David Tomás Sánchez Martínez
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(coordinadores)

Proceedings of the 7th International Seminar on ORC Power Systems

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DUAL CONDENSING PRESSURE ORC FOR CRYOGENIC ENERGY STORAGE

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ABSTRACT

A liquid air combined cycle (LACC) is a hybrid energy storage system that uses cryogenic air as a storage medium. Discharge of stored energy involves the pressurization and re-gasification of the stored liquid air, sensible heating of gaseous air, and expansion of the air through a turbine that discharges to atmosphere. During discharge of stored energy, an Organic Rankine Cycle (ORC) operates as a bottoming cycle drawing low-temperature heat from the gas turbine exhaust and using the cryogenic liquid as a heat sink. The quantity of low-grade heat that can be extracted from gas turbine exhaust is constrained by the amount of heat that must be rejected from the ORC. The overall efficiency of the LACC is the net power output divided by the energy inputs, which are the electricity used to cryogenically liquefy air during charging and the fuel consumed by the gas turbine during discharging. Power output can be increased by maximizing the heat input to the ORC, potentially including the water of combustion. To relieve the constraint imposed by the limited availability of cryogenic liquid air and to maximize the heat input and power output, a novel dual condensing pressure cycle is described. The cycle includes both high and intermediate pressure turbines, with one condenser operating at intermediate pressure and the second operating at a low pressure. A fraction of the ORC working fluid is extracted at intermediate pressure and cooled by heating regasifed air. The remainder of the working fluid expands through the intermediate pressure turbine and condenses at low pressure using the cryogenic liquid air as a low temperature heat sink. The low-pressure liquid is pumped to intermediate pressure and used to directly condense the cooled fraction, whereupon the mixture is recuperated and then heated by gas turbine exhaust.

1 INTRODUCTION

As the world transitions to zero-carbon energy systems, large-scale energy storage is expected to be needed to time-shift renewable energy on daily, weekly, and seasonal timeframes. Long duration energy storage (LDES) will be required to enable grid decarbonization via intermittent renewable resources, such as solar and wind generation. Sepulveda et al. (2021) identified key techno-economic characteristics of LDES for a zero-carbon power grid.

- Systems with storage durations greater than one hundred hours have the greatest impact on electricity cost.
- Energy storage capacity cost must be less than \$20/kWh to be cost-effective and must be less than \$1/kWh to fully displace other low-carbon technologies, such as hydrogen fueled gas turbines or carbon capture. Energy storage capacity cost reflects the cost of storage medium and container, exclusive of the charging and discharging equipment.
- High discharge efficiency (kWh-discharge per kWh-charge) is equally important.
- The cost of charge and discharge capacity and the charge efficiency play secondary roles.

Pumped Storage Hydro (PSH) represents most of the existing installed base of energy storage on both a power (MW) and energy (MWh) basis. Most PSH systems are designed for diurnal operation coupled with baseload generation, i.e., charging with off-peak power when load is low to later supply on-peak power, rather than for LDES. Electrochemical batteries (ECB), especially Lithium-based, are increasingly being deployed to provide peaking power and ancillary services, such as frequency regulation. According to Viswanathan et al. (2022), for a 1000MW x 100-hour energy storage system built in 2030, the energy storage capacity cost is $53.0/kWh for a PSH reservoir and in the range between $80.30/kWh and $100.04/kWh for a Lithium-ion DC storage block.

Alternatives to PSH and ECB are needed to meet the techno-economic requirements of LDES identified by Sepulveda et al. (2021). Since it is freely available and can be compressed or liquefied to high energy storage density, air can potentially meet the energy storage capacity cost requirement, depending on the capital cost of the storage reservoir.

Compressed Air Energy Storage (CAES) requires large underground caverns, so like PSH, has geological and geographical limitations. Cavern costs for CAES have been estimated in the range of $2.8/kWh to $13.8/kWh according to Viswanathan et al. (2022). The two commercial CAES projects are both hybrid systems (non-adiabatic), using fuel to provide heat energy for discharge.

Vecchi et al. (2021) undertook a literature search and comparison of Liquid Air Energy Storage (LAES), dividing the technical approaches into:

- Standalone LAES, shown in Figure 1a, wherein the primary energy input and output is electricity, with some waste heat rejection.
- Hybrid LAES, shown in Figure 1b, wherein additional energy supplements the electricity. The supplemental energy input could be used for additional heating of the air, as in the hybrid CAES system, or may be waste cold from an LNG regasification process.

