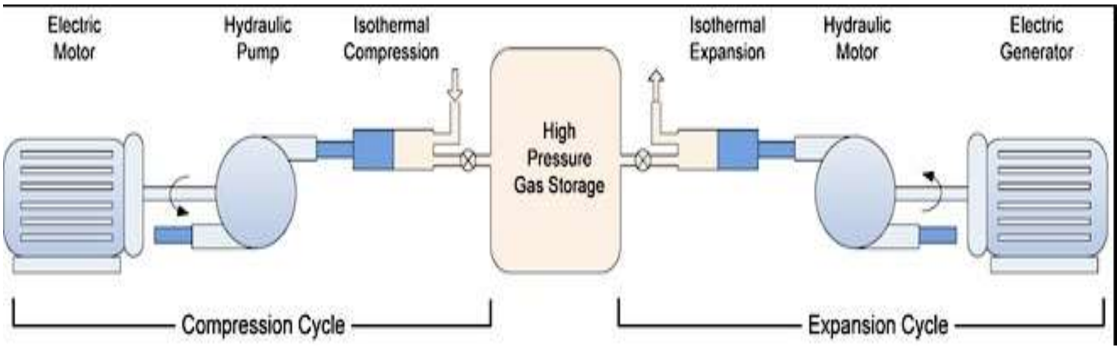

Figure 3 Conceptual representation of I-CAES system

According to the first law of thermodynamics, there will be no change in the internal energy during this process, so all the heat energy added is used to do work.

$$\Delta U = 0 = Q - W \tag{4}$$

Therefore:

$$Q = W \tag{5}$$

METHODOLOGY

The aim of this paper is to study the effect of some parameters including ambient temperature, stages number of compressor and expander and the pressure ratio of compressor and expander with compressor and turbine efficiencies on the D-CAES and A-CAES systems performance. To simplify the analysis, the following assumptions are made:

- The system operates in a steady state condition;
- All gases in the system are treated as the ideal gas;
- The kinetic and potential energy effects are negligible;
- No pressure drop in the heater exchanger.

Thermodynamic models of D-CAES

The D-CAES system is constituted of a compressor for charging the air into the reservoir, a combustion chamber and an expander. Both the compressor and expander are connected to a motor through clutches. In charging mode, the motor drives the compressor to inject the air into the reservoir. In discharging mode, the compressed air is released from the storing reservoir and fed into the combustion chamber where fuel is added to provide high temperature gases. The combustion gases expand through a turbine that drives the generator to provide peaking power [15]. The thermodynamic equations developed below consider mass and energy balances in all processes.

- Compressor model

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The compression train involves *n-1* intercooling stages and one aftercooling stage [15]. The work consumed by the compressor per unit mass of air is given by:

$$W_c({}^{kJ}/_{kg}) = \frac{C_p T_1 n}{\eta_c}(\sigma_c R^{1/n} - 1) \tag{6}$$

Where c_p is the specific heat at constant pressure, T_1 is the inlet temperature to the compressor train (in the present work T_1 refers to the Ambient pressure), η_c is the efficiency of the compressor. The parameter σ_c is a global pressure loss factor that represents pressure losses in the intercoolers.

$$\sigma_c = \mu_c^{n-1/n} \tag{7}$$

$$\mu_c = \left(\frac{P_{(i)}}{P_{(i)out}}\right)^{\frac{k-1}{k}} \tag{8}$$

The pressure values $P_{(i)}$ and $P_{(i)out}$ represent respectively the pressure values at the inlet and outlet of each intercooler/aftercooler of the compression train. $k = c_p / c_v$ is the specific heat ratio. The terminal isentropic ratio R refers to the constant pressure ratio assumed throughout the compression train.

$$R = \left(\frac{P_2}{P_1}\right)^{\frac{k-1}{k}} \tag{9}$$

- Heat exchanger model:

During both stages of compression and expansion, the intercoolers, aftercoolers, and re-heaters are treated as heat exchangers. The increasing temperature during compression is greater than the decreasing during expansion, so a greater amount of heat is generated during compression; and it is stored within heat exchangers in order to be used later for heating during the expansion process. This heat flow, which is removed for cooling the air to the inlet temperature, is given by:

$$Q(kJ) = mC_p(T_2 - T_1) \tag{10}$$

The storage process is characterized by constant volume and varying losses.

- Combustor model:

The combustor is a key component of the Diabatic – CAES system. During the discharging mode, the compressed air is released from the storage reservoir and introduced into a combustion chamber; where fuel is added and combusted in order to provide high temperature gases. The heated gases expand through the turbine; and then are reheated again by re-heaters

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to reach the previous temperature of combustion and consequently enter the next turbine. The combustion process occurs at constant pressure. The temperature ratio within a combustor is expressed by the following relationship:

$$T_{t4}/T_{t3} = (1 + \frac{f n_b Q}{C_p T_{t3}})/*(1+f) \quad (11)$$

Where T_{t4} is the temperature released after the combustion, and T_{t3} corresponds to the temperature at the throttle outlet after the storage stage. Q is the lower heating value associated to the fuel burnt within the combustor. The liquid fuel considered here is the gas oil (heating oil), which yields to a net calorific value of 42.8. The parameter n_b represents the combustion efficiency; f is the air fuel –mass ratio.

- Expander model:

The expansion train involves *(m-1)* reheating stages. The work generated by the expander per unit mass of air can be expressed as:

$$W_e(kJ/kg) = C_p T_1 \eta_e r_{me} m(1 - \frac{\sigma_e}{R^{1/m}}) \quad (12)$$

where T_1 is the temperature of the air released from the storing reservoir after passing through the throttle, η_e is the efficiency of the expander.

The parameter σ_e is a global pressure loss factor accounting for pressure losses in the reservoir, valve, piping, combustor, re-heaters and recuperator [15].

$$\sigma_e = \mu_e \mu_r^{1/m} \quad (13)$$

$$\mu_e = \left(\frac{P_{(j)}}{P_{(j)out}} \right)^{\frac{k-1}{k}} \quad (14)$$

$$\mu_r = \left(\frac{P_{6'}}{P_1} \right)^{\frac{k-1}{k}} \quad (15)$$

μ_e is the pressure ratio at each stage where P_j and $P_{(j)out}$ represent respectively the pressure values at the inlet and outlet of each re-heater/recuperator in the expansion train. μ_r is the ratio of the pressure value $P_{6'}$ released at the last expander (before entering to the recuperator) to the first pressure value P_1 of the air before entering to the compressor.

The parameter r_{me} is the ratio of the temperature released from the combustor (T_5) to the temperature at the compression stage and it is given by the following expression:

$$r_{me} = T_5 / T_1 \quad (16)$$

- Performance indicator:

In CAES, there are two different energy inputs, the electricity to drive the compressor and the fuel which is combusted to heat the air compressed. The evaluation of the performance of CAES is complicated due to the presence of these two energy inputs. The round-trip efficiency can be defined as the ratio of the output work of the expander to the sum of work consumed by compressors and the combusted fuel to heat the air compressed [17, 18]:

$$\eta_{th} = \frac{W_e}{W_c + q_f} \tag{17}$$

where q_f is the specific heat of the combusted fuel to heat the compressed air.

Thermodynamic models of AA-CAES

There are different stages in AA-CAES, which are:

- Compressor model

The compression process is assumed as an isentropic process, and the inlet temperature of the *i*-stage compressor is defined as $T_{i,inlet}$. When the unit mass of air passes through the *i*-stage compressor, the work consumed by compressor per mass unit of air is given by [17, 18]:

$$W_{c,i}\left(\frac{kJ}{kg}\right) = C_p T_{c,i}^{inlet}\left(\beta_{c,i}^{\frac{k-1}{k\eta_{poly,c}}} - 1\right) \tag{18}$$

where C_p is the specific heat of air at constant pressure, $\eta_{pol,c}$ is the polytropic efficiency of the compressor, $\beta_{c,i}$ is the pressure ratio of the *i*-stage compressor, k is the ratio of specific heat of air ($k=C_p/C_v$). $T_{c,i}^{inlet}$ is the inlet temperature.

The total work consumption per unit mass of air during compression work can be calculated by[13, 17, 18]:

$$W_c\left(\frac{kJ}{kg}\right) = \sum_{i=1}^{n} W_{c,i} \tag{19}$$

The outlet temperature can be estimated by using [17, 18]:

$$T_{c,i}^{outlet} = T_{c,i}^{inlet}\left(\beta_{c,i}^{\frac{k-1}{k\eta_{pol,c}}}\right) \tag{20}$$

- Heat exchanger model

The intercooler, aftercooler, preheater, and inter-heater are treated as heat exchangers with the counter flow type. The pressure drop through each heat exchanger affects the outlet pressures of both hot and cold streams and the outlet temperatures.

The exit temperature from the *i*-stage of compressor and inter-heater can be evaluated by using the following equation [18, 19]:

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$$T_{c,i} = (1-\varepsilon)\beta^{\frac{k-1}{k}} T_{i-1} + \varepsilon T_{\substack{S \\ AF}} \quad (21)$$

where, T_{AF} is the temperature of the cold thermal storage, (ε = 0,7) is the heat exchanger efficiency.
The hot storage temperature is related to cold storage and to the compressor exit temperature and it is calculated by using the following equation [18, 19]:

$$T_{AC} = T_{AF} + \varepsilon(T_{c,i} - T_{AF}) \quad (22)$$

- Expander model

After heat exchange in the heat exchanger, compressed air enters the turbine and undergoes the expansion process. When the unit mass of air comes through the *i*-stage turbine, the generated work is [17-19]:

$$W_{e,i}\left(\frac{kJ}{kg}\right) = C_p T_{e,i}^{inlet}(\beta^{\frac{(k-1)\eta_{pole}}{k}} - 1) \quad (23)$$

where $\eta_{pol,e}$ is the polytropic efficiency of the expander, $\beta_{c,i}$ is the pressure ratio of the *i*-stage expander, $T_{e,i}^{inlet}$ is the inlet temperature.

