Thermodynamic analysis of a hybrid energy storage system based on compressed air and liquid air

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Abstract

As renewable electricity generation capacity increases, energy storage will be required at larger scales. Compressed air energy storage at large scales, with effective management of heat, is recognised to have potential to provide affordable grid-scale energy storage. Where suitable geologies are unavailable, compressed air could be stored in pressurised steel tanks above ground, but this would incur significant storage costs. Liquid air energy storage, on the other hand, does not need a pressurised storage vessel, can be located almost anywhere, and has a relatively large volumetric exergy density at ambient pressure. However, it has lower roundtrip efficiency than compressed air energy storage technologies. This paper analyses a hybrid energy store consisting of a compressed air store at ambient temperature, and a liquid air store at ambient pressure. Thermodynamic analyses are then carried out for the conversions from compressed air to liquid air (forward process) and from liquid air to compressed air (reverse process), with notional heat pump and heat engine systems, respectively. Preliminary results indicate that provided the heat pump/heat engine systems are highly efficient, a roundtrip efficiency of 53% can be obtained. Immediate future work will involve the detailed analysis of heat pump and heat engine systems, and the economics of the hybrid energy store.

1. Introduction

It is almost certain that in the near future, electricity generation from renewable energy sources, particularly solar and wind, will account for a large portion of the overall generation capacity. Wind power accounted for ~39% of renewable power capacity added worldwide in 2012, followed by ~26% each for solar PV and hydropower [1]. The UK aims to reduce greenhouse gas emissions by 80% by 2050 [2]. To alleviate the problems caused by burning of fossil fuels, renewable generation capacity must be increased – and this calls for a secure, sustainable and reliable energy supply system. It is here that energy storage is expected to play a key role.

Compressed air energy storage (CAES), historically, has been used as a 'spinning reserve' for power smoothing applications. For CAES to be cost effective, it must be employed at large scales (e.g. underground salt caverns, depleted aquifers), but suitable geologies for large-scale CAES are not available "on demand". Thus, this paper investigates employing an above-ground compressed air energy store by supplementing it with a liquid air energy store. Although CAES has relatively high roundtrip efficiency, above-ground components in steel tanks can incur significant storage costs (see Table 1 [3]). Liquid air energy storage (LAES) has the advantage that it can be compactly stored and can be located almost anywhere. Since the efficiency of liquefaction plants depends strongly on their scale [4], the proposed system attempts to make use of the different characteristics of CAES and LAES: it comprises a compressed air store of relatively low energy storage capacity, a liquid air store of higher energy storage capacity, and machinery to transform between the two states of air. When electricity prices are low, and the compressed air tank is nearly full, electricity can still be bought by converting some amount of compressed air into liquid air. Conversely, when electricity prices are high, and the compressed air tank is nearly empty, electricity can still be sold to the grid by converting liquid air back to compressed air, and then to electricity.

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Table 1: Representative costs of CAES systems from various sources. Adapted from Ref. [3].

Source of estimate	Power related cost ($/kW)[4]	Storage cost ($/kWh)[5]
Schoenung and Hassenzahl (2003) (bulk storage)	425	3
Schoenung and Eyer (2008) (distributed gen./surface)	550	120
EPRI-DOE (2003) (salt mine 300 MW)	270	1
EPRI-DOE (2003) (surface 10 MW)	270	40
EPRI (2003) (salt/porous/hard rock/surface)	350 (all)	1/0.10/30/30
EPRI-DOE (2004) (salt/surface)	300	1.74/40

This paper concerns the design of a thermodynamic system for the conversion of compressed air to liquid air and back, with notional heat pump and heat engine systems, respectively.

