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Liquid air energy storage – analysis and first results from a pilot scale demonstration plant

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Keywords

Liquid air

Energy storage

Cryogenic

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Highlights

- A novel cryogenic energy storage system is proposed
- Classical and full cycle analysis is used to describe the process and determine the cycle efficiency
- Results from the testing of a pilot scale prototype are presented
- Scale up of the process and the characteristics of a commercial scale unit are discussed

Abstract

Energy storage is an important technology for balancing a low carbon power network. Liquid Air Energy Storage (LAES) is a class of thermo-electric energy storage that utilises a tank of liquid air as the energy storage media. The device is charged using an air liquefier and energy is recovered through a Rankine cycle using the stored liquid air as the working fluid. The cycle efficiency is greatly improved through the storage and recycling of thermal energy released during discharge and used to reduce the work required to liquefy air during charging. Analysis and results from the design and testing of novel LAES

concept at pilot scale are presented. Fundamental analysis of the LAES cycle is first described to determine the theoretical cycle performance and in particular the value of cold recycle. The pilot plant is then described together with the results of a series of comprehensive technical and commercial trials. The paper concludes with a discussion on the future potential of LAES in particular the fit with the requirements for bulk energy storage and the transition of the LAES technology from pilot to commercial scale.

1. Introduction

With the move to reduce carbon emissions from the electricity network, a higher contribution of intermittent and inflexible generation can be expected, making the balancing of the network more challenging [1]. Electric energy storage can help balance an electricity network through the time shifting of excess energy production to times of high energy demand. Evens [2] described Liquid Air Energy Storage (LAES) as a thermo-electric storage device where energy is stored as a temperature difference between two thermal reservoirs, as opposed to electrochemical or kinetic energy as with other classes of storage. In thermo-electric storage devices, work is extracted from the system by transferring thermal energy from the high to the low temperature reservoir. The process is reversed during charging by doing work on a working fluid to transfer thermal energy from the low to the high temperature reservoir. The ideal thermo electric storage process is reversible [3] but as will be discussed in this paper, a practical LAES cycle has significant irreversibility's and will achieve somewhat less than 100% efficiency (energy recovered during discharge divided by energy input during charging).

The LAES cycle operates in three discrete stages. Electrical energy is first used to liquefy air, which is stored at low pressure in an insulated tank. Cryogenic fluids can be stored for many months in low pressure insulated tanks, with losses as low as 0.05% by volume per day [4]. When power is required, liquid is drawn from the storage tank and compressed, then heated using thermal energy from the environment and expanded through a turbine to recover mechanical work which can be used to turn a generator to provide electricity back to the electricity network. The cycle efficiency is improved through 'cold recycle', which involves capturing and storing the cold thermal energy released during discharge and using the stored energy to reduce the work required to liquefy air during charging. The LAES cycle is particularly interesting to the power utilities, as the component parts are commonly found in power stations and industrial air separation plant. As such, the components are mature, have well understood maintenance requirements and are available at the scales compatible with plant sizes from 10's to 100's MW.

Variants of the LAES cycle were first described by Smith [5] and more recently by Ameel [6]. Smith proposed a cycle utilising adiabatic compression and expansion devices, resulting in a high cycle efficiency of 72% but high temperatures (1048K) and pressures (85bar) in the thermal stores. Ameel described a process utilising a combined Linde refrigeration and Rankine cycle, and calculated efficiencies of up to 43%. The cycles proposed here differ from previous studies in two respects. First, the thermal store is operated at low pressure to overcome the practical difficulties of manufacturing a large pressurised store. Secondly, the liquefier utilises the Claude cycle, where cooling is provided through isentropic expansion in one or more cold turbines as well as isenthalpic expansion through a Joule Thomson valve. The Claude cycle is the most commonly used process for large scale air liquefaction, being significantly more efficient than the Linde cycle [7].

In this paper, the role of energy storage in the power network will be discussed first, to provide market context to the LAES system and generic performance targets for a bulk storage device. The LAES cycle is then described using first classical analysis to investigate the theoretical performance and then detailed cycle analysis to understand the practical potential of the technology. Results from a pilot scale demonstration project are then presented, including performance and commercial trials. The paper concludes with a discussion on the future potential of LAES technology and the likely configuration and performance of a mature commercial scale LAES plant.

1.1 Role of storage in the power network

Recent research by Strbac [8] has shown energy storage could deliver significant efficiency savings in operating and balancing a power grid with a high contribution of intermittent renewable generation. The most valuable contribution of storage identified by Strbac was increasing the overall network efficiency, and in particular the utilisation of highly valuable renewable energy generation assets. The surprising conclusion from this work was the relatively low sensitivity of the value of storage to the efficiency of the storage device. The consequences of Strbacs analysis on the target cost and performance metrics for a large scale energy storage system were discussed in the Liquid Air report produced by the Centre for Low Carbon Futures [9]. A round trip efficiency (AC in to AC out) of >50% was proposed at a cost target 750 to 1250 £/kW. It should be noted that in some storage applications, round trip efficiency is much more important, such as in arbitrage application where the storage device shifts significant quantities of energy [10]. However, Strbac's work identified a valuable balancing service requiring a low cost – modest efficiency device that can be deployed at significant scale. As will be discussed in the rest of this paper, this application is a good fit with the characteristics of a LAES device.