The total work generated per unit mass of air during expansion work can be calculated by [19]:

$$W_e\left(\frac{kJ}{kg}\right) = \sum_{i=1}^{n} W_{e,i} \quad (24)$$

The inlet temperature $T_{in,e}$ in expander is calculated by using the following equation [17, 19]:

$$T_{in,e} = T_0 + \varepsilon(T_{AC} - T_0) \quad (25)$$

The air is expanded and cooled down in the expander, the outlet temperature of the air from stage *i* of the expander can be found by using the following equation[17, 19]:

$$T_{out,e} = [T_{i-1}(1-\varepsilon) + \varepsilon T_{AC}]\beta^{\frac{1-k}{k}} \quad (26)$$

- Performance indicators

Since in AA-CAES system, the heat generated during compression is stored and used in the expansion process to heat the compressed air, the round-trip efficiency can be simply defined as the ratio of the output work of the expander to the work consumed by compressors [17, 18]:

$$\eta_{A-CAES} = \frac{W_e}{W_c} \quad (28)$$

RESULTS AND DISCUSSION

The performance of the D-CAES and A-CAES systems is studied by using the thermodynamic model under different working conditions. The results obtained from studying the performance of D-ACES and A-CAES under the variation of the ambient temperature, stages number of compression and expansion processes and ratio of the expansion and compression are presented here below.

Thermodynamic analysis of D-CAES

- Thermodynamic analysis of D-CAES under ambient temperature

This analysis has been done under ambient temperature ranges between 250K- 420K with five different values of compressor and expander efficiencies (65%, 70%, 75%, 80% and 85%). Figures 4 and 5 show respectively the variation of the work consumption per unit mass of air of compressor and the work generated per unit mass of air of expander as a function of the ambient temperature with various of compressor and expander efficiencies respectively. It can be noticed that the work consumption of compressor and the work generated of expander per unit mass of air increased with the increasing of the ambient temperature.

For an efficiency of 65%, the work consumption per unit mass of air of compressor increases from 1530 kJ/kg to 1891 kJ/kg and the work generation of expander per unit mass of air increases from1000 kJ/kg to 1094 kJ/kg as the temperature increases from 250 K to 420 K.
For an efficiency of 85%, the work consumption per unit mass of air of compressor increases from 1170 kJ/kg to 1446 kJ/kg and the work generation of expander per unit mass of air increases from 1308 kJ/kg to 1431 kJ/kg as the temperature increases from 250 K to 420 K.
It can be noticed that the rate change of the work consumption of compressor per unit mass of air increases with the increasing of the ambient temperature while the rate change of the work generated of expander per unit mass of air decreases with the increasing of the ambient temperature.

For an ambient temperature of 298K, the work consumption per unit mass of air of compressor decreases from 1613 kJ/kg to 1233 kJ/kg when the compressor efficiency increases from 65% to 85%, while the work generation of expander increases from 1045 kJ/kg to 1233 kJ/kg when the expander efficiency increases from 65% to 85%. On the other hand, the polytropic efficiency of compressor and expander has a significant impact on the performance of the D-CAES system. It can be seen from Figures 4 and 5 that with an increase of the compressor and expander efficiencies, the work consumption per unit mas of air of compressor decreases and the work generation per unit mass of air of expander increases. The rate of change of work consumption per unit mass of air of compressor with the increasing of the compressor efficiency is higher than the rate of change of work generation per unit mass of air of expander with the increasing of the expander efficiency.

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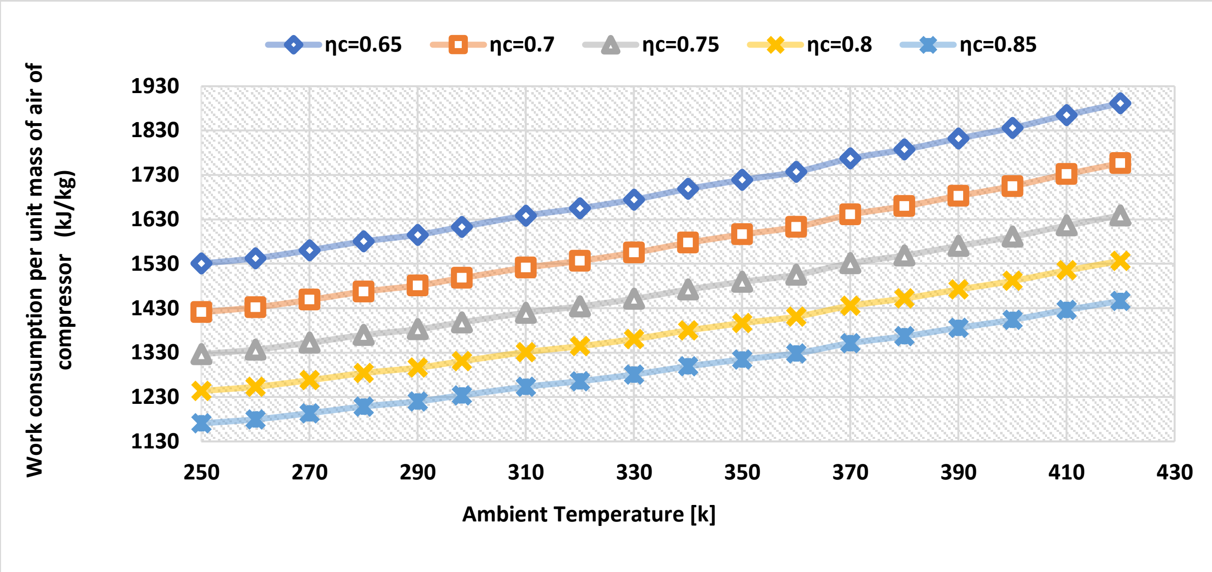

Figure 4 Work consumption per unit mass of air of compressor as a function of the ambient temperature with various compressor and expander efficiencies in D-CAES

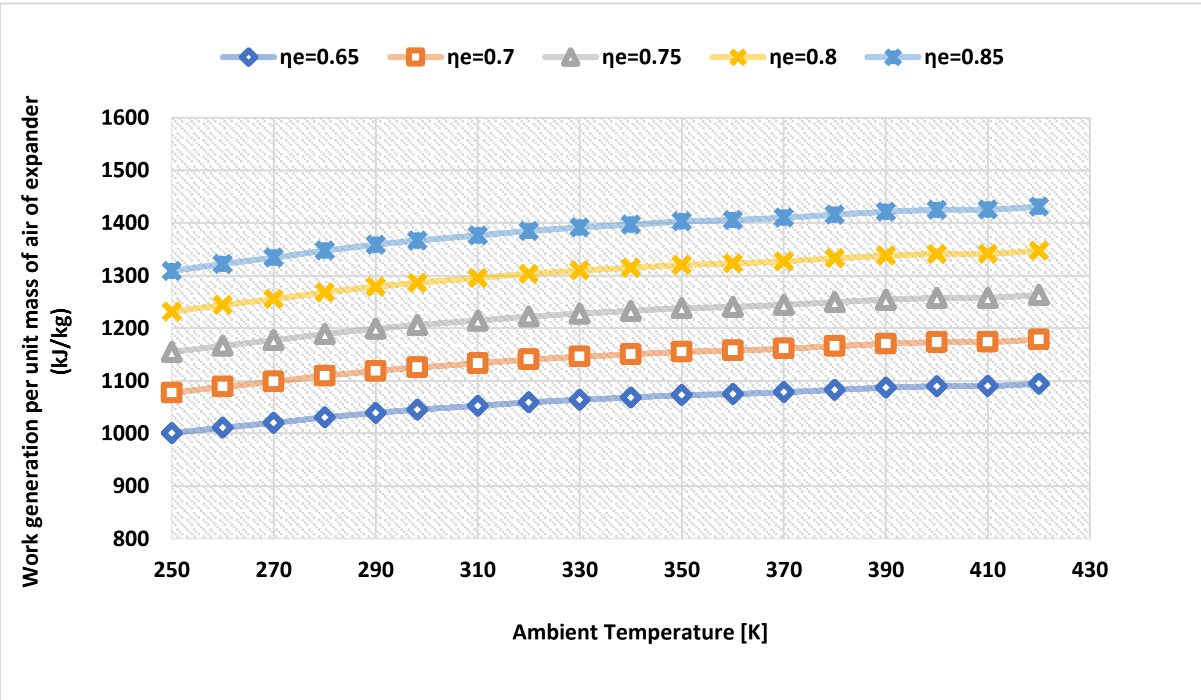

Figure 5 Work generation per unit mass of air of expander as a function of the ambient temperature with various compressor and expander efficiencies in D-CAES

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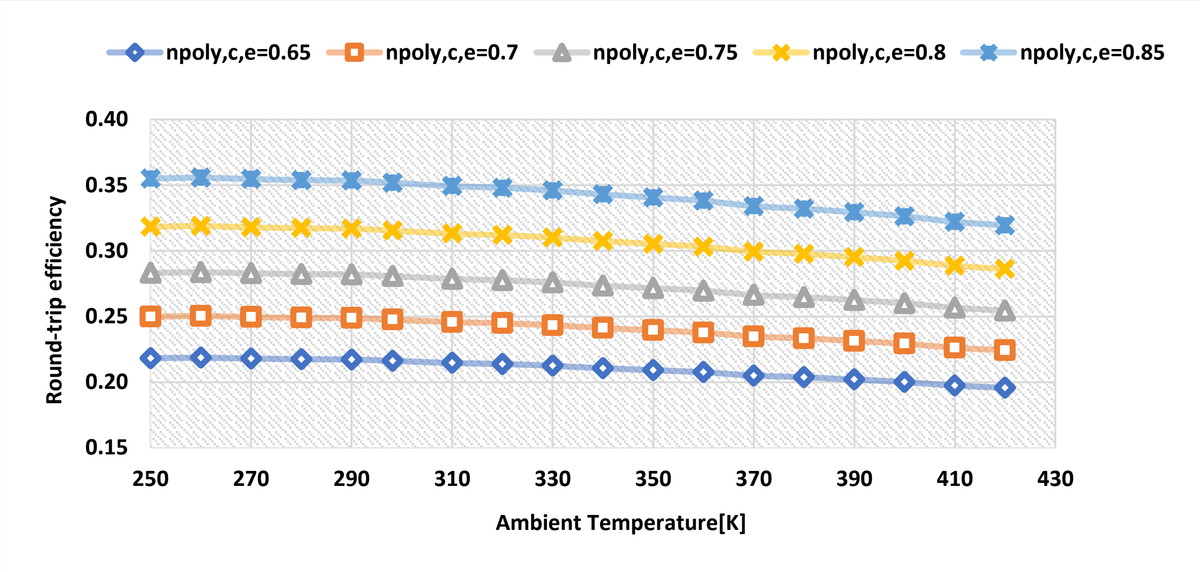

Figure 6 Roundtrip efficiency as a function of the ambient temperature with various values of compressor and expander efficiencies in D-CAES

Figure 6 displays the roundtrip efficiency of D-CAES under the variation of ambient temperature with various of compressor and expander efficiencies. For a constant polytropic efficiency of compressor and expander, the round-trip efficiency decreases as the ambient temperature increases. For a compressor and expander efficiencies of 65%, the round trip efficiency decreases from 21.6 % to 19.5 % while it goes from 35 % to 32 % for an efficiency of 85% as the ambient temperature increases from 250 K to 420 K.

