Integrated system for electricity storage and grid-temperature heat supply - LAES-HP

Piotr Krawczyk[a], Jan Surała[b], Aleksandra Dzido[a]

[a] *Institute of Heat Engineering, Warsaw University of Technology, Warsaw, Poland, aleksandra.dzido@pw.edu.pl, CA*
[b] *Institute of Applied Research of the Warsaw University of Technology Ltd., Warsaw, Poland*

Abstract:

Liquid Air Energy Storage (LAES) is increasingly considered for large-scale, long-duration electricity storage; however, its deployment is often limited by moderate round-trip electrical efficiency and incomplete utilization of internally available thermal streams. This paper presents a steady-state thermodynamic model of an integrated LAES system coupled with a heat pump (HP), designed to deliver dispatchable electrical power while simultaneously producing useful heat at elevated temperature. In the proposed configuration, air downstream of the intercoolers in the compression stage is used as the lower heat source for the HP. This enables additional air cooling before subsequent compression or liquefaction, while recovering waste heat for external use, such as district heating or industrial heat supply. The objective of the model is to quantify the trade-off between electrical performance and total energy utilization relative to a conventional LAES configuration.

For the analysed case, the LAES charging electrical power is 36.7 MW, while the discharging electrical power reaches 22.3 MW. The HP consumes 2.2 MW of electricity, absorbs 6.6 MW from the lower heat source, and delivers 8.8 MW to the upper heat sink. In addition, heat recovered from oil cooling in the first refrigeration loop provides 7.6 MW of recoverable thermal power at a minimum temperature of 80 °C. The integrated LAES–HP system achieves a round-trip electrical efficiency of 56.26%, compared with approximately 59.5% for the conventional LAES benchmark. When the exported heat is included, the overall efficiency increases to 78.41%, and further to 97.55% if the additional oil-cooling heat stream is credited as useful output, depending on the adopted system boundary. These results demonstrate that LAES–HP integration can substantially improve total energy utilization while maintaining electrical performance close to that of the conventional system.

Keywords:

Heat Pump; Heat Recovery; Liquid Air Energy Storage; Mathematical Modelling; Thermal Energy Storage.

1. Introduction

The green transition in the energy sector, driven by, among the others, the EU's "Fit for 55" regulatory package, obliges Member States to drastically reduce greenhouse gas emissions and increase the share of RES in the energy mix. In accordance with this strategy, it is planned to reduce greenhouse gases emissions by 55% by 2030 compared to 1990 and to increase the share of RES in the energy mix to at least 42.5% [1]. The implementation of such ambitious climate and energy policies is associated with a significant transformation of national electricity systems. The anticipated continued electrification of the transport and heating sectors will significantly increase peak electricity demand and alter consumption patterns, thereby intensifying the need for system flexibility [2]. At the same time, the growing share and importance of weather-dependent renewable energy sources, such as photovoltaics and wind power, introduce greater variability in electricity generation and necessitate the development of technologies capable of storing energy at a system-wide scale [3,4].

Increasingly pronounced mismatches between supply and demand are being observed — overproduction during periods of high renewable generation, when there is insufficient demand (e.g., sunny and windy days), and shortages during periods of elevated demand (e.g., cold, windless nights) [5]. Conventional fossil fuel–based power plants cannot respond quickly enough to such fluctuations. To stabilize the power system, the development of efficient energy storage is essential. While short-term, small-scale storage solutions are well established, large-scale energy storage still presents significant technical and economic challenges.

The main technologies dedicated to system-scale electricity storage include pumped hydro storage, compressed air energy storage (CAES), batteries (especially lithium-ion), and emerging solutions such as hydrogen and thermal storage. Pumped hydro offers high capacity and long lifetimes but is geographically limited. Although, PHS is currently the most widely used, best-studied way of system-scale energy store. The efficiency of this type of warehouse is between 65% and 87%. Currently, there are over 200 facilities of this type in the world with a total capacity of 160.6GW (as of 2018). PHS are valued for their long operation (usually between 40 and 60 years) and low operating costs. However, their construction is limited by specific geographical requirements (significant differences in altitude) and access to a large amount of water [6].

