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Article

Optimal Utilization of Compression Heat in Liquid Air Energy Storage

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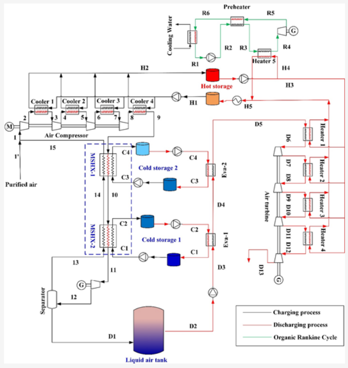
ABSTRACT: Liquid air energy storage (LAES) is regarded as one of the promising large-scale energy storage technologies due to its characteristics of high energy density, being geographically unconstrained, and low maintenance costs. However, the low liquid yield and the incomplete utilization of compression heat from the charging part limit the round-trip efficiency (RTE) of the LAES system. In this work, the organic Rankine cycle (ORC), absorption refrigeration cycle (ARC), and high temperature heat pump (HTHP) are considered to utilize the surplus compression heat in the LAES system. The ORC and the ARC are adopted to utilize high-grade compression heat in an LAES system with a four-stage compressor and a four-stage expander, while the HTHP is used to utilize medium-grade compression heat in an LAES system with a six-stage compressor and a three-stage expander. The reason is the limited working fluids available for HTHP systems. The LAES-ORC, LAES-ARC, and LAES-HTHP systems are modeled in Aspen HYSYS and optimized by a particle swarm optimization (PSO) algorithm. Optimal results indicate that the RTE of the LAES-ORC system is improved from 62.1 to 64.5% with R600a as the working fluid. For the optimized LAES-ARC system, the RTE reaches 63.5% with an increased liquid yield of air of 89.6%. In the LAES-HTHP system, the largest RTE of 58.3% is obtained when the HTHP uses R1233zd as the working fluid, and the result is an increase of 3.7% points compared to the LAES system without an HTHP. Thus, the ORC, the ARC, and the HTHP can effectively improve the performance of the LAES system using the available surplus compression heat.



1. INTRODUCTION

Renewable energy capacity has increased rapidly in recent years.[1] It is predicted that the fraction of renewables in primary energy consumption will reach 20−60% in 2050.[2] Thus, energy storage is important due to the rise of fluctuating renewables.[3] Among a number of energy storage technologies, liquid air energy storage (LAES) has certain advantages, such as being geographically unconstrained, having high energy density, and low maintenance and operational costs.[4] Therefore, LAES can be regarded as a promising electricity storage technology to be used in future energy systems.

A considerable amount of research related to liquid air energy storage has been conducted to analyze the thermodynamic performance and improve the technical and economical feasibility of this technology. The first pilot plant for an LAES system (350 kW/2.5 MWh) operated by Highview Power was located at the University of Leeds between 2011 and 2014 and later moved to the University of Birmingham.[5] In the notation (350 kW/2.5 MWh), the first number is the power recovered in the discharging mode, while the second number refers to the energy storage capacity. Since June 2018, a precommercial plant of LAES (5 MW/15 MWh), which is integrated with low-temperature waste heat from GE Jenbacher landfill gas engines, is operated in Bury. Furthermore, an LAES plant (50 MW/250 MWh) is being constructed by Highview Power and was planned to go into operation in 2022 in the UK. The results of a pilot scale LAES system are published by Morgan et al.[6] A very low round-trip efficiency of 8% was obtained for this pilot scale system. However, through the error analysis of experimental results, it is found that the low system efficiency is a result of the small size of the process. In addition, only 51% of the cold energy is recovered from liquid air. They predicted that a round-trip efficiency of up to 60% can be reached with a larger commercial scale system (100 MW/600 MWh) and a larger cold energy recovery efficiency (91%) in the cold box.

The LAES system proposed by Guizzi et al.[7] consists of two compression stages in the charging part and three expansion stages in the discharging part. Learning from adiabatic compressed air energy storage (CAES) processes, using hot and cold energy recovery cycles between the charging and discharging parts can effectively improve the performance of the system. The concept of hot and cold energy recovery cycles has also been proven to significantly improve the round-trip efficiency (RTE) to 54.4% in an LAES system. Liu et al.[8]

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studied and compared different configurations of the LAES system. The number of compressors and expanders are variables in this work. It is found that a part of the compression heat is not utilized in the reheaters when the number of compressors is larger than or equal to the number of expanders in the LAES system, and this surplus heat is utilized to drive an organic Rankine cycle (ORC) to generate extra power. By comparing different configurations of the LAES system, the system with a two-stage compressor and three-stage expander has the largest RTE of 66.7%. Liu et al.[9] also investigated seven different cold storage cycles to transfer the cold energy from liquid air regasification to the air liquefaction part. Multi-component fluid cycles (MCFCs) and ORCs are considered to replace the commonly used methanol and propane cycles for cold thermal energy storage. The optimization results show that dual MCFCs are the most energy-efficient cold energy recovery cycles for the LAES system, and the highest RTE of 62.4% is obtained for this system.

In addition to studies on the configuration of a stand-alone LAES system, there are some publications focusing on the utilization of waste heat in the LAES system. Peng et al.[10] analyzed two different methods to utilize the unused compression heat in the LAES system. One method is to use an ORC to convert this heat to power. An alternative method is to use 31% of the unused compression heat to drive an ORC, while the remaining unused heat is supplied to an absorption refrigeration cycle (ARC). It is observed that the performance of the LAES system with an ORC (62.7%) is better than the combined system with an ORC and an ARC (61.3%). She et al.[11] utilized the unused compression heat in a vapor compression refrigeration cycle (VCRC) to produce a cold stream, which is the heat sink for the ORC. It is observed that the RTE of the LAES system with ORC-VCRC can be improved to 55.5% compared to the baseline LAES system (50.3%). Tafone et al.[12] considered to use a water-LiBr absorption chiller (AC) to provide a heat sink for the ORC. It is indicated that the RTE of the LAES with only an ORC can be increased to 52.9%, while the performance of the LAES with an ORC and an AC is not improved due to the poor performance of the AC. In addition to the utilization of the unused compression heat in an ORC, Zhang et al.[13] also tried a Kalina Cycle (KC) to utilize the unused compression heat. The results suggest that the LAES system with an ORC has a higher RTE of 56.9% compared to the system with a KC (RTE of 56.1%). Nabat et al.[14] embedded a thermoelectric generator in the heat storage tank to heat up air to 1027 °C. The high temperature air is then sent to the expansion part and an ORC to generate power. The results show that the RTE of the LAES system reaches 61.1%. In addition to the literature above, some researchers used the LAES system in a polygeneration setting, rather than just providing power.[15,16] Cui et al.[15] proposed a novel LAES system, where electricity, heat, cold energy generated by an absorption refrigerator, and fresh air are provided. The cycle efficiency of the system is 75.4% when taking all outputs into account. Xue et al.[16] also studied polygeneration solutions with an LAES system. A round-trip efficiency of 69.6% for the LAES system is obtained.

In addition, a comprehensive literature review on the integration of the LAES system with external hot and cold energy sources was considered in the work of Ding et al.[17] Geothermal energy has been recognized as a heat source for the LAES system.[18] Another example is the large amount of cold energy that is available in the LNG regasification process. Some studies have been performed to indicate the possibility of integration between the LAES system and the LNG regasification process.[19−21] Overall, the performance of the LAES can be improved by the methods mentioned above, and this makes the LAES a more promising technology.[22]

As mentioned in the work of Liu et al.,[8] the optimal pressure ratios of compressors and expanders and the amount of compression heat vary with different numbers of compression stages in the charging part. The LAES with fewer compression stages and higher pressure ratios has a higher RTE. However, the operational feasibility and economy of the LAES with large pressure ratios are questioned. Thus, the system with more feasible pressure ratios of compressors is considered in this work. When the number of compression stages is greater than 3, the optimal pressure ratios for compressors are within the normal range (less than 4). The LAES system with a four-stage compressor and a four-stage expander has 17% of compression heat at 168.2 °C not being utilized in the discharging part, and this configuration is considered as the LAES with available high-grade compression heat. It is also found that an LAES with too many compression stages will result in a low RTE due to the relatively low temperature of compression heat. For the LAES system with a six-stage compressor and a three-stage expander, 54% of compression heat at 109.3 °C is not utilized in the preheaters in the expansion section. This case is selected as the LAES with available medium-grade compression heat in this work.

