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Optimization and analysis of different liquid air energy storage configurations

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ABSTRACT

Energy storage technologies are required to ensure stability of energy systems when the share of renewable energy forms (wind and solar) is increasing. Liquid air energy storage (LAES) is a promising technology for storing electricity with certain advantages, such as high energy density and being geographically unconstrained. However, one drawback of a standalone LAES is the relatively low round-trip efficiency (RTE). In this work, the performance of a standalone LAES system with different number of compression and expansion stages is studied. All cases are optimized by using a particle swarm optimization (PSO) algorithm. The optimal results show that the highest RTE of 66.7% is obtained when there is a 2-stage compressor and a 3-stage expander in the LAES system. When the number of compression stages is fixed, the highest RTE is obtained when hot and cold streams have close to parallel temperature profiles in the preheaters of the expansion section.

1. Introduction

Due to the concern of climate change, the greenhouse gas emissions should be reduced by at least 40% by 2030 compared to 1990 (European Commission 2014). Therefore, our energy systems tend to adopt lower-carbon energy sources, such as renewables. It is predicted that the share of renewable energies like solar, wind and geothermal in the total energy consumption will rise to 20-60% by 2050 compared to 5.7% in 2020 (British Petroleum 2021). In contrast, the share of fossil fuels will be reduced from 83% in 2020 to somewhere in the range 20-70% by 2050. However, this transformation of energy systems may lead to a critical issue related to instability on the power supply side. Since the availability of renewable energies like solar and wind depends on time and weather, the power generated from these energy sources is intermittent and unpredictable.

Thus, additional measures should be taken to maintain the stability of energy markets. Energy storage technologies represent an effective measure to handle the problem caused by unpredictable renewable energy sources. The principle of energy storage technologies is that available energy is collected and stored, and then provided in different forms when energy is demanded. Energy storage technologies are also able to store unstable supplied energy and smoothen variations in supply (typical for renewable sources) and demand (short-term or mid-term variations). Thus, energy storage technologies are promising in future energy markets.

Some energy storage processes have been investigated and proven to be highly efficient, such as batteries (Zakeri and Syri, 2015), pumped hydroelectric energy storage (PHES) (Rehman, Al-Hadhrami, and Alam, 2015), and compressed air energy storage (CAES) (Arabkoohsar et al., 2015). Liquid air energy storage (LAES) (Damak et al., 2020) is a promising energy storage technology that is limited by its low round-trip efficiency (RTE). These four energy storage technologies are suitable for different scenarios depending on their costs and efficiencies. It is worth noting that batteries are the most efficient alternative for storing energy, however, the cost of batteries is quite high, so they have mostly been used for small-scale energy storage (Zakeri and Syri, 2015). Currently, larger batteries have been investigated, which expands the range of applications. PHES, CAES and LAES can be considered for large-scale energy storage. Since energy densities for PHES and CAES are relatively low, large storage space is required for these two technologies. Two reservoirs at different levels and considerable sizes are needed for PHES, and suitable sites for PHES are at locations with adequate fresh water supply. Above ground tanks and underground caverns are used to store compressed air in CAES systems, and caverns are commonly adopted for economic considerations. This means that CAES tends to be located in remote areas far away from other energy conversion processes. The geographical constraints for PHES and CAES increase the

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Nomenclature

Symbols

$\dot{E}$	Exergy (kW)
$\dot{m}$	Mass flowrate (kg/s)
p	Pressure (bar)
T	Temperature (°C)
VF	Vapor fraction
$\dot{W}$	Power (kW)
w	Specific power (kJ/kg)
η	Efficiency (%)
ΔT	Minimum heat transfer approach temperature (°C)

Subscripts

air	Air
comp	Compression part or compressor
Con	Condensation
cryotur	Cryo-turbine
dir	Direct expansion in the discharging part
eva	Evaporation/Evaporator
exp	Expansion part or expander
feed	Feed stream
h	Thermal oil (working fluid in the hot thermal energy recovery cycle)
in	Inlet
liq	Liquid air
LY	Liquid yield
M	Number of expansion stages
N	Number of compression stages
net	Net power output
ORC	Organic Rankine Cycle
out	Outlet
pre	preheater
pump	Pump
rec	Cold thermal energy recovery part
RT	Round-trip efficiency
tur	Turbine

Acronyms

CAES	Compressed Air Energy Storage
LAES	Liquid Air Energy Storage
LMTD	Logarithmic Mean Temperature Difference
LNG	Liquefied Natural Gas
MTD	Minimum temperature difference
ORC	Organic Rankine Cycle
PHES	Pump Hydroelectrical Energy Storage
PSO	Particle Swarm Optimization
RTE	Round-Trip Efficiency

operating complexity and require extra cost for energy transmission. In contrast, the working fluid for the LAES system is liquid air with a high energy density, thus, the storage space is considerably reduced compared to PHES and CAES. Typical energy density values for LAES, CAES and PHES are 97 Wh/kg, 45 Wh/kg and 1 Wh/kg, respectively (Rahman et al., 2020, Aneke and Wang, 2016). Thus, the location of LAES systems is flexible enough to be placed near other energy conversion processes, which is a significant advantage of LAES over other energy storage technologies.

LAES has attracted more focus in recent years due to its characteristics of compact configurations and flexible locations. It is worth noting that the round-trip efficiency (RTE), which is defined as the ratio between the recovered energy in the discharging process and the consumed energy in the charging process, is commonly used to evaluate and compare various energy storage technologies. Highview Power is committed to developing liquid air energy storage, and the first pilot plant of an LAES with a storage capacity of 2.5 MWh was built by the University of Birmingham for academic research (Highview Power 2022). A commercial-scale plant with a storage capacity of 300 MWh is underway and will be located at Trafford Energy Park near Manchester. Guizzi et al. (Guizzi et al., 2015) proposed an LAES system based on the Claude cycle for air liquefaction. The key feature of their concept is that hot and cold thermal energy are exchanged between the charging and discharging parts. Compression heat from the charging part can be used to preheat air before expansion in the discharging part. Likewise, cold energy from regasification can be used to partly liquefy air. Such heat exchange requires energy storage since charging and discharging operate at different times, however, it achieves an RTE for the system of 54.4%.

Based on the above arguments and using the concept of Guizzi et al. (Guizzi et al., 2015), the objective of this work is to consider optimization opportunities related to the configuration of LAES systems focusing on the charging part (compression), the discharging part (expansion) and the hot thermal energy storage and recycle system. The cold thermal energy storage and recycle system has been previously studied in our group and will not be repeated here. Energy efficiency will be optimized and measured by various performance indicators. Cost issues are, however, regarded to be outside the scope of this work.

Morgan et al. (Morgan et al., 2015) used a three-stage turbine to replace the single cold turbine before the air is sent to a phase separator in their process model of the LAES. In addition, the compression heat was also recovered and used to preheat air in the discharging process. Thus, the liquid yield of air is increased and an RTE of 57% was obtained. Liu et al. (Liu et al., 2020) simulated the LAES system with different configurations in the charging and discharging processes, see Fig. 1. The number of compression stages was varied in the range of 2-6, and the number of expansion stages was varied between 3 and 6. Thermal oil is used to transport heat from the compression section to the expansion section, and the flowrate of thermal oil depends on the number of compression stages. Depending on the flowrate of thermal oil, different situations will occur in the expansion section. For high flowrates of thermal oil relative to the flowrate of air in the discharging part, the compression heat is not fully utilized. For low flowrates of thermal oil, the minimum approach temperature is shifted to the cold end of the preheaters, and the outlet temperature of air is reduced. The optimal situation is when the flowrate of thermal oil makes the temperature profiles in the preheaters parallel. The LAES system with 2-stage compression and 3-stage expansion had the largest RTE. In addition, Liu et al. (Liu et al., 2022) also tried new working fluids and novel configurations for the cold thermal energy recovery cycles. The LAES system with a 4-stage compressor and a 4-stage expander was studied. Multi-component fluid cycles (MCFCs) and Organic Rankine Cycles (ORCs) were used to transfer the cold regasification energy from the discharging process to the charging process. The optimization results of the proposed process models indicate that the LAES system with dual MCFCs has the best performance among 6 cases that were studied, and the RTE can be further improved from 62.4% to 64.7% by reducing minimum temperature differences in high-temperature heat exchangers from 10°C to 5°C (Liu et al., 2022).

In addition, the utilization of compression heat that is wasted in the discharge mode has been investigated to further increase the efficiency of a standalone LAES system. Peng et al. (Peng et al., 2018) suggested that around 2/5 of the compression heat could not be used in the discharging process and is therefore wasted. Thus, an ORC was adopted to utilize this part of compression heat to produce extra power and thereby increase the RTE of the system. Two cases related to different heat sink

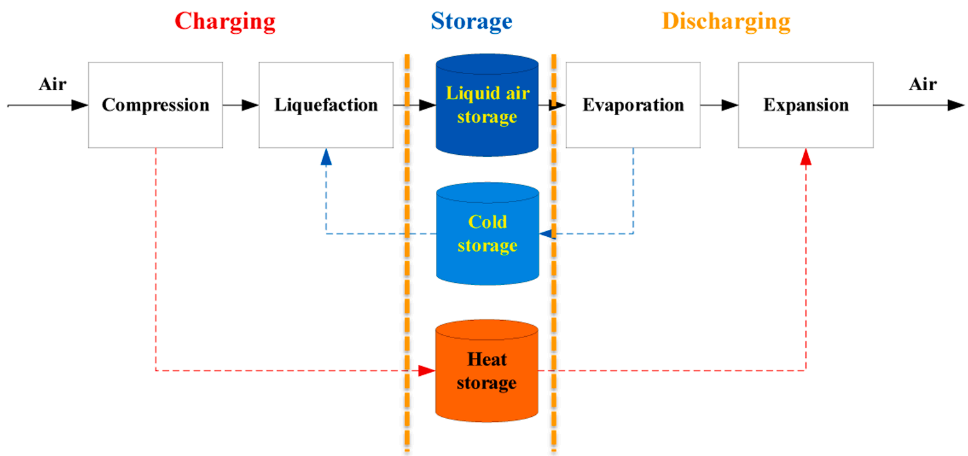

Fig. 1. Simplified block diagram of a Liquid Air Energy Storage System with charging, discharging and storage of both liquid air and thermal energy recovery fluids.

temperatures were considered for the ORCs. One at ambient temperature and the other at sub-ambient temperature. The sub-ambient temperature was achieved by consuming part of the compression heat through an Absorption Refrigeration Cycle. It was observed that the LAES system with the ORC operating at ambient temperature had a higher RTE of 62.7% compared to the LAES with the ORC operating at sub-ambient temperature. Zhang et al. (Zhang et al., 2020) studied cascaded storage of compression heat to increase the temperature level of the hot thermal energy storage. In addition, two LAES systems, one with an ORC and the other with a Kalina cycle to utilize part of the unused enhanced compression heat to generate additional power, were tested and compared. Results indicated that both the ORC and the Kalina cycle can improve the performance of the LAES. By comparing the LAES with ORC and the LAES with a Kalina cycle, the system with an additional ORC, where n-Pentane is the working fluid for the cycle, had better performance (RTE of 56.9%) than the LAES with a Kalina cycle (56.1%).