Electrochemical energy storage like various types of batteries are currently among the most widely deployed and rapidly developing technologies for grid-scale energy storage, their performance and characteristics vary depending on the specific battery chemistry used. Typically, their round-trip efficiency ranges from about 80% to over 95%, making them highly efficient compared to many other storage solutions [7]. In recent years, the global installed capacity of battery storage has grown significantly, driven by the increasing share of renewable energy sources and the need for flexible grid support. Battery systems are valued for their fast response time, modularity, and ease of installation, as well as their ability to be deployed in a wide range of locations without strict geographical constraints. However, their limitations include relatively high investment costs, limited capacity, degradation over time, limited lifetime compared to some other storage technologies, and challenges related to raw material availability and recycling.

Another store is CAES, which is the commercially available technology on a large scale. Although, as in the case of PHS, the principle of its operation is very simple, its global scale of application is much smaller. In this technology, during periods of low energy demand, excess electricity is used to compress air, which is then stored in sealed underground caverns (e.g. salt caverns). When there is an increased demand for energy, the collected air is taken from them to drive turbines. During decompression, the air cools down and must be heated to carry out the process properly, for which an external heat source is required. This technology has an efficiency of approximately 60% [8]. There are only a few large CAES installations in the world, the most famous of which are the facilities in Huntorf (321MW, Germany) and McIntosh (110MW, USA) [9].

A promising alternative to the technologies described above seems to be the Liquid Air Energy Storage (LAES) technology. Unlike PHS and CAES, LAES has a higher energy density, is geographically independent, and is scalable. The basic process consists of three main steps [10]:

- Charging – atmospheric air is purified, compressed, cooled and liquefied using surplus electricity from RES;
- Storage – liquid air is collected at a near-ambient pressure in cryogenic insulated tanks.
- Discharging – the stored liquid is pumped at high pressure, gassed and expanded again by turbines to generate electricity again.

Currently, there are only test installations based on LAES technology in the world. The largest operating facility of this type was built by Highview Power in Bury, UK. The 5 MW installation with a capacity of 15 MWh, which was launched in 2018, is the company's second project after the pilot facility with a capacity of 350 kW and a capacity of 2.5 MWh [11]. Currently, there are several projects at the development stage. The Carrington (Manchester) project is a commercial-scale LAES facility currently under construction, designed to provide grid stability and 300 MWh of long-duration energy storage with a 50 MW output starting from 2026. The Hunterston (North Ayrshire) project is a hybrid energy storage platform combining LAES and lithium-ion batteries, planned to deliver 3.2 GWh of storage and support grid stability from 2027. In Australia, Highview is developing large-scale energy systems that integrate LAES with renewable generation and battery storage to support decarbonization targets. In Japan, a 5 MW LAES demonstration plant in Hiroshima is being developed using waste cold from an LNG terminal to improve system efficiency, with operation planned for 2025 [12].

Recently, many researchers have been intensively developing liquid air energy storage (LAES) technology, recognizing it as one of the most promising solutions for long-term energy storage. These studies encompass both conventional systems based on the Linde–Hampson cycle and more advanced configurations integrated with additional thermodynamic cycles. A growing body of research indicates that a key direction for further development is improving overall system efficiency by more effectively utilizing the heat generated during compression and recovering cold energy from the expansion process [13]. To achieve this, integration of LAES systems with Organic Rankine Cycle (ORC) units [14], trans-critical CO_2 cycles [15], and LNG regasification processes is being explored [16]. Such approaches enable the use of external heat and cold sources, contributing to an increase in RTE.

This study presents a concept of a new hybrid LAES system, in which the typical LAES cycle is combined with heat pump (HP) circuit. Such an integration into LAES-HP - by leveraging the synergy effect - can improve overall system efficiency and provide additional functionalities to the energy storage system, such as the production of useful heat.

2. LAES-HP – principles of operation

The idea of the proposed system is to merge adiabatic LAES cycle with heat pump cycle by the system of heat exchangers (HE) on the air compression line. Air after each compression step goes though air-oil heat exchanger to cool down air before next compressor and reuse generated heat during LAES discharging process. In LAES-HP such heat exchangers are followed by subsequent ones, serving as lower heat source for the HP circuit. Such a solution allows for deeper air cooling which affects in lower energy expenditures for liquefaction and gives the system new, additional functionality – heat production. The scheme of the proposed system is presented in Fig. 1.

