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Research Article

Energy and Exergy Analysis of Ocean Compressed Air Energy Storage Concepts

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Optimal utilization of renewable energy resources needs energy storage capability in integration with the electric grid. Ocean compressed air energy storage (OCAES) can provide promising large-scale energy storage. In OCAES, energy is stored in the form of compressed air under the ocean. Underwater energy storage results in a constant-pressure storage system which has potential to show high efficiency compared to constant-volume energy storage. Various OCAES concepts, namely, diabatic, adiabatic, and isothermal OCAES, are possible based on the handling of heat in the system. These OCAES concepts are assessed using energy and exergy analysis in this paper. Roundtrip efficiency of liquid piston based OCAES is also investigated using an experimental liquid piston compressor. Further, the potential of improved efficiency of liquid piston based OCAES with use of various heat transfer enhancement techniques is investigated. Results show that adiabatic OCAES shows improved efficiency over diabatic OCAES by storing thermal exergy in thermal energy storage and isothermal OCAES shows significantly higher efficiency over adiabatic and diabatic OCAES. Liquid piston based OCAES is estimated to show roundtrip efficiency of about 45% and use of heat transfer enhancement in liquid piston has potential to improve roundtrip efficiency of liquid piston based OCAES up to 62%.

1. Introduction

Electricity generation from renewable energy sources plays important role in reducing dependence on fossil fuels and in curtailing greenhouse gas emissions. However, renewable energy sources such as wind, solar, tidal, and wave are sporadic in nature. The variability of power from the renewable energy sources makes it hard to integrate with the electric grid [1]. Utility-scale energy storage systems are needed to improve the utilization of renewable energy resources in electric grid [2]. Compressed air energy storage (CAES) system is a reliable large-scale energy storage method with relatively low specific investment cost [3]. In CAES system, intermittent energy is used to compress the atmospheric air to the high-pressure and compressed air is stored in the high-pressure reservoir. When electricity demand is high, the stored high pressurized air is expanded through the turbine to generate electricity. Two large-scale CAES plants are in operation, one is in Huntorf, Germany [4], and the other is in McIntosh, Alabama, USA [5]. In both the plants, compressed air is stored in an underground cavern. However, these underground caverns are constant-volume air storage reservoirs. In constant-volume air storage systems, charging and discharging processes result in pressure variation. These varying conditions can result in low efficiencies of compression and expansion due to deviation from the designed points [6]. This can be avoided by utilizing ocean depth for storage of the compressed air in which high-pressure environment under the water can be effectively used for creating constant-pressure storage system [7]. Such an ocean compressed air storage (OCAES) system can effectively integrate multiple energy sources located offshore with high system efficiency.

Various types of OCAES configurations are possibly based on the idealized thermodynamic process of compression and expansion of air. These thermodynamic processes are identified based on how heat is handled during compression and expansion processes. The three major types are diabatic OCAES, adiabatic OCAES, and isothermal OCAES. These types are discussed in the next section with more details. Roundtrip efficiency (also called end-to-end efficiency) is an important parameter to assess the efficacy of an energy storage system. As processes in the OCAES system are accompanied by heat and work transfer, it is very difficult to understand the efficiency of an OCAES system using first law of thermodynamics. This is because the first law of thermodynamics deals with heat and work equally. Therefore, roundtrip efficiency based on exergy analysis which is based on the second law of thermodynamics would be beneficial in better understanding the characteristics of the different types of OCAES systems.

Various studies have investigated different types of CAES configurations based on energy and exergy analysis. Kim et al. reviewed the main drawbacks of the existing CAES systems and presented energy and exergy analysis of various innovative CAES concepts [8]. They investigated concepts like adiabatic CAES, isothermal CAES, micro-CAES combined with air-cycle heating and cooling, and constant-pressure CAES combined with pumped hydrostorage. Their analysis illustrated that drawbacks of existing CAES systems can be addressed by employing innovative CAES concepts. Bagdanavicius and Jenkins investigated the potential for using heat generated during compression stage of CAES with a district energy system [9]. Exergy and exergoeconomic analysis of CAES and CAES with thermal storage were performed by them. Their analysis showed that utilization of waste heat increases energy efficiency from 48% for the CAES to almost 86% for CAES with thermal storage. They also observed that highest exergy destruction occurs in the heat exchangers during compression stage.

Different adiabatic CAES configurations were simulated and analyzed by Hartmann et al. [10] using energy balance. Polytropic CAES with one, two, and three stages and an isentropic CAES were considered. They observed that high value of 70% efficiency is only achieved for isentropic configuration and efficiency of the polytropic configuration is about 60%. Their analysis also suggested that developing high-temperature thermal storage (>600°C) and temperature resistant materials for compressors are key elements in achieving higher efficiency. Grazzini and Milazzo presented a thermodynamic analysis of multistage adiabatic CAES [11]. They proposed a comprehensive set of criteria for the design of adiabatic CAES based on a detailed thermodynamic analysis of the design parameters and influence on system efficiency, with attention to heat transfer devices. An exergy analysis was presented by Tessier et al. on adiabatic CAES system utilizing a cascade of phase change materials for waste heat storage and recovery [12]. Incorporation of phase change materials predicted to show 15% increased efficiencies of storage and recovery over the current design. An experimental study of CAES system with thermal energy storage by Wang et al. has shown a mere 22.6% roundtrip efficiency [13]. A recent detailed review on CAES has been presented in [14].

The existing CAES plants in Huntorf and McIntosh are of a diabatic type and show roundtrip efficiency of 42% and 54%, respectively. The Huntorf plant was primarily designed to provide reserve power and blackstart capability where high efficiency is of minor importance. The McIntosh plant was designed to perform load shifting on a weekly basis, which requires the cycle efficiency to be as high as possible. The McIntosh plant could achieve considerably higher efficiency over Huntorf plant using a recuperator to reduce exergy loss. The concept of adiabatic CAES using thermal energy storage (TES) was also considered during development of these plants; however, diabatic CAES is preferred at that time due to technical and economic advantages [14].

Adiabatic CAES needs a high-temperature TES withstanding the combination of thermal and mechanical stresses which requires special material and complex system engineering. Also, considerable engineering effort is needed to design an electrically driven compressor that operates at the high outlet temperature essential for adiabatic operation. Recent developments in TES have shown good prospects in achieving adiabatic CAES in practice [15]. An adiabatic CAES is under development under the project name "ADELE-ING" which could show roundtrip efficiency up to 70% [16]. Another approach in achieving high efficiency is through isothermal CAES. Technically, it was very difficult to achieve isothermal operation at high power density. However, recent developments in liquid piston compressor [17] and heat transfer enhancement techniques in the liquid piston [18–20] could result in near-isothermal compression and expansion at high power density. Thermodynamic and economic review of CAES by Rogers et al. [21] indicates that efficiencies of advanced adiabatic CAES and isothermal CAES have been increased by over 30% and energy storage densities have been improved by a factor of 5 using near-surface piping.

A multilevel underwater CAES system integrated with battery pack is proposed by Wang et al. [22]. Their thermodynamic analysis shows that roundtrip exergy efficiency of the multilevel underwater CAES varies from 62% to 81% in different working mode. Advanced exergy analysis of an underwater CAES by Wang et al. indicates exergy efficiency of 53.6% under real conditions with theoretical maximum exergy efficiency of 84.3% [23]. Clearly, there is a great potential for performance improvement of underwater CAES. A study by Cheung et al. indicates that pipe diameter, turbine, air compressor, and air storage depth have the greatest influence on system performance of underwater CAES [24]. Multiobjective optimization of an underwater CAES with objectives of maximizing roundtrip efficiency and operating profit and minimizing cost rate is performed by Cheung et al. using genetic algorithm [25]. Their analysis indicated roundtrip efficiency of 68.5% and operating profit of $53.5 per cycle for the preferred system designs.

Based on earlier studies on CAES, it can be observed that there could be a significant variation in efficiencies of various OCAES configurations. Comparative analysis of different OCAES systems would help in understanding these better. Preliminary studies of exergy analysis for various OCAES configurations and end-to-end efficiency of liquid piston based OCAES have been presented earlier in [26, 27]. In this paper, detailed analysis of various OCAES concepts using energy and exergy analysis is presented. This would help in assessing improvement areas in achieving higher roundtrip efficiency with OCAES.