- Thermodynamic analysis of D-CAES under different compression and expansion ratios

This analysis has been done under four values of compression ratio (5, 7, 9 and 12) and four values of expansion ratio (3, 5, 7 and 9) with five different values of compressor and expander efficiencies (65%, 70%, 75%, 80% and 85%). Figures 7 and 8 show respectively the distribution of the work consumption per unit mass of air of compressor and the distribution of the work generation per unit mass of air of expander as a function of compression and expansion ratios with various of compressor and expander efficiencies. It can be noticed from Figures 7 and 8 that the work consumption per unit mass of air of compressor increases with the increasing of compression ratio while the work generation per unit mass of air of expander decreases with the increasing of expansion ratio respectively. For a compressor efficiency of 65%, an increase of pressure ratio in compression stage from 5 to 11 leads to an increasing of work consumption per unit mass of air of compressor from 1201 kJ/kg to 2242 kJ/kg. However, it increases from 919 kJ/kg to 1715 kJ/kg under a compressor efficiency of 85%. It can be noticed from Figure 8 that an increase of pressure ratio in compression stage from 3 to 9 leads to a decreasing of work generation per unit mass of air of expander from 1468 kJ/kg to 430 kJ/kg under a expander efficiency of 65%. However, it decreases from 1920 kJ/kg to 563 kJ/kg under a compressor efficiency of 85%.

Figure 9 describes the change of the roundtrip efficiency of D-CAES system under the variation of the pressure ratios and efficiencies of the compressor and expander. It can be seen that the roundtrip efficiency of the D-CAES system decreases with the increasing of compression and expansion ratio. For a compressor and expander efficiencies of 65%, the round trip efficiency decreases from 38.5 % to 6.7% while it decreases from 61.9% to 11.1% for a compressor and expander efficiencies of 85%.

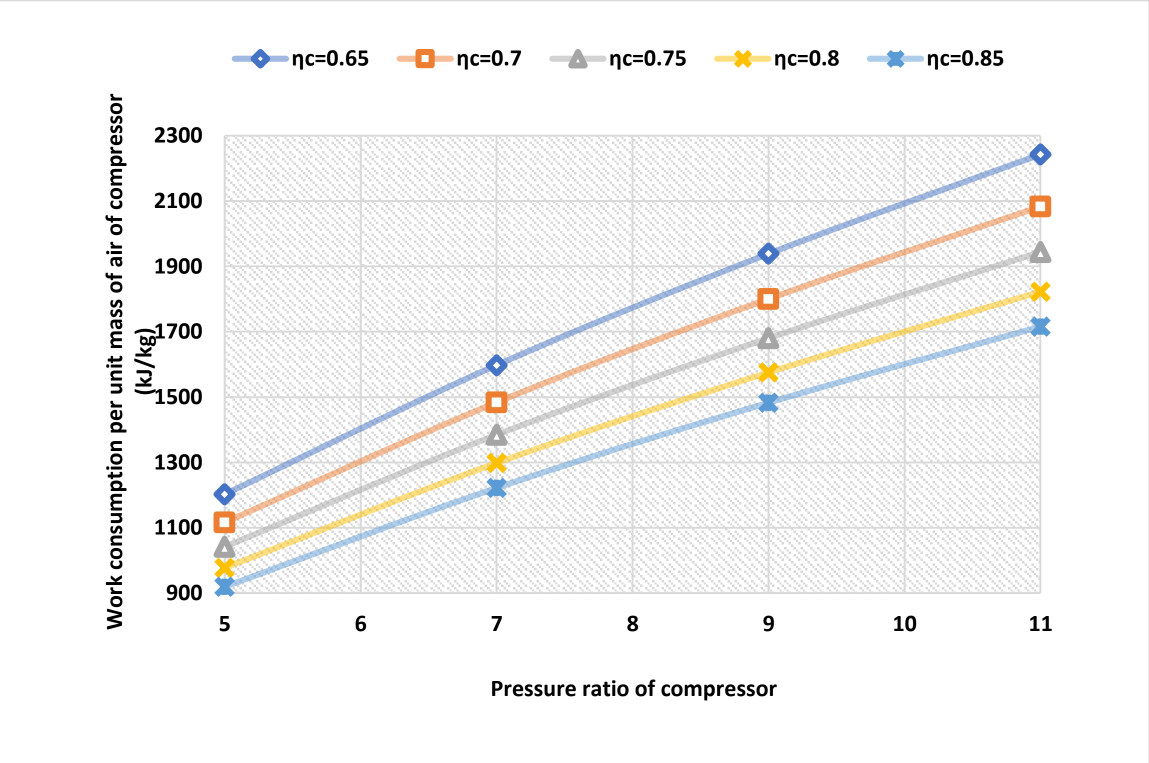

Figure 7 Work consumption per unit mass of air of compressor as a function of compression ratios with various of compressor and expander efficiencies in D-CAES

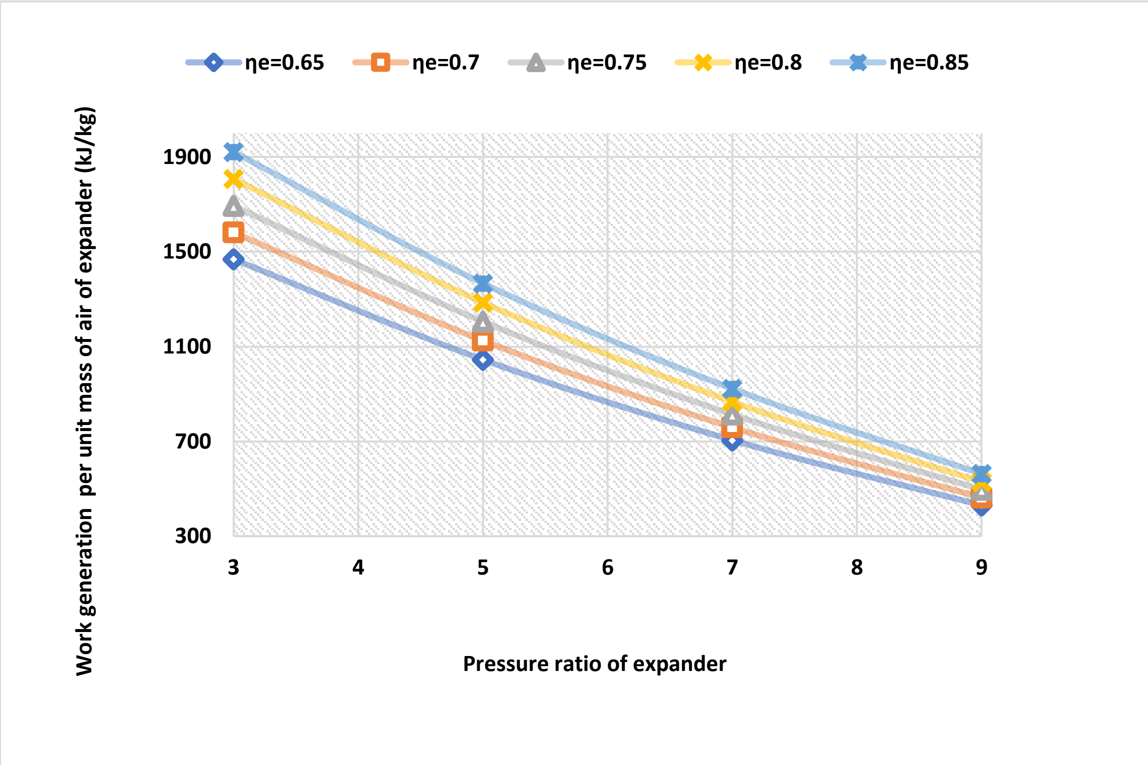

Figure 8 Work generation per unit mass of air of expander as a function of expansion ratios with various of compressor and expander efficiencies in D-CAES