ORCs and ARCs are the most common methods to utilize the high-grade compression heat in the LAES.[10−16] Most of the studies related to the ORC-based utilization of compression heat focus on the improvement of RTE for an LAES with a specific configuration. For an LAES system with a four-stage compressor and a four-stage expander, there is a lack of research on comparing and selecting different working fluids for the ORC. In addition, the ARC-based utilization of compression heat in the mentioned papers is aimed to produce a cold stream. The cold stream is either regarded as the heat sink of an ORC or provided as a cold energy product. It is known that the cold energy from the discharging part is not enough to liquefy air, so electricity is required to compress air in the charging part. In the available literature, no research has been conducted on the utilization of the cold duty produced by an ARC in the cold box for air liquefaction.

For the LAES with available medium-grade compression heat, high temperature heat pumps (HTHPs) are promising technologies to enhance waste heat and thereby increase system efficiency.[23] HTHPs can efficiently raise the process stream temperature and provide another possibility for the utilization of compression heat in the LAES. The temperature of compression heat can be upgraded in the HTHP, and the upgraded compression heat is used to preheat air to increase the power output in the discharging part. To be profitable, the additional power output must be greater than the power input to the heat pump. In addition, the investment cost for the heat pump must be considered.

In this work, three technologies for the utilization of compression heat (ORC, ARC, and HTHP) at different temperature levels are discussed. For the LAES system with a four-stage compressor and a four-stage expander, the temperature of compression heat is around 170 °C. An ORC and an ARC are considered to utilize this high temperature heat source. The ORC is used for converting excess compression heat to power, while the ARC is aimed at providing cold duty

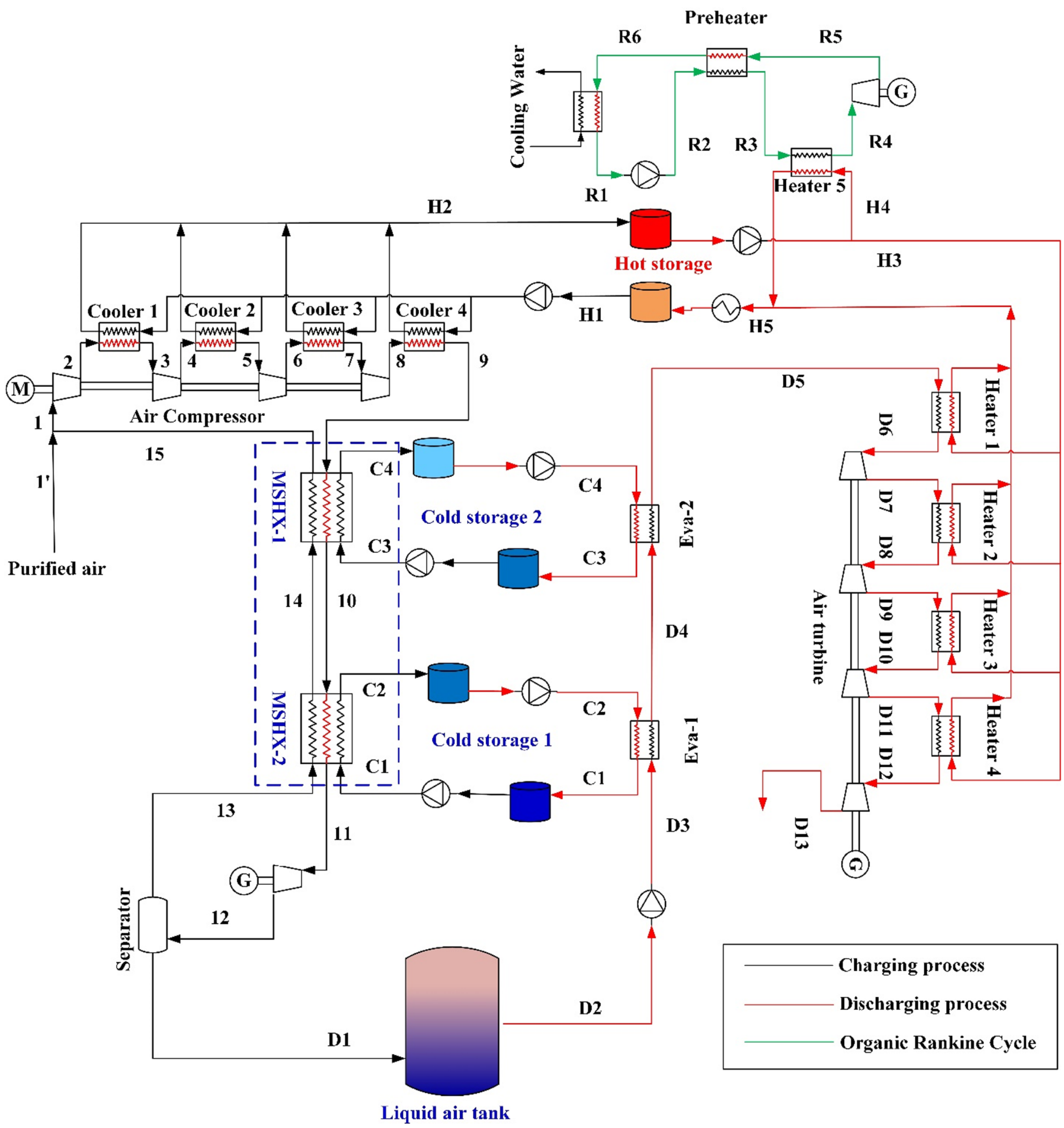

Figure 1. Flow diagram of liquid air energy storage with an ORC.

to the cold box in the charging part. An HTHP is adopted in an LAES configuration with medium-grade compression heat (around 110 °C), such as the system with a six-stage compressor and a three-stage expander. The reason is that very few working fluids are available for HTHP cycles. Furthermore, different working fluids for the ORC and the HTHP are selected and compared to identify the best working fluids for these technologies.

2. PROCESS SIMULATION AND OPTIMIZATION

First, in this section, simplifying assumptions for the model used in the analysis are provided. Then, the process descriptions for the LAES-ORC, LAES-ARC, and LAES-HTHP systems are introduced, followed by a presentation of key performance indicators (KPIs) for the system. Finally, the optimization algorithm is presented.

2.1. Assumptions. Water vapor and CO_2 are removed in the air pre-treatment process to avoid solid formation before air is sent to the energy storage process. However, the air pre-treatment process is beyond the scope of this work since focus is on thermodynamic cycles for the utilization of unused compression heat. Air feed is assumed to be composed of 78.82 mol % N_2, 21.14 mol % O_2, and 0.04 mol % argon and provided at 20 °C and 1.01 bar with a mass flow rate of 61,520 kg/h. Beyond the vapor phase air that is recycled from the flash following the cryo-turbine to the compressors in the charging part, there is no recycle of air from the discharging part, i.e., the LAES is an open, not closed, system. The simulation of the LAES is performed in Aspen HYSYS Version 10.0.[24] The thermodynamic properties of process streams are calculated by the Peng−Robinson equation of state. The following assumptions are made to simplify the system analysis:

- Adiabatic efficiencies for compressors, expanders, and pumps are assumed to be constant at 85, 90 and 80%, respectively. The adiabatic efficiency for the cryo-turbine is assumed to be 75%.
- Pressure drops and heat losses in heat exchangers, storage tanks, and the phase separator are ignored.