Investment costs associated with different types of OCAES would also vary as the technology used in these configurations differs considerably. The compressor and expander in the diabatic OCAES are mature technologies whereas the same in the adiabatic and isothermal OCAES are still in the development stages. Also, the cost of thermal energy storage (TES) used in adiabatic OCAES differs significantly based on the kind of TES system used. Broadly, there are three types of TES systems, sensible heat TES, latent heat TES, and thermochemical TES. In general, latent heat TES and thermochemical TES are more expensive than sensible heat TES; however, former could be economically viable with a high number of operating cycles [15]. Although the cost of different OCAES configurations might differ significantly, this paper only focuses on efficiency aspect without accounting any cost difference. However, an economic assessment of these OCAES configurations would be necessary before investment decisions. This study could provide a framework for an economic assessment of different OCAES by monetizing efficiencies incorporating cost difference.

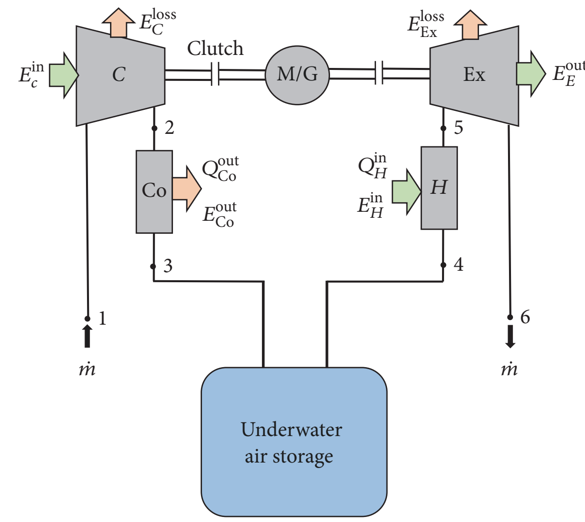
2. OCAES Configurations

OCAES configurations can be broadly distinguished depending on the targeted idealized process of compression and expansion of air. It is decided based on how heat is handled during compression and prior to expansion of the air. Three major OCAES configurations would be diabatic OCAES, adiabatic OCAES, and isothermal OCAES. These system configurations are discussed in this section.

2.1. Diabatic OCAES. In the diabatic OCAES system, in energy storage mode, air is compressed using conventional compressors and cooled to the surrounding temperature before sending it to the storage system. Compression process increases the temperature of air due to the heat of compression which is dissipated before sending air to the storage device. This results in loss of thermal energy of compression due to cooling. In energy recovery mode, the air from the storage is heated using fuel and then passed through the expander to generate electricity.

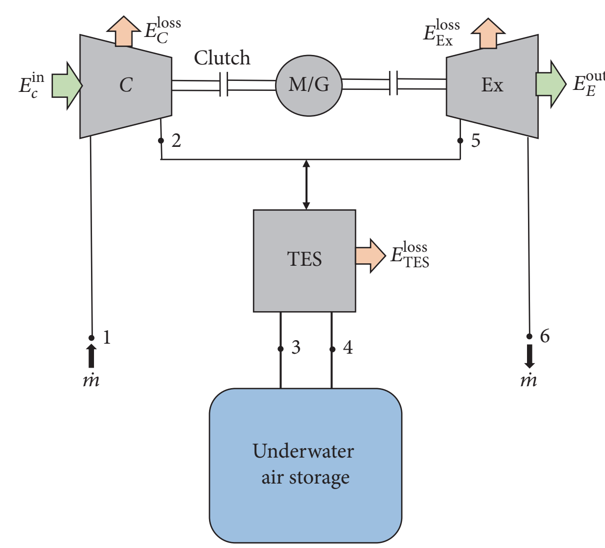
The schematic of diabatic OCAES is shown in Figure 1. Processes 1-2 represent compression of atmospheric air using air compressor run using electric motor operating on excess electric energy. The motor efficiency and losses in the compressor would result in loss of energy/exergy during this process. Compressed air from the compressor is then passed through the cooler (processes 2-3) before sending it to the underwater air storage system. A significant amount of heat energy (thermal exergy) is lost in the cooler. Processes 3-4 indicate charging and discharging from the air storage system. Mechanical exergy (energy in pressure form) in the high-pressure air is stored in the underground storage. There can be a small amount of loss of energy/exergy due to leakage and pressure drop in the storage system. In process 4-5, high-pressure air is heated using external heat input (thermal exergy) to increase expansion work. The processes 5-6 are the expansion process in which mechanical exergy in the form of electrical energy is delivered. However, processes 5-6 involve loss of energy/exergy due to inefficiencies in expansion process and also loss of energy/exergy from the exhaust gas.

2.2. Adiabatic OCAES. The construction of adiabatic system would require a compressor design delivering higher



C: compressor
Co: cooler
H: heater
Ex: expander
M/G: motor/generator
E: exergy
Q: heat transfer

FIGURE 1: Schematic of diabatic OCAES.



C: compressor
TES: thermal energy storage
Ex: expander
M/G: motor/generator
E: exergy

FIGURE 2: Schematic of adiabatic OCAES.

temperature at the outlet of compression, a reliable TES system to store thermal energy at high temperature and an expander design operating at high inlet air temperature with broad operation range. These components installed and connected with pipelines carrying air with the layout shown in Figure 2 would result in the adiabatic OCAES system.

It can be seen that cooler and heater in the diabatic OCAES are replaced with thermal energy storage (TES) in the adiabatic OCAES configuration. The diabatic OCAES uses fuel in the heater; therefore, it cannot be considered as a pure storage system and is actually a combination of storage and

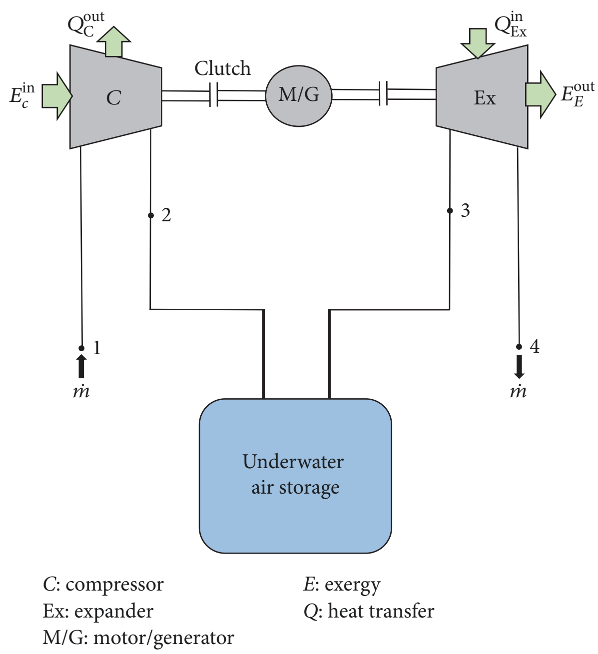


Figure 3: Schematic of isothermal OCAES.

power plant. This can be overcome in adiabatic OCAES which uses TES to store heat from the compressed air before sending it to the air storage. The stored heat in TES is used to heat the air before passing it to the expander. This eliminates the need for cooler and heater in the adiabatic system as TES works as both. Therefore, thermal energy and thermal exergy losses in the cooler and thermal heat input in the heater of diabatic OCAES are completely eliminated in the adiabatic OCAES. However, TES involves some energy/exergy losses which result in added inefficiencies in adiabatic OCAES.

2.3. Isothermal OCAES. Isothermal OCAES would eliminate the need for fuel and high-temperature thermal energy storage. This can be done by isothermal compression and expansion process which minimizes compression work and maximizes expansion work. Figure 3 shows a schematic of isothermal OCAES system. The compressor in the isothermal OCAES dissipates heat energy during compression process resulting in the conversion of electrical energy into mechanical exergy form in the compressed air. Ideal isothermal compression does not add any thermal exergy in the compressed air; therefore, loss of exergy is totally avoided. Similarly, in the ideal isothermal expansion, mechanical exergy from the compressed air is completely converted into electrical energy.