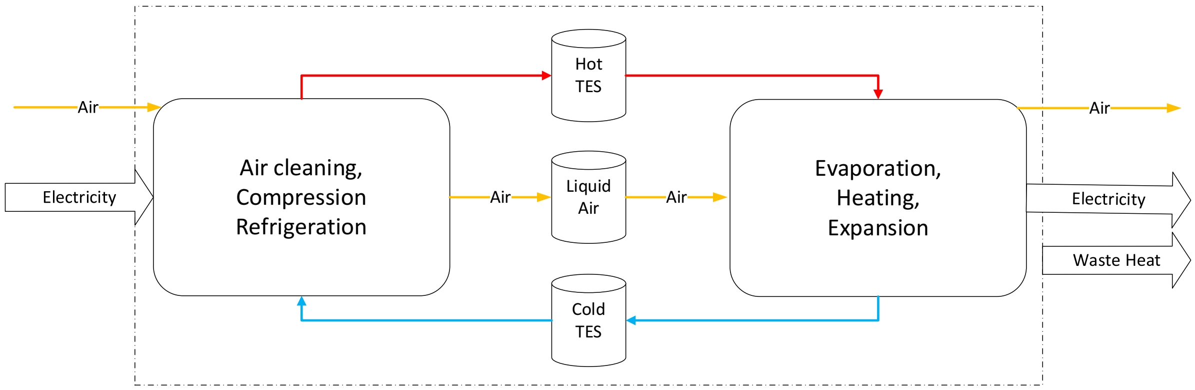

Figure 1a: Standalone LAES process, adapted from Vecchi et al. (2021).

744

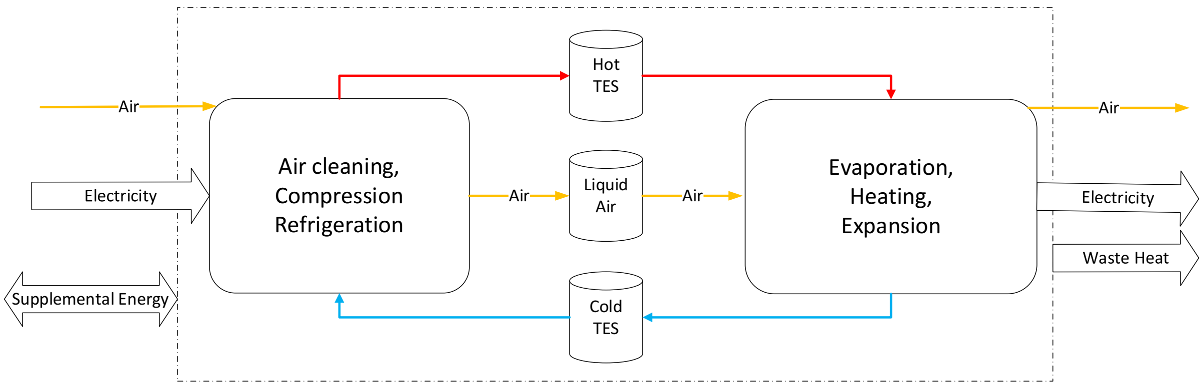

Figure 1b: Hybrid LAES processes adapted from Vecchi et al. (2021).

A notable aspect of these LAES approaches is the use of hot and cold thermal energy storage (TES) units in addition to cryogenic liquid air storage. The hot TES captures heat of compression during charging to be used for air heating during discharging. The cold TES is cooled by the liquid air during discharge with the cold recycled to the cryogenic refrigerator to reduce the energy intensity of the charging process.

Vecchi et al. (2021) report the liquefaction work ranged from 0.2 to 0.3 kWh/kg using cold recycle to produce liquid air at atmospheric pressure, compared to 0.32 to 0.33 kWh/kg without cold recycle. Liquefaction work could be reduced to 0.163 kWh/kg if the liquid air were pressurized (and stored at higher temperature), but that would lose economy of scale due to the need for multiple smaller diameter pressure vessels compared to a single atmospheric pressure storage tank. Strahan (2013) provide a more conservative estimate for liquefaction work without cold recycle using the best available technology as 0.439 kWh/kg.