In addition to the mentioned studies on the configuration improvements of a standalone LAES system, various integration strategies between the LAES and external hot and cold thermal energy sources have proven to be effective methods for enhancing the performance of the LAES. Li et al. (Li et al., 2014) studied the integration with a nuclear power plant as a way to increase the efficiency of LAES system. The high temperature steam (287°C) from the nuclear power plant was used to preheat air in the discharging process, and thereby increasing the power generated from the turbines. The RTE was increased from 34% to 71%. Cetin et al. (Cetin et al., 2019) considered to use geothermal heat to increase the turbine inlet air temperatures in the LAES system. The geothermal heat is supplied at temperatures in the range of 150-250°C, and the corresponding RTE of the LAES system is reported to be between 40% and 55%. Qi et al. (Qi et al., 2020) investigated the integration between the LAES system and an LNG regasification process. When energy is demanded, both the discharging process in the LAES system and the LNG regasification process are activated and used to generate power as two separate systems. The LAES discharging process produces power by expanding pressurized air, and the LNG regasification process produces power by operating as heat sink for an ORC system. When excess energy is available and needed to be stored, the LNG is first sent to the cold box and then to aftercoolers in the compression part of the LAES system. After delivering part of the cold regasification energy to the LAES, LNG is used as a heat sink in an ORC. The novel integrated system has an extremely high RTE of 129.2%; larger than 100% because of the “free” cold energy from LNG regasification. Park et al. (Park et al., 2020) also considered to use the cold energy from an LNG regasification process to liquefy air. It was found that the specific energy consumption is reduced, and the liquid air yield is increased with increasing number of compressor stages, however, the economic analysis of the cases indicated that the minimum total cost per liquid air is obtained when there is a 4-stage compressor in the liquefaction process. In addition, liquid air has been framed as an emerging energy vector towards carbon neutrality (Qi et al., 2022).

In a standalone LAES system, when the number of compression stages is greater than or equal to the number of expansion stages, part of the compression heat is wasted rather than utilized (Liu et al., 2020). To improve the performance of LAES systems, the wasted part of compression heat can in certain cases be utilized to produce power. Organic Rankine Cycles (ORCs) (Yu et al., 2016), which represent an effective heat to work technology, are applied in this study to utilize this wasted part of the compression heat. The temperature difference between thermal oil and ambient temperature is the driving force for the ORC.

It is well known that power consumption and production will vary with different number of compressors and expanders when the outlet pressures of compression and expansion are fixed. Most of the existing literature discuss the expansion section and compression section separately for the LAES system. However, the expansion section is strongly influenced by the compression section when there is a hot thermal energy storage between these two sections in the LAES. The amount and temperature of thermal oil are changing with different number of compressor stages, which will directly affect the inlet air temperature to expanders and the power output. Thus, there is a lack of studies related to the complex relations of different number of compression and expansion stages and how this affects the performance of the LAES system. This paper focuses on the potential of system efficiency improvements for different configurations of the LAES system, where the term configuration relates to having different number of compression and expansion stages, as well as the potential use of ORC.

To have a fair comparison between these different configurations of the LAES system, all cases have been optimized to maximize their RTE by using a particle swarm optimization (PSO) method (Eberhart and Kennedy, 1995). Different configurations of the LAES system are optimized and analyzed in Section 5. It is worth noting that the corresponding pressure ratios of compressors and expanders are changing with different number of compression and expansion stages. The pressure ratios will be relatively high when the number of compression (or expansion) stages is small, which may affect the efficiencies of compressors (or expanders). However, this work emphasizes purely on a thermodynamic analysis, which is why mechanical limitations and cost issues related to pressure ratios and outlet temperatures of compressors and expanders will not be considered.

2. System description

Flowsheets of the liquid air energy storage system are illustrated in Figs. 2 and 3. The liquid air energy storage is commonly divided into charging, storage and discharging processes based on its operating mode. However, for a standalone LAES, the overall system can also be decomposed into three parts, which are the compression, the hot and cold thermal energy recovery cycles, and the expansion sections, according to the function of each part. In the compression section, air is compressed in stages by using available electricity. Air is cooled to 30°C after each compressor stage and the compression heat is collected and stored by the hot thermal energy recovery cycle. The high-pressure air is then sent to the cold thermal energy recovery part. It is precooled in the cold box by working fluids in cold thermal energy recovery cycles before it is expanded in the cryo-turbine to atmospheric pressure. The name of this turbine refers to the very low (i.e. cryogenic) temperature, and the unit is operating with a liquid inlet stream and a two-phase outlet stream. The cryo-turbine can produce additional refrigeration capacity and power, and thereby improve the efficiency of the system. The partially evaporated stream is then separated into a vapor stream, which is sent back to the compression part, and a liquid stream that is sent to storage. When there is a need for power, liquid air is first pumped to a high pressure before it is sent to evaporators, where liquid air is heated to be regasified by working fluids in cold thermal energy recovery cycles. After delivering the cold regasification energy, air is sent to the expansion part, where air is expanded through a series of turbine stages to generate power. There is a preheater before each expander (Heater 1-M) to increase the power generation.

The hot thermal energy storage between the compression and expansion parts is used to collect the compression heat and release it to increase the inlet air temperature to the turbine stages. The working fluid for the hot energy recovery cycle is Therminol 66 with a wide operating temperature range from -3 to 350°C. This fluid is widely used in the hot storage cycle for LAES systems (Tafone et al., 2018). Two cold thermal energy recovery cycles are used to transfer the cold regasification energy of liquid air to the compressed air. In our previous study (Liu et al., 2022), different cold thermal energy recovery cycles are proposed and compared to identify the most suitable cold cycles for the LAES when the configurations of the compression and expansion parts are fixed. It was found that two multi-component fluid cycles gave the highest performance (RTE) among six studied cases. Since the focus here is on configurations of the compression and expansion parts rather than working fluids, a slightly simpler set of working fluids with a marginal decrease in RTE was used in order to reduce the computational efforts. The cold thermal energy recovery cycles are therefore using a single-component cycle with methanol and a multi-component fluid cycle, which consists of 70 mol% propane, 20 mol% ethane and 10 mol% n-butane. These working fluids are operated at different temperatures. For different LAES processes in this work, the configuration and the components of cold thermal energy recovery cycles are the same, while the compression and expansion parts are changed by varying the number of compression and expansion stages.

The process flowsheet of the LAES system with N compression stages ($2 \leq N \leq 6$) and M expansion stages ($3 \leq M \leq 5$) is shown in Fig. 2. When the number of compression stages N is greater than or equal to the number of expansion stages M, there will always be a part of the compression heat that cannot be utilized in the expansion part. This is due to the fact that the amount of compression heat and the temperature of the air are reduced when the number of compression stages is increased. As a result, the temperature of the thermal oil is reduced. Despite the fact that the compressor duty is decreased with increasing number of stages, the repeated cooling of air will increase the flowrate of thermal oil since it must be split into more branches. In order to improve the performance of the system, ORCs are considered to utilize the unused part of compression heat. For the additional ORC, R152a ($C_2H_4F_2$ with critical temperature 113.3°C and critical pressure 45.2 bar) is chosen as the working fluid. The reason for selecting R152a as the working fluid in the ORC is that it is environment-friendly and non-toxic to humans, and it is also reported to be more efficient than several other working fluids, such as R134a, R143a, and R32 (Bellos and Tzivanidis, 2019). In addition, the critical temperature of R152a is within the temperature range of the compression heat in different LAES configurations. The critical pressure of R152a is relatively large and the saturation pressure at ambient temperature is only 3.7 bar, therefore, we can take advantage of the

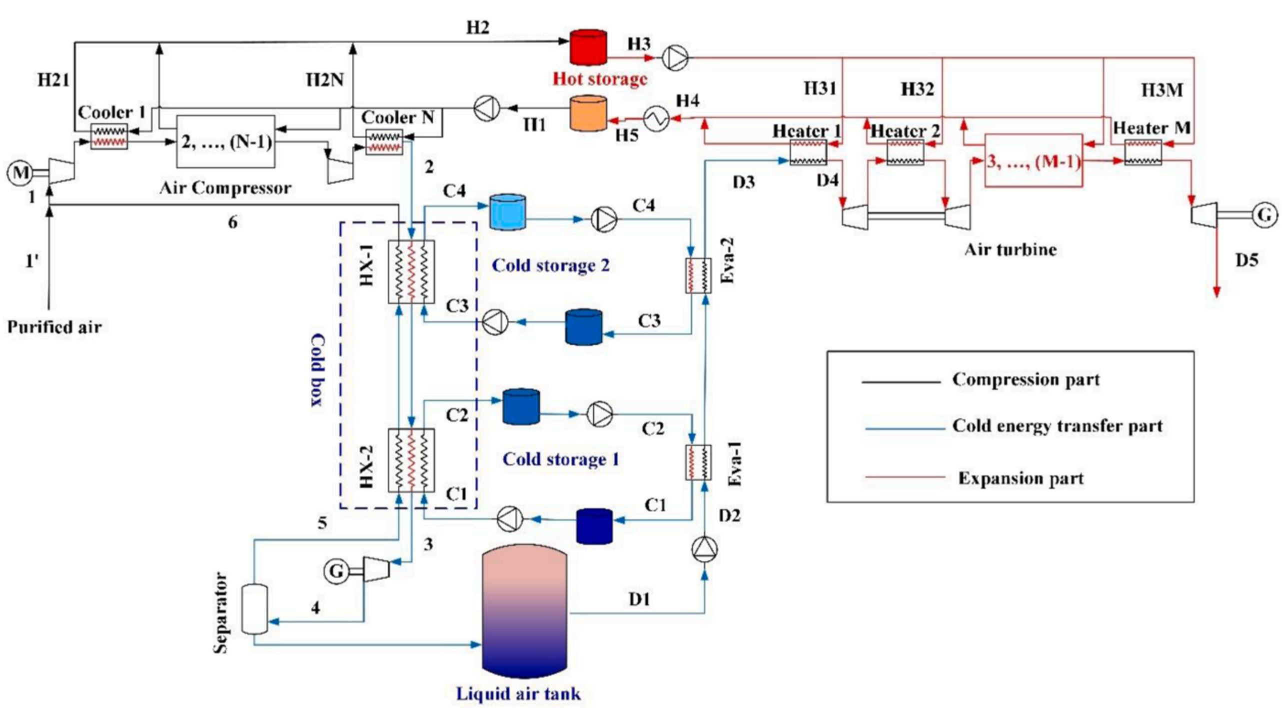

Fig. 2. Process flowsheet of the liquid air energy storage (LAES) system.