2. Background on Liquid Air Energy Storage (LAES)

If one removes sufficient heat from an isolated mass of air, it will liquefy. A simple air liquefaction cycle, the Linde-Hampson cycle, is shown in Fig. 1, and it employs the Joule-Thomson effect to produce liquid air. At ambient pressure, air becomes completely liquid at 78.9K. There has recently been a surge of interest in using liquid air as an 'energy carrier', i.e. an energy storage medium, as reported in [5-7], owing to its relatively high exergy density (competitive with existing battery technologies [7]) and its potential as a clean transport fuel [5]. Thus, LAES can be thought of as a thermo-electric storage device which stores energy as a temperature difference between two thermal reservoirs [8]. Generally, the LAES cycle involves [9]: (a) The charging of the liquid air store (i.e., the liquefaction process), with the liquid air then stored in a thermally insulated tank at near-ambient pressures; (b) The discharging of the liquid air store, where power is recovered by first pressurising the liquid air, then supplying thermal energy to the fluid, and subsequently expanding to generate work output. This in turn drives a generator to feed electricity back to the grid; (c) 'Cold recycle', where cold thermal energy released during discharge is stored, and is used to minimise the liquefaction work during charging.

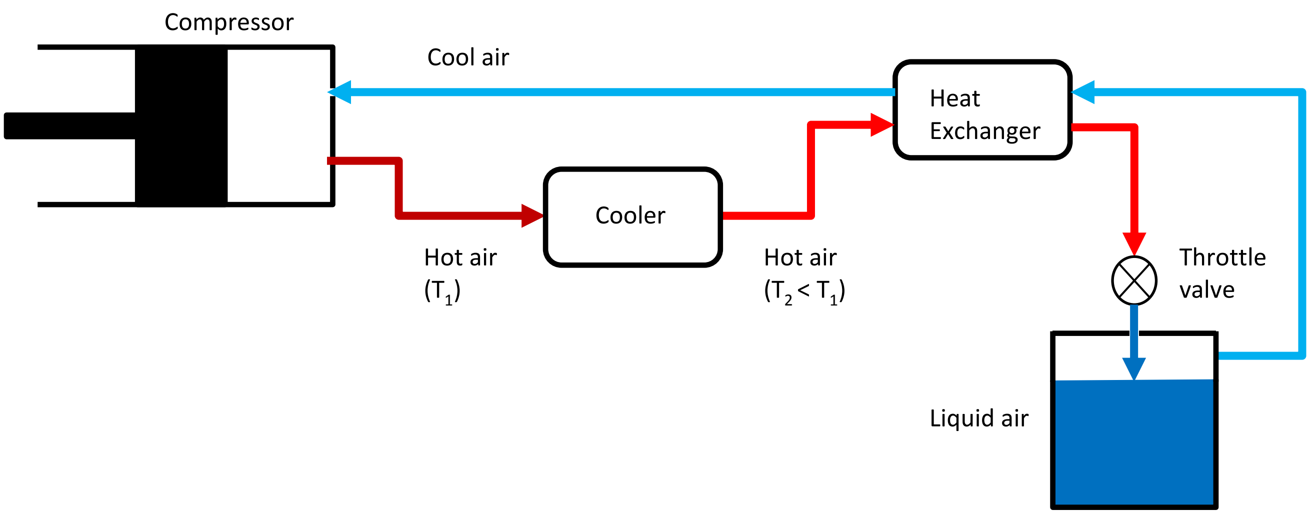

Figure 1: A simple air liquefaction cycle.

Interest in LAES goes back as far as 1977 when Smith [10] proposed a cycle using adiabatic compression and expansion, and reported an energy recovery efficiency of 72%. But this configuration required, most importantly, a regenerator which could withstand temperatures between -200C and 800C, pressures up to 100bar, and allow contact with both compressed air and liquid air. Ameel *et al*. [11] analyse a combined Rankine cycle with Linde liquefaction process, and report that 43% of the energy can be recovered from liquid air. Power recovery from cryogen via an

[4] Investment cost of the storage technology per unit of rated output power.
[5] Investment cost of the storage technology per unit of energy storage capacity.

indirect Rankine cycle is one of four major methods of extraction of cold exergy [12], the other three being: (a) 'Direct expansion' cycle where pressurised cryogen is supplied with thermal energy from ambient or waste heat sources, and then expanded to extract work; (b) Indirect Brayton cycle where the cryogen cools down the gas at the inlet to a compressor, then the compressed gas is heated further before expansion. Here, the cryogen is used to minimise compression work; (c) Combination of either Rankine cycle with direct expansion or Brayton cycle with direct expansion.