Vecchi et al. (2021) also reported the discharge performance of standalone LAES as specific work output in the range of 333 to 552 kJ/kg, which is equivalent to specific discharge air consumption (SDAC) of 10.8 to 6.5 kg/kWh. They report that LAES discharge cycles operating at higher pressure have lower SDAC, but also less cold recycle available to reduce the liquefaction work.

Focusing on atmospheric pressure storage, and associating the higher SDAC with the lower liquefaction work, the ratio of charge to discharge energy is in the range of 2.16 kWh/kWh (0.2 kWh/kg x 10.8 kg/kWh) to 1.95 kWh/kWh (0.3 kWh/kg x 6.5 kg/kWh), which is an electrical round-trip efficiency of 46.3% to 51.3%. Using a charging cost of \$0.042/kWh as assumed by Lazard (2021), the marginal cost of discharge electricity would range from \$0.091/kWh to \$0.082/kWh.

Salazar and Bulte (2022) estimated that a 200,000 m^3 atmospheric pressure cryogenic storage tank would cost \$72.3 million. Liquid air density is 878.4 kg/m^3, so a full tank would contain 175.7 million kg of liquid air. Based on the above SDAC, a standalone LAES system could discharge between 16.3 and 27.0 million kWh of electricity, corresponding to an energy storage capacity cost between \$4.44/kWh and \$2.68/kWh, respectively.

The hybrid LAES concepts surveyed by Vecchi et al. (2021) featured coupling to external processes but retained the use of cold recycle. Of pertinent interest are those coupled to upstream combustion, where the specific work output was reported to exceed 900 kJ/kg (SDAC of 4 kg/kWh). The reduction of SDAC increases the energy capacity of the storage tank to 43.9 million kWh with a corresponding reduction of energy storage capacity cost to \$1.65/kWh.

Vecchi et al. (2021) also note that the round-trip efficiency of hybrid LAES can exceed 100%. N.B., round trip efficiency is defined as the electrical energy output per electrical energy input and is distinct from the thermodynamic efficiency that accounts for all energy inputs. Efficiency greater than 100% is often confusing, so in its Performance Test Code for Energy Storage Systems (PTC-53), ASME adopted the term primary energy rate, analogous to fuel heat rate, which is the ratio of electrical energy input to electrical energy output, i.e., the inverse of round-trip efficiency.

2 HYBRID LIQUID AIR COMBINED CYCLE

Conlon et al. (2022) described a hybrid LAES system called the Liquid Air Combined Cycle (LACC), illustrated in Figure 2, using gas turbine exhaust heat for driving the discharge process. To reduce complexity and capital cost, the LACC approach eliminates the intermediate cold and hot TES, so only liquid air is used for storage. Without cold recycle, the charging energy consumption is increased, but the SDAC is reduced so the overall charge energy and charge cost is less than systems with cold recycle. Decoupling the charging and discharging processes also simplifies startup and off-design operation, since pre-charging a cold store is not needed to begin producing liquid air.

The LACC uses any type of cryogenic refrigeration process (1) to produce liquid air to charge the liquid air storage tank (2). During discharge, the liquid air is pressurized (3) and heated by the condensing ORC working fluid (4), heated by low-pressure working fluid exhausted by the ORC turbine (5), and further heated by gas turbine exhaust (6). The condensed working fluid is pressurized (11), recuperated (12) and warmed by cooling the gas turbine inlet air (13), and then absorbs gas turbine exhaust heat (14) before expansion (15) to produce power. Fuel powers the gas turbine (9), whose exhaust heat (10) drives the discharge process. Ethane was selected as the working fluid because of its low freezing point, positive pressure at the condensing temperature (about -80°C) to avoid air ingress, and its environmental characteristics in the event of a leak, with nil Ozone Depletion Potential and moderate Global Warming Potential.