It is very difficult to achieve isothermal compression and expansion using conventional compressors and expansion devices as conventional compressors and expanders work at high speed with the nearly adiabatic process. Special types of compressors and expanders are required to achieve isothermal OCAES in reality. Liquid piston compressor can be used to achieve near-isothermal compression and expansion operation. In the liquid piston compressor, a column of liquid (usually water) is utilized to compress a gas in the fixed volume chamber. A hydraulic pump is used to generate the flow of the liquid for the liquid pistons. The liquid flow in and out of compression chamber is controlled with valves. As a liquid can conform to an irregular chamber volume, the surface area to volume ratio in the gas chamber can be maximized using a liquid piston. This results in increasing the heat transfer during the gas compression/expansion which facilitates near-isothermal operation [17]. Various heat transfer enhancement techniques like the use of porous media inserts [19], spray cooling [18], and use of hollow spheres [20] can be effectively used in the liquid piston to achieve near-isothermal operation.

A typical liquid piston based isothermal OCAES would have electric motor/generator, hydraulic pump/motor, liquid piston compressor/expander, air cooler/heater, pipelines connecting various components, control valves, and underwater air storage. Figure 4 shows a schematic of a liquid piston based OCAES system. Although liquid piston compression is efficient compared to the existing compressors technologies, the added components like hydraulic pump/motor and hydraulic lines have some inefficiencies which would affect the overall efficiency of OCAES system.

3. Energy and Exergy Analysis

Inefficiencies in various components of the OCAES contribute to the loss of energy in the storage system. Energy efficiency of a component is the ratio of energy out from the component to energy into the component. In energy analysis, energy efficiencies of various components in the OCAES are modeled and used to evaluate the energy efficiency of overall OCAES system.

The exergy transfer to the system can happen by work, heat, and mass transfer. The exergy transfer by heat is given by

$$\dot{E}_q = \int 1 - \frac{T_0}{T}\delta\dot{Q}, \quad (1)$$

where T is temperature, $\dot{Q}$ is heat transfer rate, and subscript 0 indicates properties at environmental conditions [28].

Exergy transfer by mass flow ($\dot{m}$) is given by

$$\dot{E}_m = \dot{m}e, \quad (2)$$

where specific exergy (e) of an ideal gas is given by

$$e = C_p\left(T - T_0\right) - T_0\left[C_p \ln\left(\frac{T}{T_0}\right) - R\ln\left(\frac{P}{P_0}\right)\right], \quad (3)$$

where C_p is specific heat at constant pressure, R is the specific gas constant, and P is pressure.

The above specific exergy consists of two parts, mechanical exergy and thermal exergy. The mechanical exergy is associated with the system pressure and is the exergy change when the system is brought to the state $[T_0, P_0]$ from the state $[T_0, P]$. The thermal exergy is associated with the system temperature and is the exergy change when the system is

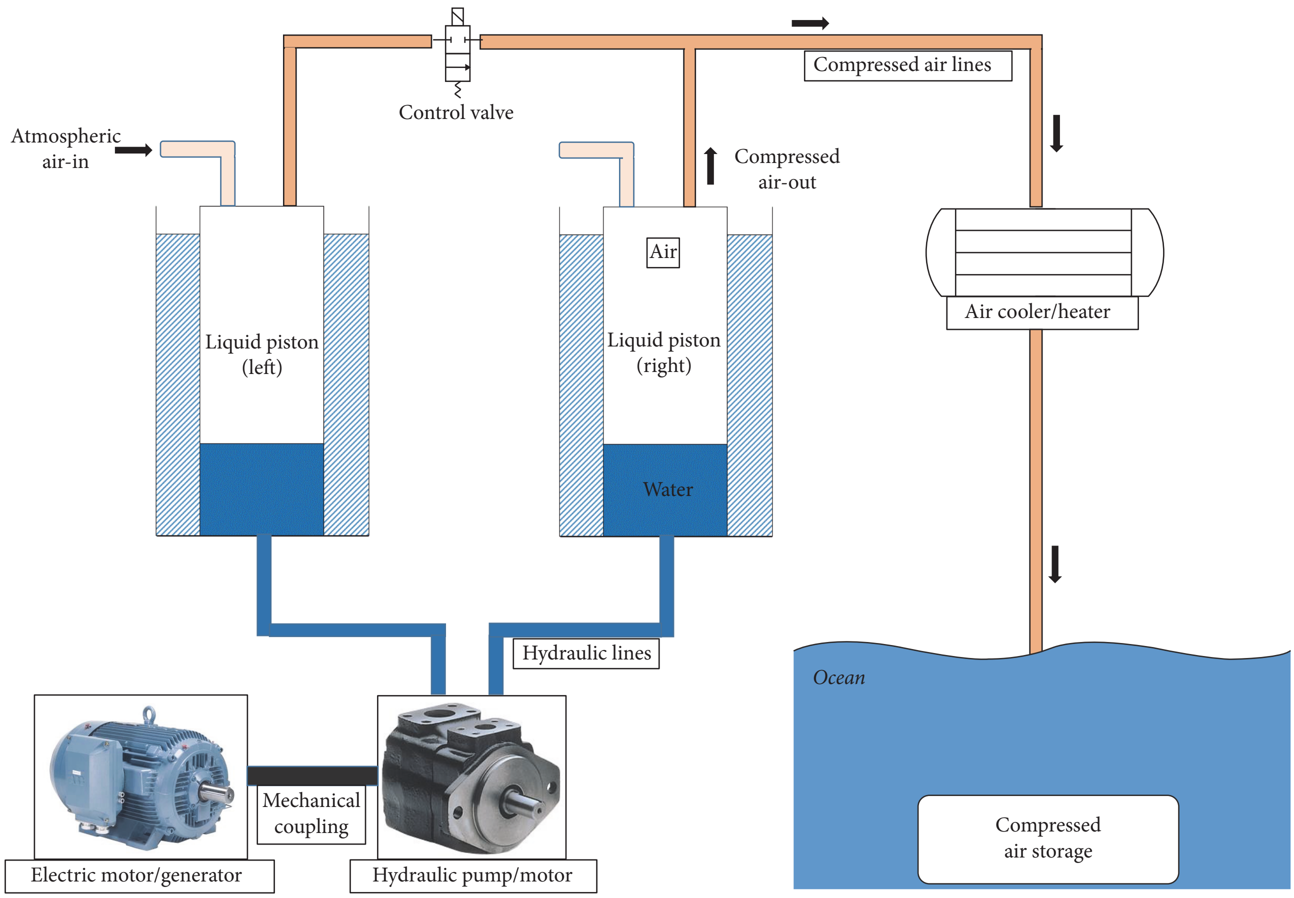


FIGURE 4: Schematic of a liquid piston based isothermal OCAES system.

brought to the state $[T_0, P]$ from the state $[T, P]$. Mechanical and thermal exergies are given by (4) and (5), respectively.

$$e^M = RT_0 \ln\left(\frac{P}{P_0}\right), \tag{4}$$

$$e^T = C_p\left(T - T_0 - T_0 \ln\left(\frac{T}{T_0}\right)\right), \tag{5}$$

where superscripts M and T indicate mechanical and thermal parts, respectively.

Individual components in the OCAES system can be analyzed based on exergy analysis. Exergy efficiency of a component is given by

$$\varepsilon_{\text{com}} = \frac{\dot{E}^-_{\text{com}}}{\dot{E}^+_{\text{com}}}, \tag{6}$$

where ε denotes exergy efficiency and subscripts indicates component.

Energy efficiency and exergy efficiency of individual components in OCAES are discussed below.

3.1. Electric Motor and Generator. Electric motor/generator has mechanical and electrical losses during its operation. Energy efficiency of electric motor/generator (M/G) is the ratio of power output from M/G to the power input to M/G as calculated using

$$\eta_{\text{M/G}} = \frac{P_n \times \text{Load}}{P_i}. \tag{7}$$

As electric motor/generator deals with work transfer (electrical energy and shaft work) only, exergy efficiency of electric motor/generator is its energy efficiency.

3.2. Hydraulic Pump/Motor. Energy efficiency of hydraulic pump/motor depends on its displacement, speed of operation, and the pressure differential between inlet and outlet. The overall efficiency of a hydraulic pump for a particular power input is given by

$$\eta_{\text{HP}} = \frac{D \times N \times \Delta p}{P_{\text{in}}}. \tag{8}$$

Exergy efficiency of hydraulic pump/motor is also same as its energy efficiency because it deals only with work transfer.