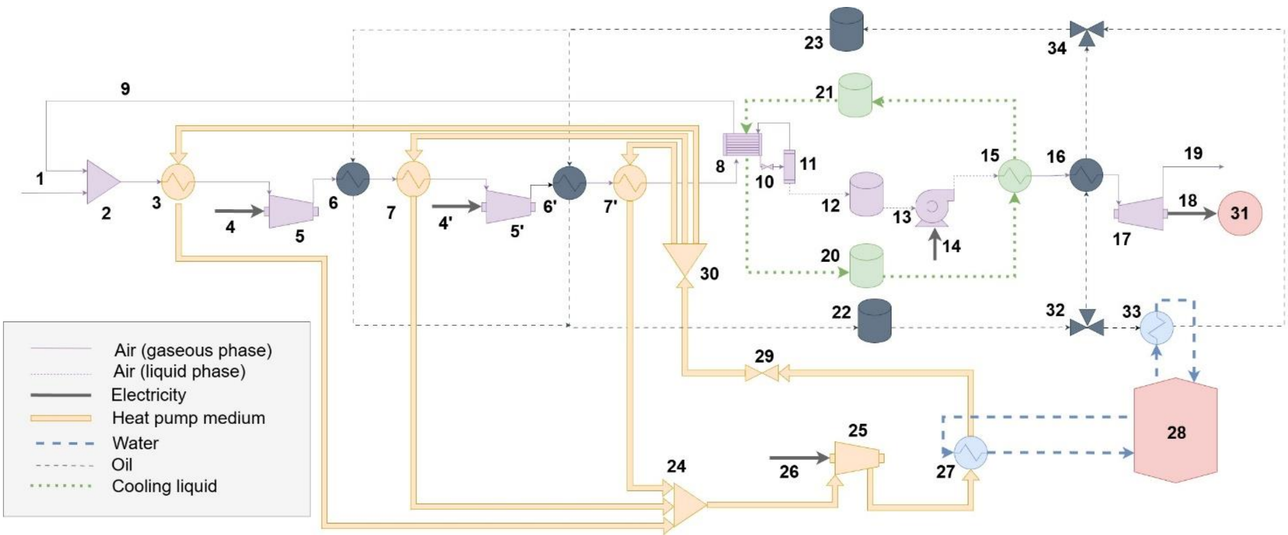

Figure. 1. *LAES-HP scheme. 1 – inlet air stream, 2 – air stream mixer, 3, 7, 7' –evaporator of the HP cycle, 4, 4' – electrical energy, 5, 5' – compressors, 6, 6' – oil-air HE, 8 – multi-stream HE, 9 – recirculated air stream, 10 – throttling valve, 11 – phase separator, 12 – insulated liquid air storage tank, 13 – cryogenic pump, 14 – electrical energy stream supplying the cryogenic pump, 15 – cooling cycle HE, 16 – oil-air HE, 17 – expander, 18 – electrical energy output, 19 – outlet air stream, 20 – hot refrigerant storage, 21 – cold refrigerant storage, 22 – hot oil storage, 23 – cold oil storage, 24 – heat pump working fluid mixer, 25 – HP compressor, 26 – electrical energy stream supplying the HP, 27 – HP condenser, 28 – heat receiver, 29 – throttling valve of the HP cycle, 30 – HP working fluid separator, 31 – electric generator, 32 – three-way valve I, 33 – oil–water heat exchanger, 34 – three-way valve II*