Nomenclature

C, c	Compressor
LAES	Liquid air energy storage
h	Enthalpy
h_l	Enthalpy at bubble line
MAC	Main air compressor
P, p	Pump
P_r	Pressure ratio
Q_r	Cold recycle energy
RAC	Recycle air compressor
STOR	Short Term Operating Reserve
T	Turbine
TS	Thermal store
V	Valve
W	Work
y	Yield
χ	Round trip efficiency

2. Theoretical performance of a liquid air storage plant

2.1 The Liquid Air Energy Storage cycle

The LAES cycle contains three principal parts (fig. 1); a charging device, a liquid and various thermal stores and a generation device. Thermal energy is captured, stored and recycled between the charge and discharge cycles. In a LAES system, unlike a battery these are three physically different components that can be independently sized, i.e. a storage device with a slow charging rate and large storage capacity is possible leading to the opportunity to optimise the LAES system for different applications. The LAES be analysed as follows, referring to the figure 2.

Fig. 1. Block diagram of the LAES system

Fig 2. Flow and TS diagram of the LAES cycle

The round trip efficiency χ can be expressed as:

$$\chi = y\frac{(W_t - W_p)}{W_c} \qquad (1)$$

Where y is the yield (mass of liquid/total mass) of the isenthalpic expansion process (3-4), W_t is the turbine work (2-1), W_p is the pump work (4'-3) and W_c is the compressor work (1-2). Referring to figure 2b, assuming the gas compression (1-2) and expansion (2-1) processes are isothermal and the working fluid behaves as an ideal gas, the main losses in the cycle are the adiabatic work (4'-3) in compressing the cryogenic fluid and isenthalpic expansion (3-4) resulting in incomplete condensation of the working fluid. Using the analysis described by Haywood [11] for a Linde refrigeration cycle modified to include the cooling contribution of the cold recycle (2-3), the yield at (4) is:

$$y = \frac{h_1 - h_2 + Q_r}{h_1 - h_l} \qquad (3)$$

Where h_1 and h_2 are the enthalpies at (1) and (2), h_l is the enthalpy at the bubble line at the same pressure as (4), and Q_r is the cooling contribution from the recovered thermal energy from the heating of the working fluid during discharge:

$$Q_r = h_2 - (h_l + W_p) \qquad (4)$$

The isothermal compression and expansion work can be calculated from the pressure ratio P_r of the process, temperature, T_1 and gas constant R:

$$W = RT_1 lnP_r \qquad (5)$$

Hence, from equations 2-5, the round trip efficiency (1) can be calculated as a function of the charge and discharge pressure ratio as shown in fig. 3. It should be noted that the cycle analysed in this manner does not achieve mass equilibrium from cycle to cycle as the mass of working fluid re-liquefied during the charging phase of the cycle reduces slightly with every repeat cycle due to incomplete condensation at (4). To balance the stored mass over repeat charge and discharge cycles, a supplementary refrigeration device would be required to top up the storage tank. Bearing this limitation in mind, it is apparent from inspection of fig. 3 that the cycle efficiency improves with increasing discharge pressure but beyond 120bar, the benefit is reduced and a theoretical round trip efficiency of the cycle trends to a maximum of 86%. From equation 4, we can see that increasing the discharge cycle working pressure reduces the quantity of cold recycle, but this is more than offset by the increase in work recovery during discharge. Higher discharge pressures also increase round trip efficiency by increasing the yield at (4).

Fig. 3. Theoretical effect of charging pressure on round trip efficiency for different discharge pressures

2.2 Cycle analysis

The analysis described above although useful in illustrating the principles of the LAES cycle but does not account for second law losses in the heat transfer processes and air is normally liquefied using more efficient refrigeration processes, such as the Claude cycle [7]. To capture these effects, full cycle analysis is required to provide a more accurate prediction of the cycle performance. For this analysis, the Aspen HYSYS [12] process simulation code was selected.

A diagram of the LAES cycle model is shown in fig. 4. In this case, a two turbine Claude cycle is used for the liquefier and four expansion stages with interstage ambient re-heating for discharge. The following assumptions were made regarding machine efficiency (table 1).

Fig.4. Baseline LAES cycle diagram

Table 1 Modelling assumptions

The model was run at a range of peak liquefier pressures and peak first stage turbine inlet pressures to investigate the optimal process conditions. The effect of the pressure at the inlet to the Joule Thomson valve on the specific work of the liquefier is shown in figure 5a. A minima is observed at 54bar. Inspection of the modelling results showed little improvement in the liquid yield from the 95% at 54bar for higher pressures, for the penalty of increased compression work. Most modern air liquefiers operate at between 50 and 60bar and so cold recycle has not fundamentally changed the optimal operating pressure of the liquefier, implying current components are likely to be suitable for a LAES plant.