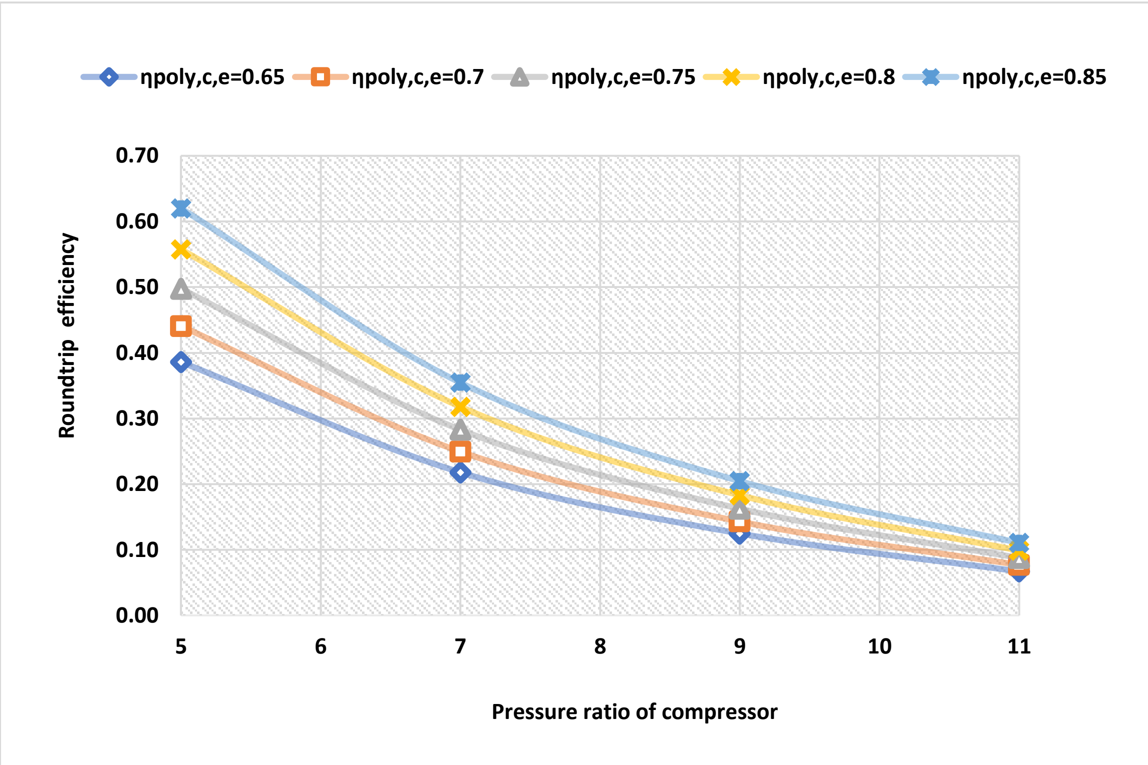

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Figure 9 Roundtrip efficiency as a function of expansion and compression ratios with various of compressor and expander efficiencies in D-CAES

- Thermodynamic analysis of D-CAES under different stage number of compression and expansion

This analysis has been done under four different stage number values of compression (2, 3, 4 and 5) and four different stage number values of compression (3, 4, 5, 6) with five different values of compressor and expander efficiencies (65%, 70%, 75%, 80% and 85%).

Figures 10 and 11 show the distribution of the work consumption per unit mass of air of compressor and the distribution of the work generation per unit mass of air of expander versus the number of stages in compression and expansion trains with various expander and compressor efficiencies respectively. The work consumption and generation per unit mass of air is proportional to the increase of number of stages in both cases. For a compressor efficiency of 85%, the work consumption of compressor rises from 1221 kJ/kg to 3053 kJ/kg as the number of stages increases from 2 to 5. However for a compressor efficiency of 65%, it rises from 1595 kJ/kg to 3993 kJ/kg.

The work generation of expander rises from 1336 kJ/kg to 3416 kJ/kg as the number of stages increases from 2 to 5 for an expander efficiency of 85%. However, it rises from 1045 kJ/kg to 2612 kJ/kg for an expander efficiency of 65%. It can be noticed that the rate of change of the work consumption of compressor decreases with the increasing of the compressor efficiency.

It can be seen in Figure 11 that the rate of change of the work consumption of compressor increases with the increasing of compressor efficiency.

Figure 12 shows the distribution of the round-trip efficiency of D-CAES as a function of stage stages number of compression and expansion trains under various polytropic efficiency of compressor and expander. The increase of number of compression from 3 to 6 and expansion stages from 2 to 5 leads to increase the roundtrip efficiency from 35.4 % to 40.5 % for a compressor and expander efficiencies of 85%. However, it increases from 21.7 % to 24.33 % for a compressor and expander efficiencies of 85%.

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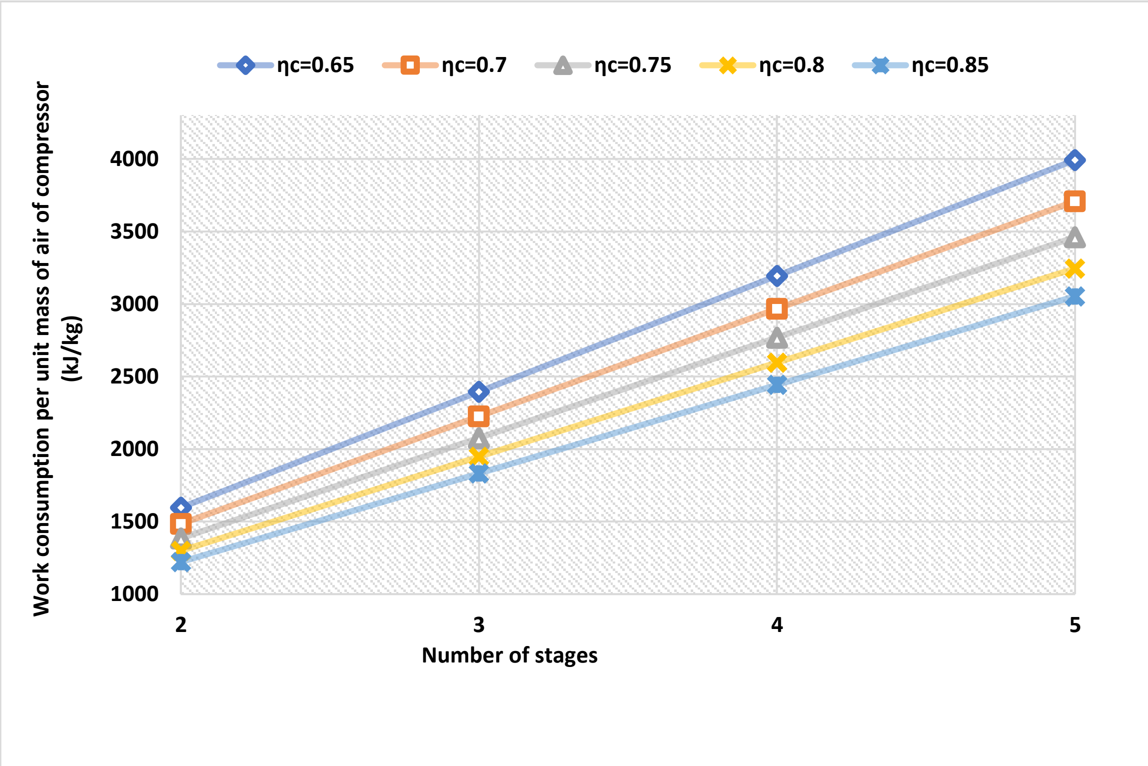

Figure 10 work consumption per unit mass of air of compressor as a function of stage numbers of compression with various of compressor and expander efficiencies in D-CAES

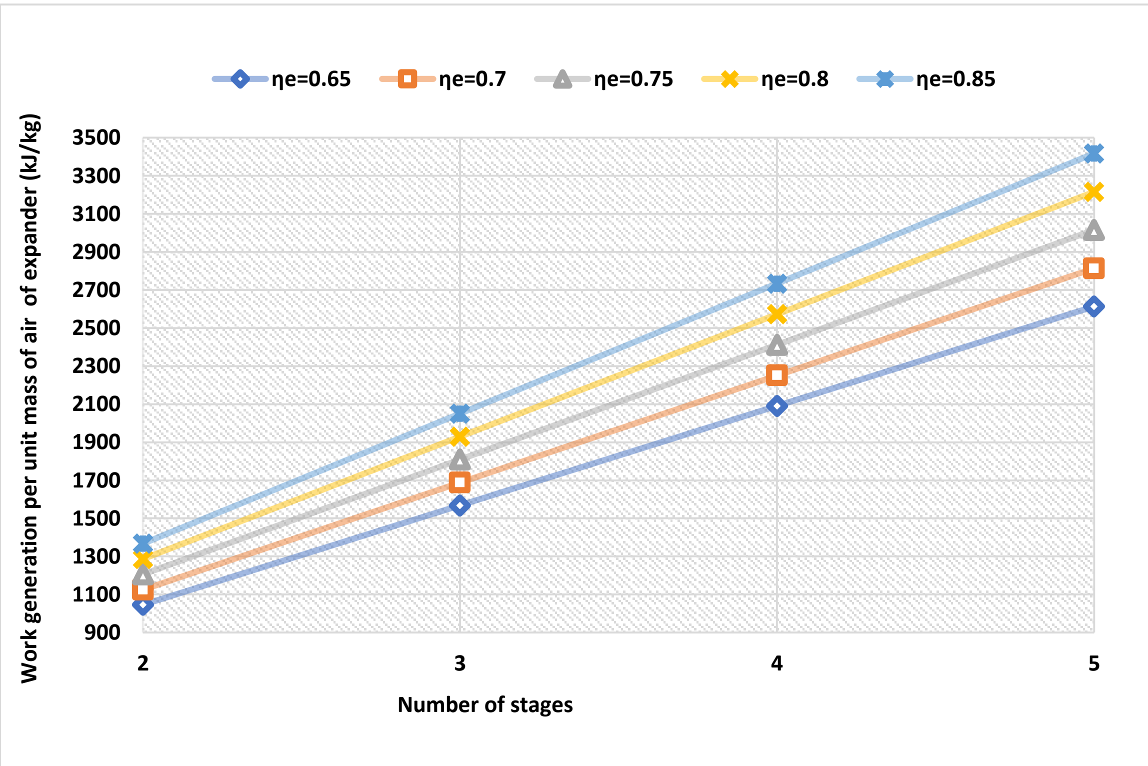

Figure 11 Work generation per unit mass of air of expander as a function of stage number of expander with various of compressor and expander efficiencies in D-CAES