3.3. Compressor. The exergy transfer to the compressor ($\dot{E}^+_C$) is in the form of shaft work from the motor whereas exergy transfer from the compressor ($\dot{E}_C^-$) is due to air mass transfer from the compressor at high pressure and temperature. There are exergy losses in the compressor due to mechanical losses. Isentropic efficiency [29] and mechanical efficiency of the compressor can be used to evaluate exergy efficiency of compressor.

In case of isothermal OCAES, liquid piston compressor efficiency is defined as the ratio of stored energy to work input. Storage energy is the amount of work extracted from the isothermal expansion of compressed air to the atmospheric pressure. Work input consists of compression work, cooling work and friction work. Atmospheric air compressed to high-pressure results in increasing temperature. This compressed air is cooled to initial temperature to maintain storage pressure [30]. This adds cooling work. In case of liquid pistons, friction work is comparatively small (unless diameter is too small) and can be neglected [17]. Liquid piston compressor efficiency (η_C) after neglecting viscous friction is given by

$$\eta_C = \frac{\overbrace{\ln(P_r) + 1/P_r - 1}^{E_{\text{Storage}}}}{\underbrace{\left(P_r^{(n-1)/n} - 1\right)/(n-1) + P_r^{-1/n} - 1}_{W_{\text{compression}}} + \underbrace{(P_r - 1)\left(P_r^{-1/n} - 1/P_r\right)}_{W_{\text{Cooling}}}}, \tag{9}$$

where P_r is the pressure ratio (ratio of storage pressure to the atmospheric pressure) and n is a polytropic index of compression. Storage pressure (hence P_r) depends on the underwater air storage depth and n depends on the magnitude of heat transfer in liquid piston compressor.

3.4. Cooler. The compressed air from the compressor contains both mechanical and thermal exergy. In the cooler, the compressed air is cooled to the atmospheric temperature at a constant pressure by dissipating thermal exergy of the compressed air to cooling media. The output compressed air from the cooler would contain only mechanical exergy. Therefore, exergy efficiency of the cooler neglecting pressure losses in the cooler is given by

$$\begin{aligned}\varepsilon_{\text{co}} &= \frac{\dot{m}e^M}{\dot{m}\left(e^M + e^T\right)} \\ &= \frac{RT_0 \ln\left(P_{\text{co}}/P_0\right)}{RT_0 \ln\left(P_{\text{co}}/P_0\right) + C_p\left(T_{\text{co}} - T_0 - T_0 \ln\left(T_{\text{co}}/T_0\right)\right)},\end{aligned} \tag{10}$$

where P_{co} and T_{co} are pressure and temperature of compressed air at inlet of the cooler, respectively.

3.5. Air Pipeline Connecting Various Components. Energy/exergy loss inside the pipeline carrying air is equal to product of pressure drop and flow rate. Pressure drop for steady, fully developed, incompressible flow in the pipe can be calculated using Darcy –Weisbach equation [31].

3.6. Thermal Energy Storage (TES). In adiabatic OCAES, high-temperature high-pressure compressed air is passed through the TES to store thermal exergy of compressed air in the TES. This stored thermal energy is used to increase thermal exergy of compressed air before sending it through the expander. The thermal and pressure losses in the TES result in the loss of energy/exergy. The exergy efficiency of TES is given by

$$\varepsilon_{\text{TES}} = \frac{RT_0 \ln\left(P_{\text{TES}}^{\text{out}}/P_0\right)}{RT_0 \ln\left(P_{\text{TES}}^{\text{in}}/P_0\right) + C_p\left(T_{\text{TES}}^{\text{in}} - T_0 - T_0 \ln\left(T_{\text{TES}}^{\text{in}}/T_0\right)\right)}, \tag{11}$$

where $P_{\text{TES}}^{\text{in}}$ and $T_{\text{TES}}^{\text{in}}$ are pressure and temperature of compressed air at inlet of the TES. $P_{\text{TES}}^{\text{out}}$ and $T_{\text{TES}}^{\text{out}}$ are pressure and temperature of compressed air at outlet of the TES, respectively.

3.7. Air Storage. Leakage and pressure losses in the underwater air storage system result in energy/exergy losses. Leakage per unit volume per unit time can be calculated by measuring pressure drop in an isolated air storage system given by equation [32]

$$\dot{L}_S = \frac{\rho_s}{t_e} \frac{(\Delta P)_S}{P_S}, \tag{12}$$

where $\dot{L}_S$ is the leakage rate (kg/hr·m^3), ρ_s is the density of air at storage pressure and temperature, $(\Delta P)S$ is the pressure drop in the isolated air storage system in time t_e, and P_S is the storage pressure.

The energy/exergy efficiency of an air storage system is given by [27]

$$\varepsilon_S = 1 - \frac{t_0 \times \dot{L}_S}{\rho_s}, \tag{13}$$

where t_o is the operation time of the OCAES.

3.8. Heater. In the diabatic OCAES, external fuel is used to heat the air. The heat (thermal exergy) added in the heater increases exergy of the air. Exergy efficiency of a heater neglecting pressure losses in the heater and considering the constant rate of heat transfer is given by

$$\varepsilon_H = \frac{C_p\left(T_H - T_0 - T_0 \ln\left(T_H/T_0\right)\right) + RT_0 \ln\left(P_{H,\text{out}}/P_0\right)}{Q_s\left(1 - T_0/T_s\right) + RT_0 \ln\left(P_{H,\text{in}}/P_0\right)}, \tag{14}$$

where $P_{H,\text{in}}$ and $P_{H,\text{out}}$ are pressures at inlet and outlet of the heater, respectively, T_H is the temperature of air at heater output, T_S is the temperature of heat source, and Q_S is the heat transfer per unit mass of air.

3.9. Expander. The exergy transfer to the expander ($\dot{E}_{\text{Ex}}^+$) is by compressed air inlet whereas exergy transfer from the expander ($\dot{E}_{\text{Ex}}^-$) is in the form of shaft work delivered to the generator. Similar to compressor, exergy efficiency of

Table 1: Stochastic assignments in Monte Carlo simulation.

Variable	Mean/max value [μ] (%)	Standard deviation[a] or max/min value (%)	Distribution
$\varepsilon_{\mathrm{M/G}}$ [34]	96	0.5	Normal
$\varepsilon_{\mathrm{HP/HM}}$ [8]	93	1	Normal
$\eta_{\mathrm{C,isen}}$	85	2	Normal
$\eta_{\mathrm{C,mech}}$	95	1	Normal
η_P	$(P_{\mathrm{in}} - \Delta P)/P_{\mathrm{in}}$	Max = μ, Min = $\mu - 0.5$	Triangular
$\varepsilon_{\mathrm{TES}}$	80	2	Normal
ε_S	Using (13)	Max = $\mu + 0.5$ Min = $\mu - 0.5$	Triangular
ε_H	95	1	Normal
$\eta_{\mathrm{Ex,isen}}$	85	1	Normal
$\eta_{\mathrm{Ex,mech}}$	95	1	Normal

[a]For normal distribution.

expander can be calculated using isentropic efficiency [29] and mechanical efficiency of the expander.

In case of isothermal OCAES using a liquid piston, liquid piston expansion efficiency (η_E) for polytropic expansion index n is given by

$$\eta_E = \frac{\overbrace{\left(1 - (1/P_r)^{(n-1)/n}\right)/(n-1) - (1/P_r)^{(n-1)/n} + 1/P_r}^{W_{\mathrm{Expansion}}}}{\underbrace{\ln(P_r) + 1/P_r - 1}_{E_{\mathrm{Storage}}}}. \quad (15)$$

Table 2: Thermodynamic properties and mass flow rates of air at different state points for diabatic OCAES system (see Figure 1 for state points).

State point	Pressure (kPa)	Temperature (K)	Mass flow rate (kg/s)
1	101.3	293	1.33
2	1119.6	633	1.33
3	1114.6	293	1.33
4	1109.6	293	1.32
5	4200.0	823	1.32
6	101.3	364	1.32

4. Numerical Simulations

Different types of OCAES systems are modeled based on energy and exergy analysis of individual components in the OCAES system. Storage pressure of 10 bar gauge (100 m ocean depth) is considered for analysis. Various components specifications designed for maximum power capacity of 0.5 MW with 2 MWh energy storage were used [33]. Efficiencies of motor/generator and hydraulic pump/motor are considered from the industry standards [8, 34]. Pipelines connecting cooler to air storage and air storage to the heater are considered with 1000 m in length, 0.2 m in diameter, and of 15 μm surface roughness [35]. For air storage, the leakage rate of 0.01 kg/hr·m^3 [36] and operation time of 8 hours were assumed [37].