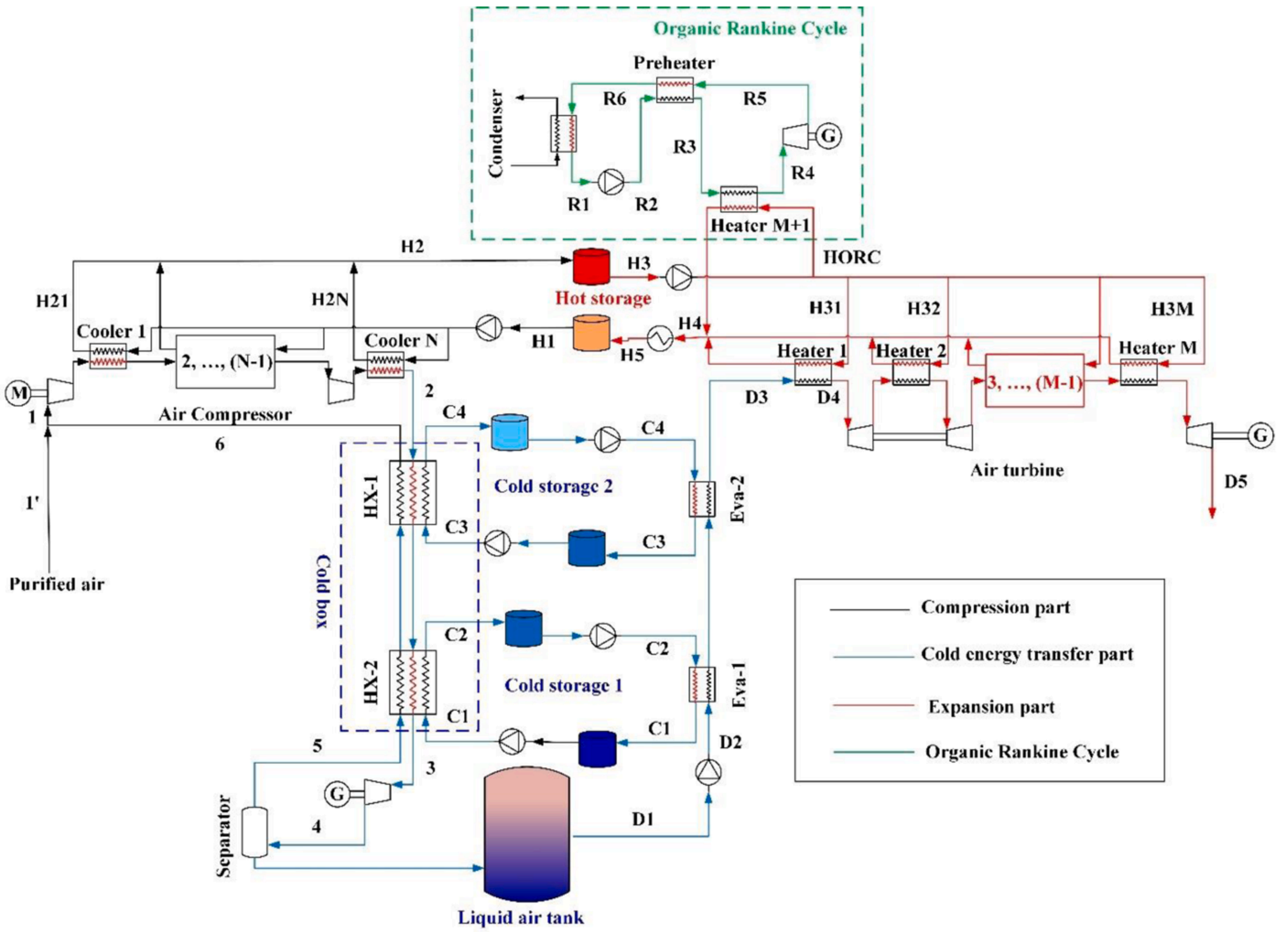

Fig. 3. Process flowsheet of the LAES system with an additional ORC.

large pressure drop during expansion. Fig. 3 illustrates the LAES system with an additional ORC, where the working fluid of the ORC is assumed to be cooled to 20°C by cooling water.

3. Process simulation

In this section, the assumptions and key parameter settings for main components during the simulation of the LAES system are provided. Although the air pre-treatment steps for the removal of CO_2, H_2O and other trace components are important for the operation of LAES systems, they are not included in our simulation and optimization studies, since the main focus is on the LAES configurations. The LAES system is modeled and simulated by Aspen HYSYS Version 10.0 (HYSYS, 2017) with the Peng-Robinson equation of state to calculate thermodynamic properties of process streams. It is assumed that the air feed consists of 78.82 mole% nitrogen, 21.14 mole% oxygen and 0.04 mole% argon. The mass flowrate of air feed is 61,520 kg/h, which is aimed at a 10 MW scale energy storage system, and air is supplied at 20°C and 1 bar. In this work, the configuration of the cold thermal energy recovery part is identical for all cases. This also applies to the working fluids of the two cycles, where methanol and a multi-component fluid are adopted to transfer the cold regasification energy of liquid air. As mentioned in Section 2, each compressor stage is followed by an aftercooler and a preheater is placed before each expander stage. Pressure drops and heat losses of heat exchangers, the phase separator and storage tanks are neglected in the simulation models. To simplify the thermodynamic analysis of the LAES system in this work, constant isentropic efficiencies for the compressors and expanders are used in the modeling of the LAES. The effect of different pressure ratios and internal stage pressures and temperatures in LAES systems with varying number of compression and expansion stages on the performance of compressors and expanders is not considered. The isentropic efficiencies of compressors and expanders are assumed to be 85% and 90%, respectively. Other simulation conditions for process units are given in Table 1.

4. Process evaluation, validation and optimization

This section introduces the key performance indicators (KPIs) that are used to evaluate the different process configurations, followed by a validation of the process model that is used against the results presented by Guizzi et al. (Guizzi et al., 2015). Finally, the optimization formulation is presented.

4.1. Process evaluation

Various key performance indicators (KPIs) are selected to evaluate the thermodynamic performance of different LAES configurations. As mentioned in Section 1, the most widely used parameter for energy

Table 1
Simulation conditions for the LAES configurations.

Design parameters	Unit	Value
Ambient temperature	°C	20
Ambient pressure	bar	1
Isentropic efficiency of compressors	%	85
Isentropic efficiency of expanders	%	90
Isentropic efficiency of cryo-turbine	%	75
Isentropic efficiency of pumps	%	80

storage technologies is the round-trip efficiency, which is commonly used for comparison with other technologies in the literature. The specific power consumption is a performance parameter to indicate the efficiency of the liquefaction process in the LAES system. In addition, exergy efficiency (such as the Exergy Transfer Effectiveness - ETE), which evaluates the quality of work and heat in a consistent way, is considered as a performance indicator for the charging and discharging parts of the LAES system.

The definition of round-trip efficiency (RTE) is shown in Eq. (1).

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

Here, $\dot{m}_{liq}$ and $\dot{m}_{comp}$ denote the mass flowrate of liquid air produced and air that is sent to compressors, respectively. w_{comp} and w_{tur} represent the specific work of compressors in the compression part and turbines in the expansion part, respectively. η_{LY} is the liquid yield of air. As mentioned in Section 2, the outlet stream of the cryo-turbine that is downstream of the cold box is split into a vapor and a liquid stream. The vapor stream is returned back to the compression part, which leads to a larger mass flowrate of air entering the compressors than the mass flowrate of liquid air. The definition of liquid yield of air is the ratio between the mass flowrates of liquid air produced and air compressed.

Specific power consumption (SPC) is the net work consumption in the charging part divided by the mass flowrate of liquid air produced, see Eq. (2).

$$SPC = \frac{\dot{W}_{net}}{\dot{m}_{liq}} = \frac{\sum \dot{W}_{comp} - \dot{W}_{cryotur}}{\dot{m}_{liq}} \tag{2}$$

Here, $\dot{W}_{comp}$ is the total work consumed in the compression part and $\dot{W}_{cryotur}$ is the expansion work produced by the cryo-turbine. The above-mentioned performance parameters evaluate the system in terms of energy efficiency.

However, changes in temperature, pressure, and composition of process streams also have significant effects on the performance of the system. This information cannot be revealed by energy performance parameters. Exergy analysis is a comprehensive method to include both the 1st and 2nd laws of thermodynamics. All variations in process streams, work and thermal energy in the system are considered in the concept of exergy. For process streams, exergy represents the maximum work that can be obtained by varying the stream temperature, pressure and composition in reversible processes to equilibrium with environmental conditions. The exergy of heat is a simple function of temperature through the Carnot factor, while work is pure (100%) exergy. Not including electrical, potential, kinetic, and nuclear exergies, the exergy of process streams consists of thermo-mechanical (or physical) exergy ($\dot{E}^{TM}$) and chemical exergy ($\dot{E}^{Ch}$) (Kotas, 2012), see Eq. (3).

$$\dot{E} = \dot{E}^{TM} + \dot{E}^{Ch} \tag{3}$$

Thermo-mechanical exergy is the maximum work produced in ideal processes when the stream is taken to a situation that has the same temperature and pressure as the environment, as is given by Eq. (4).

$$\dot{E}^{TM} = \dot{H}(T,p) - \dot{H}(T_0,p_0) - T_0[\dot{S}(T,p) - \dot{S}(T_0,p_0)] \tag{4}$$

Chemical exergy is the maximum work obtained in ideal processes when the stream composition is changed to equilibrium with the composition of the environment under ambient conditions. The effect of chemical exergy in the LAES system is relatively small since there are no chemical reactions; only the separator after the cryo-turbine has compositional changes, and the LAES system experiences minor changes in chemical exergy.