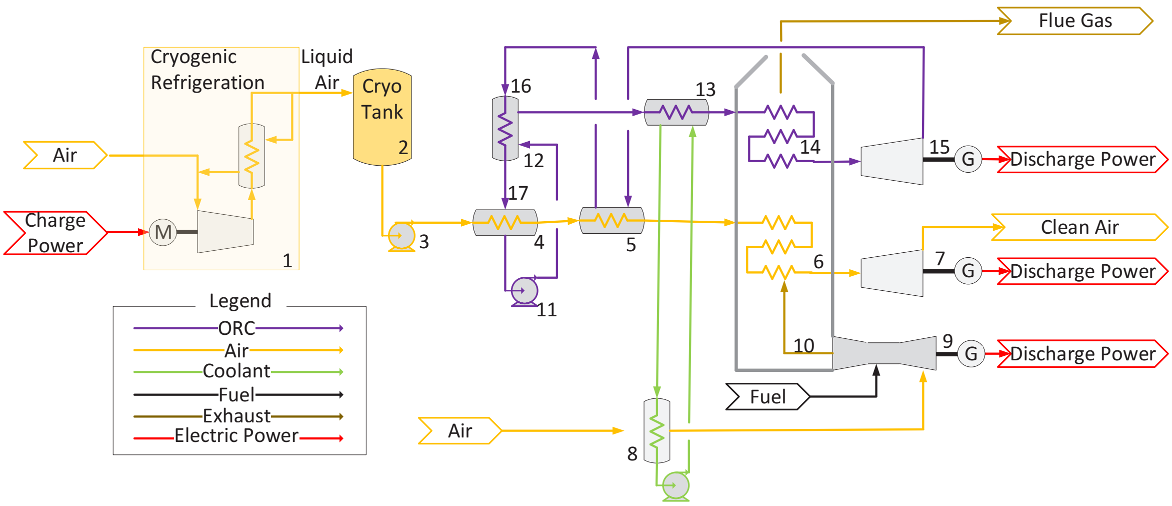

Figure 2: Liquid Air Combined Cycle from Conlon et al. (2022).

746

The discharge performance was modeled using Ebsilon Professional version 15.03. LACC operating conditions were selected based on a techno-economic analysis that included thousands of cases (Rimpel and Conlon 2023). With a Siemens SGT-800 gas turbine, Conlon et al. (2022) reported net power of 116984 kW at an SDAC of 2.32 kg/kWh, which corresponds to 75.7 million kWh of storage in a 200000 m^3 tank, and an energy storage capacity cost of \$0.95/kWh. This is below the \$1/kWh threshold identified by Sepulveda et al. (2021) for LDES to displace other dispatchable generation.

The relatively high charge energy of LACC without cold recycle is offset by the lower SDAC, so the charge to discharge energy ratio is 1.02 kWh/kWh (0.439[1] kWh/kg x 2.32 kg/kWh), which is approximately 98% round-trip efficiency. By reducing liquid air use, the charging cost is reduced to about \$0.043/kWh, substantially less than the systems with cold recycle. On the other hand, the LACC combusts fuel at net heat rate of 5041 kJ/kWh (HHV). Assuming a fuel cost of \$7.03/GJ (\$1.00/kg of hydrogen, which is the target of the U.S. Department Hydrogen Shot[2]), an additional \$0.035/kWh would be added to the cost of discharge energy, bringing the marginal cost of discharge energy to \$0.078/kWh, less than the standalone LAES systems with cold recycle.

Except for the ORC, the LACC is largely comprised of commercially available equipment. As shown in Table 1, the ORC operates at significantly lower temperatures and higher pressures than is typical for ORC power cycles. Accordingly, both the pump and the turbine require development.

[1] The more conservative estimate from Strahan (2013) is used.
[2] https://www.energy.gov/eere/fuelcells/hydrogen-shot accessed 25 April 2023

Table 1: ORC turbine conditions for LACC of Figure 2

Parameter	Turbine
Isentropic Efficiency	0.88
Shaft Power	17873 kW
Mass Flow	44.75 kg/s
Inlet Volume Flow (ACMH)	2057 m^3/h
Inlet Pressure	127.4 bar
Inlet Temperature	335°C
Exit Pressure	4.5 bar
Exit Temperature	165°C
Exit Volume Flow (ACMH)	42977 m^3/h
Condensing Temperature	-73.9°C

Pryor et al. (2022) evaluated axial and radial turbomachinery for LACC and concluded that either an axial (8- or 9-stage) or radial (four stages at two shaft speeds) turbine would be capable of handling the pressure ratios. The axial expander is expected to have somewhat higher efficiency and reduced complexity.