More recently, a cryogenic energy storage system for electrical energy storage which uses liquid air/nitrogen as the energy carrier coupled with a natural gas-fuelled closed Brayton cycle was proposed [13]. The carbon dioxide produced in the cycle is captured as dry ice, and the roundtrip efficiency is reported as 54%. Here, helium is used as the 'blending gas' (it circulates within the system and is not consumed) to control the temperature of the natural gas after combustion in an oxygen rich environment, and before it enters a gas turbine. It is reported in Ref. [14] that for the system proposed in Ref. [13], capital costs dominate, and the air liquefaction unit accounts for a large part of the capital costs. The authors report that both the capital and peak electricity costs of the system are comparable with combined cycle gas turbine (CCGT) plant. The cost of the cryogenic tank depends, of course, on the capacity, and in terms of cost per rate of liquefaction, $30,000/(tonne/day) for a liquefaction plant with capacity of 500 tonne/day is suggested.

A demonstration LAES plant (350kW/2.5MWh) was built in 2008 in Slough, UK, and detailed analysis and results from the testing of this pilot plant can be found in Ref. [9].

3. Thermodynamic analysis of a 'series' hybrid energy storage system based on compressed air and liquid air

It should be noted that the term 'series' here means that all energy transactions with the grid will be via the compressed air energy store. Thus, the liquid air store acts as *overflow* capacity. The forward process (charging of the liquid air store), is an air liquefaction process, and the reverse process (discharging of the liquid air store) is the energy recovery process, where the recovery is accomplished by converting liquid air to compressed air and finally to electricity, rather than from liquid air to electricity via processes similar to those described in the previous section.

3.1 Ideal, reversible hybrid energy storage system

Storing liquid air is storing exergy because as ambient heat is allowed back into the air, it will evaporate and thus expand, and so can be used to do work. Thus, all pumped thermal electricity storage systems are implicitly exergy storage systems. This section describes the configuration of an ideal reversible system and outlines an exergy analysis for this system. Based on this ideal system, considerable insight is gained into what features a practical system should have. It should be noted that an isobaric compressed air store is assumed for the analyses carried out.

In the ideal (reversible) case for the hybrid energy storage system shown in Figure 2, there is no loss of exergy – all of the flow exergy of the compressed air is transformed to flow exergy of liquid air during the forward conversion. Similarly, all of the flow exergy of the liquid air is transformed back to flow exergy of ambient-temperature compressed air in the reverse conversion.

It is convenient to assign a notation to some relevant temperatures:
T_1: Ambient temperature
T_2: A temperature just above dew-point for atmospheric pressure air
T_3: A temperature just below bubble-point for atmospheric pressure liquid air

The forward conversion process is explained as follows. High-pressure air is drawn from the compressed air store in two different streams: *primary air*, which is all ultimately converted into liquid form, and *secondary air*, whose exergy is exploited to provide additional cooling for the primary air.

A straightforward exergy balance reveals the required mass-flow ratio between primary and secondary air. Taking ambient temperature to be 290K, and assuming that the high pressure air is stored at 50bar, 1kg of liquid air at ambient pressure and 78.9K has the same flow exergy as ~2.2kg of compressed air. In that case, the ratio of mass flow rates of air in the primary and secondary streams is 1:1.2.