Exergy Transfer Effectiveness (*ETE*) is adopted to calculate the exergy efficiency of the system, since this is a suitable performance parameter for energy conversion processes. The concept of *ETE* was first proposed by Marmolejo-Correa and Gundersen (Marmolejo-Correa and Gundersen, 2015) who only considered the thermo-mechanical exergy. The *ETE* was further developed to include chemical exergy by Kim and Gundersen (Kim and Gundersen, 2018). The *ETE* is defined as the ratio of exergy sinks and exergy sources, and thereby focusing, as the name indicates, on exergy transfer, see Eq. (5).

$$\eta_{\dot{E}} = ETE = \frac{\sum ExergySinks}{\sum ExergySources} \tag{5}$$

Any increment in exergy (produced exergy) in the process is regarded as exergy sinks, and exergy decreases (consumed exergy) are exergy sources. As mentioned in Section 2, the LAES system is decomposed into three parts: compression, hot and cold thermal energy recovery, and expansion sections. Thus, exergy efficiencies of the three parts are evaluated to reveal the performance of each part. The exergy efficiency of the compression part $\eta_{\dot{E}_{comp}}$ is calculated by Eq. (6).

$$\eta_{\dot{E}_{comp}} = \frac{\dot{E}_{out,air,comp} + \dot{E}_h}{\dot{W}_{comp} + \dot{E}_{feed,air}} \tag{6}$$

Here, $\dot{W}_{comp}$ denotes the work consumed by compressors in the compression part. $\dot{E}_{out,air,comp}$, $\dot{E}_h$ and $\dot{E}_{feed,air}$ represent the thermo-mechanical exergy of outlet air of the compression part, working fluid in the hot thermal energy recovery cycle (thermal oil), and the air feed. The thermo-mechanical exergy of streams, which can be calculated by Eq. (4), is obtained by applying Visual Basic codes in Aspen HYSYS simulations as proposed in the work of Abdollahi-Demneh et al. (Abdollahi-Demneh et al., 2011). Similar to exergy efficiency for the compression section, exergy efficiencies of the cold energy recovery $\eta_{\dot{E}_{rec}}$ and expansion section $\eta_{\dot{E}_{exp,dir}}$ or $\eta_{\dot{E}_{exp,dir+ORC}}$ are given by (Eqs. 7, 8 and 9). $\eta_{\dot{E}_{exp,dir}}$ represents the exergy efficiency of the expansion part that only considers the multistage turbine, while $\eta_{\dot{E}_{exp,dir+ORC}}$ is the exergy efficiency of the expansion section when an additional ORC is part of the LAES system.

$$\eta_{\dot{E}_{rec}} = \frac{\dot{E}_{out,air,rec} + \dot{W}_{cryotur}}{\dot{E}_{out,air,comp} + \dot{W}_{pump}} \tag{7}$$

$$\eta_{\dot{E}_{exp,dir}} = \frac{\dot{W}_{tur,exp}}{\dot{E}_{out,air,rec} + \dot{E}_h} \tag{8}$$

$$\eta_{\dot{E}_{exp,dir+ORC}} = \frac{\dot{W}_{tur,exp} + \dot{W}_{tur,ORC}}{\dot{E}_{out,air,rec} + \dot{E}_h + \dot{W}_{pump,ORC}} \tag{9}$$

Here, $\dot{E}_{out,air,rec}$ represents the thermo-mechanical exergy of outlet air from the cold thermal energy recovery part and $\dot{W}_{tur,exp}$ is the work produced by the multistage turbine in the expansion part. $\dot{W}_{tur,ORC}$ and $\dot{W}_{pump,ORC}$ are the expansion work and pump work in the additional ORC.

4.2. Process validation

The validation of the simulation model that is used for the analysis and optimization of the various configurations in this work has been done by comparing with the work by Guizzi et al. (Guizzi et al., 2015). As described in our previous work (Liu et al., 2022), RTE values for different charging pressures in the flowsheet shown in Fig. 2 with N=2 compression stages and M=3 expansion stages have been compared with values presented in (Guizzi et al., 2015). Fig. 4 shows the corresponding RTE values, and the difference between the results is within 1.4%. This means that our process model has an acceptable accuracy.

4.3. Process optimization

Due to the thermal energy storage cycles, the charging and discharging parts are closely connected. The power recovery ratio is crucial for an energy storage technology, and measures to increase the RTE of

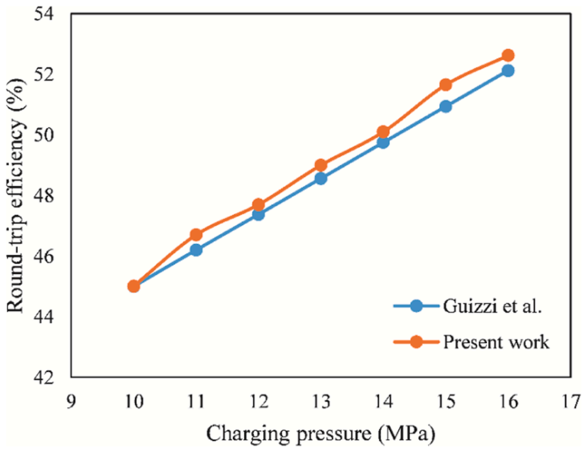

Fig. 4. Model validation by comparing RTE with numbers from Guizzi et al. (Guizzi et al., 2015).

the system are strongly required. This can be solved by using optimization algorithms in combination with thermodynamic analysis to improve the process performance.

The optimization of the LAES involves highly nonlinear and non-convex equations, which is challenging for deterministic optimization algorithms due to the complex thermodynamic property calculations. Surrogate models or simplified models can be considered as alternatives for the replacement of rigorous thermodynamic models when performing the optimization. However, accurate thermodynamic models are prerequisite for reliable results. It is also important to identify the best decision variables, which form a multi-dimensional space during optimization. Thus, it is computationally intensive to find global optimal solutions. The built-in optimizer in HYSYS is too weak to get satisfactory results. To deal with such challenges, stochastic search algorithms such as Particle Swarm Optimization (PSO) have been suggested used in energy intensive systems such as liquefaction processes and LNG regasification processes (Mofid et al., 2019, Cao et al., 2017, Sun et al., 2018). Compared to other stochastic search algorithms, the PSO is shown to be a technique that is computationally faster and provides better optimal solutions (Eghbal et al., 2011, Elbeltagi et al., 2005). Another advantage of using the PSO is that few adjusting parameters are required. A Genetic Algorithm (GA) is another popular evolutionary optimization method to be used in the optimization of chemical engineering problems (Jin and Lim, 2018). Unlike the GA, the PSO is mainly applied in continuous problems and does not require coding of variables and operations of crossover and mutation. Thus, the computing time is reduced, and the implementation is easier for the PSO compared to the GA. For these reasons, the PSO algorithm is applied in this study to optimize LAES systems.

The PSO algorithm is implemented in Matlab, where the relative convergence of variables or the fitness function values being unchanged after 45 iterations are termination conditions. Table 2 lists the parameters for the PSO algorithm. Acceleration factors affect the movement of particles and the inertia weight decides the convergence behavior of the PSO algorithm during optimization. These parameters were suggested in a recent work (Hamedi et al., 2018) related to the optimization of nitrogen rejection processes from natural gas using PSO. The PSO algorithm is connected to Aspen HYSYS, performing the simulation of each input generated by the algorithm. The inputs to Aspen HYSYS are generated randomly by the PSO algorithm, and certain inputs may lead to infeasible solutions and non-convergence of the Aspen HYSYS simulation. In this case, the objective function is set to a large value.

Table 2
Parameters for the PSO algorithm.

Parameters	Value
Number of particles	150
Global acceleration factor	1
Personal acceleration factor	1
Minimum inertia weight	0.5
Maximum inertia weight	1

$$\begin{aligned}
&\min_x -\eta_{RT} = -f(x) = -\frac{\dot{m}_{liq} w_{tur}}{\dot{m}_{comp} w_{comp}} \\
&\text{subject to } \Delta T_a \geq 10 \quad a = \{\text{aftercoolers } 1,2,\ldots,N, \text{preheaters } 1,2,\ldots,M\} \\
&\qquad \Delta T_b \geq 1 \quad b = \{\text{HX}-1,2, \text{Eva}-1,2\} \\
&\qquad VF_{ORC,pump,in} = 0 \\
&\qquad VF_{ORC,tur,in} = 1 \\
&\qquad x_{LB} \leq x \leq x_{UB}
\end{aligned} \tag{10}$$

Here, ΔT_a is the minimum temperature difference (MTD) of aftercoolers and preheaters, while ΔT_b is the minimum temperature difference (MTD) for heat exchangers and evaporators. $VF_{ORC,pump,in}$ and $VF_{ORC,tur,in}$ denote the vapor fraction of inlet streams of the pump and the turbine in the ORC. $\boldsymbol{x}$ represents decision variables in the LAES system, while $\boldsymbol{x}_{LB}$ and $\boldsymbol{x}_{UB}$ are the lower and upper bounds of the variables.

The objective function and the constraints of the optimization formulation are provided in Eq. (10). The purpose of optimizing the system is to identify the most promising option among different LAES configurations. The decision variables in this work include the pressure ratios of compressors and expanders, the outlet temperature of thermal oil from coolers, the outlet temperature of air and the recycled air stream from the cold box, and the molar flowrates and temperatures of the working fluids in the cold thermal energy recovery cycle. In the system with an additional ORC, the evaporation temperature and pressure and the molar flowrate of the working fluid are also selected as variables. The degrees of freedom for design are different in different cases, since the number of compression and expansion stages are varying in the LAES.