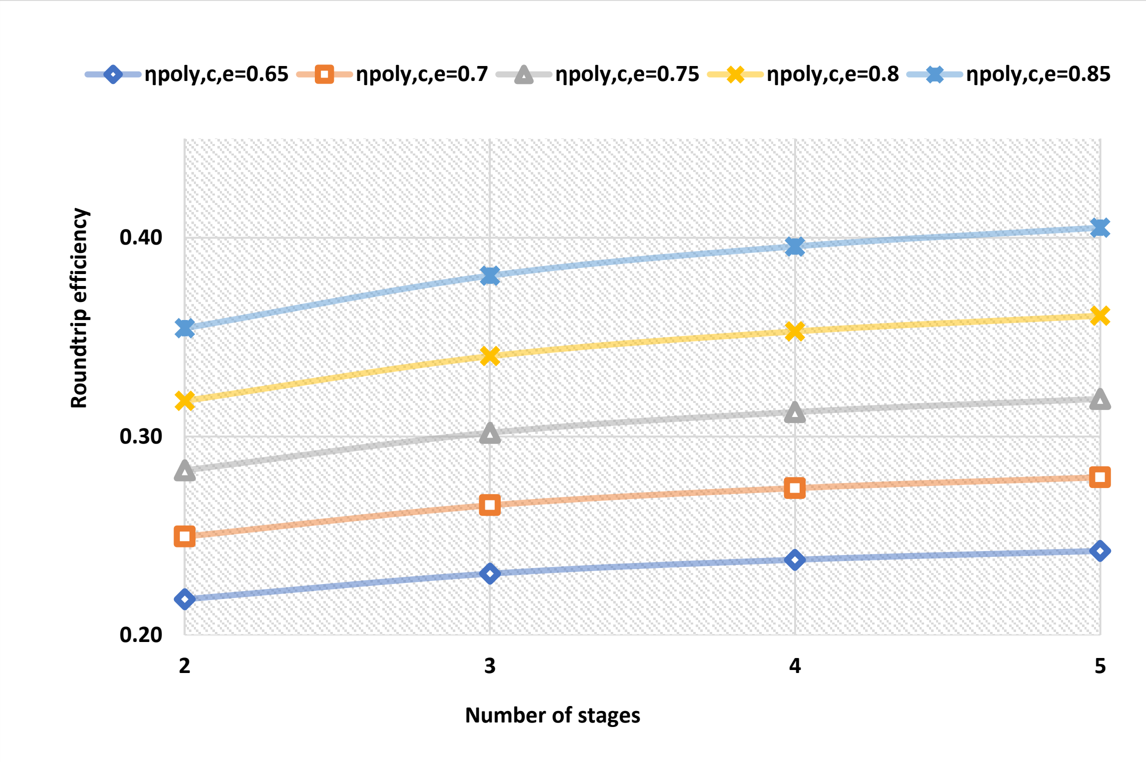

Figure 12 Round-trip efficiency as a function of stage numbers of compression with various of compressor and expander efficiencies in D-CAES

Thermodynamic analysis of AA-CAES

- Thermodynamic analysis of AA-CAES under ambient temperature

This analysis has been done under ambient temperature range between 250 K- 420 K with five different values of compressor and expander efficiencies (65%, 70,% , 75%, 80% and 85%). Figures 12 and 13 show the distribution of the work consumption of compressor per unit mass of air and work generation of expander per unit mass of air at the different ambient temperature with various of compressor and expander efficiencies respectively.

For a compressor and expander efficiencies of 85%, The work consumption of compressor per unit mass of air increases from 477 kJ/kg to 690 kJ/kg while the work generation of expander per unit mass of air increases from 360 kJ/kg to 473 kJ/kg as the ambient temperature increases from 250 K to 420 K. While for a compressor and expander efficiencies of 65%, the work consumption of compressor per unit mass of air increases from 722 kJ/kg to 1044 kJ/kg while

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the work generation of expander per unit mass of air increases from 307 kJ/kg to 405 kJ/kg as the ambient temperature increases from 250 K to 420 K.

For an ambient temperature of 250 K, The work consumption of compressor per unit mass of air increases from 722 kJ/kg to 477 kJ/kg while the work generation of expander per unit mass of air increases from 307 kJ/kg to 360 kJ/kg as the compressor and expander efficiencies increases from 65% to 85%. While for an ambient temperature of 420 K, The work consumption of compressor per unit mass of air increases from 1044 kJ/kg to 690 kJ/kg while work generation of expander per unit mass of air increases from 405 kJ/kg to 473 kJ/kg as the compressor and expander efficiencies increases from 65% to 85%. The results indicate that the ambient temperature has a strong impact on the work consumption and work generated per unit mass of air in the A-CAES system.

Figure 14 shows the distribution of the round-trip efficiency of AA-CAES as a function of ambient temperature with various polytropic efficiency of compressor and expander. The results show that the round-trip efficiency decreases with the increase of the ambient temperature. It is obvious since the rate of increasing of energy density consumption of compression is bigger than the rate of increasing of energy density generation of expander.
For a compressor and expander efficiencies of 85%, the round-trip efficiency decreases from 75% to 68 % as the ambient temperature increases from 250 K to 420 K. While, for a compressor and expander efficiencies of 65%, it decreases from 46.5% to 42.5% as the ambient temperature increases from 250 K to 420 K. It can be concluded that the impact of ambient temperature on the round-trip efficiency declines when the polytropic efficiency of compressor and expander decreases.

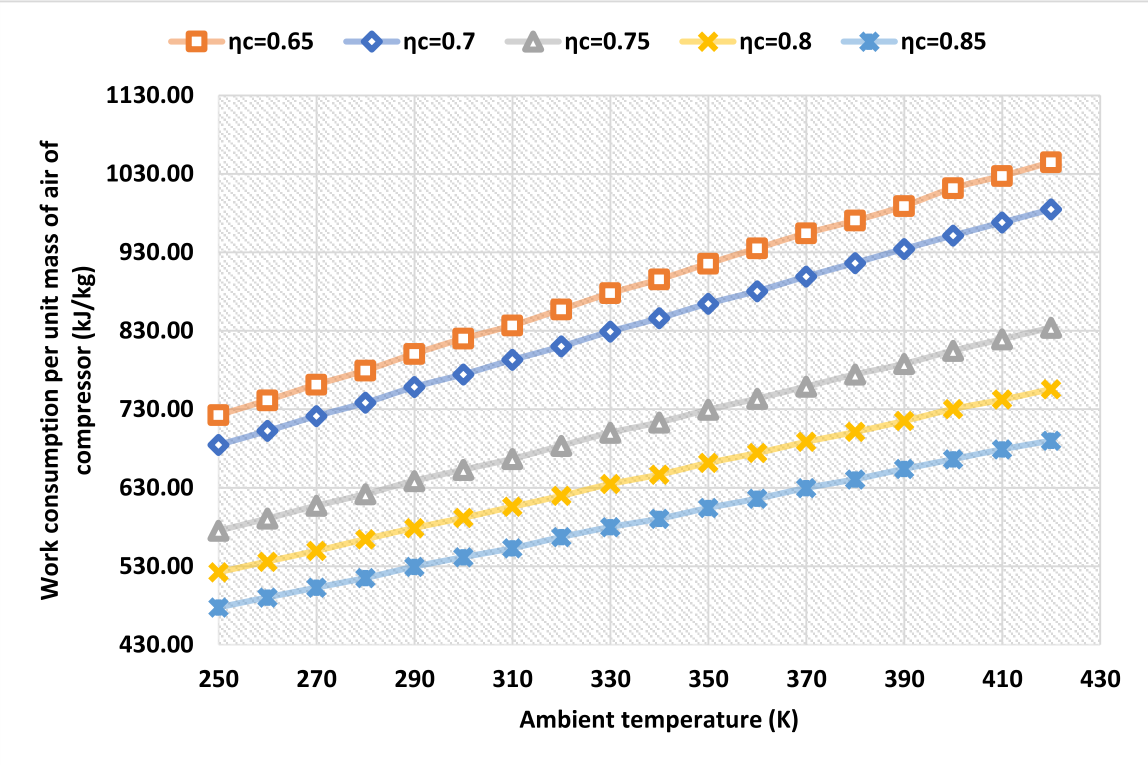

Figure 12 Work consumption of compressor per unit mass of air as a function of the ambient temperature with various of compressor and expander efficiencies in AA-CAES

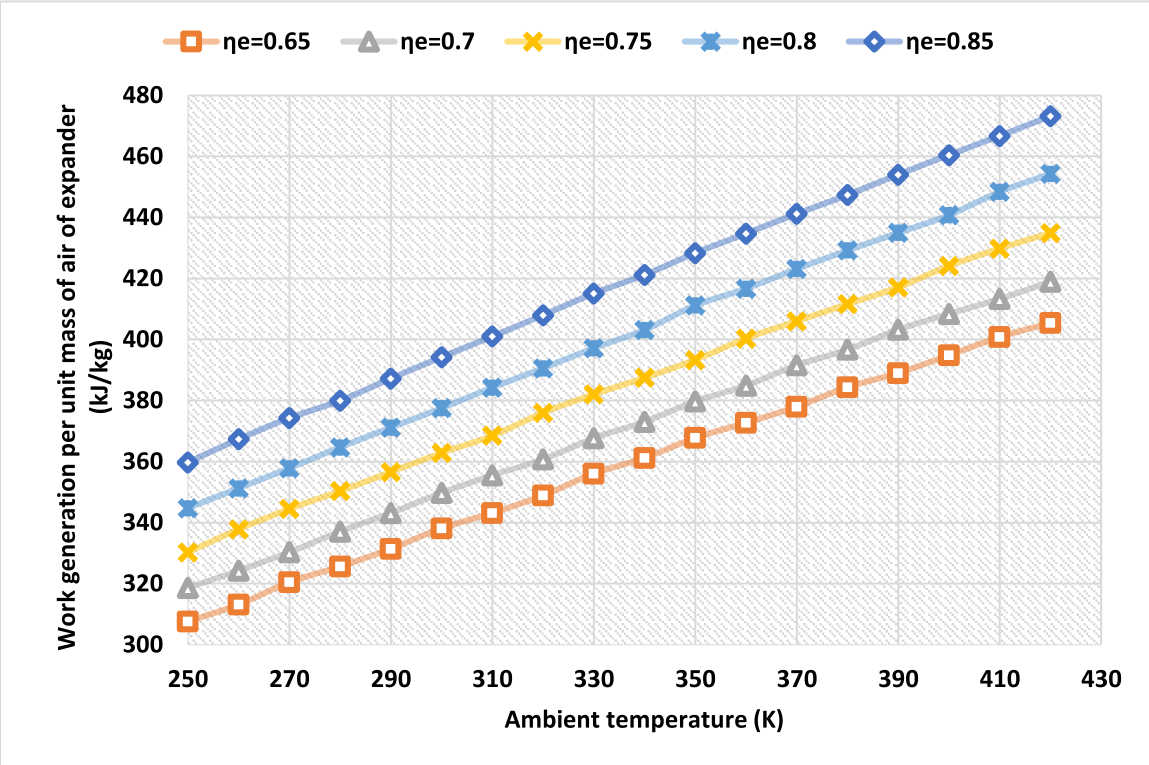

Figure 13 Work generation of expander per unit mass of air as a function of the ambient temperature with various of compressor and expander efficiencies in AA-CAES