Moving to the power recovery unit, higher peak cycle pressures improve work recovery but as shown in section 2, reduce the quantity of cold recycle. Fig. 5b shows the effect of peak power recovery cycle pressure on overall net cycle efficiency. The increased work recovery from higher pressures more than offsets the reduction in cold recycle, but above 150bar, the benefit is limited. This would suggest the optimal LAES power recovery cycle pressure is between 150 and 200bar, which is comparable with the peak operating pressures of modern steam plant expanders which could form the basis for a large air expander.

Fig.5. Effect of liquefier (a) and power recovery (b) pressures on specific work and net round trip efficiency

3. Pilot Plant

To prove the LAES concept, a pilot scale demonstration plant was constructed. The project started in 2008, with the discharge part of the process being built and run first, and then the liquefier being added to demonstrate the complete cycle. The discharge plant was run on tanker supplied liquid nitrogen until the liquefier was available to provide liquid air. The site was located on Scottish and Southern Energy's Slough Heat and Power station in Slough, United Kingdom.

Fig. 6 Pilot plant P&ID

Fig. 7 Pilot plant layout

A P&ID of the process is shown in fig. 6 and diagram of the plant in fig. 7. Key process parameters are shown in table 2. The peak process pressure in the discharge unit was 60bar and 13bar in the air liquefier. These are below the expected optimal working pressures of a mature LAES system, 150bar and 55bar being more typical, but were selected to fit in with available equipment at pilot scale and the project budget. Only 51% of the available cold thermal energy (defined as the enthalpy recovered from the cold store during charging divided by the enthalpy added to the working fluid during charging between the cryo pump outlet to 273K) was recycled, again to keep the project within the available financial resources.

Table 2 Pilot plant process parameters

The discharge unit consisted of a pair of reciprocating cryogenic pumps, connected via a thermosyphon to the cryogenic liquid storage tank. This ensured the pump's cold head was maintained at the cryogenic liquid temperature. The pumps were driven through a variable speed motor to enable the process pressure to be varied. The cryogenic fluid was heated in three stages, the first two by the exhaust gas from the final turbine stage, and finally by a water-glycol warmed heat exchanger. Cold recycle was only recovered from the first heat exchanger. The process fluid was expanded in four stages through a series of radial inlet turbines, supplied by Concepts NREC. The process gas was re-heated between stages to achieve quasi isothermal expansion. The re-heaters and superheaters were heated using a water-glycol heating circuit. Thermal energy was supplied to the process from steam supplied from an adjacent power station. This enabled the process temperature at the turbine stage inlets to be varied from 288K to 343K, simulating the use of heat from the environment or from various waste heat sources.

The liquefier was supplied by Chengdu Air Separation Corporation and was of a single turbine Claude design. A liquid submerged sub cooler was used at the low temperature end of the cold box. Although this is not the most efficient process for liquid production, it was selected to reduce the risk of ignition of condensed hydrocarbons due to boiling of locally oxygen enriched liquid. Two screw compressors, supplied by Atlas Copco were used to supply and recycle air in the process. Both machines had variable speed motors to enable the air mass flow to be controlled.   Cold was captured and stored in a series of eight packed gravel beds, filled with a quartzite based river shingle sized by a 15mm sieve. The columns of gravel were installed inside a shipping container and insulated using perlite. The columns were interconnected through a series of valves that allowed columns to be connected in parallel and series to optimise cold recovery and minimise pressure drop. This concept of parallel series operation of the cold store is described more completely in [13].

4. Results

4.1 Discharge Unit Testing

The effect of peak cycle pressure and temperature were investigated. Fig. 8a shows the electrical energy generated at a 289K turbine inlet temperature. A linear relationship between turbine work and pressure was observed. Further testing was performed at 48bar turbine inlet temperature to investigate the impact of recycled or waste heat on the cycle (fig. 8b). 45% of the supplied thermal energy (referenced to ambient temperature) was converted to work. This high conversion rate of low grades of waste heat is a key features of the LAES and compares very favourably with the 5-15% efficiency from other systems such as an organic Rankine cycle [14].

The outlet temperature of the low pressure stream of HX1 is critical to the overall performance of the cycle, as it indicates the quality of the cold thermal energy captured for storage and recycle. A 1 to 2K approach temperature between the high pressure liquid and low pressure gas outlet was observed across the full operating range of the pilot plant, demonstrating very efficient cold recovery from the process.

Fig. 8a Effect of stage 1 turbine inlet pressure on generated power at 289K inlet pressure

Fig. 8b Effect of turbine inlet temperature on generated power at 48 bar stage 1 inlet pressure

4.2 Liquefier Performance (cold recycle)

The liquefier was commissioned during April 2010. The target bulk thermal store temperature of 115K was achieved and the specific work of the liquefier was measured across a range of specific cold recycle rates, expressed as the kWhrs per unit mass of liquid production. The cold recycle rate was controlled by varying the mass flow of the blower circulating dry air between the cold store and cold box, at a constant cold store exit temperature. The results of these tests and the predicted results from a cycle simulation model are shown in fig. 9. Close agreement between simulation and test results was observed. As predicted, the specific liquefier work reduced as the cold recycle rate was increased. This demonstrates liquefier efficiency can be controlled by varying the discharge rate of the cold store. This flexibility would enable a commercial plant to exploit low energy prices to generate liquid without depleting the cold store, which could be used to reduce the energy input to the liquefier when energy prices are high, whilst still enabling the liquefier to be operated economically to charge the system.