The primary air stream begins at high pressure and at temperature T_1. This air is expanded isentropically to atmospheric pressure via the expander E_{pri} so that its temperature reduces to T_2. The primary air then passes through heat exchangers present on the cold side of the notional heat pump. When sufficient further heat has been removed from the primary air by the heat pump, it has become fully liquid with temperature T_3 and it can be stored at atmospheric pressure. The secondary air stream also begins at high pressure and temperature T_1. This air is expanded isentropically to atmospheric pressure via the expander E_{sec} to yield a stream of cool air which passes through a heat exchanger on the hotter side of the heat pump in order to draw heat from that heat pump.

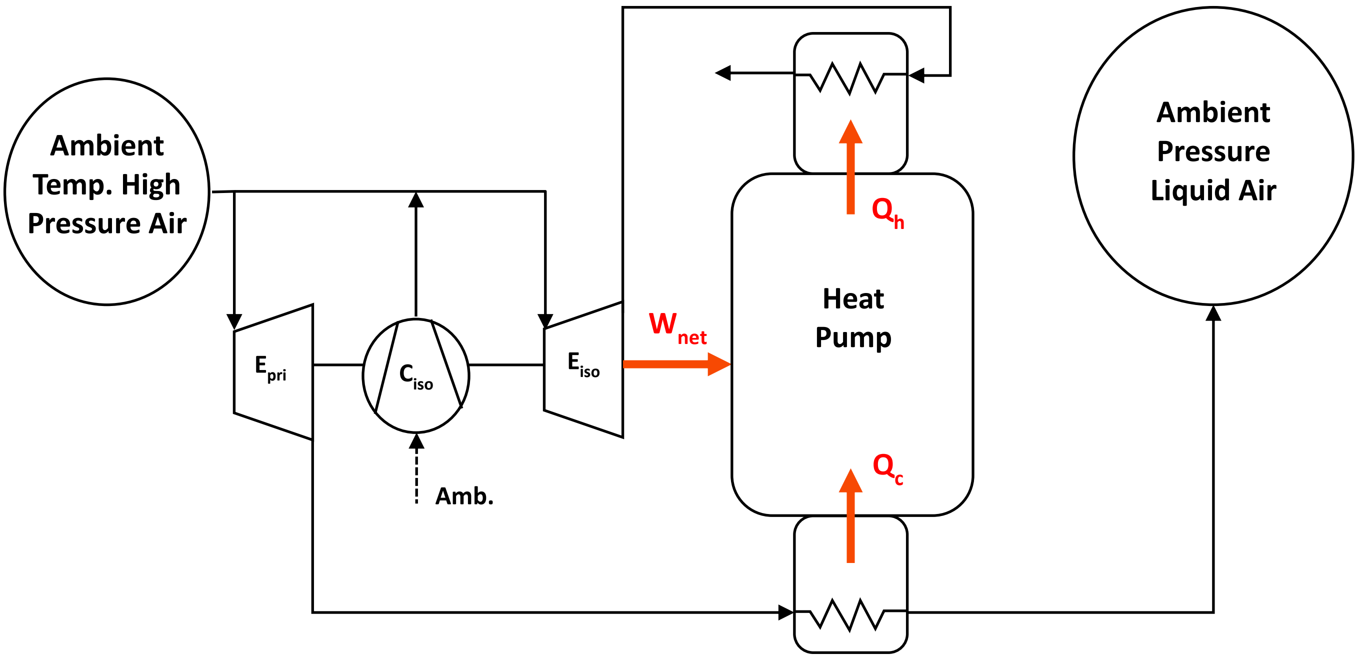

Figure 2: Schematic for the ideal (reversible) hybrid energy storage system.