3 ORGANIC RANKINE CYCLE

Pryor et al. (2023) describe a development and demonstration program at pilot (about one-tenth) scale, which would be followed by full-scale development and demonstration. This approach would entail the serial development and testing of new turbomachinery at both pilot and full-scale with the attendant expenses, risks, and timelines.

This paper investigates an accelerated commercialization path that foregoes development of custom turbomachinery in favor of commercially available Generator Loaded Expanders (GLX). GLX units couple a radial turbo-expander to a generator, typically via a speed reducing gearbox, are available from a range of manufacturers and are marketed for applications such as Air Separation Units and natural gas liquefaction where they improve efficiency compared to Joule-Kelvin valves. In the LACC application, two GLX units would be operated in series as suggested by Pryor et al. (2022).

One vendor's commercially available radial GLX specifies maximum inlet pressure of 206 bar_g and maximum turbine inlet temperature of 260°C. The maximum volumetric flow rate is 10000 m^3/h for a 10000 kW expander and 16000 m^3/h for a 14000 kW expander. The vendor suggested 80% isentropic efficiency for modeling purposes.

The lower turbine efficiency compared to Table 1 impacts cycle design by reducing power output and increasing the quantity of heat rejected from the ORC to the cryogenic air. To accommodate the lower ORC turbine efficiency in the cycle of Figure 2, liquid air flow rate could be increased to provide more cooling or ORC working fluid flow rate could be reduced. An aftercooler could also be employed to reject heat from the ORC turbine exhaust in order to maintain the heat balance. Each of these approaches would increase SDAC.

Another approach was proposed by Conlon (2022), wherein a fraction of the working fluid is extracted from the exhaust of a high pressure (HP) turbine, used to heat the regasified air and then condensed at an intermediate pressure (IP). The balance of the working fluid continues to an IP turbine to produce additional power and is then condensed at low temperature using the liquid air as the heat sink.

In the arrangement shown in Figure 3, liquid air is pressurized and regasified by absorbing the latent heat of condensation from working fluid exhausted from the IP turbine. The air is then heated by the coolant used to chill the gas turbine inlet air and by the working fluid fraction extracted between the HP and IP turbines, before being heated by the gas turbine's exhaust.

Turning to the ORC, refer to the corresponding numerals on the simplified process flow diagram of Figure 3 and the Temperature-entropy (Ts) diagram of Figure 4. The low-pressure ORC working fluid is desuperheated, condensed, and subcooled (1) by the liquid air. The fluid is pumped to intermediate pressure (2) and flows to the IP condenser, where it desuperheats and condenses the fraction exiting the IP Cooler (10). Subcooled liquid (3) from the IP Condenser is then pumped to high pressure (4), heated in the recuperator (5), and further heated by exhaust gas (6) to be expanded to intermediate pressure (7) by the HP turbine. Approximately 30% of the working fluid exhausted from the HP turbine is sent to the IP Cooler with the balance expanded to low pressure by the IP turbine (8). This larger fraction of working fluid is cooled (9) in the recuperator and then flows to the LP Condenser, completing the cycle.