MTDs for heat exchangers are economic parameters trading off investment cost and operating cost. Transferring heat with large temperature differences increases irreversibilities in the plant, and these exergy losses are paid for by increased compressor work. In sub-ambient processes, these exergy losses depend on both the temperature difference and the absolute temperature level, making it more important to reduce temperature driving forces at lower temperatures. As a result, MTDs of aftercoolers and preheaters that are operating above ambient temperature are assumed to be 10°C, while MTDs for HX-1,2 and Eva-1,2 that are operating significantly below ambient temperature are assumed to be 1°C (Higginbotham et al., 2011). This is a simplification, but the focus of this work is not cost minimization, emphasis is on energy efficiency. It is shown in the literature that below ambient, UA_{max} is a better design specification than MTD, where UA is the lumped parameter of heat transfer coefficient and heat transfer area, also referred to as the heat exchanger conductance (Jensen and Skogestad, 2008, Austbø and Gundersen, 2015, Kim and Gundersen, 2017). In the LAES system with an additional ORC to utilize the unused compression heat, the inlet stream to the pump should be totally liquid and the inlet stream to the turbine should be totally vapor. Thus, the vapor fraction of the inlet stream to the pump and the turbine in the ORC should be 0 and 1, respectively. In reality, the outlet temperature of compressors should be considered as constraints during optimization. However, constraints related to the pressure ratio and outlet temperature of compressors and expanders are not considered in this thermodynamic analysis of the LAES system. The decision variables and their lower and upper bounds are listed in Table 3.

It is worth noting that the pressure ratios of compressors would be relatively high when the number of compression (or expansion) stages is

Table 3
Decision variables with lower and upper bounds.

Variables	Lower Bounds	Upper Bounds
Pressure ratio for compressors [a, b]	1	20
Pressure ratio for expanders [a, b]	1	10
Thermal oil temperature ($T_{H21\text{-}H2N}$) (°C) [a, b]	100	230
Cold box outlet air temperature (°C) [a, b]	-185	-165
Cold box outlet recycled air temperature (°C) [a, b]	-10	29
Working fluid operating temperature (higher) (°C) [a, b]	-90	-20
Working fluid molar flowrate (kmol/h) [a, b]	0	200
Working fluid operating temperature (lower) (°C) [a, b]	-186	-166
Working fluid molar flowrate in ORC (kmol/h) [b]	0.01	800
Working fluid evaporation pressure in ORC (bar) [b]	1.1	41
Working fluid evaporation temperature in ORC (°C) [b]	10	113.6

[a] variable bounds for the LAES when the number of compression stages is less than the number of expansion stages (N<M)
[b] variable bounds for the LAES with additional ORC when the number of compression stages is greater than or equal to the number of expansion stages (N≥M)

small in the LAES. In the ADELE and ADELE-ING projects (RWE Power, 2010, Zunft, 2015), the combination of an axial LP compressor and a radial HP compressor has been suggested to elevate the pressure of air from 1 bar to 100 bar with an outlet temperature of 600°C. Axial and radial compressors have been used in some aerospace applications with a pressure ratio of up to 40 and in some industrial applications with a pressure ratio of up to 30 (U.S. Department of Energy 2006). The upper bound for the pressure ratio of compressors is arbitrarily set to 20 in this work, which is achievable according to the literature review.

5. Results and discussion

In this work, the LAES system with different number of compression and expansion stages is optimized and compared. It is worth noting that an additional ORC is utilized only when the number of compression stages is greater than or equal to the number of expansion stages.

5.1. Performance of different LAES configurations

Fig. 5 shows the round-trip efficiency of the LAES system with different number of compression and expansion stages. It can be seen that when there is a 3-stage turbine in the expansion part, the RTE of the system reduces with increasing number of stages in the compression part. The trend for an LAES system with a 4-stage turbine is different. The RTE first increases when the number of compression stages is changed from 2 to 3, and then, the RTE decreases when the number of compression stages continues to increase from 3 to 6. For the system with 5 stages expansion, the RTE that has a maximum at 4 compression stages is increased with changing number of compression stages from 2 to 4, and then it is reduced with further increasing number of compression stages from 4 to 6. Optimization results show that there exists an optimal match between the number of compression stages and expansion stages. When the number of expansion stages is 3, 4 and 5, the highest RTE is obtained with 2, 3, and 4 compression stages in the LAES system, respectively. This is due to the fact that the hot storage cycle connects the compression and expansion parts in a standalone LAES system. With varying numbers of compression stages, the temperature and flowrate of thermal oil change, which influences not only the temperature of the air entering expanders but also the location of pinch points in preheaters before each stage of the expander. The overall best performance for the LAES system is with a 2-stage compression and a 3-stage expansion, and the highest RTE is found to be 66.7%. The temperature-entropy diagram for the energy storage and release mode of the three best matches between compression and expansion stages is provided in Fig. 6.

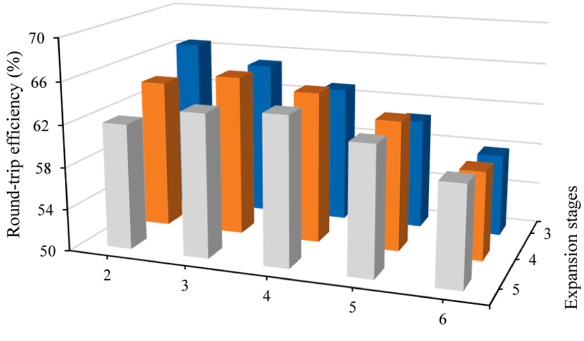

Fig. 5. Round-trip efficiency of the LAES for different configurations.

The properties of the thermal oil transferring heat of compression to the expansion section depend only on the configuration of the compression section. With increasing number of compressor stages, both the total compressor duty (and thereby the compression heat, see Table 4) and the outlet temperature of air from the compressor stages are reduced. This obviously also reduces the outlet temperature of thermal oil from the aftercoolers, as shown in Fig. 7. As mentioned in Section 2, even though the compressor duty is decreased with increasing number of stages, the repeated cooling of air increases the flowrate of thermal oil since it must be split into more branches. The effect of repeated cooling of air is more important than the reduced compressor duty. As a result, the flowrate of thermal oil increases with the number of compression stages, and this is also shown in Fig. 7.

Focusing on the expansion section, the fact that the flowrate and temperature of thermal oil have opposite trends with respect to the number of compression stages also means that they have opposite effects on the preheating of air in the expansion section, and therefore also on the power generation and round-trip efficiency. Thus, there is a trade-off between mass flowrate and temperature of thermal oil, which is why there exists an optimal match between the number of compression stages and the number of expansion stages. This optimal match has already been presented based on the results in Fig. 5. For 3, 4 and 5 expansion stages, the optimal number of compression stages is 2, 3 and 4, respectively. The best combination with 2 compression stages and 3 expansion stages has the highest RTE of 66.7%.

The easiest way to explain the trends in Fig. 5 is to consider a case with a fixed number of expansion stages, such as 5. Then the thermal oil must be split into 5 branches and sent to the air preheaters. With few

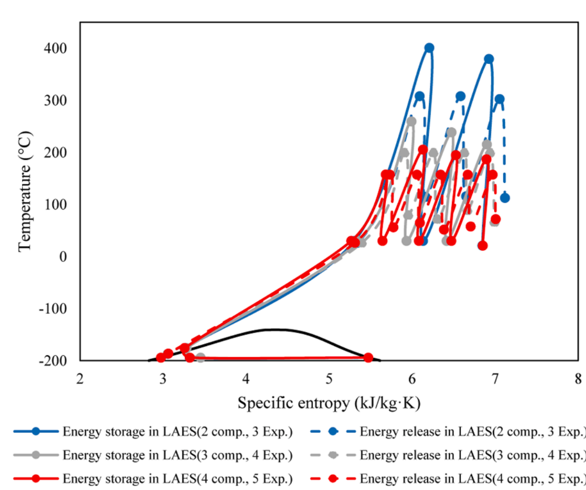

Fig. 6. Energy storage and release mode in a T-S diagram for the three best matches between compression and expansion stages.

Table 4
The compression heat recovered by the thermal oil for configurations with 3 expansion stages and 2-6 compression stages.

Compression stages		2	3	4	5	6
Recovered heat duty	GJ/h	56.88	47.78	48.11	46.49	41.37

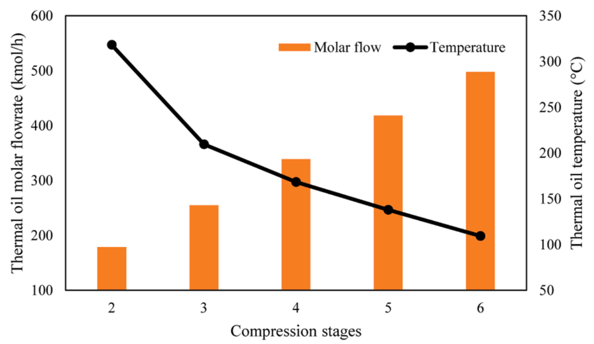

Fig. 7. Molar flowrate and temperature of thermal oil for configurations with 3 expansion stages and 2-6 compression stages.

compression stages, such as 2 or 3, the temperature of thermal oil is relatively high, but the flowrate of thermal oil is too low, causing a pinch in the cold end of the preheaters. This means that the expansion part is not able to take advantage of the high thermal oil temperature, although the situation is somewhat improved from 2 to 3 compression stages. With 4 compression stages, the composite curves in the preheaters are almost parallel, and the RTE reaches its maximum value for the case with 5 expansion stages. Increasing the number of compression stages further to 5 or 6 will result in a too high flowrate of thermal oil, and the pinch will move to the hot end of the preheaters. Air can now be preheated to a temperature that is $\Delta T_{\min}$ below the thermal oil temperature, however, this temperature is reduced due to the relatively large number of compression stages, and the large flowrate of thermal oil also means that the compression heat cannot be fully utilized.

Fig. 8 shows the scope for using ORC to produce power from compression heat that is not fully utilized in the expansion section. As already explained, such surplus heat will be available when the number of compression stages is equal to or larger than the number of expansion stages. The split of thermal oil between the preheaters and the additional ORC is shown as a function of number of compression stages for 3 expansion stages (Fig. 8(a)), 4 expansion stages (Fig. 8(b)) and 5 expansion stages (Fig. 8(c)). Unfortunately, the power from the ORC is not enough to compensate for the reduction in power production in the expansion section in these cases, partly caused by reduced thermal oil temperature. As a result, for the entire system (charging and discharging), the RTE will be reduced when the number of compression stages is increased beyond the optimal number for the given number of expanders (the 2-3, the 3-4, and the 4-5 matches). This explains the trends in Fig. 5.