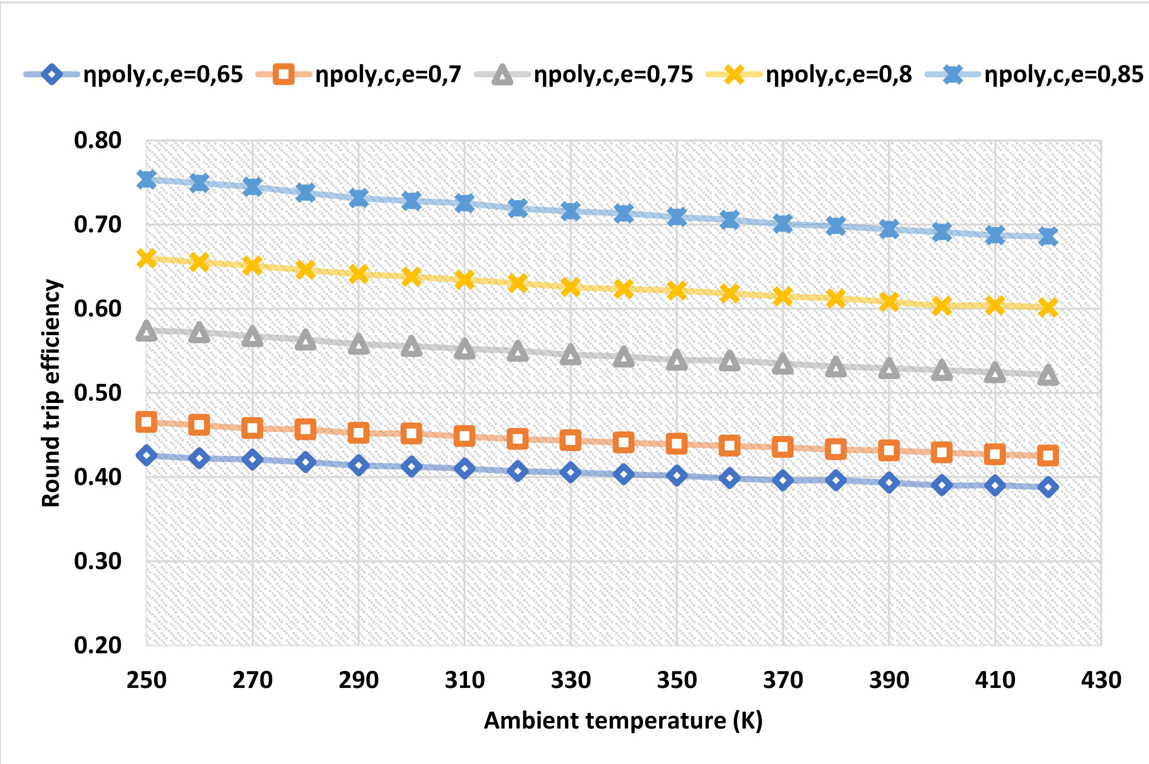

Figure 14 Round-trip efficiency as a function of the ambient temperature with various of compressor and expander efficiencies in AA-CAES

- Thermodynamic analysis of AA-CAES under different compression and expansion ratios

This analysis has been done under four values of compression ratio (5, 7, 9 and 11) and four values of expansion ratio (3, 6, 8 and 9) with five different values of compressor and expander efficiencies (65%, 70%, 75%, 80% and 85%).

Figures 15 and 16 show the work consumption of compressor per unit mass of air and the work generation of expander per unit mass of air, versus the change of the compression and the expansion ratios with various compressor and expander efficiencies.

It can be noticed that for a compressor efficiency of 65%, the work consumption of compressor per unit mass of air increases from 758 kJ/kg to 1603 kJ/kg and the work generation of expander per unit mass of air increases from 110 kJ/kg to 344 kJ/kg as the compression and expansion ratios increase respectively from 5 to 11 and from 3 to 9 respectively. While for a compressor efficiency of 85%, the work consumption of compressor per unit mass of air increases from 527 kJ/kg to 1061 kJ/kg and the work generation of expander per unit mass of air increases from 148 kJ/kg to 477 kJ/kg as the compression and expansion ratios increase respectively from 5 to 11 and from 3 to 9 respectively. It can be noticed that the variation of the work consumption of compressor per unit mass of air and the work generation of expander per unit mass of air increases with the increase of compression and expansion ratios.

Figure 17 shows the round-trip efficiency of the system AA-CAES versus compression and expansion ratio with various values of compressor and expander efficiencies. The round-trip efficiency increases with the increasing of compression and expansion ratio. For a compressor and expander efficiencies of 0.85, the round-trip efficiency of AA-CAES system increases from 28 % to 45 % as the compression ratio increases from 5 to 11 and the expansion ratio increases from 3 to 9. For a compressor and expander efficiencies of 0.65, the round-trip efficiency of AA-CAES system decreases from 14 % to 21 % as the compression ratio increases from 5 to 11 and the expansion ratio from 3 to 9.

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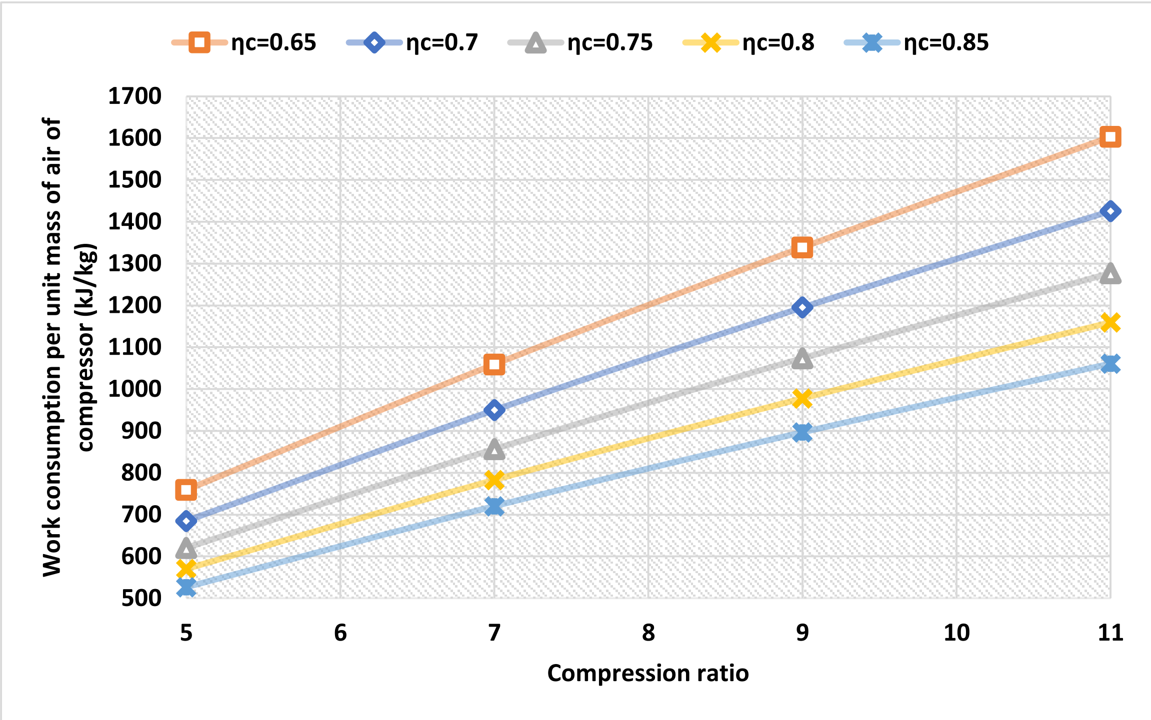

Figure 15 Work consumption per unit mass of air of compressor as a function of compression ratios with various of compressor and expander efficiencies in AA-CAES

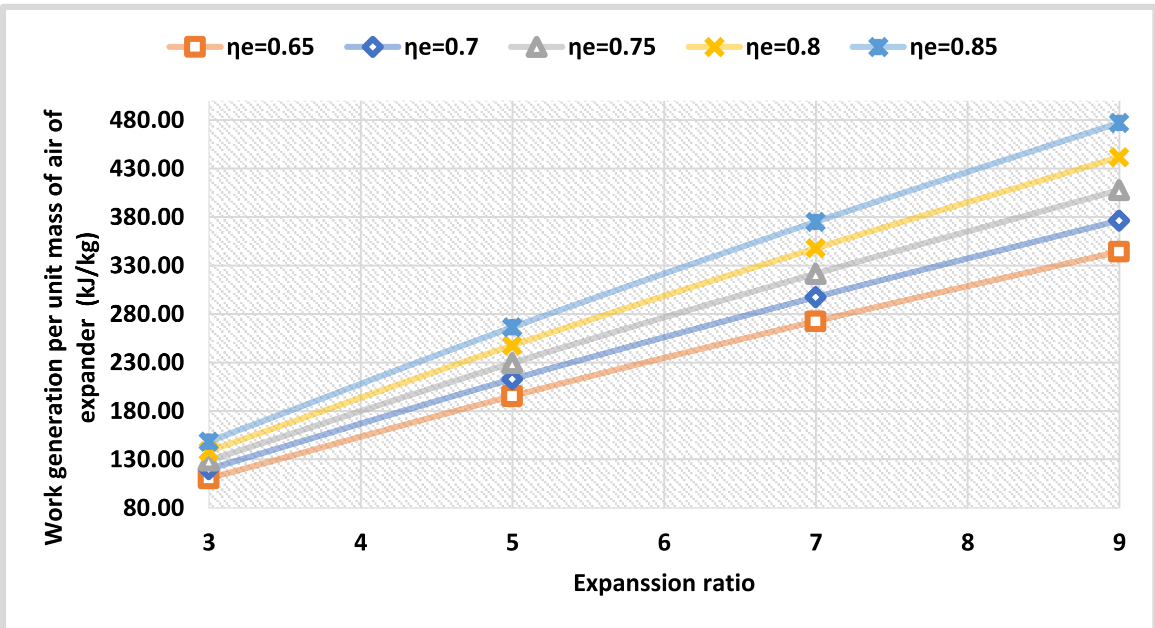

Figure 16 work generation per unit mass of air of expander as a function of expansion ratios with various of compressor and expander efficiencies in AA-CAES