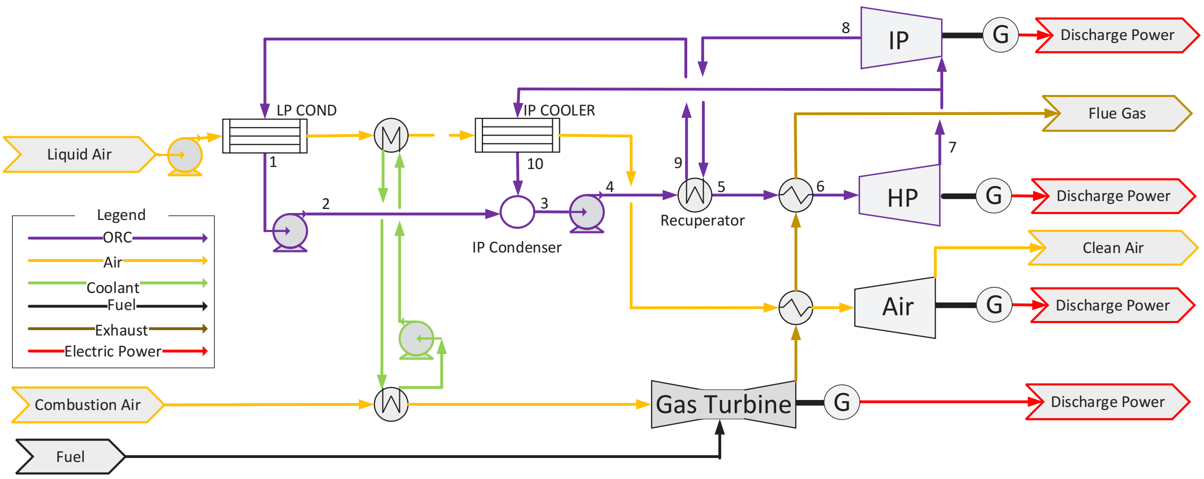

Figure 3: Dual Condensing Pressure ORC for LACC

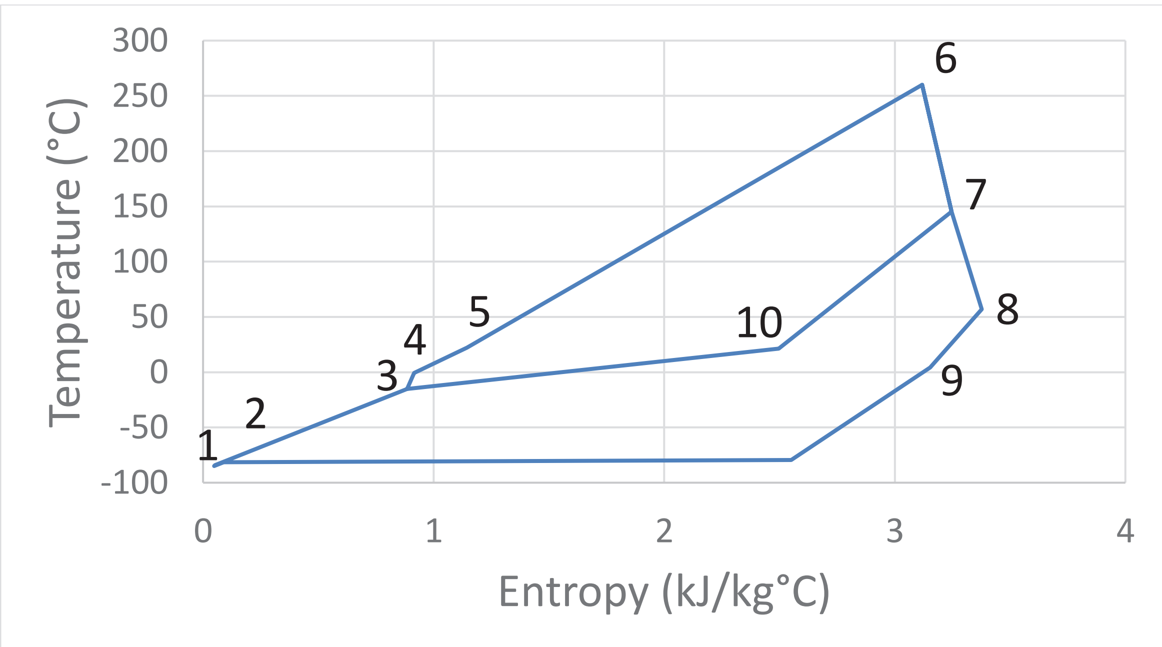

Figure 4: Temperature-entropy diagram for Dual Condensing Pressure ORC

748

The system was modeled using Ebsilon Professional version 15.03, with similar equipment assumptions as the LACC of Figure 2. However, it was necessary to increase the flue gas exit temperature by about 18°C to reduce the heat input into the ORC and balance the available cooling. On the other hand, the ORC condensing pressure was reduced by moving the air heater to the intermediate pressure loop.

Table 2 compares the performance of the LACC of Figures 2 and 3 at the same ambient, gas turbine, and air turbine conditions. For the dual condensing pressure system of Figure 3, the net power output is about 3.3% less due to lower ORC output, which causes the fuel heat rate to increase by 2.6%, and the SDAC by about 3.4%. The marginal cost of energy, using the same cost assumption, increases 2.6%. Notably, the cycle analyses used the same liquid air flow rate and air turbine inlet conditions, which may not be optimal for the dual condensing pressure cycle.