The logarithmic mean temperature differences (LMTDs) of preheaters in different LAES configurations are listed in Table 5. When the LAES system has 3 stages of expansion, a relatively good match between the temperature profiles of thermal oil and air is obtained with 2 stages of compression. The flowrate of thermal oil is slightly less than required to have parallel profiles, and the pinch is in the cold end of the preheaters. As a result, the LMTD is 10.8°C in this case. It is observed that the LMTDs for the cases with surplus thermal oil are 10.3°C. Since $\Delta T_{\min}$ equals 10°C, an LMTD of 10.3°C indicates a case with close to parallel composite curves in preheaters. For the same reason, when the LAES has 4 or 5 stages of expansion, systems with 3 or 4 stages of compression have better performance compared to other combinations. The composite curves for the cases mentioned above are shown in Fig. 9. In Fig. 9

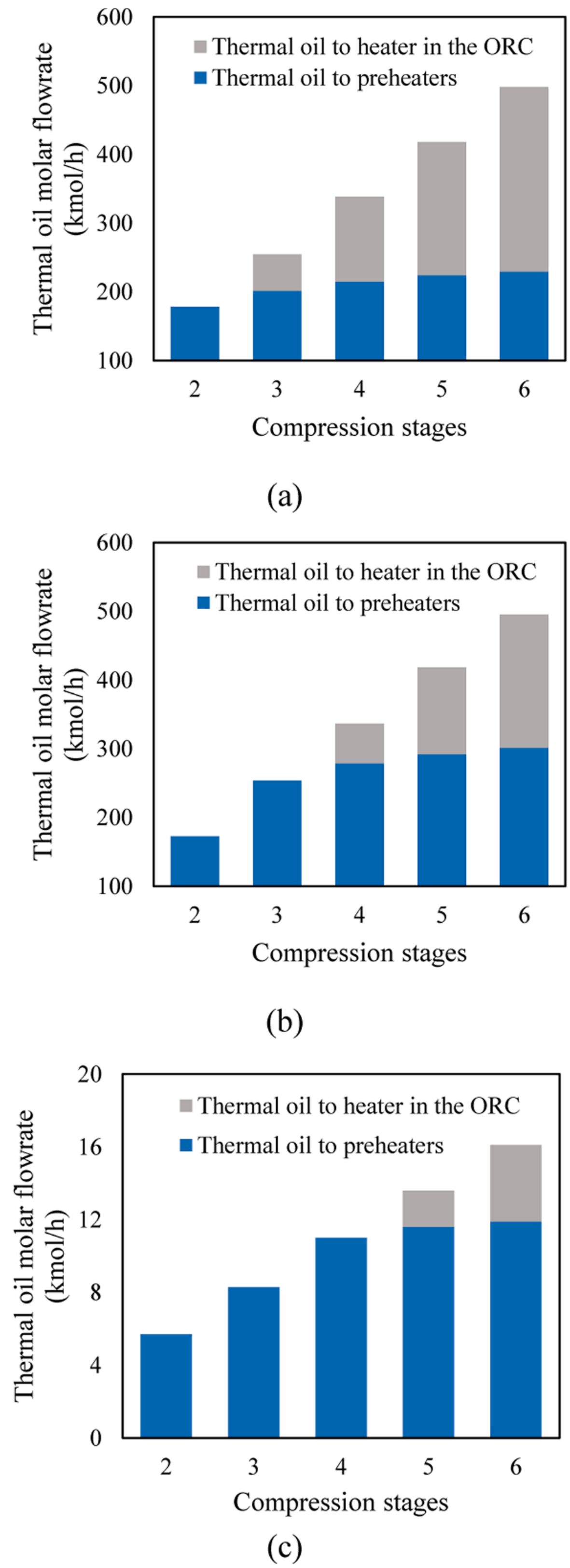

Fig. 8. Optimized distribution of thermal oil between the expansion preheaters and the heater in the ORC in different configurations of the LAES: (a) 3-stage turbine; (b) 4-stage turbine; (c) 5-stage turbine.

Table 5
Optimal values for some variables and key performance indicators in different LAES configurations.

Exp. stages	Comp. stages	pr_{comp}	pr_{exp}	$LMTD_{pre}$ (°C)	$LMTD_{heater,\ ORC}$ (°C)	η_{LY} (%)	SPC (kWh/t)	$\dot{W}_{ch}$ (MW)	$\dot{W}_{dc}$ (MW)	η_{RT} (%)
3	2	13.11	5.29	10.78	-	86.21	243.95	15.01	10.00	66.65
	3	5.24	4.98	10.27	32.85	86.19	204.79	12.60	8.19	64.99
	4	3.70	5.43	10.27	20.46	86.62	203.41	12.51	7.90	63.12
	5	2.86	5.45	10.27	18.11	86.64	195.93	12.05	7.30	60.60
	6	2.25	4.82	10.27	20.17	85.98	176.83	10.88	6.30	57.88
4	2	12.86	3.45	29.11	-	86.34	241.59	14.86	9.55	64.23
	3	5.31	3.37	11.62	-	86.26	207.08	12.74	8.32	65.32
	4	3.70	3.56	10.27	21.50	86.57	203.05	12.49	8.05	64.41
	5	2.86	3.57	10.27	18.24	86.65	195.96	12.06	7.52	62.40
	6	2.23	3.21	10.27	19.26	85.93	175.39	10.79	6.30	58.42
5	2	14.19	2.79	39.95	-	86.68	252.99	15.56	9.64	61.93
	3	5.34	2.65	21.42	-	86.29	207.85	12.79	8.13	63.61
	4	3.72	2.77	10.59	-	86.65	204.07	12.55	8.05	64.10
	5	2.89	2.79	10.27	18.26	86.65	197.88	12.17	7.57	62.22
	6	2.22	2.54	10.27	19.56	85.91	174.70	10.75	6.40	59.53

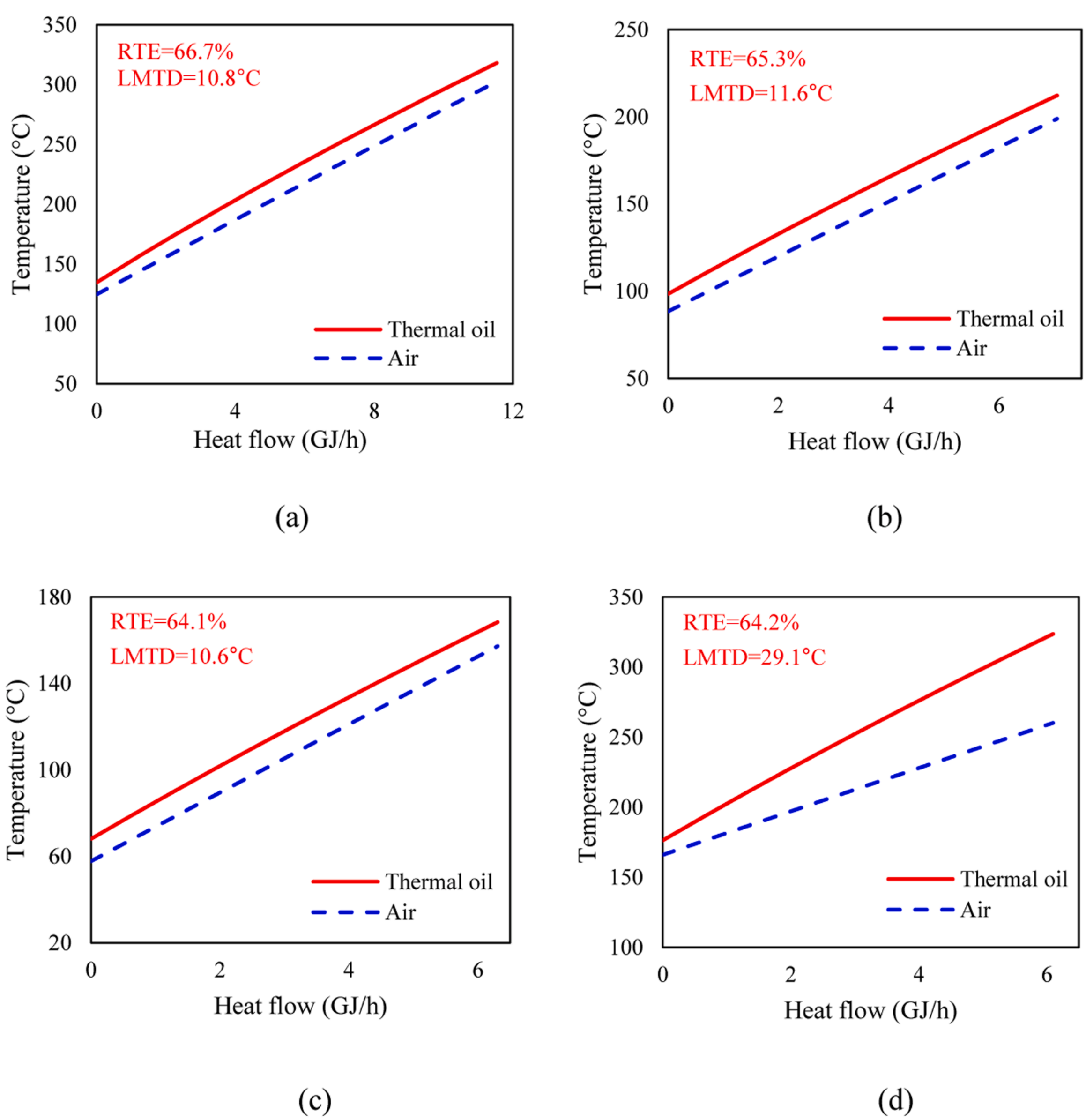

Fig. 9. Composite curves of preheaters in the LAES system: (a) 2-stage compressor and a 3-stage turbine; (b) 3-stage compressor and a 4-stage turbine; (c) 4-stage compressor and a 5-stage turbine; (d) 2-stage compressor and a 4-stage turbine.

(a)-(c), the composite curves are close to parallel as indicated by the listed LMTD values, and they have higher RTE values than other cases having the same number of expansion stages. Fig. 9(d), in contrast, illustrates an example of composite curves for preheaters when the system has insufficient amounts of thermal oil in the expansion part. In this particular case, the LAES system has a 2-stage compressor and a 4-stage turbine, and the LMTD of the preheaters is 29.1°C. As a result, the relatively large temperature difference between hot and cold streams lead to a poor performance and a lower RTE of the system.