After expansion, the secondary air is also at temperature T_2. The latent heat of vaporization of liquid air is very much larger than the total heat required to raise the temperature of air from T_2 to T_1. Moreover, the heat discharged from a heat pump on its hotter side will necessarily be larger than the heat drawn in on its colder side. Thus, it becomes obvious that the secondary air stream will not provide enough capacity for heat removal from the heat exchanger if the amount of secondary air suggested by an exergy-balance is used directly for cooling. The isothermal compressor, C_{iso}, shown in Figure 2 between E_{pri} and E_{sec} provides one means by which the notional system can be made to fully liquefy air while minimising output shaft power (which is not desired when charging the liquid air store, typically when electricity prices are low). By drawing some of the shaft power produced by E_{pri} and E_{sec} to drive C_{iso}, the total mass flow rate of cool ambient-pressure air exiting E_{sec} can be multiplied such that there is now sufficient thermal capacity to remove all heat from the heat pump with all air discharged being at ambient temperature.

The heat pump in this system has a rather interesting configuration. Temperatures on its hot side range from T_2 to T_1 – a range of more than 200K for air that started at 50bar prior to expansion. By contrast, temperatures on its cool side range from just T_3 to T_2 – a range of ~14K. If this heat pump uses an ideal gas as a working fluid, the number of expansion stages with inter-stage re-heats must be large compared with the number of compression stages.

Properties of air were calculated based on the mixture model of Lemmon *et al.* [15] which is explicit in Helmholtz energy.

Assumptions for the analyses that follow are listed in Table 2.

Table 2: Assumptions for analyses with notional heat pump and heat engine systems

Ambient temperature	290K
Temperature of liquid air	78.9K
Ambient pressure	1atm
Pressure of air in compressed air store	50bar
Mass flow rate of primary air	1kg/s
Isentropic efficiency of air expanders	95%
Effectiveness of heat exchanger unit	95%
Exergy efficiency of heat pump	90%
Exergy efficiency of heat engine	90%
Efficiency of cryogenic pump	80%
Efficiency of near-isothermal compressor	85%
Compressed air to electricity conversion efficiency (or vice versa)	85%

3.2 Practical implementation of the hybrid energy storage system

3.2.1 Forward conversion process

The system for the conversion of HP air to liquid air is shown in Figure 3. It comprises an air-to-air pre-cooler for secondary air to reduce the work output after multi-stage expansion. It is assumed that the secondary air expander expands in two stages of equal pressure ratios, and that the pressure drop in the pre-cooler is negligible.

An iterative procedure was devised to model the forward conversion process. Key “residuals” – difference between the two sides of an equation that is to be solved – were identified based on the governing equations for heat exchanger units, compressors/expanders, and the heat pump. The Newton-Raphson method was then employed, which iteratively finds closer approximations to the roots of the governing equations and so reduces the residuals towards zero.

In the present analysis of the forward process, a notional heat pump liquefies the primary air. The secondary air stream provides cooling capacity in multi-stream heat exchangers. The heat pump system is driven by power output from the air expanders, and it is characterised by a measure of exergy efficiency. As primary air is cooled to form liquid air, exergy is added to it. The secondary air stream releases exergy as it heats up, and the heat pump is supplied with exergy in the form of rate of work, W_{HP}. Thus, the exergy efficiency residual for the heat pump can be described as:

$$r_1 = \eta - \left(\frac{\dot{B}_{air,cold}}{\dot{B}_{air,hot} + \dot{W}_{HP}} \right)$$

Where:

- $\dot{B}_{air,cold}$ is the flow exergy added into the cold, primary air efficiency.
- $\dot{B}_{air,hot}$ is the flow exergy that is given up by the secondary air.
- $\dot{W}_{HP}$ is the rate of work consumed by the heat pump.

The residuals for the remaining components are listed below.

Heat exchanger units (heat balance and effectiveness):

$$r_2 = \Delta \dot{Q}_{hot} - \Delta \dot{Q}_{cold}$$

$$r_3 = \varepsilon - \left(\frac{\Delta \dot{Q}}{\Delta \dot{Q}_{\max}} \right)$$

Where:

- $\Delta \dot{Q}_{cold}$ is the heat flow associated with the cold stream(s).
- $\Delta \dot{Q}_{hot}$ is the heat flow associated with the hot stream(s).
- $\Delta \dot{Q}_{\max}$ is maximum possible heat transfer in the heat exchanger unit.