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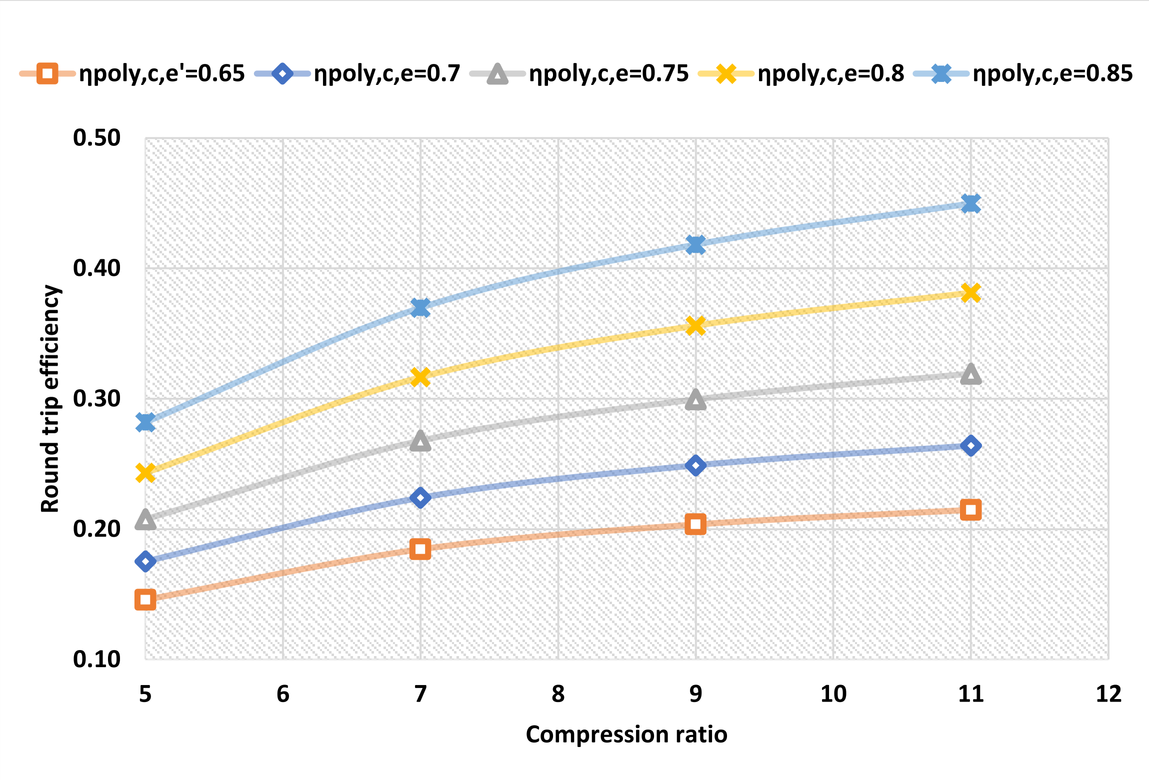

Figure 17 Round-trip efficiency as a function of the expansion and compression ratios with various of compressor and expander efficiencies in AA-CAES

- Thermodynamic analysis of AA-CAES under different stages number of compression and expansion

This analysis has been done under four different stages number values of compression and expansion (2, 3, 4 and 5) with five different values of compressor and expander efficiencies (65%, 70,%, 75%, 80% and 85%).

Figures 18 and 19 show the distribution of work consumption per unit mass of air of compressor and work generation per unit mass of air of expander, versus the stages number of compression and the expansion with various compressor and expander efficiencies.

For a compressor and expander efficiencies of 65%, the work consumption per unit mass of air of compressor decreases from 807.9 kJ/kg to 668 kJ/kg and the work generation of expander decreases from 295 kJ/kg to 145 kJ/kg as the stages number of compression and expansion increases from 2 to 5.

For a compressor and expander efficiencies of 85%, the work consumption per unit mass of air of compressor decreases from 535 kJ/kg to 472 kJ/kg and the work generation of expander decreases from 342 kJ/kg to 185 kJ/kg as the stages numbers of compression and expansion increases from 2 to 5.

The rate of change of energy density consumption of compressor increases with the increase of stages number but and it decreases with the increase of compressor efficiency.

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Figure 20 shows the round-trip efficiency of the system AA-CAES versus the number of stages of compression and expansion with various of compressor and expander efficiencies. The round-trip efficiency decreases as the stage number of both compression and expansion trains gets larger. For a compressor and expander efficiencies of 85%, the round-trip efficiency of AA-CAES system decreases from 77 % to 44 % as the stages number of compression and expansion increase from 2 to 5. For a compressor and expander efficiencies of 65%, the round-trip efficiency of AA-CAES system is reduced from 42 % to 24 % as the stages number of compression and expansion increase from 2 to 5. It can be noticed that the rate of decreasing of round-trip efficiency gets smaller with the decrease of compression and expander efficiencies.

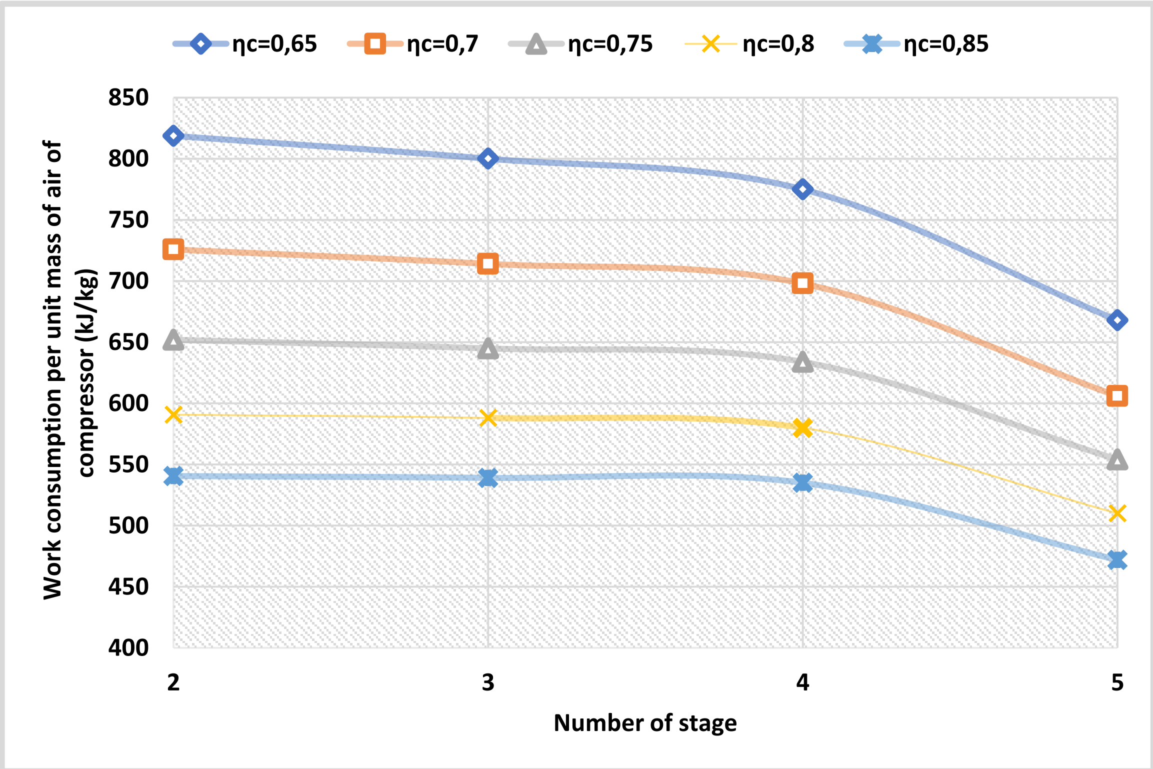

Figure 18 Work consumption per unit mass of air of compressor as a function of stages numbers of compression with various of compressor and expander efficiencies in AA-CAES

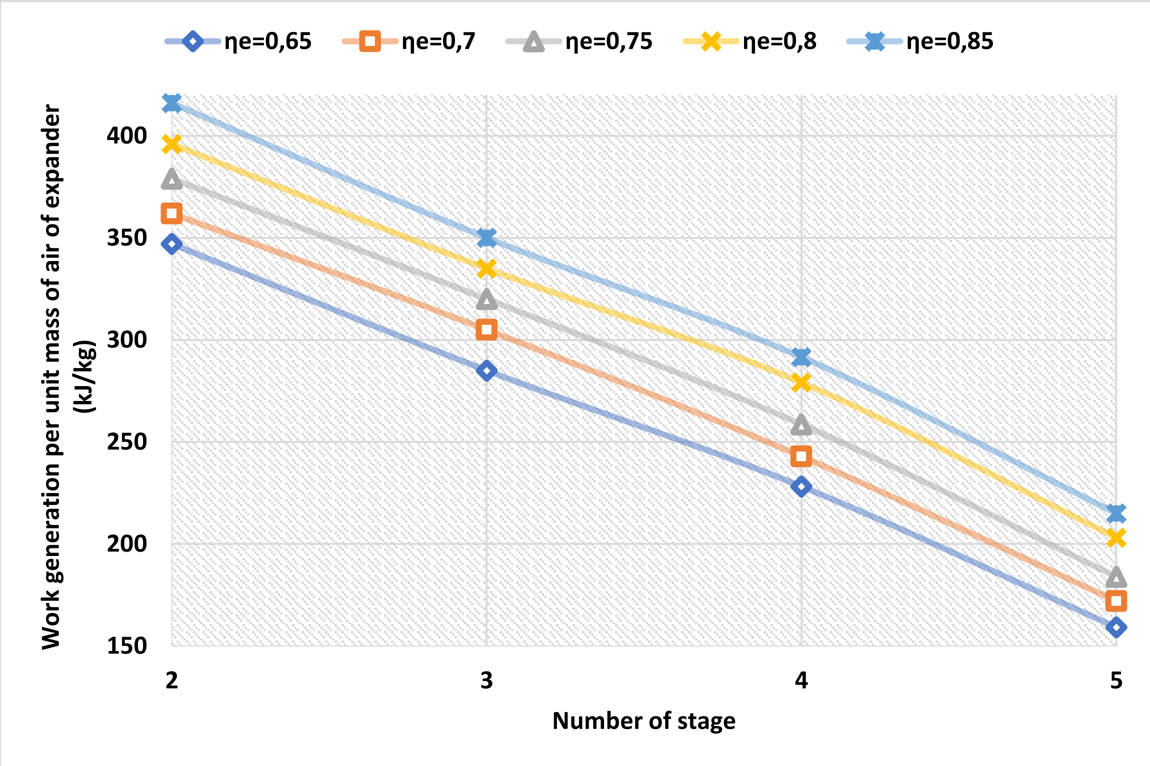

Figure 19 Work generation per unit mass of air of expander as a function of stages number of expansion with various of compressor and expander efficiencies in AA-CAES