Table 2: ORC performance comparison Single (Fig. 2) vs. Dual Condensing Pressure (Fig. 3)

Parameter	Single Pressure	Dual Pressure
Net Power (kW)	116984	113148
HHV Net Fuel Heat Rate (kJ/kWh)	5041	5172
SDAC (kg/kWh)	2.32	2.4
Flue gas temperature (°C)	49	67
ORC Condensing Pressure (bar)	2.1	1.58
ORC shaft power (kW)	17919	14819
Marginal cost of energy ($/kWh)	0.078	0.081

The operating conditions for the HP and IP turbines in the dual condensing pressure system of Figure 3 are shown in Table 3. The pressure ratios for the HP and IP turbines were 9.2:1 and 8.6:1, respectively, each with assumed 80% isentropic efficiency. Total shaft power from the two turbines is 14819 kW, 3100 kW less than the higher efficiency turbine in Table 1. Importantly, all parameters were within the range of the vendor's commercially available GTX units.

Table 3: ORC turbine conditions for Dual Condensing Pressure LACC of Figure 3

Parameter	HP Turbine	IP Turbine
Isentropic Efficiency	0.80	0.80
Shaft Power	9495 kW	5324 kW
Mass Flow	46 kg/s	32 kg/s
Inlet Volume Flow (ACMH)	1350 m^3/h	7071 m^3/h
Inlet Pressure	165 bar	18 bar
Inlet Temperature	260°C	145°C
Exit Pressure	18 bar	2.1 bar
Exit Temperature	149.8°C	57.0°C
Exit Volume Flow (ACMH)	10165 m^3/h	49202 m^3/h
Condensing Temperature	-12.0	-79.9 °C

4 CONCLUSIONS

This paper has described an alternative Organic Rankine Cycle for use in Liquid Air Combined Cycle energy storage systems. The objective is to avoid the cost, time, and risk of developing custom turbomachinery by employing commercially available Generator Loaded Expanders. To accommodate power, pressure, temperature, and flow rate specifications, and a reduced isentropic efficiency, the ORC has been modified to extract a working fluid fraction at intermediate pressure, where it is used for heating regasified air prior to condensation by mixing with sub-cooled working fluid.

The overall performance impact is modest (about 3%), despite a substantial (about 17%) reduction in the ORC turbine power output and an 8% reduction in its isentropic efficiency. Cycle optimization may improve performance, but the isentropic efficiency of the ORC turbines is clearly the most important performance factor.

From a commercial perspective, the dual condensing pressure ORC provides a means to employ proven turbomachinery and pumps and thereby increase the Technology Readiness Level of LACC.

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This book contains the compilation of works contributed to the *7th International Seminar on Organic Rankine Cycle Power Systems* (ORC 2023), held in Seville between the 4th and 6th of September 2023. The event was hosted by Universidad de Sevilla on behalf of the Knowledge Centre on Organic Rankine Cycle Technology (KCORC), incorporated in The Netherlands.

The ORC conference, organized biennially, stems as the only conference that is specific to ORC technology, therefore gathering a diverse community whose affiliation spans across all the interested stakeholders, not only in this particular technology but also and in a broader context, in the energy transition. Original equipment manufacturers, professional associations, end-users, investors, policy makers, academics, scientists feel at home at ORC 2023.

The almost 100 proceedings in this book cover a wide variety of topics, from fundamentals to system integration through component design, accounting for thermodynamic performance as well as component design. In addition to this, and as a new track in 2023, works on heat pump technology were also accepted in order to raise awareness of the strong ties between both technologies, specifically in energy storage applications.

This book provides an excellent overview of the current maturity of power systems based on Organic Rankine Cycle technology for applications as diverse as geothermal and waste heat recovery in industry or downstream of other prime movers (e.g., marine applications). It is also an excellent source of information to understand the current challenges faced by the technology, stemming from a very competitive market and increasingly stringent environmental regulations.

The organizers of ORC 2023 hope that the reader finds this work as exciting as the attendees to the conference and, maybe, make the decision to join the 8th edition to the conference in 2025.

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