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- Outlet temperatures of coolers for compressed air are assumed to be 30 °C.
- Storage tanks for working fluids in the hot and cold energy recovery cycles and liquid air are at atmospheric pressure.

2.2. System Description. The LAES system with a four-stage compressor and a four-stage expander has a relatively high-grade compression heat (Case I), while the system with a six-stage compressor and a three-stage expander has a medium-grade compression heat (Case II). These configurations have been thoroughly studied in the work of Liu et al.,[8] and they are regarded as reference cases in this work.

For a typical standalone LAES system, during the charging period, the air feed is pressurized in a multi-stage compressor with aftercooling before it is fully liquefied in the cold box (consisting of multistream heat exchangers MSHX-1 and MSHX-2, see Figure 1). In the aftercoolers, compression heat is transferred from air to thermal oil, which is sent to a high temperature storage tank in the heat recovery cycle. The cold box consists of two low temperature heat exchangers, where the cold duty is provided by the working fluids in two cold energy recovery cycles. The composition of the working fluids in the two cold energy recovery cycles is obtained from our previous study,[9] where two multi-component fluid cycles proved to have the best performance. However, the fractions of nitrogen and ethane in these working fluids were relatively small, so these two components were not included to reduce the computational work in previous work. An even simpler set of working fluids used in this work is provided in Table 1. With the consideration of operational safety and economics, the high pressure liquid air is first expanded to nearly atmospheric pressure, and then, it is separated into gas and liquid streams. The gas stream is recycled to the compression part, while the liquid air is sent to a cryogenic storage tank. During the discharging period, liquid air is pumped and then evaporated in heat exchangers, where the cold energy from liquid air regasification is recovered by working fluids in the cold energy recovery cycles. Then, the high pressure air enters a multi-stage expander to generate power. Thermal oil carrying the compression heat is used to preheat air before each stage of the expansion section to increase power production. The recovery of compression heat and cold energy from liquid air regasification is crucial for the improvement of system performance.

Table 1. Composition of Working Fluids in the Two Cold Energy Recovery Cycles

component	mole fraction	
	1st cycle	2nd cycle
methanol	1.00	
ethane		0.20
propane		0.70
n-butane		0.10

As mentioned in Section 1, a part of compression heat is not fully utilized in the discharging part for the two reference cases. In this work, an ORC and an ARC are introduced to utilize the surplus high-grade compression heat in Case I, while an HTHP is selected to utilize the surplus medium-grade compression heat in Case II.

2.2.1. LAES System with an ORC for the Utilization of Surplus High-Grade Compression Heat. The process flow diagram of the LAES system with an ORC is shown in Figure 1. Surplus compression heat (stream H4) is used as the heat source for the ORC. The working fluid in the ORC is first pumped to high pressure, and then, it is preheated and evaporated by recovering surplus compression heat. After the working fluid is fully regasified, it is expanded to generate power. The exhaust stream from the expander provides heat to a preheater before the stream is liquified by a water cooler. In this work, subcritical ORCs are selected with the consideration of operational feasibility, safety, and investment cost. To ensure stable operation of the ORC, the pressure of working fluids in the ORC should be below the near-critical region, here set to be below 90% of the critical pressure.[25] Eight substances have been considered as working fluids for the ORC in the LAES system. The performance of different working fluids of the ORC is discussed in Section 3.1.

2.2.2. LAES System with an ARC for the Utilization of Surplus High-Grade Compression Heat. The flowsheet of the LAES system with an ARC is shown in Figure 2. Ammonia—water is the working fluid of the ARC. The ammonia solution (A1) is pumped and preheated before entering a distillation column to be separated. The pure ammonia stream obtained at the top of the column is first precooled by seawater and then throttled to generate refrigeration capacity for the cold box. After the cold energy of the ammonia regasification is delivered to the cold box, the ammonia stream is sent back to an absorber. The product in the bottom of the distillation column is a lean ammonia solution, which is used to preheat the feed stream to the column. The lean ammonia stream is next throttled before it is sent back to the absorber to be mixed with pure ammonia. It is worth noting that the heat source of the ARC is surplus compression heat (stream H4), which provides the heat duty in the reboiler of the column. The cold energy generated from the ARC is sent to the cold box to liquefy more air.

2.2.3. LAES System with an HTHP for the Utilization of Surplus Medium-Grade Compression Heat. As discussed in the work of Liu et al.,[8] medium-grade compression heat at 109.3 °C is obtained when there is a six-stage compressor in the charging part. The schematic of the LAES system with an HTHP to utilize the medium-grade compression heat is shown in Figure 3. The medium-grade compression heat is split into two streams H3 and H4. Stream H4 is sent to heat exchanger HP-Eva, where surplus compression heat is used to evaporate the working fluid of the HTHP. The vapor is then pressurized in a compressor, and the outlet stream of the compressor is at high temperature, which is used to increase the temperature of stream H3. The enhanced stream H3 is sent to heat exchangers to preheat air before expansion stages and thereby increase the power production. Eight substances have been selected as potential working fluids of the HTHP, and the results will be discussed in detail in Section 3.3.

2.3. System Evaluation. The performance of the LAES system can be evaluated by various key performance indicators (KPIs). The round-trip efficiency and exergy efficiency are commonly selected and used for the comparison of different energy storage processes. In addition, the thermal efficiency of the ORC and the Coefficient of Performance (COP) of the ARC and the HTHP are calculated to indicate the performance of different thermodynamic cycles.

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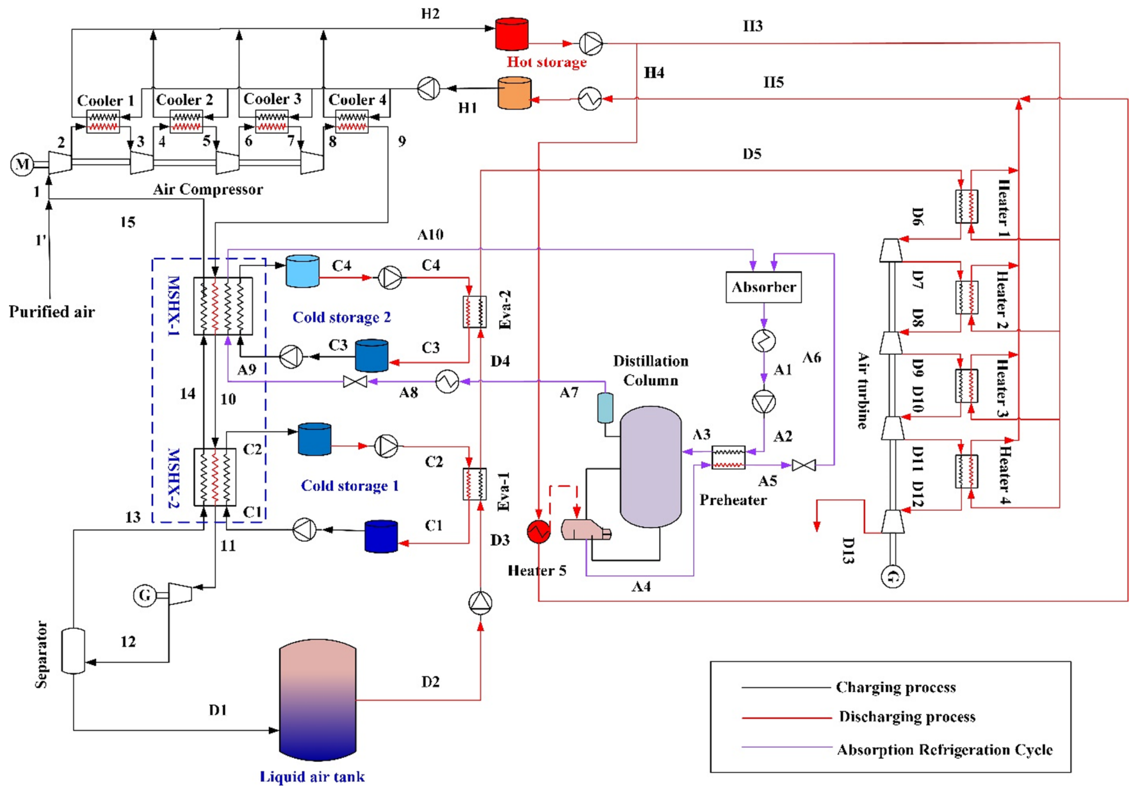

Figure 2. Flow diagram of liquid air energy storage with an ARC.