During periods of surplus electricity in the grid, representing the charging phase of the LAES storage—the inlet air stream (1) is combined with the recirculated air stream (9) via the air mixer (2). This combined stream is then directed to the first evaporator (3) of the heat pump circuit. The pre-cooled air subsequently enters the compression train (5), which is powered by the electrical energy input (4). Following the compression stage (5), the air stream reaches high pressure but also exhibits an elevated temperature, which is thermodynamically unfavorable for the subsequent process stages. To address this, the compression system (5) is followed by air-oil HE (6) and HP evaporator (7). To obtain higher RTE, as high pressure is required, process can be divided into stages (i.e. next steps like 4'-7' can be involved). Air, after being cooled, enters the multi-stream heat exchanger (8) for further temperature reduction. In the next stage, the cooled air stream is directed to the Joule-Thomson throttling valve (10). The throttling process reduces the air temperature, leading to partial liquefaction. This two-phase mixture is then separated in the phase separator (11). The gaseous fraction is recirculated through the multi-stream heat exchanger (8), where it is heated while providing cooling capacity, and eventually flows as stream (9) to the mixer (2). The liquefied air is stored in the insulated cryogenic tank (12) at near-atmospheric pressure.

Simultaneously, thermal energy is extracted via the heat pump evaporators (3) and (7). The vaporized working fluid streams are combined in the mixer (24) and directed to the compressor (25), powered by electricity (26). The high-pressure refrigerant then enters the condenser (27), where it rejects heat at a temperature higher than the evaporator exit temperature. This upgraded heat is transferred to an external heat sink or district heating network (28). The refrigerant cycle is completed as the fluid passes through the expansion valve (29) and the splitter (30) before returning to the evaporators.

During periods of high electricity demand, the system enters the discharging phase. Liquid air is extracted from the cryogenic tank (12) and pressurized by the cryogenic pump (13), driven by electrical energy (14). The pressurized air is then regasified in the discharge heat exchanger (15), which is part of an internal cooling loop.

The regasified air stream from the HE (15) is further heated in a oil HE (16). The high-pressure, heated air is then directed to the expander (17), potentially incorporating interstage reheating using heat from thermal oil circuit. The expander (17) is coupled to a generator, supplying electricity to the grid via stream (18). Finally, the expanded air is released to the atmosphere as exhaust stream (19) at near-ambient conditions.

3. Methods

To analyze the efficiency of this proposed solution, a mathematical model was developed to simulate the main processes occurring within the system. Mathematical modeling allows for the prediction and optimization of complex processes in various configurations without spending significant resources on building multiple physical prototypes.

The developed model is based on fundamental physical principles to ensure an accurate representation of the system's behavior. The analysis involves solving mass and energy balance equations in conjunction with phase equilibrium data and fluid property correlations. The focus was placed on evaluating the system's efficiency under steady-state conditions. To account for the real gas behavior of both air and the heat pump refrigerants, especially under cryogenic and high-pressure conditions, advanced thermodynamic property packages were used. This approach ensures high precision when calculating the energy exchange between the air liquefaction circuit and the heat pump module.

The thermophysical properties for both air and the heat pump refrigerant were determined using the Peng–Robinson equation of state Eq. (1). This equation is a standard choice for cryogenic applications as it reliably predicts liquid-vapor equilibrium and density at high pressures:

$$p = \frac{RT}{v-b} - \frac{a(T)}{v(v+b)+b(v-b)}. \qquad (1)$$

where: p — pressure [Pa], R — individual ideal gas constant [J/kg·K], T — absolute temperature [K], v — specific volume [m^3/kg], a, b — constants.

The fluid flow through the expansion valves was modeled based on the pressure differential and the hydraulic conductance:

$$\dot{m} = k\sqrt{\Delta p}. \qquad (2)$$

where: $\dot{m}$ — mass flow [kg/s], k — coefficient representing the inverse of flow resistance (conductivity) [$\sqrt{kg \cdot m}$)], Δp — pressure drop [Pa].

The relationship between mass flow rate and flow resistance was further characterized by the coefficient:

$$\dot{m} = f(p_{in}, p_{out}, C_V) \qquad (3)$$

where: p_{in}, p_{out} — inlet and outlet pressure [Pa], C_V — flow resistance coefficient.