Isentropic expander/compressor (outlet temperature):

$$r_4 = (T_{out})_n - T_{out}$$

Where:

- $T_{out} = T_{in} \times \left(P_{in} / P_{out} \right)^{\left(1-\gamma / \gamma \right)}$
- γ is the ratio of specific heats of air.

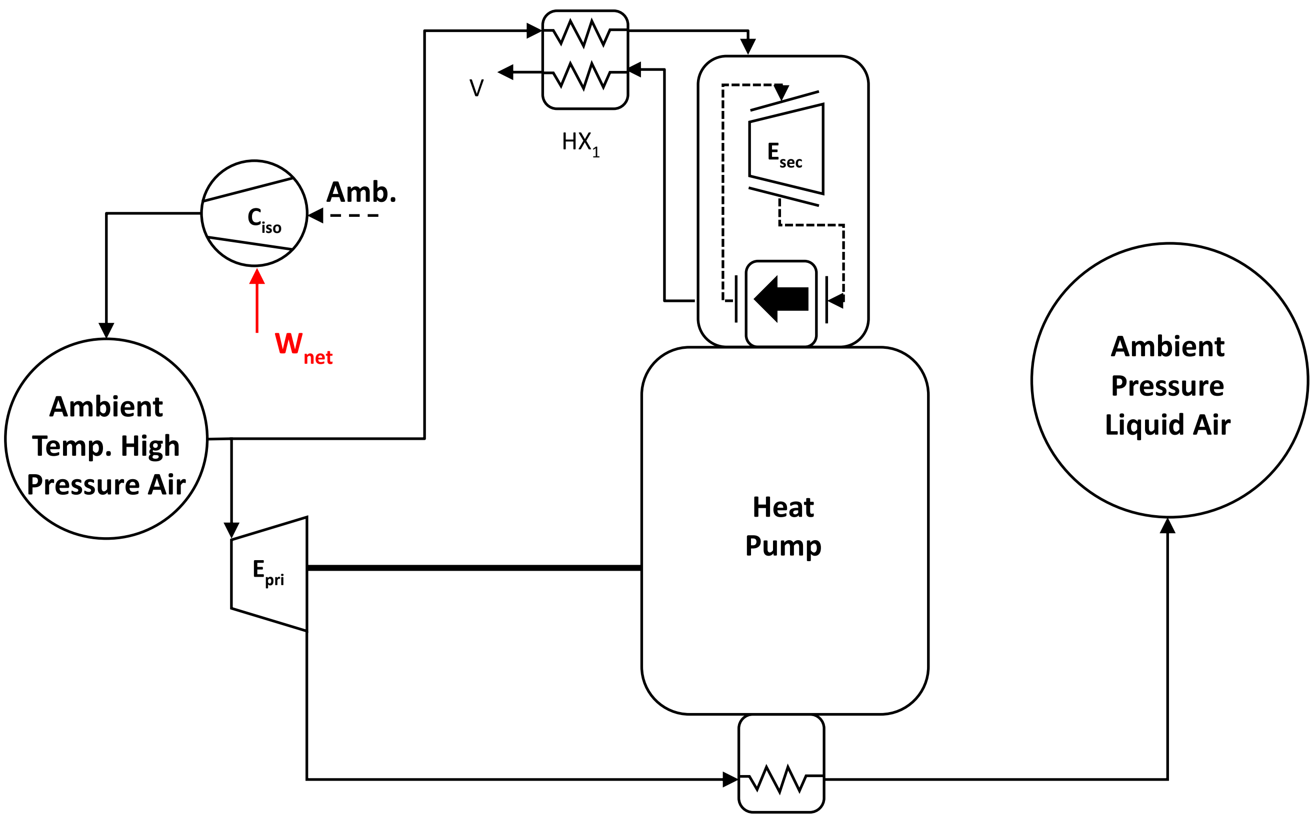

Figure 3: Schematic for the forward conversion process of the hybrid energy storage system.