Uncertainty analysis is performed using Monte-Carlo Simulations (100000 runs) to estimate mean and confidence interval values of energy efficiency and exergy efficiency. Stochastic assignments considered are given in Table 1.

All the simulations were performed considering 1 atm and 20°C environmental conditions. In all the configurations, single stage compression and single stage expansion were considered. Adiabatic and isothermal OCAES systems are considered without the use of fuel. In the diabatic configuration, heat source temperature of 1500°C is considered. The amount of heat transfer from the heat source is evaluated to achieve inlet conditions to expander with 42 bar and 550°C. These values are referred from HP turbine operating conditions of Huntorf plant [38]. In adiabatic OCAES configuration, TES storage temperature of 327°C is considered [39] and inlet air conditions to the expander of 10 bar and 327°C are considered. In isothermal OCAES configuration, liquid piston based compression and expansion are considered.

To simplify the thermodynamic model, all the analysis is performed for steady state operation of the system. Also, the kinetic and potential energy of the fluids are assumed to be negligible and the air is considered as an ideal gas [23]. For these system considerations, the thermodynamic properties and mass flow rate of air at different state points of the system are evaluated. Those are listed in Tables 2, 3, and 4 for diabatic, adiabatic, and isothermal OCAES, respectively. Finally, different types of OCAES systems are compared using energy/exergy flow, energy efficiency, and exergy efficiency evaluations.

5. Results and Discussion

Figures 5, 6, and 7 show exergy flow in various OCAES configurations. In figures, each box represents a component in the OCAES system. The height of the box represents qualitative exergy flow to the component. The arrows indicate the exergy flow direction and the amount of exergy flow is

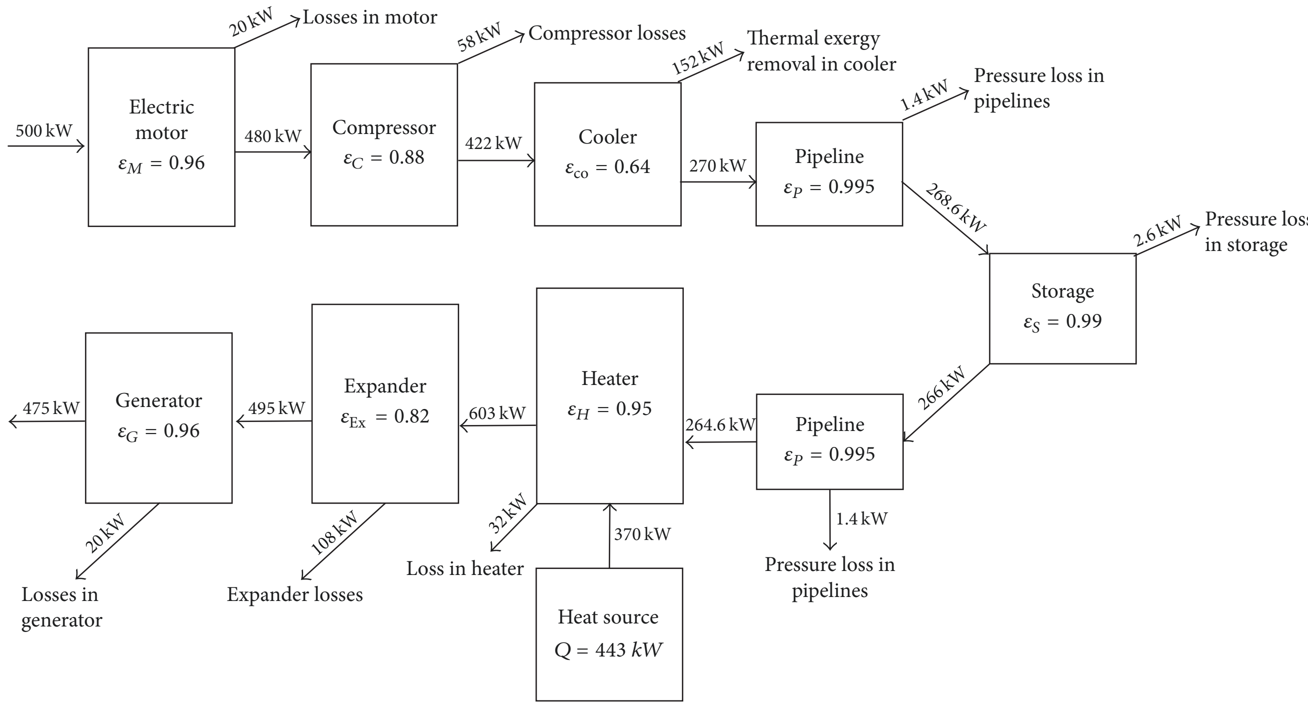


Figure 5: Exergy flow in diabatic OCAES.

Table 3: Thermodynamic properties and mass flow rates of air at different state points for adiabatic OCAES system (see Figure 2 for state points).

State point	Pressure (kPa)	Temperature (K)	Mass flow rate (kg/s)
1	101.3	293	1.33
2	1119.6	633	1.33
3	1114.6	293	1.33
4	1109.6	293	1.32
5	1104.6	600	1.32
6	101.3	354	1.32

Table 4: Thermodynamic properties and mass flow rates of air at different state points for isothermal OCAES system (see Figure 3 for state points).

State point	Pressure (kPa)	Temperature (K)	Mass flow rate (kg/s)
1	101.3	293	2.10
2	1119.6	300	2.10
3	1114.6	293	2.08
4	101.3	286	2.08

mentioned on each arrow. The arrows pointing away from the boxes indicate exergy loss to the environment. The exergy efficiencies of the individual components are mentioned in the boxes. The input power of 500 kW is considered for the analysis.

5.1. Diabatic OCAES. Exergy flow in the diabatic OCAES is presented in Figure 5. Inefficiencies in electric motor result in exergy loss of 20 kW. It is observed that a significant amount of exergy is lost from the cooler to the surrounding. Losses in the pipelines and storage are very small compared to other losses. The addition of heat energy to the heater from the heat source adds exergy to the system. Although energy input from the heat source to the heater is 443 kW, the exergy value of this heat is only about 370 kW. This addition of exergy in heater compensates the loss of exergy in the cooler to achieve the same level of power output. The expander is the next exergy inefficient component after the cooler in the diabatic system. Inlet air to the expander contains exergy in both thermal and mechanical forms. Irreversibility in expansion process results in exhaust air with a significant amount of thermal exergy, which results in higher exergy loss in the expander. The recuperator is not considered in the current configuration. In recuperative process waste heat from the exhaust of the expander can be used to heat expander inlet air. This would reduce exergy burden from the heat source and hence improve overall exergy efficiency of OCAES system.

The overall exergy efficiency of diabatic OCAES is 55% whereas energy efficiency is 50%. As diabatic OCAES has fuel input in the heater system, it is not a pure energy storage system but the combination of energy storage and power plant. Therefore, exergy efficiency is a good measure of diabatic OCAES for comparison with other systems.