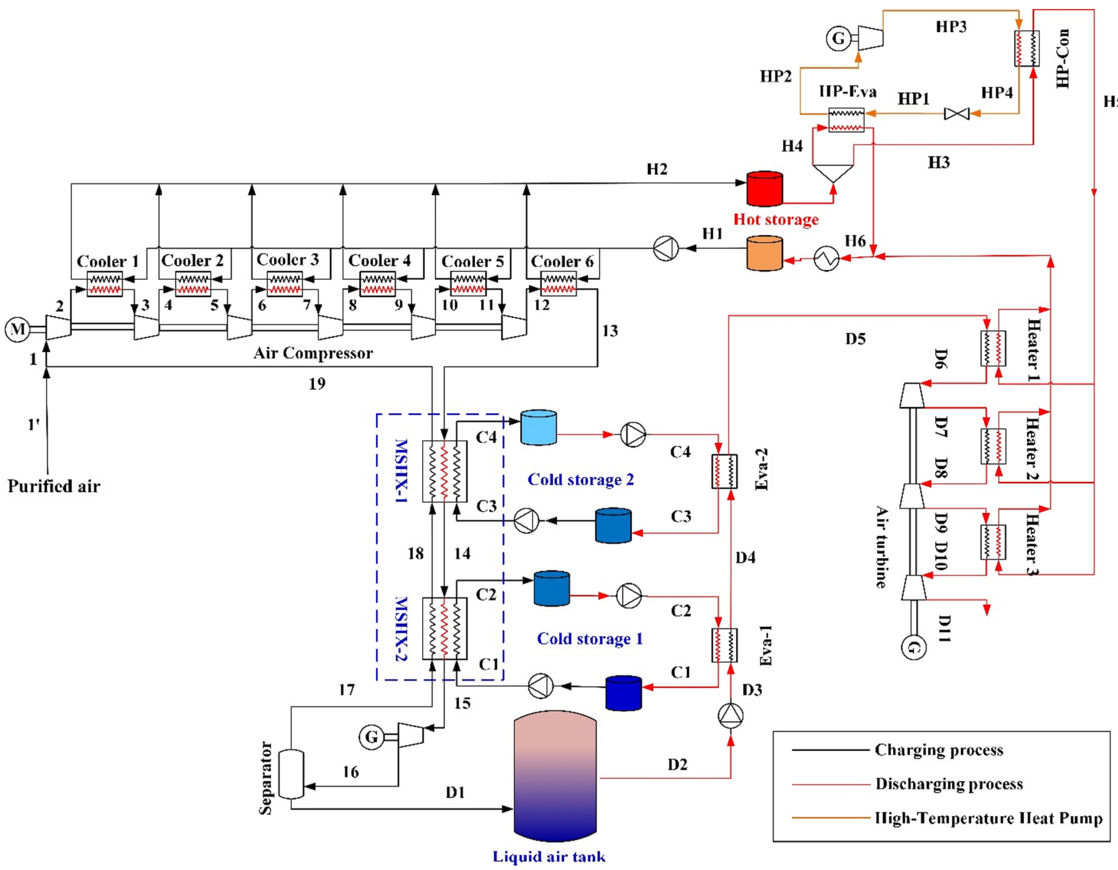

Figure 3. Flow diagram of liquid air energy storage with an HTHP.

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The round-trip efficiency (RTE) is defined as the ratio between the work output of the discharging part and the work input of the charging part, as shown in eq 1.

$$\eta_{\mathrm{RT}} = \frac{\dot{W}_{\mathrm{out}}}{\dot{W}_{\mathrm{in}}} = \frac{\dot{m}_{\mathrm{liq}} w_{\mathrm{tur}}}{\dot{m}_{\mathrm{comp}} w_{\mathrm{comp}}} = \eta_{\mathrm{LY}} \cdot \frac{w_{\mathrm{tur}}}{w_{\mathrm{comp}}} \tag{1}$$

Here, $\dot{m}_{\mathrm{liq}}$ and $\dot{m}_{\mathrm{comp}}$ represent the mass flow rate of the produced liquid air and the air entering the compression part. The mass flow rate of the air entering the compression part is larger than that of the liquid air ($\dot{m}_{\mathrm{comp}} > \dot{m}_{\mathrm{liq}}$). The reason is that air is not fully liquefied in the charging part, and a phase separator is used before the storage tank. Liquid air is stored in a cryogenic tank, while the gaseous air is sent back to the compression part. η_{LY} is the liquid yield of air. w_{comp} and w_{tur} are the specific work of compressors in the charging part and turbines in the discharging part, respectively.

Exergy efficiency, which considers both the 1st and 2nd laws of thermodynamics, is another significant parameter to reveal the performance of processes. Exergy is the maximum work obtained when the process stream is taken from its present state to thermodynamic equilibrium with the environment. Neglecting kinetic, electrical, potential, and nuclear exergies, the exergy of process streams comprises thermo-mechanical exergy ($\dot{E}^{\mathrm{TM}}$) and chemical exergy ($\dot{E}^{\mathrm{Ch}}$).[26] Change in chemical exergy is related to the chemical reaction, mixing, and separation. The variation of chemical exergy in the LAES system is very small since it is only related to the phase separator before the liquid air storage tank, and changes of the composition is relatively small since nitrogen and oxygen are close-boiling components.

A considerable number of exergy efficiencies has been proposed in the literature, and they can be classified as input–output and consumed–produced. However, most of these efficiencies suffer from two major deficiencies: (1) since chemical exergy tends to be much larger than thermo-mechanical exergy, it has a diluting effect on exergy efficiency in equipment without change in composition, making it impossible to evaluate on an objective basis the changes in thermo-mechanical exergy. (2) Most of the proposed efficiencies are unable to handle subambient processes. To overcome this, Marmolejo-Correa and Gundersen[27] proposed the Exergy Transfer Effectiveness (*ETE*), which focuses on (as the name indicates) the exchange or transfer of exergy between process streams and energies in the various equipment types. This is done by a careful identification of exergy sources and sinks, and the definition of *ETE* is the ratio between the exergy sinks (exergy increases) and the exergy sources (exergy reductions) in a process, as is given by eq 2. While the original *ETE* only considered thermo-mechanical exergy, the concept of *ETE* was further extended by Kim and Gundersen[28,29] to also consider chemical exergy.

$$\eta_{\dot{E}} = ETE = \frac{\sum \text{Exergy Sinks}}{\sum \text{Exergy Sources}} \tag{2}$$

The exergy efficiencies of the charging and discharging parts ($\eta_{\dot{E}_{\mathrm{ch}}}$ and $\eta_{\dot{E}_{\mathrm{dc}}}$) of the LAES-ORC, LAES-ARC, and LAES-HTHP systems are given by eqs 3–8.