When there is excess shaft power, it is used to drive a near-isothermal air compressor which sucks in air from ambient and produces compressed air to be fed back into the store. Thus, the forward conversion efficiency is calculated as:

$$\eta_F = \frac{\dot{B}_{LA}}{\left(\dot{B}_{CA} - \eta_{iso}\dot{W}_{net}\right)}$$

Where:

- η_F is the forward conversion efficiency.
- $\dot{B}_{LA}, \dot{B}_{CA}$ are flow exergies of liquid air and compressed air (from the store), respectively.
- $\dot{W}_{net}$ is the excess shaft power produced by the system (which the near-isothermal compressor makes use of).
- η_{iso} is the efficiency of the near-isothermal compressor C_{iso}.

The performance summary of the forward conversion process and the exergy loss distribution are shown in Tables 3a and 3b, respectively.

Table 3a: Performance summary of the forward conversion process

Flow exergy of liquid air	701 kW
Feedback flow exergy of compressed air	338 kW
Flow exergy of compressed air	1293 kW
Conversion efficiency	**73 %**

Table 3b: Exergy loss distribution

Machine	Loss	Fraction of total loss
Primary air expander (E_{pri})	30 kW	12 %
Secondary air expanders (E_{sec})	71 kW	28 %
Secondary air precooler (HX_1)	29 kW	11 %
Heat pump	65 kW	26 %
Isothermal compressor (C_{iso})	60 kW	23 %
Vent loss (V)	0.2 kW	~0 %

3.2.2 Reverse conversion process

In the reverse process, shown in Figure 4, liquid air is first pressurised using a cryogenic pump. A heat engine transforms the high pressure liquid air into high pressure compressed air at ambient temperature, whilst also extracting some work. The hot reservoir of the heat engine is a large mass of water, e.g. a river, at almost constant (ambient) temperature.

Since the exergy flow into the hot reservoir is negligible, the exergy efficiency of the heat engine can be described as:

$$\eta = \left(\frac{\dot{W}_{HE}}{\dot{B}_{air,cold}}\right)$$

Where:

- $\dot{W}_{HE}$ is the rate of work output from the heat engine.
- $\dot{B}_{air,cold}$ is the exergy flow released by the high pressure liquid air when it transforms into high pressure compressed air at ambient temperature.

The temperature rise after liquid air is pumped to a higher pressure can be determined by applying first law to the cryogenic pump and using the correlation between enthalpy, pressure, density, and temperature from Lemmon *et al.* [11]. Application of the combination of first law and second law to the heat engine gives the heat content associated with the hot reservoir, and the net work output.

The reverse conversion efficiency is then calculated as:

$$\eta_B = \frac{\left(\dot{B}_{CA} + \left(\dot{W}_{net} / \eta_{CE}\right)\right)}{\dot{B}_{LA}}$$

Where:

- η_B is the reverse conversion efficiency.
- η_{CE} is the conversion efficiency of compressed air to electricity.