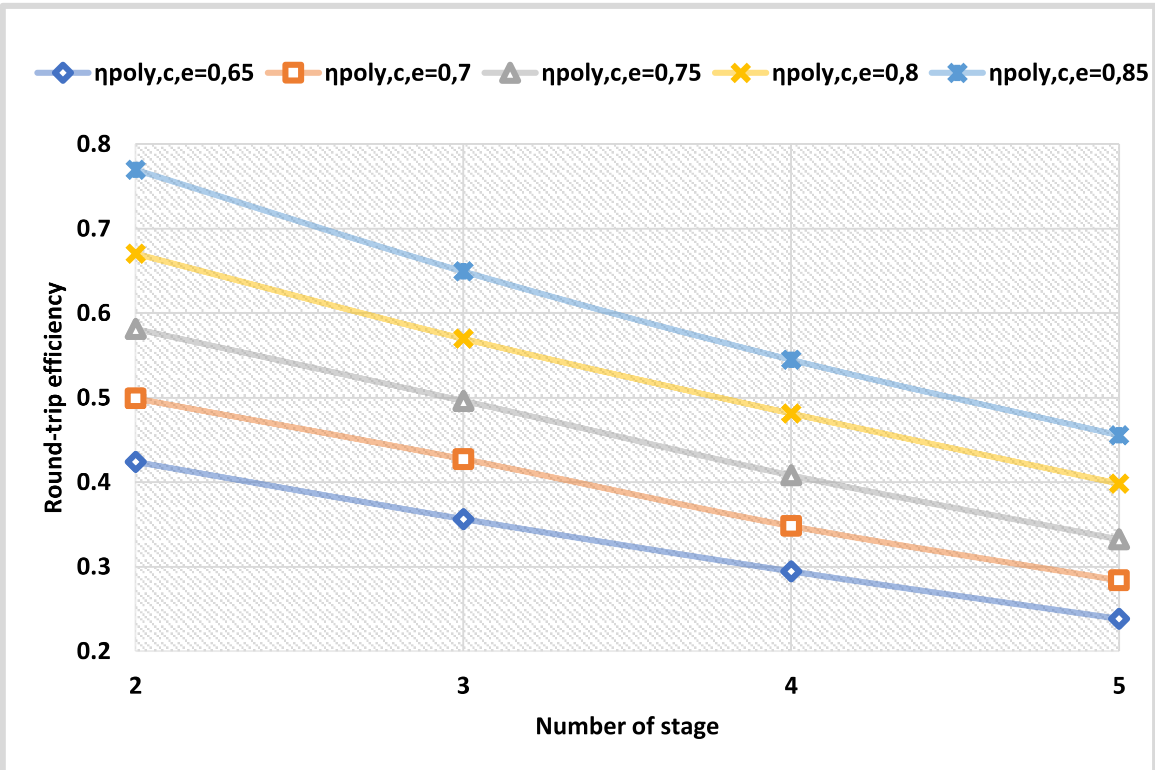

Figure 20 Round-trip efficiency as a function of stages number of expansion and compression* with various of compressor and expander efficiencies *in AA-CAES

Comparison between D-CAES and A-CAES

A comparison between the obtained results from studying the effect of ambient temperature, compression and expansion ratios and the stages number of expansion and compression trains with various of compressor and expander efficiencies on the D-CAES and A-CAES systems is presented here.

- Comparison between D-CAES and AA-CAES under different ambient temperature

The work consumption per unit mass of air of compressor and work generation per unit mass of air of expander increase with the increase of the ambient temperature in both D-CAES and A-CAES system. The roundtrip efficiency decreases with the increasing of ambient temperature in both systems.

- Comparison between D-CAES and AA-CAES under different compression and expansion ratios

In D-CAES, the work consumption per unit mass of air of compressor increases while the work generation per unit mass of air of expander decreases with the increase of the compression and expansion ratios. In AA-CAES, the work consumption per unit mass of air of compressor and the work generation of unit mass of air of expander increase with the increase of the compression and expansion ratios. The round trip efficiency decreases with the increasing of compression and expansion ratios in D-CAES but it increases in AA-CAES.

- Comparison between D-CAES and A-CAES under different stages number

The work consumption per unit mass of air of compressor and the work generation per unit mass of air of expander increase with the increase of the compression stages number in D-CAES while the work consumption per unit mass of air of compressor and the work generation of expander decrease with the increase of the compression stages number in AA-CAES. The round trip efficiency increases with the increase of the stage number of compression and expansion trains in D-CAES but it decreases in AA-CAES.

CONCLUSION

A thermodynamic analysis of D-CAES and AA-CAES under different parameters has been carried out in this paper. The studied parameters are the ambient temperature, expansion and compression ratios and the number of stages of compression and expansion. It has been found that:

- D-CAES system: The work consumption per unit mass of air of compressor increases with the increase of the ambient temperature, the increase of the compression and expansion ratios, and the increases of the compression stages number. The work generation per unit mass of air of expander increases with the increase of the ambient temperature and with the expanssion stages number while it decreases with the increase of expansion ratio. The round trip efficiency of D-CAES systems decreases with the increasing of ambient temperature, and expansion and compression ratios but it increases with the increase of the stages number of compression and expansion trains.

- AA-CAES system: The work consumption per unit mass of air of compressor increases with the increase of the ambient temperature and the increase of the compression ratios while it decreases with the increases of the compression stages number. The work generation per unit mass of air of expander increases with the increase of the ambient temperature and the expansion ratios while it decreases with the increase of the expansion stages number. The round trip efficiency decreases with the increase of ambient temperature and stages number of compression and expansion trains while it increases with the increase of the expansion and compression ratios.

REFERENCES

1. Masaud, T.M., K. Lee, and P. Sen. *An overview of energy storage technologies in electric power systems: What is the future?* in *North American Power Symposium 2010*. 2010. IEEE.
2. Hino, T. and A. Lejeune, *Pumped storage hydropower developments.* 2012.
3. Crotogino, F., K.-U. Mohmeyer, and R. Scharf, *Huntorf CAES: More than 20 years of successful operation.* Orlando, Florida, USA, 2001.
4. Budt, M., et al., *A review on compressed air energy storage: Basic principles, past milestones and recent developments.* Applied Energy, 2016. **170**: p. 250-268.
5. Ding, Y., et al., *Liquid air energy storage*, in *Storing Energy*. 2016, Elsevier. p. 167-181.
6. Li, Y., et al., *A real-time dispatch model of CAES with considering the part-load characteristics and the power regulation uncertainty.* International Journal of Electrical Power & Energy Systems, 2019. **105**: p. 179-190.
7. Kalhammer, F.R. and T.R. Schneider, *Energy storage.* Annual Review of Energy, 1976. **1**(1): p. 311-343.
8. Venkataramani, G., et al., *A review on compressed air energy storage–A pathway for smart grid and polygeneration.* Renewable and Sustainable energy reviews, 2016. **62**: p. 895-907.
9. Garvey, S.D. and A. Pimm, *Compressed Air Energy Storage*, in *Storing Energy*. 2016, Elsevier. p. 87-111.
10. Drury, E., P. Denholm, and R. Sioshansi, *The value of compressed air energy storage in energy and reserve markets.* Energy, 2011. **36**(8): p. 4959-4973.

11. Zhao, P., et al., *Energy efficiency analysis and off-design analysis of two different discharge modes for compressed air energy storage system using axial turbines.* Renewable Energy, 2016. **85**: p. 1164-1177.
12. Steen, E. and M. Torestam, *Compressed air energy storage: Process review and case study of small scale compressed air energy storage aimed at residential buildings*. 2018.
13. Mozayeni, H., et al., *Performance study of an advanced adiabatic compressed air energy storage system.* Energy Procedia, 2017. **110**: p. 71-76.
14. Helsingen, E.M., *Adiabatic compressed air energy storage*, in *Faculty of Engineering Science and Technology*. 2015, Norges teknisk-naturvitenskapelige universitet: Norway.
15. Vadasz, P., *Compressed air energy storage: optimal performance and techno-economical indices.* International Journal of Thermodynamics, 1999. **2**(2): p. 69-80.
16. Succar, S. and R.H. Williams, *Compressed air energy storage: theory, resources, and applications for wind power.* Princeton environmental institute report, 2008. **8**: p. 81.
17. Peng, H., et al., *Thermodynamic analysis of an improved adiabatic compressed air energy storage system.* Applied energy, 2016. **183**: p. 1361-1373.
18. Zhang, Y., et al., *Thermodynamic analysis of energy conversion and transfer in hybrid system consisting of wind turbine and advanced adiabatic compressed air energy storage.* Energy, 2014. **77**: p. 460-477.
19. Grazzini, G. and A. Milazzo, *Thermodynamic analysis of CAES/TES systems for renewable energy plants.* Renewable energy, 2008. **33**(9): p. 1998-2006.