In addition, when the number of compression stages is greater than or equal to the number of expansion stages, composite curves of preheaters in LAES systems with an additional ORC are similar to and slightly better than Fig. 9(a)-(c). The reason is that one part of the thermal oil flowrate is sent to the ORC, while the remaining part has a cooling curve that is parallel with the air preheating curve. The composite curves of the LAES system with a 4-stage compressor and a 4-stage turbine are used as an example and shown in Fig. 10. However, even if there is a good match between the thermal oil and air temperature profiles, the RTE is reduced when the number of compression stages is increased from 3 to 4 and the number of expansion stages is 4. This means that despite the power production in the ORC, there is a larger reduction in the power production in the expansion part compared to the case with 3 compression stages. The thermal oil temperature is reduced when increasing the number of compression stages from 3 to 4, and the RTE is reduced.

Optimal results for compression and expansion pressure ratios and key performance indicators (liquid yield, specific power consumption, exergy efficiency and round-trip efficiency) for different LAES configurations are listed in Table 5. It can be seen that optimal pressure ratios, the power consumption of compressors and SPCs are reduced with increasing number of compression stages for a given number of expansion stages in the LAES. This is because a near isothermal operation of compression requires near minimum power consumption. Moreover, the power production of the discharging part has the same trend as the power consumption of the charging part. This is because a higher expansion pressure results in more work produced, but the expansion pressure is constrained to be less than the compression pressure due to the cold thermal energy recovery cycles. The cold regasification energy from liquid air is needed to liquefy compressed air in the LAES system. In order to have driving forces for cold regasification energy transfer, the liquid air must be at a lower temperature than the compressed air. After the cold box, the optimal phase separation temperature is around -176°C for all cases, since the amount and temperature of cold thermal energy in the cold energy recovery part are insufficient. With almost

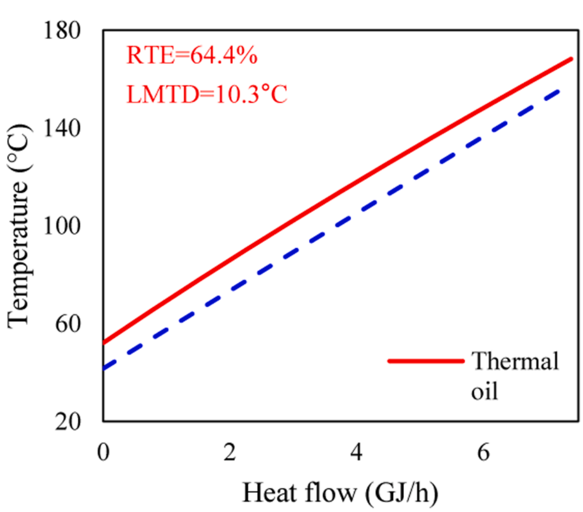

Fig. 10. Composite curves of preheaters in the LAES system with a 4-stage compressor and a 4-stage turbine.

constant temperature at the outlet of the cold box, compression pressure is the only factor that affects the liquid yield of air. A higher charging pressure increases the pressure drop in the cryo-turbine, with atmospheric pressure at the outlet. This results in a lower outlet temperature and reduced fraction of air in vapor phase, which improves liquid yield of air. For the entire system, the use of the hot thermal recovery cycle has decisive effects on the performance (the inlet temperature of air to expanders and the location of pinch points in preheaters), and the highest energy recovery ratio of the LAES (RTE) is obtained when the composite curves of hot and cold streams in preheaters are close to parallel. The major stream data in different optimized LAES configurations are provided in the Supplementary Material.

The detailed optimization results for an additional ORC in different LAES configurations are listed in Table 6. The heat source is stream HORC (thermal oil, see Fig. 3) in the LAES system. The condensation pressure of the ORC is 5.30 bar for all the cases, which is the saturation pressure of the working fluid at ambient temperature. When the number of compression stages is less than 6 in the LAES system, the evaporation pressure of the ORC reaches the upper bound, which is set to 41 bar, i.e. 90% of the critical pressure of the working fluid, resulting in the largest power output. When the number of compression stages is 6, the heat source temperature (thermal oil) is less than the saturation temperature of the ORC working fluid at critical pressure. Thus, the pressure and temperature of the working fluid in the ORC are less than for the other cases that have 2-5 compression stages. It is shown that the largest net work output of 605.01 kW is obtained when there is a 4-stage compressor and a 3-stage turbine in the LAES system. It is worth noting that the net power output in the ORC is affected by the physical properties of the working fluid (critical pressure and temperature). Thus, the performance of the ORC may be different for other working fluids.

5.2. Exergy analysis of different LAES configurations

Exergy efficiencies of the compression, cold thermal energy recovery and expansion parts in the LAES system with different number of compression and expansion stages are shown in Fig. 11. From Fig. 11(a), it can be seen that the exergy efficiency of the compression part is reduced with increasing number of compression stages. This is mainly caused by the increased exergy losses related to irreversibilities in the aftercoolers. The same trend is observed in Fig. 11(b) and (c). It is worth noting that the exergy efficiencies of the cold thermal energy recovery part are almost the same in the various cases. The number of compression and expansion stages has only marginal effects on the cold thermal energy recovery part. The somewhat obvious reason is that the configuration of this part is the same in all cases. The exergy efficiencies of the expansion part, however, show significant differences. These exergy efficiencies are affected by both the heat transfer efficiency in preheaters and the performance of the ORC.

For the direct expansion part, which means the multistage turbine part, it is observed that when the number of expansion stages is 3, the highest exergy efficiency is obtained with a 2-stage compressor in the system. For the system with a 4-stage or 5-stage turbine, the best performance is obtained when there is a 3-stage or 4-stage compressor. This is in line with the previous discussion and conclusion based on composite curves in the preheaters.

As already established in Section 5.1, the best performance measured by the RTE is obtained when the flowrate of thermal oil is large enough to have close to parallel temperature profiles in the preheaters of the expansion part of the LAES. If the flowrate of thermal oil is too small, the pinch in the preheaters is in the cold end, and air preheat cannot take advantage of the thermal oil inlet temperature. If the flowrate of thermal oil is too large, the pinch in the preheaters is in the hot end, which is an advantage for air preheat, but then there is a part of the compression heat transferred by the thermal oil that is not utilized. In such cases, one could envisage that the surplus flowrate of thermal oil could be sent to

Table 6
Performance of the additional ORC for utilizing surplus compression heat.

Exp. stages	Comp. stages	$\dot{W}_{net}$ (kW)	$p_{orc,con}$ (bar)	$p_{orc,eva}$ (bar)	$T_{orc,eva}$ (°C)	$\dot{W}_{orc,tur}$ (kW)	$\dot{W}_{orc,pump}$ (kW)
3	2	-	-	-	-	-	-
	3	363.86	5.30	41.00	112.50	394.96	31.10
	4	605.01	5.30	41.00	112.00	657.02	52.01
	5	576.67	5.30	41.00	110.32	627.33	50.66
	6	361.94	5.30	18.64	69.17	379.90	17.96
4	2	-	-	-	-	-	-
	3	-	-	-	-	-	-
	4	281.80	5.30	41.00	113.03	305.74	23.94
	5	374.21	5.30	41.00	110.44	407.03	32.82
	6	216.20	5.30	21.22	92.48	226.80	10.60
5	2	-	-	-	-	-	-
	3	-	-	-	-	-	-
	4	-	-	-	-	-	-
	5	185.89	5.30	41.00	110.37	202.21	16.32
	6	141.81	5.30	20.82	89.56	148.75	6.94

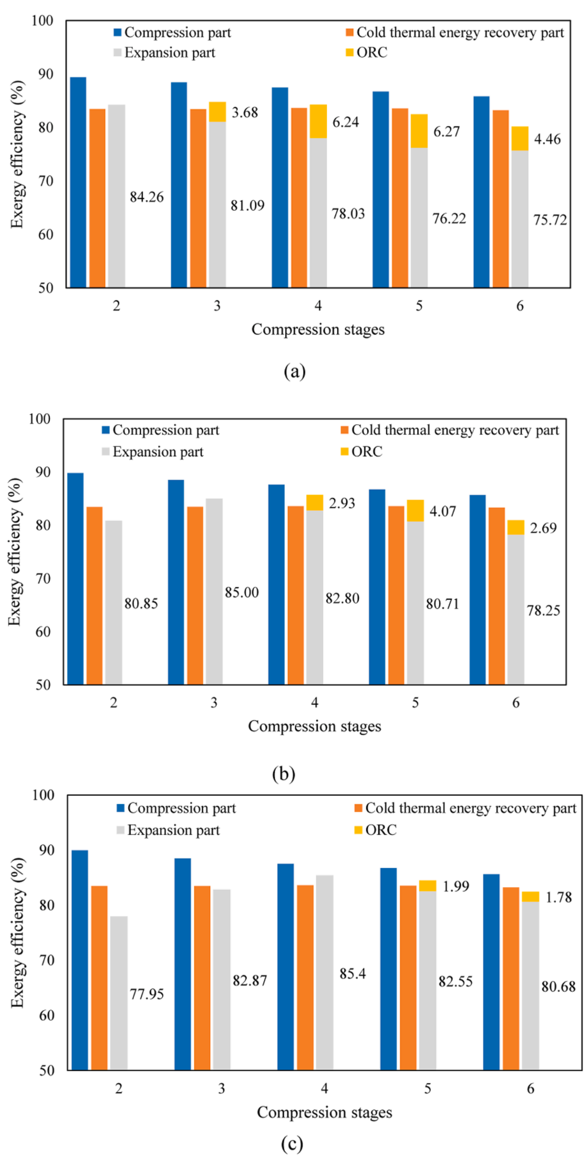

Fig. 11. Exergy efficiencies of the compression, cold thermal energy recovery and expansion parts in different LAES configurations: (a) 3-stage turbine; (b) 4-stage turbine; (c) 5-stage turbine.

an ORC to produce additional power while making the composite curves parallel.

The following conclusions can be made based on the results in Table 5 and Fig. 11(a)-(c) and explained by exergy analysis, as well as the observations in Section 5.1:

- Increased flowrate of thermal oil is a result of increased number of compression stages, and despite a reduction in compression work, irreversibilities in the aftercoolers are increased. Repeated cooling of air increases with more compression stages, thereby increasing irreversibilities due to heat transfer with temperature differences larger than zero.
- When the flowrate of thermal oil is larger than required in the expansion section, the use of ORC both makes the temperature profiles in the preheaters parallel (and thereby reduce irreversibilities) and produces additional power by utilizing otherwise wasted compression heat.
- Since the work produced by the ORC is quite small, it cannot compensate for the additional irreversibilities in the compression part by having more stages.