Fig. 9 Measured and predicted gross liquefier specific work

4.3 Commercial trials

Tests were undertaken to evaluate the performance of the LAES technology against a range of reserve and response services. These services are used today to balance differences between supply and demand over a range of timescales from seconds to hours, and provide reserve capacity in the event of the loss of a generating asset. It is expected more of these ‘ancillary services’ will be required in the future to balance the power network when there is a significant contribution from intermittent generation [15].

4.3.1 Reserve Service Trials

Reserve services provide capacity to balance differences between predicted and actual demand. The United Kingdom National Grid STOR service [16] was used to assess the reserve capability of the pilot plant. The STOR service requires the generating assets to be on load within 15 minutes of a call from the network operator with a very high level of reliability and availability. A simulated STOR service trial was undertaken over a ten day period during which the pilot plant was made available at pre-defined times and calls were made at random to bring the pilot plant on line. The results of the trial are shown in table 3.

Table 3 Results of STOR trial

The STOR trials indicate even at pilot stage impressive reliability and good fit with a reserve type of service. At larger scale, it would take longer to load up the turbines and generate power from a standby condition due to the higher rotational inertia of larger turbomachinary. However, in the case of a LAES plant thermal inertia is much less of an issue during start up compared with other thermal generating technologies such as gas turbines. The thermal inertia in the system provides a heat sink to heat the cold liquid and would allow more rapid heating than in the steady state condition. The power turbines also experience a narrower temperature range compared to a gas turbine, operating from around ambient to around 210K, allowing more rapid loading without risk of thermal fatigue damage.

Table 3

4.3.2 Response Service Trials

The ability to follow rapid variations in demand through load modulation is another critical service for electricity grid balancing. To assess the suitability of the LAES system to this type of service, a 'PJM' [17] test was undertaken. The PJM test is a series of 5 minute duration load ramps, which the generating assets have to follow. The closeness of the actual power dispatched to demand provides a measure of the suitability of LAES to a response service. The pilot plant achieved a mark of 99% against the PJM criteria pass mark of 75% for this service. This illustrates the ability of the LAES technology to load follow.

5. Discussion

5.1 Pilot Plant

The pilot plant project successfully demonstrated the viability of liquid air as an energy storage media, and the value of cold recycle. The process modelling tools developed during the project were also validated against test data. Considerable operating experience was gained during the testing phase of the project, and is discussed below.

5.1.1 Liquid Air Storage

One of the concerns at the start of the pilot plant project was stratification of the liquid air in the storage tank. This could result in local oxygen enrichment of the working fluid and the risk of fire or an explosion if the enriched fluid came into contact with hydrocarbons, such as the lubricating oil for the power turbine. This was managed in the pilot plant by ensuring all parts in contact with the working fluid were oxygen cleaned. Some enrichment was observed in feed pipes to the cryogenic pumps after 5-7 days of down time through preferential boiling of the nitrogen content of the liquid air. Purging of these pipes before operation cleared the enriched fluid from the system. No evidence of stratification or enrichment of the bulk liquid air in the storage tank was observed during the project over two years of operation.

5.1.2 Cold Storage

The cold store was the only component that was not available ‘off the shelf’ from the supply chain. The use of low cost bulk gravel is vital to achieving a low price point, as considerable quantities of cold storage media would be required at commercial scale. Over two years of operation, the quartzite gravel used in the store was thermally stable, and no dust or shattering or the material was observed. This was despite approximate 100 thermal cycles from ambient to cryogenic conditions.

One observation from the cold store testing was ‘settling’ of cold at the bottom of the store due to free convection, particularly if the store was part charged. This was not accounted for in the prototype design, but attention to this in a commercial scale cold store would be relatively straight forward, by careful arrangement of inlet and outlet ducts to the storage cells to ensure maximum energy recovery from the store and minimal dissipation of a part charged store over time.

5.2 Process Efficiency

At commercial scale, a round trip efficiency of >50% is desirable to maximise economic return. Review of the pilot plant design showed the level of cold recycle needed to be increased from 51% to around 91% of the available cold and the process optimised to maximise cooling from adiabatic expansion to achieve this target. Analysis of the baseline cycle shown in fig. 4 showed a pinch at the cold end of the cold box and poor recovery of cold at the warm end of the cold box above 72% cold recycle. This is illustrated in fig. 10, with a flattening of the line above 250kJ/kg cold recycle. This poor utilisation of the cold recycle restricts the roundtrip efficiency to 36%. Re arrangement of the cycle to increase the number of expansion turbines and arrange the turbines to expand in series as well as parallel (see fig. 11 for the revised cold box layout) relieved the ‘pinch’ and enabled 91% of the available cold to be efficiently utilised, as shown in the lower line in fig 10. Analysis by White [18] showed an overall efficiency of 94% (thermal energy recovered / energy supplied) could be achieved from this type of thermal store and so further optimisation of the process to capture more cold recycle was unnecessary.