For the heat exchangers, such as the HP evaporators and the condenser, energy conservation was expressed through differential balance equations for the shell (3) and tube (4) sides to track heat transfer and potential losses:

$$\dot{m}_{shell} \cdot (h_{in} - h_{out})_{shell} - q_{loss} + q = \rho \frac{d(V \cdot h_{out})_{shell}}{dt}, (4)$$

where: $\dot{m}_{shell}$— mass flow at the shell side [kg/s], h_{in}, h_{out} — inlet and outlet enthalpy od the flow [kJ/kg], q_{loss} — heat losses [kW], q — heat provided to the shell side [kW], ρ — density [kg/m^3], t — time [s], V — volume [m^3]

$$\dot{m}_{tube} \cdot (h_{in} - h_{out})_{tube} - q_{loss} + q = \rho \frac{d(V \cdot h_{out})_{tube}}{dt}, (5)$$

where: $\dot{m}_{tube}$— mass flow at the tube side [kg/s], h_{in}, h_{out} — inlet and outlet enthalpy od the flow [kJ/kg], q_{loss} — heat losses [kW], q — heat provided to the tube side [kW], ρ — density [kg/m^3], t — time [s], V — volume [m^3]

The power consumption of the compressors and the work generated by the expanders were calculated using a polytropic model adjusted for real gas behavior:

$$P = \dot{n}_{in} \cdot M \cdot \left(\frac{n}{n-1}\right) \cdot CF \cdot \left(\frac{p_{in}}{\rho_{in}}\right) \cdot \left[\left(\frac{p_{out}}{p_{in}}\right)^{\frac{n-1}{n}} - 1\right] \qquad (6)$$

where: P— power [kW], $\dot{n}_{in}$— inlet molar flow rate [kmol/s], M— molar mass [kg/kmol], n — polytropic exponent, CF — correction factor, p_{in}, p_{out} — inlet and outlet pressure [Pa], ρ_{in} — inlet density [kg/m³].

To account for the deviation of real gases from the ideal model, the CF correction factor was determined as follows:

$$CF = \frac{h'_{out} - h_{in}}{\left(\frac{n-1}{n}\right)\left(\frac{p_{out}}{\rho'_{out}} - \frac{p_{in}}{\rho_{in}}\right)} \qquad (7)$$

where: h_{in}, h'_{out} — inlet and isentropic outlet enthalpy [kJ/kg], ρ'_{out} — isentropic outlet density [kg/m³].

The total power transferred to the working medium during the compression process was defined as:

$$P = \dot{n}_{in} \cdot M \cdot (h_{out} - h_{in}) \qquad (8)$$

In the case of the expansion process, the power delivered by the working medium was calculated accordingly:

$P = \dot{n}_{in} \cdot M \cdot (h_{in} - h_{out})$ (9)

The power required by the cryogenic pump was determined based on the pressure increase across the device:

$P = \frac{(p_{out} - p_{in}) \cdot \dot{n}_{in} \cdot M}{\rho}$ (10)

The overall efficiency of the storage system was evaluated as the ratio of generated power to the sum of power consumed by the compression and pumping units:

$\eta = \frac{P_{espander}}{P_{compressor} + P_{pump}}$ (11)

4. Model implementation & results

For the numerical analysis of the hybrid system, the Aspen HYSYS® simulation environment was employed. The choice of this platform was primarily driven by its proven accuracy in handling multi-component energy systems and its widespread use in both academic research and the cryogenic industry. Because LAES technology involves complex phase changes at extremely low temperatures, the advanced thermodynamic property packages provided by Aspen HYSYS were essential for ensuring the precision of the results.

The inlet air was assumed at a temperature of 35°C and a pressure of 1 bar, with a volumetric flow rate of approximately 200 000 m^3/h and a composition of 21% oxygen and 79% nitrogen. The main component efficiencies were defined as follows: compressors – 82% isentropic efficiency, expanders – 90% isentropic efficiency, and the cryogenic pump – 80% efficiency. Two cases were analysed: standalone LAES (reference case) and hybrid LAES-HP solution.

Main simulations results are shown in the Tab. 1. The obtained values indicate that the reference LAES system achieves an electrical round-trip efficiency (RTE) of approximately 59.5%, which is consistent with values reported in the literature. The integration of a heat pump (LAES-HP system) slightly reduces the electrical RTE to 56.26%, primarily due to the additional energy consumption required to operate the heat pump cycle. The RTE drop is also caused by lower electricity output caused by decrease in air temperature at the expander inlet.