Considering first the cases without ORC, the maximum exergy efficiency of the expansion part coincides with the maximum RTE (i.e., 2 compression stages for 3 expansion stages, 3 compression stages for 4 expansion stages, and 4 compression stages for 5 expansion stages). However, while the RTE is reduced for these cases with increased number of expansion stages (from 66.7% via 65.4% to 64.2%), the expansion section exergy efficiency is increased with increased number of expansion stages (from 84.3% via 85.0% to 85.4%). Contributing to the overall system, exergy efficiency of the compression part is reduced with increasing number of compression stages (from 89.4% via 88.5% to 87.5%). The larger reduction in exergy efficiency of the charging part compared to the discharging part explains why there is a reduction in RTE for the overall system.

As explained before, use of an additional ORC only makes sense when the number of compression stages is greater than or equal to the number of expansion stages. As can be seen in Fig. 11, the maximum exergy efficiency of the expansion section for LAES systems with an additional ORC is obtained when the number of compression stages is equal to the number of expansion stages. However, Fig. 11 also indicates that the exergy efficiency with ORC is only marginally better than for the optimal case without ORC. For 3 expansion stages, the exergy efficiency is improved from 84.3% to 84.8%. For 4 expansion stages, a similar improvement from 85.0% to 85.7% is observed. Finally, for 5 expansion stages, the exergy efficiency with the use of an ORC is actually reduced from 85.4% to 84.5%. In summary then, the power produced by the additional ORC does not justify the investment cost and added complexity.

As for a standalone LAES system without external heat sources, the exergy efficiency of the overall system would be very similar to the definition of the round-trip efficiency, which is why it has not been calculated in this work. The major terms in the exergy efficiency for the overall system are the work produced in the discharging and the work consumed in the charging. The minor terms are the thermo-mechanical exergy of the inlet air (which is constant) and the outlet air, and both terms are negligible. On the other hand, the individual exergy efficiencies of the charging and discharging parts can help explain the change in RTE. However, the economic feasibility of the system should be analyzed for a comprehensive evaluation of the LAES system before any project is implemented, and this is beyond the scope of this work.

5.3. Effects of additional ORCs

In addition to the ORC for the unutilized part of compression heat, an ORC can also be used to collect the heat carried by the exhaust air from the last stage expander for further improvement of the LAES. In this study, we used the same working fluid (R152a, as discussed in Section 2) as for the ORCs for unutilized compression heat. As mentioned in Section 5.1, the LAES system with a 2-stage compressor and a 3-stage expander has the best performance, and this process configuration is selected as the design basis. The process flow diagram of this configuration is shown in Fig. 12. In this process, the temperature of exhaust air (stream D11) is 103.6°C. Optimization results indicate that only 78.2 kW power is produced by the ORC, and this added power does not justify the investment in an ORC. The RTE of the LAES system can only be improved from 66.7% to 67.2% with this additional ORC. The improvement by producing power from the heat of exhaust air is marginal compared to the system where wasted compression heat is utilized by ORCs.

6. Conclusions

The scope of this work has been to investigate opportunities for improving the energy performance of Liquid Air Energy Storage (LAES) systems. While previous work in our group considered improving the cold thermal energy recovery cycles, this work has focused on the different configurations of the compression and expansion sections, meaning the number of compressor and expander stages as well as the hot thermal energy recovery cycle.

The main assumptions in this work are:

- Constant isentropic efficiencies for varying compressor and expander duties and number of stages.
- Fixed ΔT_{min} for heat exchangers, however, adjusted values for above (10°C) and below (1°C) ambient temperature.
- Fixed set and composition of the two cold thermal energy recovery cycles.
- R152a has been selected as the working fluid for all cases involving ORC.
- Pressure ratios have been allowed to vary between 1 and 20.
- No heat losses to the surroundings or pressure drops in piping and equipment.

One important observation from this work is that when the number of compression stages is greater than or equal to the number of expansion stages, the expansion section is not able to fully utilize the compression heat that is carried by the thermal oil in the hot energy recovery cycle. This can easily be explained by relative slopes of the temperature profiles in the air preheaters. An Organic Rankine Cycle (ORC) has been used to turn this unused compression heat into power. The LAES configuration with the best performance has been identified by a systematic optimization-based comparison of the various cases. The

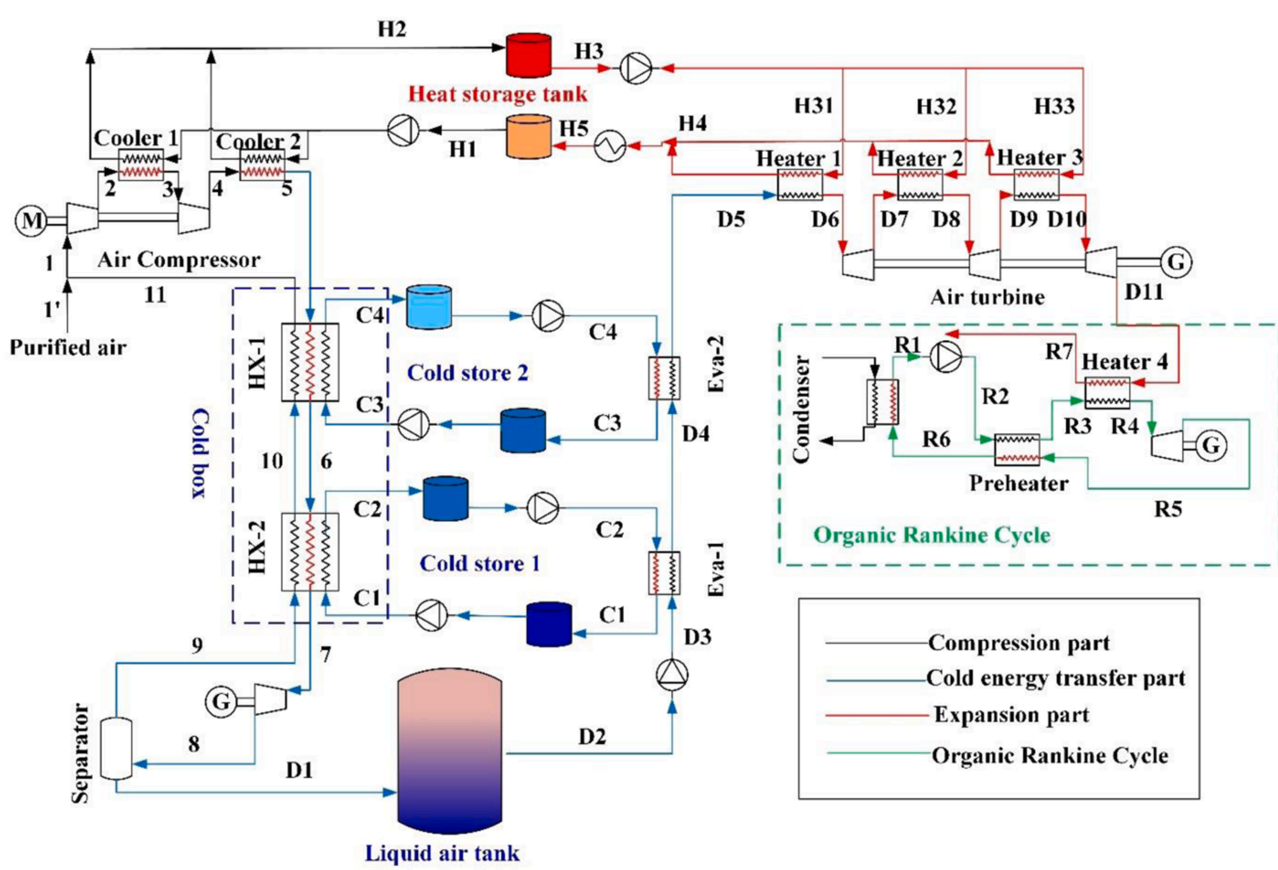

Fig. 12. Flow diagram of the liquid air energy storage with an additional ORC for exhaust air.

following conclusions are obtained in this study:

- There exists an optimal match between the number of compression stages and expansion stages in a standalone LAES system. When the number of expansion stages is 3, 4 and 5, the highest RTE is obtained with 2, 3, and 4 compression stages, respectively. Among these, the LAES system with a 2-stage compression and a 3-stage expansion has the highest RTE of 66.7%, which compares nicely with the original work of Guizzi et al. in 2015 (54.4%).
- ORCs are used to recover compression heat that is not fully utilized for preheating air in the expansion section. The largest net work output obtained for the case with 4-stage compression and 4-stage expansion is 605.01 kW, unfortunately this work is less than the reduced expansion work when increasing the number of compression stages from 3 to 4, i.e. the optimal match for the case with 4-stage expansion. This shows that ORCs can never improve the energy efficiency of standalone LAES systems.
- For cases with optimal matches between the number of compression stages and expansion stages, the exergy efficiency of the compression part is reduced with increasing number of compression stages (from 89.4% via 88.5% to 87.5%), while the expansion section exergy efficiency is increased (from 84.3% via 85.0% to 85.4%). The larger reduction in exergy efficiency of the charging part compared to the discharging part explains the reduction in RTE (from 66.7% via 65.37% to 64.17%). This clearly indicates that exergy efficiency is a valuable and comprehensive performance indicator for LAES systems.

The best configuration of the LAES is the system with 2-stage compression and 3-stage expansion (66.7%). Such high RTE makes the LAES more competitive among various energy storage technologies in terms of energy efficiency. However, the pressure ratios of compressors in the best configuration of the LAES are relatively high, and the cost for unconventional compressors with high pressure ratios is obviously higher than the conventional ones. Thus, there are several challenges that can be considered in future work:

- A cost analysis can be conducted and used to indicate the profitability and feasibility of the LAES system with high pressure ratio compressors.
- More case studies are needed to analyze the LAES with additional thermodynamic cycles.
- Going beyond the standalone LAES system, integration with external hot and cold thermal energy sources can significantly boost the RTE of the system.

Author contribution statement

Zhongxuan Liu is the corresponding author and main researcher behind the manuscript. She has developed the concept, built the simulation models and carried out the optimizations; Donghoi Kim has been a discussion partner, in particular on the simulation models and the organization of the optimization; Truls Gundersen has been responsible for the quality of the language and the structure as well as the layout of the manuscript.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data Availability

Data will be made available on request.

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Supplementary materials

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