$$\eta_{\dot{E}_{\mathrm{ch,ORC}}} = \frac{\dot{W}_{\mathrm{cryotur,ch}} + \dot{E}_{\mathrm{liq}} + \dot{E}_{\mathrm{h}}}{\dot{W}_{\mathrm{comp,ch}} + \dot{E}_{\mathrm{c}} + \dot{E}_{\mathrm{fa}}} \tag{3}$$

$$\eta_{\dot{E}_{\mathrm{dc,ORC}}} = \frac{\dot{W}_{\mathrm{tur,dc}} + \dot{E}_{\mathrm{c}} + \dot{W}_{\mathrm{ORC}}}{\dot{W}_{\mathrm{pump,dc}} + \dot{E}_{\mathrm{liq}} + \dot{E}_{\mathrm{h}}} \tag{4}$$

$$\eta_{\dot{E}_{\mathrm{ch,ARC}}} = \frac{\dot{W}_{\mathrm{cryotur,ch}} + \dot{E}_{\mathrm{liq}} + \dot{E}_{\mathrm{h}}}{\dot{W}_{\mathrm{comp,ch}} + \dot{E}_{\mathrm{c}} + \dot{E}_{\mathrm{fa}} + \dot{W}_{\mathrm{ARC}}} \tag{5}$$

$$\eta_{\dot{E}_{\mathrm{dc,ARC}}} = \frac{\dot{W}_{\mathrm{tur,dc}} + \dot{E}_{\mathrm{c}}}{\dot{W}_{\mathrm{pump,dc}} + \dot{E}_{\mathrm{liq}} + \dot{E}_{\mathrm{h}}} \tag{6}$$

$$\eta_{\dot{E}_{\mathrm{ch,HTHP}}} = \frac{\dot{W}_{\mathrm{cryotur,ch}} + \dot{E}_{\mathrm{liq}} + \dot{E}_{\mathrm{h}}}{\dot{W}_{\mathrm{comp,ch}} + \dot{E}_{\mathrm{c}} + \dot{E}_{\mathrm{fa}} + \dot{W}_{\mathrm{HTHP}}} \tag{7}$$

$$\eta_{\dot{E}_{\mathrm{dc,HTHP}}} = \frac{\dot{W}_{\mathrm{tur,dc}} + \dot{E}_{\mathrm{c}}}{\dot{W}_{\mathrm{pump,dc}} + \dot{E}_{\mathrm{liq}} + \dot{E}_{\mathrm{h}}} \tag{8}$$

Here, $\dot{W}_{\mathrm{cryotur,\,ch}}$ represents the work produced by the cryo-turbine and $\dot{W}_{\mathrm{comp,\,ch}}$ is the work consumed by compressors in the charging process. $\dot{E}_{\mathrm{liq}}$, $\dot{E}_{\mathrm{h}}$, $\dot{E}_{\mathrm{c}}$, and $\dot{E}_{\mathrm{fa}}$ denote the thermo-mechanical exergy of liquid air, thermal oil, working fluids in the cold energy recovery cycles, and the air feed, respectively. $\dot{W}_{\mathrm{tur,\,dc}}$, $\dot{W}_{\mathrm{pump,\,dc}}$, and $\dot{W}_{\mathrm{ORC}}$ are the expansion work of expanders, the work consumed in the pump, and the net work production in the ORC in the discharging part, respectively. $\dot{W}_{\mathrm{ARC}}$ is the work consumed by the pump in the ARC, and $\dot{W}_{\mathrm{HTHP}}$ is the compression work in the HTHP.

The performance of the ORC, the ARC, and the HTHP is provided by eqs 9–11.

$$\eta_{\mathrm{ORC}} = \frac{\dot{W}_{\mathrm{ORC}}}{\dot{Q}_{\mathrm{ORC}}} \tag{9}$$

$$\mathrm{COP}_{\mathrm{ARC}} = \frac{\dot{Q}_{\mathrm{ARC}}}{\dot{W}_{\mathrm{ARC}}} \tag{10}$$

$$\mathrm{COP}_{\mathrm{HTHP}} = \frac{\dot{Q}_{\mathrm{HTHP}}}{\dot{W}_{\mathrm{HTHP}}} \tag{11}$$

Here, $\dot{Q}_{\mathrm{ORC}}$ represents the heat duty of the evaporator (Heater 5) in the ORC in Figure 1. $\dot{Q}_{\mathrm{ARC}}$ is the cold energy produced in the ARC and then delivered to the cold box in Figure 2. $\dot{Q}_{\mathrm{HTHP}}$ is the heat duty of the condenser (HP-Con) in the HTHP, as shown in Figure 3.

2.4. System Optimization. The LAES system with different thermodynamic cycles is optimized by using a particle swarm optimization (PSO) algorithm. Derivatives of mathematical functions (gradients) are not required in the PSO algorithm. Consequently, the algorithm can be applied to complex black box models.[30] The PSO is executed in Matlab, version R2018a,[31] which provides the input to the HYSYS model and collects stream data from the simulated LAES system. The objective is to maximize the RTE of the LAES with different thermodynamic cycles to utilize the surplus compression heat, as formulated by eq 12.

$$\min_{\boldsymbol{x}} -\eta_{\mathrm{RT}} = -f(\boldsymbol{x}) = -\frac{\dot{m}_{\mathrm{liq}} w_{\mathrm{tur}}}{\dot{m}_{\mathrm{comp}} w_{\mathrm{comp}}} \tag{12}$$

The constraints for different LAES systems are listed in Table 2. It is worth noting that the optimization is performed only for the ORC and the HTHP cycle in the LAES-ORC and the LAES-HTHP systems, since only hot oil stream H4 is

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Table 2. Constraints for the LAES-ORC, LAES-ARC, and LAES-HTHP Systems

	constraints
LAES-ORC	$\Delta T_{con/eva} \geq 10$ °C
	$VF_{in,pump,ORC} = 0$
	$VF_{in,tur,ORC} = 1$
	$VF_{out,tur,ORC} = 1$
LAES-ARC	$\Delta T_{heater/cooler} \geq 10$ °C
	$\Delta T_{MSHX/eva} \geq 1$ °C
	$x_{NH3} \geq 0.999$
	$T_{out,Heater\ 5} - T_{A4} \geq 10$ °C
LAES-HTHP	$\Delta T_{con/eva} \geq 10$ °C
	$VF_{in,comp,HP} = 1$
	$VF_{out,comp,HP} = 1$
	$T_{out,comp} \leq T_{cr}$
	$p_{out,comp} \leq p_{cr}$

affected. However, the situation is different for the LAES-ARC system. The cold duty generated from the ARC is provided to the cold box, which affects the performance of the overall system. Thus, the optimization is implemented for the entire LAES-ARC system. In order to balance heat transfer efficiency and costs of heat exchangers, in all three cases, (ORC, ARC, and HTHP), the minimum temperature difference (ΔT_{min}) of heaters and coolers is assumed to be 10 °C, while the ΔT_{min} of low temperature heat exchangers (the cold box and evaporators) is set to 1 °C.[32] For the LAES-ARC, the purity of ammonia in the top of the distillation column is constrained to be larger than 99.9 mol %. In addition, to ensure that the surplus compression heat is transferred to the reboiler, the outlet temperature of Heater 5 is assumed to be 10 °C larger than the bottom stream of the column. The decision variables for the LAES system with different thermodynamic cycles are listed in Table 3, where the upper and lower bounds for the variables are provided.

Table 3. Lower and Upper Bounds for Decision Variables of the LAES-ORC, LAES-ARC, and LAES-HTHP Systems

variables	lower bounds	upper bounds
condensation pressure (bar)[a]	1.0	$0.9p_{cr}$
evaporation pressure (bar)[a]	1.1	$0.9p_{cr}$
working fluid evaporation temperature (°C)[a]	50	T_{cr}
working fluid molar flowrate (kmol/h)[a]	0.1	800
pressure ratio for compressors[b]	1.0	5
pressure ratio for expanders[b]	1.0	10
thermal oil temperature (°C)[b]	100	230
cold box outlet air temperature (°C)[b]	−185	−165
cold box outlet recycled air temperature (°C)[b]	−10	29
working fluid operating temperature (higher) (°C)[b]	−90	−20
working fluid molar flowrate (kmol/h)[b]	0.1	3000
working fluid operating temperature (lower) (°C)[b]	−186	−166
condensation pressure (bar)[c]	1.1	p_{cr}
evaporation pressure (bar)[c]	1.0	p_{cr}
working fluid evaporation temperature (°C)[c]	50	T_{cr}
working fluid molar flowrate (kmol/h)[c]	0.1	800

[a]Variable bounds for the LAES-ORC system. [b]Variable bounds for the LAES-ARC system. [c]Variable bounds for the LAES-HTHP system.