5.2. Adiabatic OCAES. The thermal exergy lost from the cooler in the diabatic OCAES is stored in the adiabatic OCAES using TES which can improve exergy efficiency of

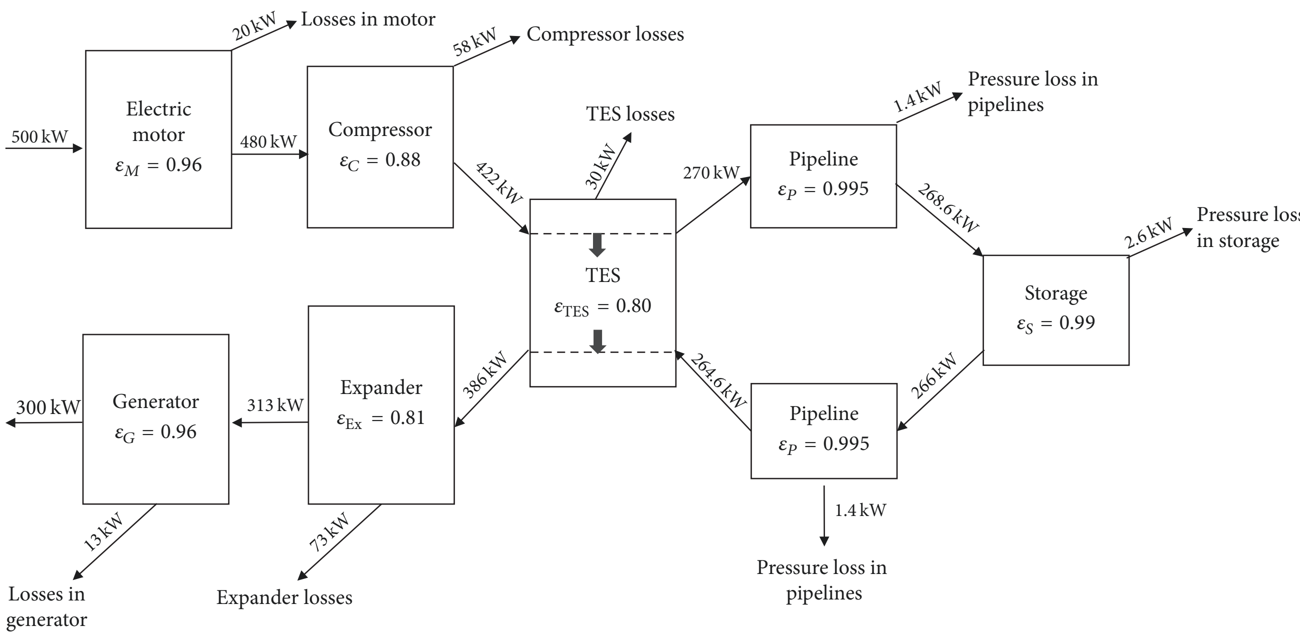


Figure 6: Exergy flow in adiabatic OCAES.

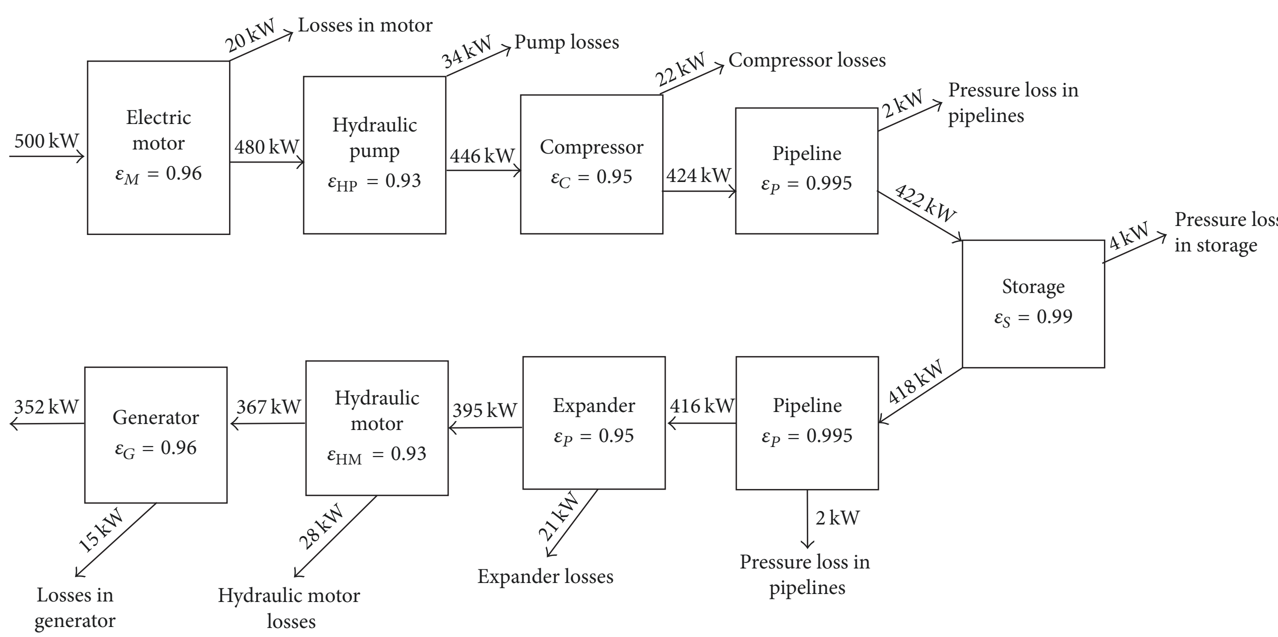


Figure 7: Exergy flow in isothermal OCAES.

adiabatic OCAES significantly over diabatic OCAES. Figure 6 shows exergy flow in adiabatic OCAES. Exergy loss in the cooler of diabatic OCAES can be avoided by use of TES which stores a significant amount of thermal energy. TES supplies heat to the air before the expander thus increasing the exergy potential of air. With the use of TES, the external heat source is removed in the adiabatic OCAES configuration. However, inefficiencies in TES account for exergy loss which contribute to reduction energy output. Comparison of diabatic and adiabatic exergy flow reveals that adiabatic CAES gives less exergy output (electric energy) from the generator. This is because the fuel source used in the diabatic OCAES allows the expander to be operated with higher power capacity.

Overall exergy efficiency of adiabatic OCAES is 60% which is 5% higher than that of diabatic OCAES. This improvement is due to reuse of thermal exergy of the air using TES. Improvement in TES efficiency from current consideration of 80% would further improve the efficiency of adiabatic OCAES. As external fuel is not used in adiabatic OCAES, overall energy efficiency of adiabatic OCAES is same

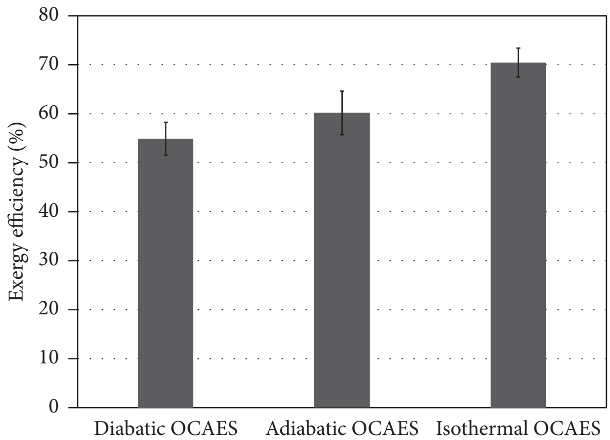


Figure 8: Overall exergy efficiency of different types of OCAES.

as overall exergy efficiency. Careful observation of exergy losses in adiabatic OCAES shows that losses in compressor and expander are major contributors of inefficiencies in adiabatic OCAES.

5.3. Isothermal OCAES. The need of TES and fuel source is eliminated in the isothermal OCAES. Exergy flow in the isothermal OCAES is shown in Figure 7. Liquid piston based compressor and expander requires a hydraulic pump and a hydraulic motor as added components in the isothermal OCAES. Inefficiencies of these components would contribute to the exergy loss in the system. It can be observed in Figure 7 that hydraulic pump/motor show high exergy losses in comparison with losses in other components in isothermal OCAES. However, the use of liquid piston in conjunction with hydraulic pump and motor has the potential to improve efficiencies of the compressor and expander significantly. This results in a reduction of exergy losses in compressor and expander. Also, the absence of TES and heater eliminates losses associated with those which helps in improving the overall efficiency of the system. Overall exergy efficiency of the isothermal OCAES is about 70% which is significantly higher than diabatic and adiabatic OCAES.

The overall exergy efficiencies of all the three configurations with 95% confidence interval bounds are shown in Figure 8 for comparison. The uncertainties in various assumptions show about 3-4% variation in exergy efficiencies. Clearly, isothermal OCAES is the most efficient and diabatic OCAES is the least efficient among three configurations considered based on exergy analysis. Energy efficiency of diabatic OCAES is about 50% whereas that of adiabatic and isothermal OCAES is same as their exergy efficiencies. Energy efficiency might not be a reliable comparative parameter as it would undervalue the efficiency of the diabatic system.