However, when the system is evaluated from a broader perspective, considering both electrical energy and useful heat recovery, the total efficiency increases significantly to 78.41%. This demonstrates the advantage of the hybrid configuration, where part of the input energy is effectively recovered in the form of thermal energy. A particularly notable result is the further increase in total efficiency to 97.55% when low-temperature heat (with a minimum temperature of 80°C) recovered from oil cooling in the first refrigeration cycle is also utilized. This highlights the strong potential of waste heat recovery in improving overall system performance and energy utilization.

In terms of power balance, the system requires 36.67 MW of electrical power during the charging phase and delivers 22.3 MW during discharging. The heat pump consumes 2.19 MW of electrical power, while providing 6.57 MW at the evaporator side and 8.76 MW at the condenser, indicating effective heat upgrading. Additionally, 7.57 MW of heat is recovered from oil cooling, contributing significantly to the total thermal output of the system.

Overall, the results confirm that although the integration of a heat pump slightly decreases electrical efficiency, it substantially enhances total system efficiency by enabling the recovery and utilization of thermal energy, making the LAES-HP system a highly promising solution for combined energy storage and heat production.

Table 1. *Main results.*

Parameter	Value	Unit
RTE reference LAES system (electrical)	59.5	%
RTE LAES-HP system (electrical)	56.26	%
LAES-HP total efficiency (including electrical energy and heat)	78.41	%
LAES-HP total efficiency incl. low-temp heat (min. 80°C) from oil cooling	97.55	%
Electrical power for LAES-HP charging	36.67	MW
Electrical power for LAES-HP discharging	22.3	MW
Electrical power input for the HP	2.19	MW
HP evaporators power	6.57	MW
HP condenser power	8.76	MW
Additional heat from oil cooling	7.57	MW
Low-temperature heat temperature (min.)	80	°C

5. Outcomes

The proposed concept represents a novel approach to hybrid energy storage, combining LAES technology with a HP in a way that enables simultaneous electricity storage and heat production. This solution aligns well with the ongoing green energy transition, supporting the integration of renewable energy sources and improving overall system flexibility. By enhancing energy utilization and enabling sector coupling, the LAES–HP system contributes to the development of more sustainable and efficient low-carbon energy systems.

Mathematical model of the hybrid LAES-HP store was built and implemented in Aspen Hysys numerical environment. The results demonstrate that the integration of a heat pump with the LAES system leads to a slight decrease in electrical round-trip efficiency (from ~59.5% to 56.26%), mainly due to additional auxiliary energy consumption and modified thermodynamic conditions in the expansion stage. Despite this reduction, the electrical performance remains comparable to the reference system, confirming that the hybridization does not significantly compromise the primary energy storage function.

At the same time, the system exhibits a substantial improvement in overall energy utilization when thermal outputs are taken into account. The total efficiency increases to 78.41% when useful heat from the heat pump is included, indicating effective recovery and upgrading of low-grade thermal energy. This confirms the benefit of coupling LAES with a heat pump in applications where both electricity and heat are required.

Furthermore, when low-temperature heat (≥80°C) recovered from oil cooling is also considered, the total efficiency rises to as much as 97.55%. This highlights the strong potential of integrating waste heat streams into the system and demonstrates that nearly all input energy can be utilized in combined energy systems.

Overall, the proposed LAES–HP configuration proves to be a highly efficient and flexible solution, particularly suitable for applications involving sector coupling, where simultaneous electricity storage and heat supply are required.

Nomenclature

a Peng-Robinson attraction parameter

b Peng-Robinson co-volume parameter

CF correction factor

CV flow resistance coefficient

h specific enthalpy, kJ/kg

k hydraulic conductance coefficient

M molar mass, kg/kmol

n polytropic exponent

p pressure, Pa

P power, kW

q heat transfer rate, kW

R individual gas constant, J/(kg K)

T temperature, K

t time, s

v specific volume, m3/kg

V volume, m3

Greek symbols

Δp pressure drop, Pa

η efficiency

ρ density, kg/m3

Subscripts and superscripts

in inlet

out outlet

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