3. RESULTS AND DISCUSSION

The ORC, the ARC, and the HTHP are selected to utilize the waste compression heat at different temperature levels in the LAES system. The performance of these thermodynamic cycles is discussed in the following.

3.1. Analysis of the LAES System with an ORC.

As illustrated in Figure 1, a part of the high temperature compression heat in Case I will be recovered by the ORC rather than wasted. Stream data for the heat recovery cycle in the LAES is shown in Table S1. From this table, it is observed that around 17% of the compression heat at 168.2 °C is in surplus. The ORC is regarded to be one of the solutions to utilize this surplus high temperature compression heat in the LAES.

The thermodynamic properties of the working fluid are crucial for the ORC and will thereby affect the performance of the LAES system. A comparison of working fluids has been conducted to identify the most suitable medium that leads to the largest net work production of the ORC and the highest RTE of the LAES. Working fluids for typical ORCs have been investigated widely, and a large number of candidates can be considered for the selection in this work. Thus, a pre-selection is needed to reduce the workload of screening working fluids for the ORC in the LAES system. The criteria for the selection of working fluid for the ORC take physical properties, environmental impact, economics, and operational feasibility into consideration, and the most important ones are the following:

(1) Freezing point of the working fluid should be lower than ambient temperature to avoid the formation of solids in the ORC.
(2) Ozone depletion potential (ODP) of the working fluid should be 0 and the global warming potential (GWP) of the fluid should be low.
(3) The working fluids should be noncorrosive, nontoxic, and innoxious to humans.
(4) Low cost working fluids are more favorable.
(5) In order to avoid a pinch limitation in the evaporation heat exchanger (Heater 5) in the ORC, the critical temperature of the working fluid should be lower than the temperature of the heat source (stream H4). However, the critical temperature should not be too low since that would result in large exergy losses in Heater 5.

Based on the criteria listed above, eight working fluids that are studied as candidates for the ORC are listed in Table 4. As mentioned in Section 2.2.1, subcritical operation is applied for the ORC. Figure 4 illustrates the net power output of ORCs

Table 4. Properties of Working Fluids Investigated in This Study

fluids	chemical formula	T_{cr} (°C)	p_{cr} (bar)	T_{nb} (°C)	ODP	GWP_{100}
R125	C_2HF_5	66.02	36.19	−48.11	0	3500
R143a	$C_2H_3F_3$	72.73	37.64	−47.34	0	4470
R1270	C_3H_6	92.44	46.64	−47.69	0	2
R290	C_3H_8	96.67	42.42	−42.08	0	3
R134a	$C_2H_2F_4$	101.03	40.56	−26.07	0	1100
R152a	$C_2H_4F_2$	113.55	45.00	−24.95	0	124
R600a	C_4H_{10}-2	134.83	36.54	−11.79	0	3
R600	C_4H_{10}-1	152.00	37.96	−0.51	0	4

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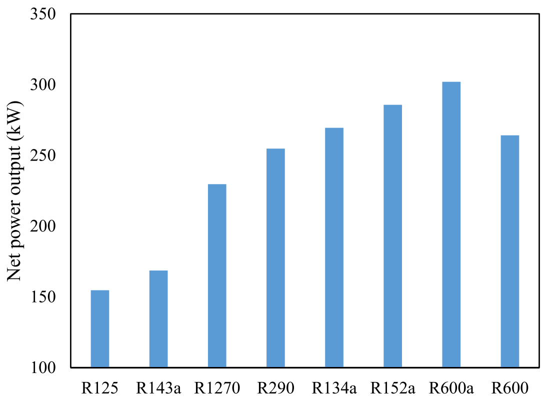

Figure 4. Net power output of different working fluids for the ORC when utilizing the surplus high temperature compression heat in the LAES.

with different working fluids that utilize the high temperature surplus compression heat in the LAES system. It is shown that R600a and R152a are the top two working fluids in terms of net power output. Table 5 shows the optimal operating variables for the eight working fluids, such as condensation and evaporation pressure, temperature, heat duty, and logarithmic mean temperature difference (LMTD) of the evaporator, and work of the turbine and pump. The smallest LMTD of the evaporator and the largest pressure ratio between evaporation and condensation are obtained in the cycle using R600a as the working fluid. This leads to the smallest exergy loss in the evaporator and the largest net power output and thermal efficiency of the cycle. This also results in the largest RTE of 64.5% in the LAES-ORC system.

It can be noticed that the LMTD of the evaporator in the cycle with R134a is slightly smaller than the cycle using R152a as the working fluid, while the net power output is larger for the cycle with R152a. This is because the critical pressure of working fluid R152a is quite high (45 bar, see Table 4), and the pressure ratio of the expander in the ORC using R152a as the working fluid is larger than the cycle with R143a. As a result, the power output of the cycle with R152a is larger than the cycle with R134a. The working fluid R600 has the second largest thermal efficiency for the ORC in converting compression heat to power. However, the power output of the cycle with R600 is lower compared to the cases using R600a, R152a, and R134a as working fluids since the heat recovered in the evaporator (1663.6 kW) is the smallest. This is due to the fact that there exists a trade-off between the amount of recovered heat and the pressure ratio of the turbine in the ORC. A higher evaporation pressure is preferred to have a larger power output. However, to avoid pinch limitation in the evaporator, the amount of recovered heat in the evaporator may not be satisfactory, as the case using R600 as the working fluid shows. For other working fluids with a low critical temperature (e.g., R125, R143a, and R1270), the amount of recovered heat in the evaporator is large, while the pressure ratio of the turbine in the ORC is limited, and therefore, the power output and thermal efficiency of the cycle is low. In addition, the high GWP_{100} for R125, R143a, and R134a makes them less competitive than other working fluids in our study. Thus, it is the working fluid with proper critical temperature and pressure (R600a) and a moderate amount of recovered heat in the evaporator that has the maximum power output.

3.2. Analysis of the LAES System with an ARC.

Some data of the optimized LAES-ARC are provided in Table S2. Figure 5 shows the composite curves of the cold box in the LAES system without and with an ARC. It is noticed that the LMTD for MSHX-1 is larger for the LAES system with an ARC (4.05 °C) compared to the system without an ARC (1.71 °C) due to constant temperature during ammonia phase changing. However, the increased cold energy from ammonia regasification in MSHX-1 leads to a larger liquid yield and a slightly lower charging pressure in the charging part of the LAES-ARC system, as shown in Table 6. A higher liquid yield of air means that less air needs to be recompressed and less compression work is required in the charging part of the LAES-ARC system compared to the system without an ARC. Thus, with the ARC to utilize the surplus compression heat, the RTE of the system is increased to 63.5% due to the reduced compression work in the charging part.

3.3. Analysis of the LAES System with an HTHP.

A lower temperature of thermal oil at 109.3 °C is obtained in the LAES with a six-stage compressor compared to the system with a four-stage compressor (168.2 °C). Stream data for the heat recovery cycle in Case II, which is illustrated in Figure 3, are shown in Table S3. In this work, the HTHP is selected to utilize the medium-grade compression heat in the LAES system. As for the selection of working fluid for the HTHP, in addition to the mentioned concerns about physical properties and economics mentioned in Section 3.1, three significant criteria should be considered for the working fluid selection:

(1) Operational feasibility: The evaporation temperature of the heat pump should be lower than the temperature of the medium-grade compression heat in the LAES. Moreover, the critical temperature of the working fluid of the heat pump should be higher than the temperature of the compression heat.
(2) Coefficient of Performance: The COP reveals the efficiency to convert work to heat in heat pumps.
(3) Environmental impact: To mitigate the progress of global warming, potential greenhouse gases should be avoided.