5.4. Liquid Piston Based OCAES. Although isothermal OCAES shows high exergy efficiency, such a high level of efficiency is contingent upon near-isothermal compression and expansion. Liquid piston compressor is experimentally tested to investigate its effectiveness in achieving near-isothermal

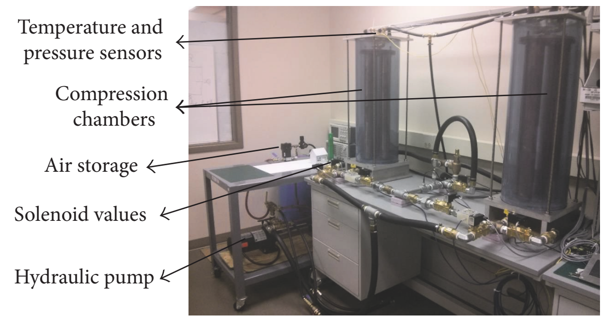


Figure 9: Experimental setup of liquid piston compressor.

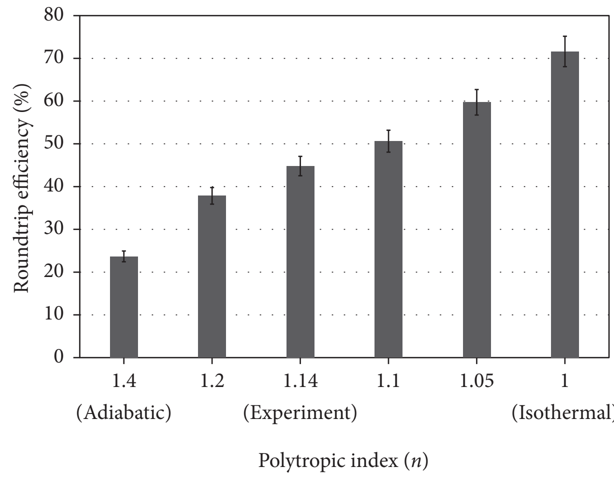


Figure 10: Roundtrip efficiency of liquid piston based OCAES for different polytropic indices.

compression. Figure 9 shows the experimental setup of a liquid piston compressor. The compression chamber was divided into four parallel copper pipes, each having an inner diameter of 76 mm and length of 760 mm. Two compression chambers were used in the experimental setup to ensure continuous production of compressed air. The compression chambers were enclosed in a plastic cylinder filled with water to maintain the temperature of the outer copper wall a constant. A hydraulic pump was used to alternatively drive water from one compression chamber to the other. The pump delivers a constant flow rate of 10 gpm at a maximum pressure of 13.1 bar gauge (190 psi). A K-type thermocouple of diameter 0.0508 mm (0.002 in) and a pressure sensor were installed at the top of each compression chamber as indicated in Figure 9. Experiments were performed with a pressure ratio of 6 and the stroke time of 10 s.

The liquid piston compressor and expansion efficiency can be calculated using the polytropic index of compression and expansion in (9) and (15), respectively. The polytropic index is calculated from the P-T curve (pressure-temperature curve) considering $P^{(1-n)}T^{n}$ = constant relation.

Roundtrip efficiencies of liquid piston based OCAES system with various polytropic indexes of compression/expansion are shown in Figure 10. Uncertainty bars represent

95% confidence interval valves. It can be observed that estimated mean value of end-to-end efficiency increases from 24% to 72% with a decrease in the polytropic index of compression from 1.4 (adiabatic process) to 1 (isothermal process). This clearly indicates liquid piston compression and expansion efficiency has a major influence on the end-to-end efficiency. For a polytropic index of 1.14 observed with an experimental liquid piston, a roundtrip efficiency of 45% was shown. Noticeably, this efficiency value is way below efficiency level of isothermal OCAES. However, the liquid piston setup used in the experimental investigation did not involve any heat transfer enhancement mechanism to abate temperature rise. Various designs of liquid pistons leading to a lower polytropic index of compression/expansion would increase roundtrip efficiency for the OCAES system.

5.5. OCAES with Heat Transfer Enhancement in Liquid Piston. Various heat transfer enhancement techniques in liquid piston have been observed to be effective in improving efficiency and power density. Various heat transfer enhancement techniques considered are use of optimal trajectories, use of hollow spheres, spray cooling, and use of porous media. The effectiveness of these heat transfer enhancement techniques in achieving an efficiency level of isothermal OCAES is further investigated. Heat transfer enhancement techniques considered in this analysis are discussed below.

5.5.1. Optimal Trajectories. The Pareto optimal trajectories for the liquid piston compressor/expander that maximizes efficiency for a given power have been found by Saadat et al. [40]. They considered general heat transfer models, the viscous friction, and system constraints in the optimization process. These optimal trajectories were experimentally tested by Shirazi et al. [41]. It was observed that optimal profiles show up to 4% higher efficiency for the same power density or 30% higher power density for the same the efficiency compared to ad hoc constant flow rate profiles.

5.5.2. Use of Hollow Spheres. Hollow spheres floating at the liquid-air interface in the liquid piston have been observed to be effective in reducing the temperature of the compressed air in the liquid piston compressor. Hollow spheres made of Silicon Carbide (SiC), High-Density Polyethylene (HDPE), and Polypropylene (PP) were tested by Ramakrishnan et al. in a liquid piston compressor [20]. It was observed that those hollow spheres are effective in bringing down the temperature of the compressed gas and hence enhance heat transfer in the system. The polytropic index of compression with use of hollow spheres reduces to 1.08 from 1.15 which corresponds to increase in compression efficiency from 79% to 87% for a compression ratio of 10.

5.5.3. Spray Cooling. In the spray cooling concept, small water droplets and high mass loading create a large interfacial surface area for heat transfer. The droplet spray heat transfer in the liquid piston compressor has been investigated by Qin and Loth [42]. They developed a detailed multiphase thermodynamic model and validated with experimental data.

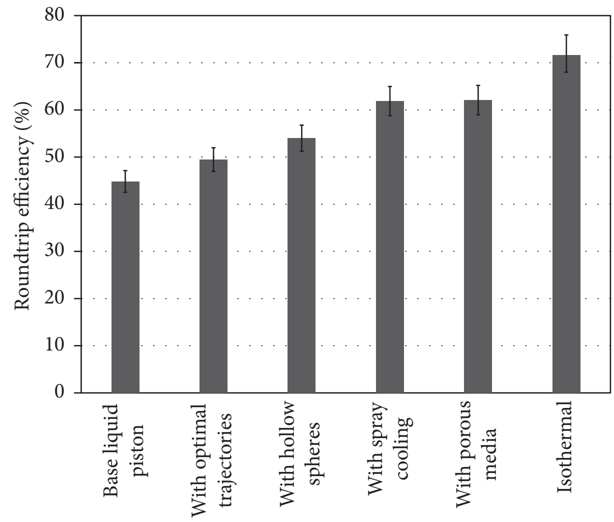


Figure 11: Roundtrip efficiency of liquid piston based OCAES with various heat transfer enhancement techniques.

It was observed that the total surface area of aloft droplets is critical to achieving high performance in a liquid piston and the best can be achieved with small droplets and high mass loading combined with direct injection. It is shown that compression efficiency could be increased from 71% for adiabatic compression to as high as 98% with spray injection for a compression ratio of 10. The mass loading, droplet diameters, and direct versus premixed injection influence effectiveness of spray cooling and hence compression efficiency. For a compression ratio of 10 and mass loading of 1, the compression efficiency of 93% is shown with spray cooling in the liquid piston.

5.5.4. Porous Media. The porous media inserts increase heat transfer surface area significantly; hence their addition to the liquid piston compression/expander increases the compression efficiency at a fixed power density. Experimental investigations on heat transfer with porous media in a liquid piston during compression and expansion have been carried out by Yan et al. [19]. A baseline case without inserts and five cases with different porous inserts were tested in a compression experiment. It was found that, in compression, porous inserts increase power density by 39-fold at 95% efficiency and increase efficiency by 18% at 100 kW/m^3. Similarly, in the expansion, porous media inserts increase power density threefold at 89% efficiency and increase efficiency by 7% at 150 kW/m^3.