The heat of compression (or warm recycle) from the RAC and MAC was also recycled to increase the turbine inlet temperatures and improve work recovery during discharge. The peak discharge pressure was fixed at 200bar and the efficiency assumptions listed in table 1 were used. Including all parasitic losses (regeneration air, circulation pumps, cooling fans etc.), a round trip efficiency of 49% at 373K warm recycle and up to 60% at 448K warm recycle was predicted for the ‘best build’ configuration. The best build, described more fully in [19] is more complex than typical smaller scale liquefiers of a few hundred tonnes capacity, but is similar in complexity to large mixed refrigerant LNG liquefiers of 1000 tonne per day or more capacity [20]. A bulk storage device would require a large liquefier of this scale and so the best build is representative of the commercial liquefiers of this size.

Fig. 10 Effect of specific cold recycle on liquefier efficiency for baseline and 'best' build

Fig. 11 Best build cold box process layout

5.3 Scale up

The best build model was used to calculate indicative component metrics for a 100MW/ 600MWhr power storage device that can be re-charged at 20% of the discharge rate. This asymmetry was selected to (a) demonstrate the independence of charge, discharge storage for LAES and (b) is representative of the sort of configurations that could be expected, where a high discharge capacity is advantageous with the ability to re-charge over a more extended period of time. Key component characteristics are summarised in table 4, assuming a cold store filled with quartzite and a liquid warm store filed with ethylene glycol. The calculated component sizes are within the feasible range for current technologies suggesting a low risk scale up of the technology, particularly when compared to other technologies such as flow batteries where scale up has been an issue. The current supply chain is also able to support a large scale LAES plant without significant development of new components, giving a clear path for moving from pilot to first commercial to large scale units [9]. Cost projections for LAES were presented in [9] and indicated a mature cost at scale of £500-750/kW for LAES. This is comparable with fossil fuel power plant costs and very competitive with the cost of other storage technologies [8] and competitive with the target cost for storage. Compared to the metrics for a bulk storage device presented in [9], the LAES achieves the efficiency and cost targets and is scalable to the required plant sizes.

Table 4 Component sizes for a 100MW / 600MWhr LAES device

6 Conclusions

A novel liquid air energy storage concept is described. The cycle efficiency is greatly improved by recycling and storing thermal energy between the charging and discharging parts of the cycle. Testing of a pilot scale prototype demonstrated:

- 45% conversion of low grade heat to power,
- Effective, durable storage of cold in a low cost quartz base packed bed store,
- Long term storage of liquid air without significant enrichment or stratification,
- Effective reduction in liquefier work through cold recycle,
- Rapid response and load following in line with the requirements of grid support ancillary services,
- Close agreement of measured and predicted benefit of cold recycle, indicating good fundamental understanding of the cold recycle process.

Analysis showed conventional twin turbine Claude cycle liquefier process design restricted the level of cold recycle that could be efficiently utilised without convergence of the stream temperatures within the cold box and inefficient utilisation of the cold. Modification of the process design in particular the number and position of cold expansion turbines enabled more cold to be used more efficiently. The optimised process model predicted a round trip efficiency of up to 60%, above the target of >50%.

Analysis of key component sizes at scale indicated the current power and process industry supply chains could service a LAES system of at least 100MW capacity. Efficiency and projected costs are comparable with the projected requirements for bulk storage in a power network with a high contribution of intermittent generation. LAES therefore presents an attractive storage technology for balancing a future low carbon power network.