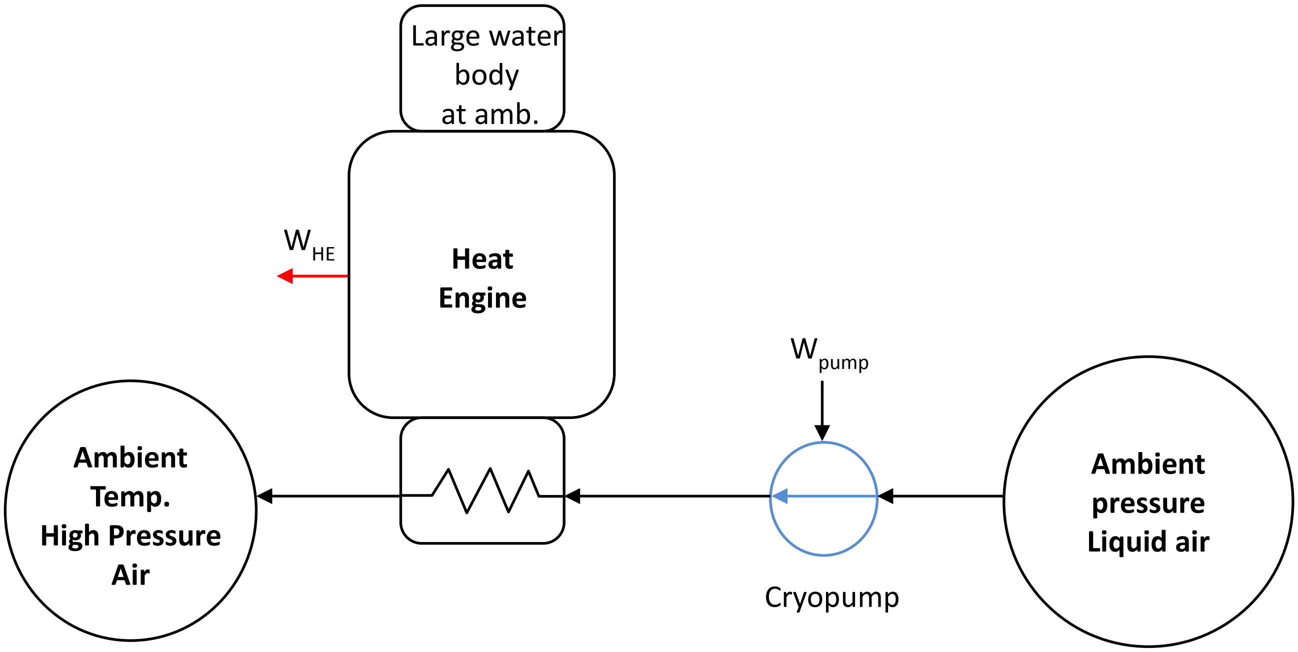

Figure 4: Schematic for the reverse conversion process of the hybrid energy storage system.

The performance summary of the reverse conversion process and the exergy loss distribution are shown in Tables 4a and 4b, respectively.

Table 4a: Performance summary of the reverse conversion process

Temperature of pressurised liquid air	80.8 K
Net shaft power output	297 kW
Flow exergy of liquid air	701 kW
Flow exergy of compressed air	323 kW
Conversion efficiency	**96 %**

Table 4b: Exergy loss distribution

Machine	Loss	Fraction of total loss

Cryopump	5 kW	6 %
Heat Engine	76 kW	94 %

4. Conclusions

A novel hybrid energy storage system, comprising a compressed air store supplemented with a liquid air store of relatively higher energy storage capacity, is proposed. CAES offers high roundtrip efficiency, but above-ground storage of compressed air in a pressurised steel tank has significant costs associated with it. Liquid air energy storage on the other hand is not geographically constrained; it does not need a pressurised vessel for storage, but a very well thermally insulated container, which facilitates the storage of the cryogen for many months with negligible heat loss. Surplus shaft power output is used to convert compressed air at ambient temperature in the already nearly-full compressed air store to liquid air (the forward conversion process), and during times of high demand, liquid air is converted back to compressed air at ambient temperature, and the contained energy is then converted to electricity and sold to the grid.

In this paper, thermodynamic analyses of both the conversion processes are carried out. Preliminary results indicate that provided the heat pump/heat engine systems are highly efficient, roundtrip efficiencies of: (a) ~70% for the conversion of compressed air-to-liquid air and back, and (b) ~53% for the conversion of electricity-to-compressed air-to-liquid air and back can be obtained.

Future work will include the detailed analysis of the heat pump and heat engine systems, and the economics of the hybrid energy store.

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