The end-to-end efficiency of liquid piston based OCAES system with various heat transfer enhancement techniques in liquid piston is shown in Figure 11. The base liquid piston without any heat transfer enhancement technique considers experimental observed liquid piston compressor/expander efficiency. The isothermal OCAES is one in which compression and expansion happen isothermally which indicates 100% compression and expansion efficiency. Uncertainty bars represent 95% confidence interval valves.

It can be observed that estimated mean value of end-to-end efficiency is consistently higher with the use of heat transfer enhancement technique over the base liquid piston. The optimal trajectories can improve OCAES efficiency by about 5%. Although this improvement is comparatively a small value, this improvement happens without introducing any other media in the liquid piston. Therefore, optimal trajectory technique can be considered as an efficiency improvement technique without affecting other performance of the liquid piston like volumetric efficiency, power density, and long-term reliability. The use of hollow spheres has potential to improve the efficiency of OCAES system by about 9%. The addition of a layer of floating spheres in liquid piston increases heat transfer area and helps in temperature abatement in the liquid piston during compression. This results in improved efficiency of compression and similarly for expansion.

Further, spray cooling and porous media inserts can show about 17% improvement in the OCAES efficiency achieving end-to-end efficiency of about 62%. Both of these methods increase heat transfer medium inside the liquid piston and hence help to curtail the temperature rise in the liquid piston during compression and temperature fall during expansion. The high surface area and high specific heat of the water spray help in heat transfer enhancement and hence improving compression/expansion efficiency of liquid piston. In porous media inserts, higher heat transfer surface area in porous media helps in absorbing the thermal energy from the air during compression and dissipates it to the liquid. In expansion mode, the same porous media transfers thermal energy from the liquid to the air. Although spray cooling and porous media improves the efficiency of OCAES significantly, these heat transfer enhancement techniques require other media to be introduced in the liquid piston. This results in a reduced volumetric efficiency of compression/expansion. Also, spray cooling and porous media inserts might require special care for continuous reliable heat transfer performance. This is because of the change in spray characteristics and porous media characteristics over time due to their degradation.

An isothermal liquid piston compressor would define an upper limit for roundtrip efficiency of OCAES, which is about 72% for the given system considerations. This indicates that inefficiencies in the motor/generator, hydraulic pump/motor, pipelines, control valves, and storage result in about 28% of energy loss. End-to-end efficiencies of existing compressed air energy storage (CAES) plants in Huntorf (Germany) and McIntosh AL (USA) are 42% and 54%, respectively [14]. Clearly, liquid piston based OCAES with the use of heat transfer enhancement technique such as spray cooling or porous media inserts in liquid piston can show significantly higher end-to-end efficiency over existing CAES plants.

6. Conclusions

In pursuit of developing efficient economical large-scale energy storage, ocean compressed air energy storage can play an important role. Various OCAES concepts are possible, namely, diabatic, adiabatic, and isothermal OCAES. Energy and exergy analysis of these concepts is performed for OCAES system of the maximum power capacity of 0.5 MW and 2 MWh energy storage with storage pressure of 10 bar (100 m of ocean depth). Analytical models for energy and exergy analysis of various components in OCAES are presented. The exergy flow, energy efficiency, and exergy efficiency of various OCAES concepts are analyzed for comparative assessment.

The analysis shows that energy efficiency of diabatic OCAES is about 50% whereas its exergy efficiency is about 55%. Clearly, energy efficiency undervalues efficiency of diabatic OCAES; therefore, exergy efficiency would be a good measure of efficiency for comparison with other storage concepts. Adiabatic OCAES shows about 5% improvement in exergy efficiency over diabatic OCAES. Isothermal OCAES shows significantly higher efficiency over diabatic and adiabatic OCAES. Analysis of liquid piston based OCAES with the use of experimental liquid piston compressor indicated roundtrip efficiency of 45%. With heat transfer enhancement in the liquid piston, roundtrip efficiency of about 62% is possible with the use of either spray cooling or porous media inserts. Overall, liquid piston based OCAES with use of heat transfer enhancement has potential to show significantly higher efficiency than existing compressed air energy storage plants.

Disclosure

Portions of this study have been presented in a poster form at 2017 MAE Graduate Research Symposium of North Carolina State University, Raleigh, USA. Also, part of the work in this paper was presented in ASME 2017 Power Conference and TechConnect World Innovation Conference 2017. Dr. Paul I. Ro served in the capacity of technical advisor for this work.

Conflicts of Interest

The authors declare that there are no conflicts of interest regarding the publication of this paper.

Acknowledgments

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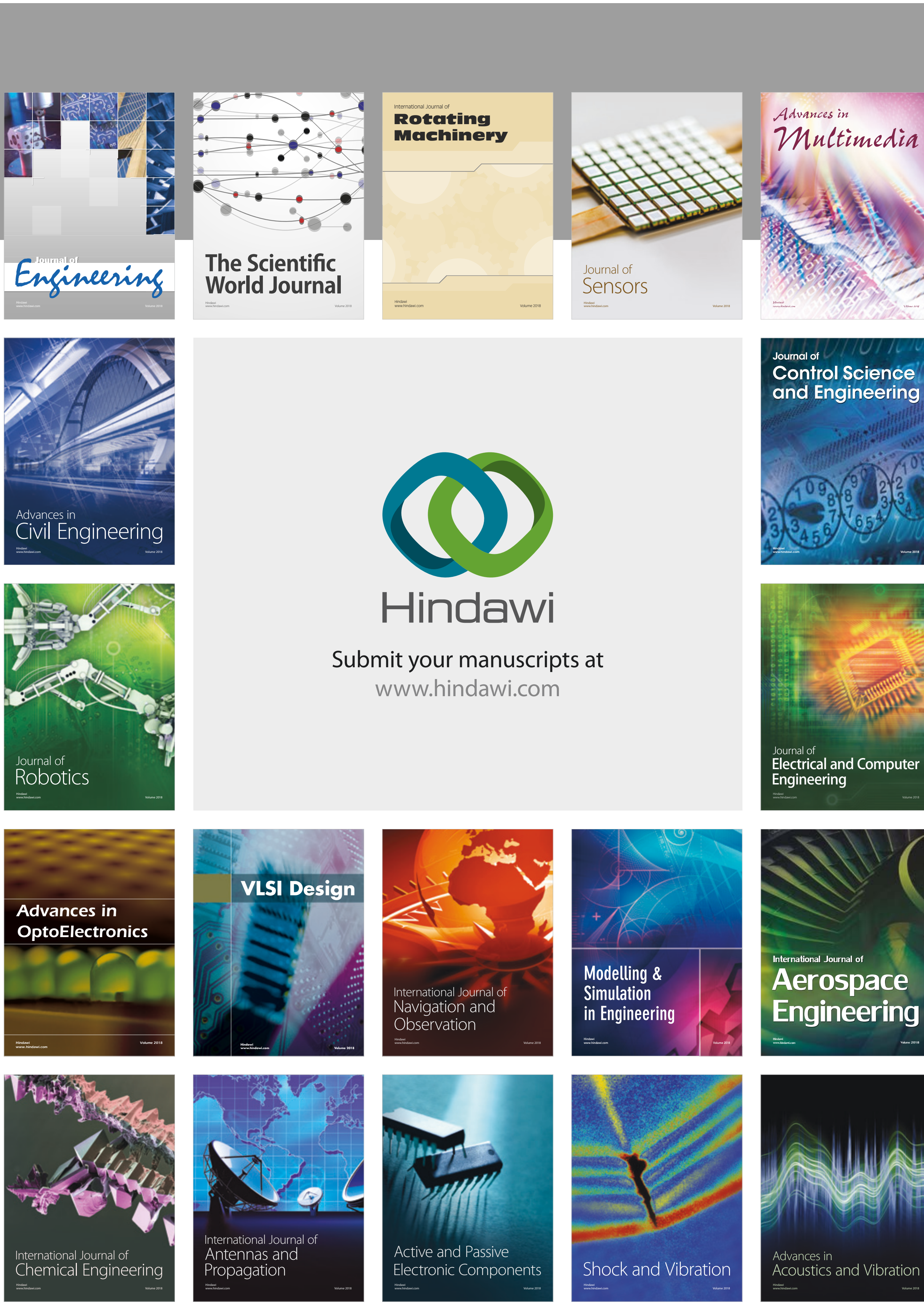
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