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Figures

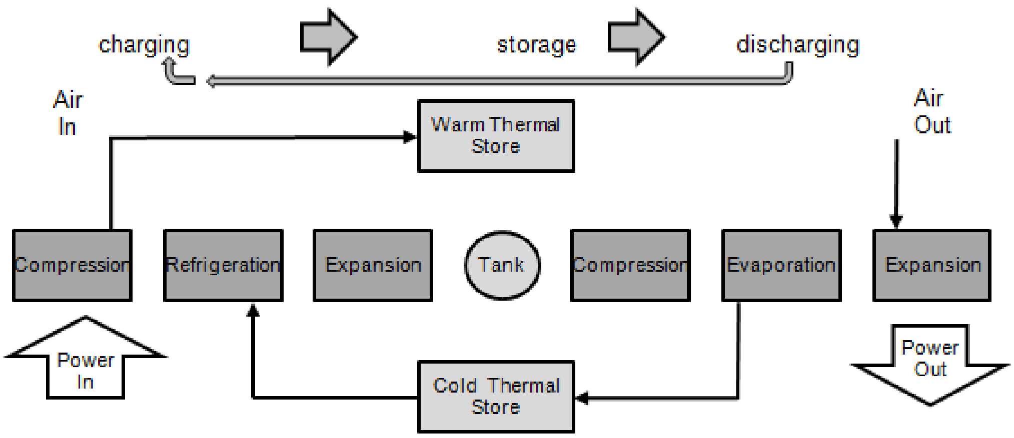

Fig. 1. Block diagram of the LAES system

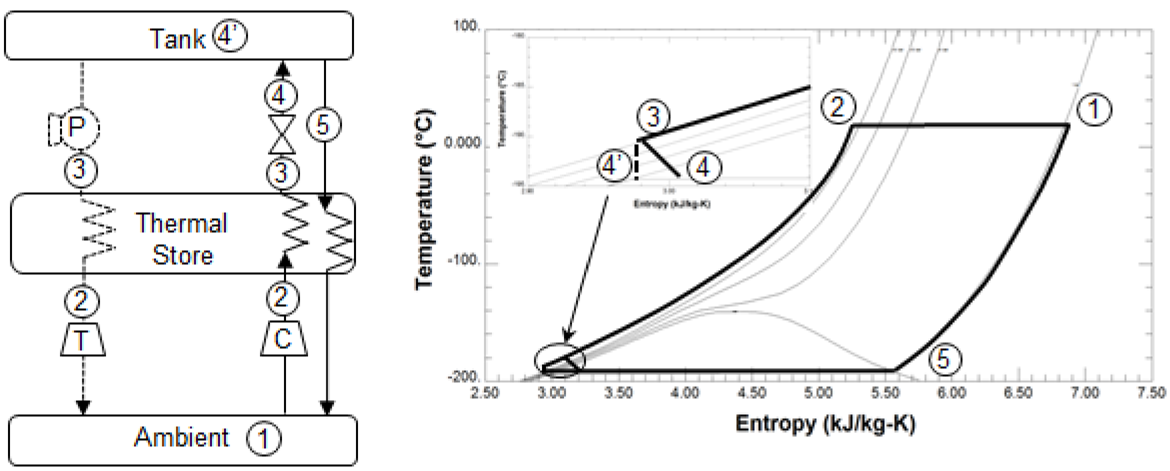

Fig 2. Flow and TS diagram of the LAES cycle

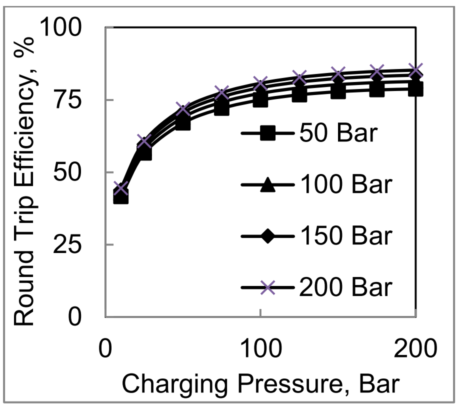

Fig. 3. Theoretical effect of peal charging pressure on round trip efficiency for different liquefier peak pressures