Table 5. Performance of the ORC with Different Working Fluids in the LAES System

fluids	$\dot{W}_{net}$ (kW)	p_{con} (bar)	p_{eva} (bar)	$\dot{m}$ (kmol/h)	$\dot{Q}_{eva}$ (kW)	T_{eva} (°C)	$LMTD_{eva}$ (°C)	$\dot{W}_{tur}$ (kW)	$\dot{W}_{pump}$ (kW)	η_{ORC} (%)	RTE (%)
R125	154.65	12.10	27.55	451.34	2074.77	66.32	34.65	178.54	23.89	7.45	63.36
R143a	168.60	12.00	28.24	450.52	2077.38	72.99	31.09	187.19	18.59	8.12	63.47
R1270	229.64	10.31	32.04	427.15	2068.38	91.42	21.40	256.07	26.43	11.10	63.96
R290	254.79	8.42	32.65	401.67	2066.45	96.27	19.86	284.62	29.83	12.33	64.16
R134a	269.43	5.82	31.00	323.89	2073.97	101.24	18.15	293.14	23.72	12.99	64.28
R152a	285.78	5.34	41.00	270.23	1859.20	113.22	18.47	310.01	24.23	15.37	64.41
R600a	301.95	3.11	29.45	263.01	1856.82	121.79	15.84	327.07	25.12	16.26	64.54
R600	264.11	2.14	17.01	216.67	1663.57	105.49	23.33	275.38	11.27	15.88	64.24

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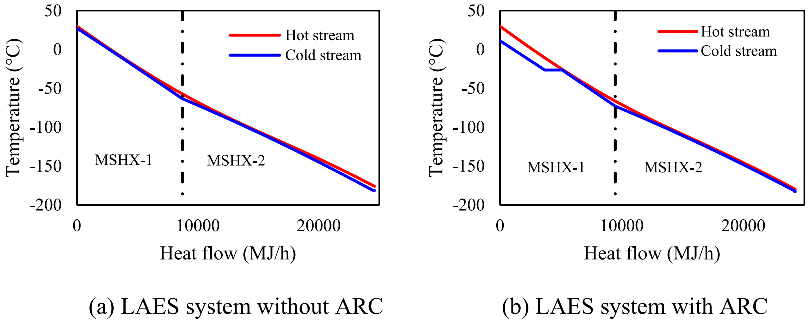

Figure 5. Composite curves of the cold box in LAES configurations (a) without and (b) with an ARC system.

Table 6. Performance of the LAES System with an ARC

	LAES	LAES-ARC
$\dot{Q}_{coldbox}$ (kJ/kg)	346.24	353.26
$LMTD_{MSHX\text{-}1}$ (°C)	1.71	4.05
$LMTD_{MSHX\text{-}2}$ (°C)	2.50	2.41
p_{ch} (bar)	187.12	186.02
p_{dc} (bar)	159.81	148.26
η_{LY} (%)	86.57	89.60
$\dot{W}_{ch}$ (MW)	12.50	12.06
$\dot{W}_{dc}$ (MW)	7.76	7.65
RTE (%)	62.13	63.47

Taking the above concerns into consideration, eight working fluid candidates with critical temperature higher than 109.3 °C are considered and listed in Table 7. Some key parameters and the performance of the HTHP with different working fluids are shown in Table 8, such as the mass flow rate of the working fluid, heat duties and temperatures of evaporators and condensers, the work consumed in the compressor and the heat pump, the work consumed in the charging part, and the work produced in the discharging part of the LAES system. It is observed that the system using R1233zd as the working fluid in the heat pump has the highest upgraded temperature of the compression heat (155.1 °C), and thereby the highest RTE of the LAES-HTHP system. The reason is the large heat duty of the evaporator in the heat pump, which is the main factor that influences the temperature of the upgraded compression heat and the power production in the LAES-HTHP system as well. The heat source for the heat pump is a part of surplus compression heat in the LAES system, which is regarded as free. The largest heat recovery in the evaporator will lead to the largest condensation heat in the heat pump, which is beneficial for the power generation in the discharging part of the LAES system. For the cycle using R1233zd as the working fluid, the heat recovered in the evaporator achieves a maximum of 2.0 MW, and the largest heat duty of 2.7 MW is obtained in the condenser to be released to upgrade the compression heat. Although compression work is the largest and COP is the smallest for the heat pump cycle with R1233zd, the additional work produced in the discharging part of the LAES system can justify the compression work in the heat pump due to the upgraded compression heat. Another advantage of selecting R1233zd is that it is a hydrofluoroolefin (HFO) with low environmental impact, achieving the phase-out of hydrofluorocarbons (HFCs).[33]

3.4. Comparison of LAES-ORC, LAES-ARC, and LAES-HTHP.

Table 9 compares the performance of different LAES configurations studied in Liu et al.[8] and the work presented in this chapter. Based on the analysis of different thermodynamic cycles in the LAES system in Sections 3.1−3.3, the best performance of LAES-ORC, LAES-ARC, and LAES-HTHP systems is selected to be compared with the standalone LAES system. It is seen that the optimal LAES case with a two-stage compressor and a three-stage expander still has the best performance. However, the operational feasibility and economics of this optimal case can be questioned due to the large pressure ratios in compressors.

For the LAES systems with more feasible pressure ratios and available surplus compression heat (i.e., four-stage compressor and four-stage expander), the exergy efficiency of the discharging part in the system with an ORC is increased from 84.3 to 86.4%, while the exergy efficiency of the charging part is unchanged. The reason is that the power produced in the ORC leads to an increase in the total power production and therefore also an increase in the exergy efficiency in the discharging part. The RTE of the LAES system with an ORC is increased from 62.1 to 64.5%.

For the system with an ARC to utilize the surplus compression heat, the RTE of the system is increased from 62.1 to 63.5% compared to the standalone LAES system with the same number of compression and expansion stages. It is

Table 7. Properties of Working Fluid Candidates for the HTHP

fluids	chemical formula	T_{cr} (°C)	p_{cr} (bar)	T_{nb} (°C)	ODP	GWP_{100}
R600a	C_4H_{10}-2	134.83	36.54	−11.79	0	3
R600	C_4H_{10}-1	152.00	37.96	−0.51	0	4
R1234ze	$C_3H_2F_4$-N2	150.12	35.37	9.80	0	1
R245fa[a]	$C_3H_3F_5$	154.05	36.40	14.90	0	858
R1233zd	$C_3H_2CLF_3$-N1	165.60	35.70	18.30	0.00034	1
R601a	C_5H_{12}-2	187.25	33.34	27.88	0	4
R601	C_5H_{12}-1	196.45	33.75	36.06	0	4
n-hexane	C_6H_{14}	234.75	30.32	68.73	0	<6

[a]Even though R245fa has a large GWP_{100}, it is selected as a traditional refrigerant to be compared with other low-GWP refrigerants in this work.