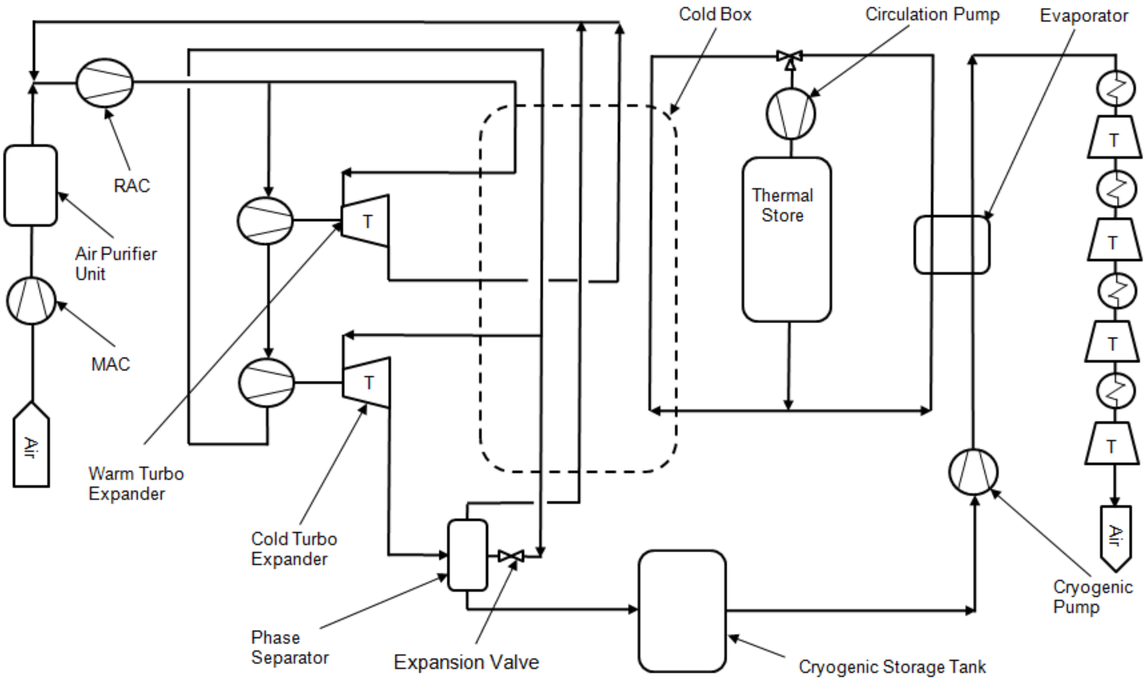

Fig.4. Basic LAES cycle diagram

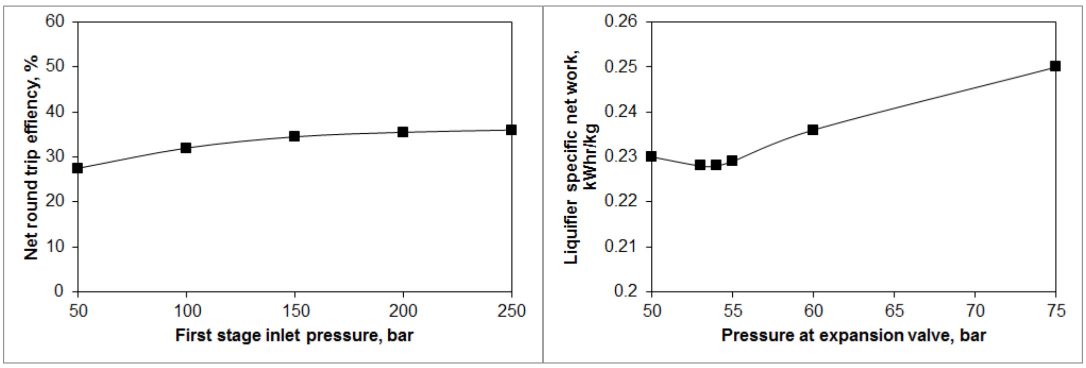

Fig.5. Effect of liquefier and power recovery peak pressures on specific work and round trip efficiency

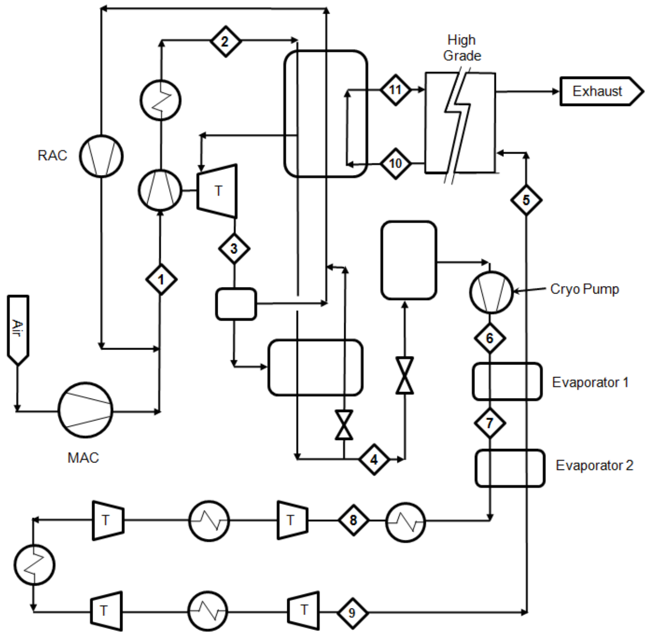

Fig. 6 Pilot plant process layout

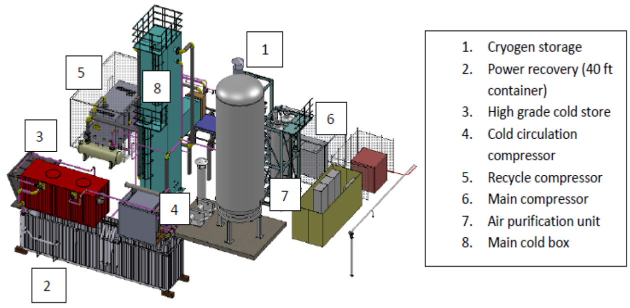

Fig. 7 Pilot plant layout

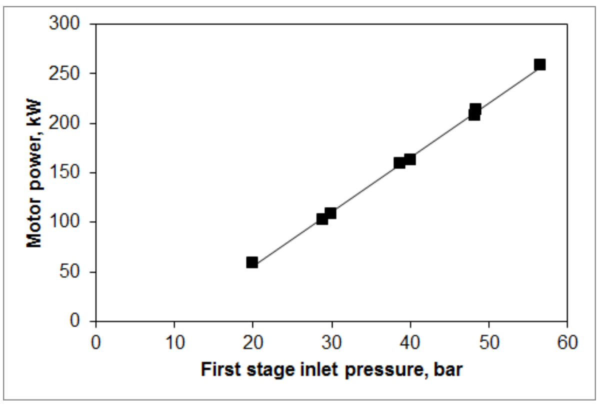

Fig. 8a Effect of stage 1 turbine inlet pressure on generated power at 289K inlet pressure

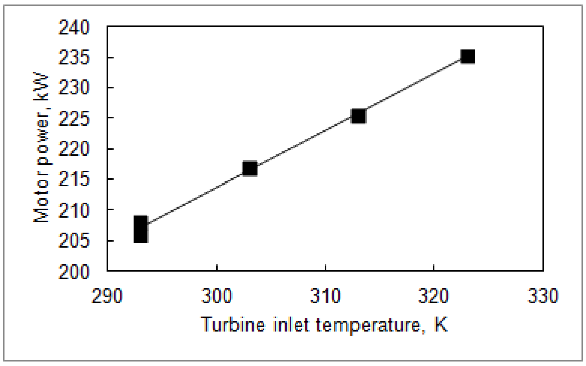

Fig. 8b Effect of turbine inlet temperature on generated power at 48 bar stage 1 inlet pressure