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Table 8. Performance of the LAES System with HTHP Using Different Working Fluids

fluids	$\dot{m}$ (kmol/h)	$\dot{Q}_{eva}$ (MW)	T_{eva} (°C)	$\dot{Q}_{con}$ (MW)	T_{con} (°C)	T_{H5} (°C)	$\dot{W}_{comp}$ (MW)	$\dot{W}_{ch}$ (MW)	$\dot{W}_{dc}$ (MW)	COP	RTE (%)
R600a	255.03	0.59	90.49	0.71	129.19	120.92	0.11	10.99	6.14	6.45	55.89
R600	507.89	1.51	78.60	1.96	149.10	141.97	0.45	11.33	6.52	4.35	57.55
R1234ze	368.05	1.28	81.77	1.61	142.80	136.09	0.33	11.21	6.42	4.88	57.24
R245fa	442.67	1.59	77.64	2.08	151.00	143.99	0.48	11.36	6.56	4.33	57.71
R1233zd	532.91	2.01	72.11	2.73	161.60	155.19	0.72	11.59	6.76	3.79	58.27
R601a	246.31	1.13	79.33	1.40	142.59	132.57	0.27	11.15	6.35	5.19	56.96
R601	225.54	1.14	79.34	1.41	142.69	132.67	0.27	11.15	6.35	5.21	56.99
n-hexane	78.36	0.50	79.34	0.59	129.05	119.02	0.09	10.97	6.11	6.55	55.68

Table 9. Comparison between Different LAES Configurations Investigated by Liu et al.[8] and the Work Presented in This Chapter

KPIs	optimal LAES	LAES	LAES-ORC	LAES-ARC	LAES	LAES-HTHP
compression stages	2	4	4	4	6	6
expansion stages	3	4	4	4	3	3
additional cycle			ORC	ARC		HTHP
$\dot{W}_{ch}$ (MW)	15.00	12.50	12.50	12.06	10.88	11.59
$\dot{W}_{dc}$ (MW)	10.00	7.76	8.07	7.65	5.94	6.76
η_{LY} (%)	86.21	86.57	86.57	89.60	85.98	85.98
$\eta_{\dot{E}_{ch}}$ (%)	88.31	86.67	86.67	84.45	86.06	84.44
$\eta_{\dot{E}_{dc}}$ (%)	86.65	84.28	86.35	86.76	80.29	85.91
RTE (%)	66.65	62.13	64.54	63.47	54.56	58.27

worth noting that a part of surplus compression heat is required to provide heat duty to the reboiler of the distillation column, and pump work is also needed to pressurize the working fluid (ammonia-water solution) in the ARC. Thus, the decreased compression heat from the charging part and the additional pump work in the ARC result in a reduced exergy efficiency in the charging part of the LAES-ARC system compared to the system without an ARC. The exergy efficiency of the discharging part is increased due to the reduction of the exergy input associated with reduced compression heat.

HTHPs are considered to utilize medium-grade compression heat in the LAES system. It is observed that the exergy efficiency is reduced for the charging part of the LAES-HTHP system. This is due to the fact that the compressor in the HTHP provides heat to upgrade the compression heat in the LAES system by consuming work, which will be added to the work consumed in the charging part. Thus, the power consumption in the charging part is increased and, as a result, the exergy efficiency of the charging part is decreased. However, the enhanced compression heat results in an increase in the work production in the discharging part. The increased work production can justify the increased work consumption in the LAES-HTHP system. The RTE of the LAES-HTHP system reaches 58.3%, which is an increment of 3.7% points compared to the system without an HTHP.

4. CONCLUSIONS

It is known that there is surplus of compression heat at different temperature levels available in the LAES with different numbers of compression stages for realistic operations, meaning practical pressure ratios of current compressor technologies. Three thermodynamic cycles have been adopted in different LAES configurations, depending on the quality (temperature) of the heat. The three thermodynamic cycles are the ORC, ARC, and HTHP. The ORC and ARC are considered for the utilization of relatively high temperature compression heat in an LAES system with a four-stage compressor and a four-stage expander, while the HTHP is utilizing medium-grade compression heat in an LAES system with a six-stage compressor and a three-stage expander. The three processes are simulated in Aspen HYSYS and optimized by using a PSO algorithm.

The main conclusions from this work are the following:

- The ORC is a promising method to utilize relatively high temperature compression heat, increasing the RTE by 2.4% points.
- The ARC delivers cold energy to the cold box and therefore improves the liquid yield compared to the other alternatives. The RTE is, however, 1.1% points below the LAES with ORC.
- The HTHP also improves the system efficiency by utilizing medium-grade compression heat. However, temperature limitations for available working fluids forced us to use six compression stages. The use of HTHP was not able to compensate for the lower RTE value of having six compression stages.
- With additional thermodynamic cycles to improve the RTE of the system, the LAES is increasingly competitive in terms of energy efficiency compared to other energy storage technologies.

The process efficiency of the LAES system with additional cycles could be further improved by an optimal heat integration, which has not been the scope of this work. However, it was found that the LAES system with additional thermodynamic cycles studied in this work is not competitive from an energy efficiency point of view with a two-stage compressor and a three-stage expander, where there is no surplus compression heat. Since, however, the pressure ratios of compressors in the thermodynamically optimal LAES system with a 2-stage compressor and 3-stage expander are relatively large, the practical feasibility in real applications can be questioned. Thus, there are several challenges that can be considered in future work:

- Cost evaluation of the optimal LAES case with three compressor stages and two expander stages, including a critical assessment of current and emerging compressor technologies with respect to high compression ratios.
- Cost evaluation of an LAES system with ORC, ARC, and HTHP should be conducted to identify the most profitable thermodynamic cycles for the LAES.
- Other cycles, such as the Kalina Cycle and the Brayton cycle, can also be evaluated for the utilization of surplus

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compression heat and thereby improve the performance of the overall system.

■ ASSOCIATED CONTENT

Supporting Information

The Supporting Information is available free of charge at https://pubs.acs.org/doi/10.1021/acs.iecr.2c04059.

Stream data for the heat recovery cycles studied in this paper; ORC, ARC; and HTHP (PDF)

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Author Contributions

Z.L. has been the main researcher in this project, identifying the objects to study, developing simulation models, performing optimization studies, and drafting the manuscript. T.H. contributed with her knowledge about Absorption Refrigeration Cycles and developed the design for this part. D.K. contributed with his knowledge about low temperature processes, advanced thermodynamics, and process knowledge, and he also contributed to the structure of the manuscript. T.G. has acted as the supervisor for the first author, and he has given numerous suggestions to the work. He has also contributed to the final manuscript.

Notes

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■ NOMENCLATURE

Symbols

$\dot{E}$	Exergy rate (kW)
$\dot{H}$	Enthalpy rate (kW)
$\dot{m}$	Mass flow rate (kg/s)
p	Pressure (bar)
$\dot{Q}$	Heat rate (kW)
$\dot{S}$	Entropy rate (kW/°C)
T	Temperature (°C)
VF	Vapor fraction
$\dot{W}$	Power (kW)
w	Specific power (kW/kg)
x	Fraction (−)
η	Efficiency (%)
ΔT_{min}	Minimum heat transfer approach temperature (°C)

Subscripts

air	Air
ARC	Absorption Refrigeration Cycle
c	Cold energy recovery cycles
ch	Charging part
cr	Critical
cryotur	Cryo-turbine
comp	Compression part or compressor
con	Condenser
dc	Discharging part
eva	Evaporator
fa	Air feed
HTHP	High Temperature Heat Pump
h	Heat recovery cycle
in	Inlet
liq	Liquid air
LMTD	Logarithmic mean temperature difference
LY	Liquid yield
MSHX	Multistream heat exchanger
nb	Normal boiling point
net	Net power output
out	Outlet
ORC	Organic Rankine Cycle
pump	Pump
RT	Round-trip efficiency
tur	Turbine

Acronyms

ARC	Absorption Refrigeration Cycle
CAES	Compressed Air Energy Storage
COP	Coefficient of Performance
ETE	Exergy Transfer Effectiveness
GWP	Global Warming Potential
HTHP	High Temperature Heat Pump
KC	Kalina Cycle
KPI	Key Performance Indicator
LAES	Liquid Air Energy Storage
LMTD	Logarithmic Mean Temperature Difference
LNG	Liquefied Natural Gas
MCFC	Multi-component Fluid Cycle
ODP	Ozone Depletion Potential
ORC	Organic Rankine Cycle
PHES	Pumped Hydroelectrical Energy Storage
PSO	Particle Swarm Optimization
RTE	Round-Trip Efficiency
VCRC	Vapor Compression Refrigeration Cycle

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