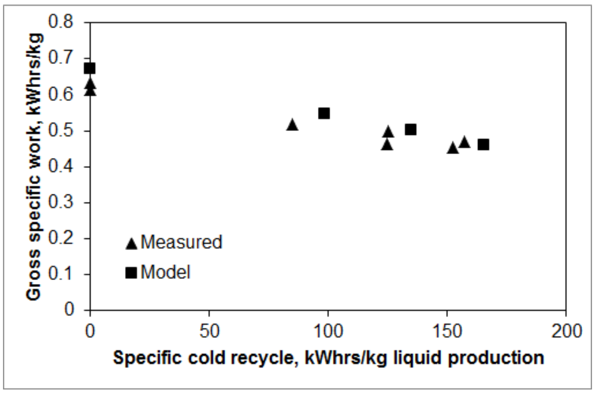

Fig. 9 Measured and predicted gross liquefier specific work

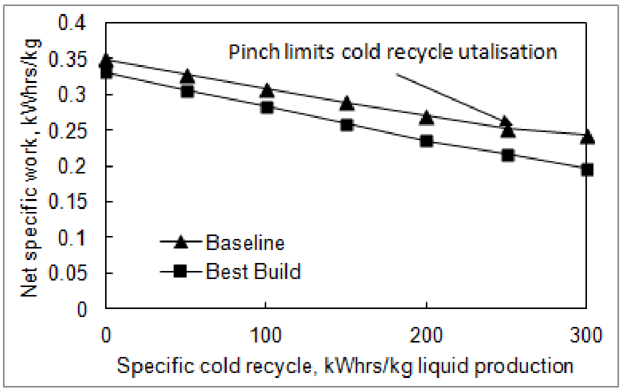

Fig. 10 Effect of specific cold recycle on liquefier efficiency for baseline and 'best' build

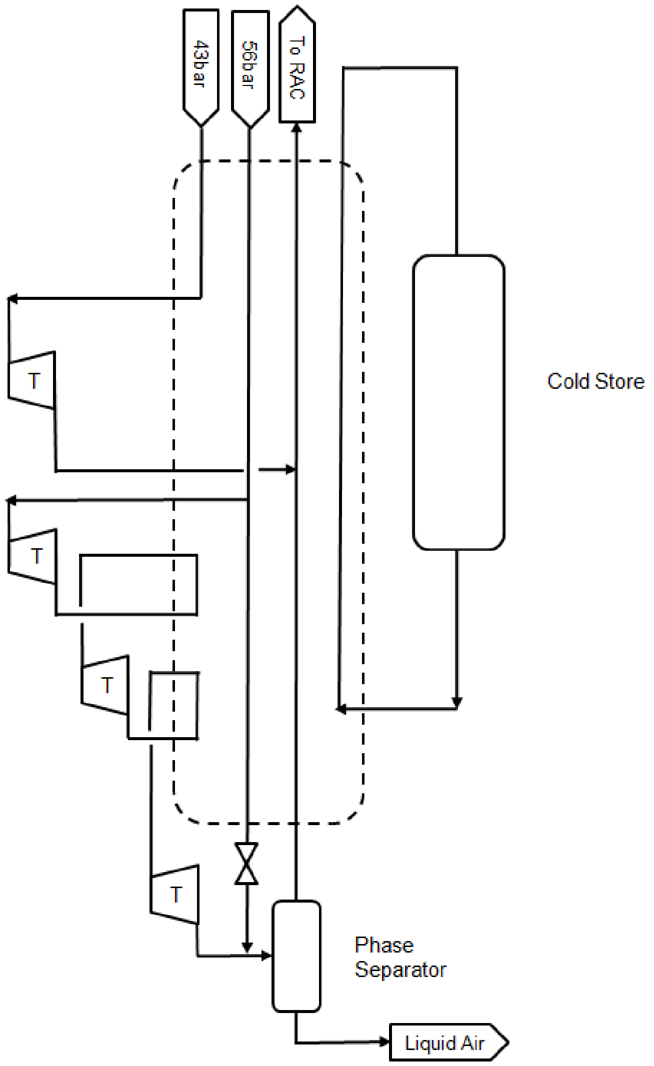

Fig. 11 Best build process layout

Tables

Compressor efficiency	89%
Turbine efficiency	90%
Electrical machine efficiency	97%
Cryogenic Pump Efficiency	80%
Minimum approach temperature in heat exchangers	1K

Table 1 Modelling assumptions

	Pressure	Temperature	Mass Flow
1	10bar	301K	2.3kg/s
2	12bar	298K	2.3kg/s
3	1.3bar	84K	1.8kg/s
4	12bar	98K	0.4kg/s
5	1.2bar	106K	2kg/s
6	56bar	104K	2kg/s
7	56bar	109K	2kg/s
8	51bar	337K	2kg/s
9	1.7bar	262K	2kg/s
10	1.2bar	115K	0.4kg/s
11	1.0bar	296K	0.4kg/s

Table 2 Pilot plant process conditions

Response time (to load set point)	100s	
Total time on call	2160mins	100%
Generating time	245mins	10.6%
Recovery time (shutting down and re-starting before and after generation	108mins	5%
On standby	1800mins	84%
Unplanned outage	7mins	0.3%
Availability (either on standby or generating)	2045mins	94.6%
Reliability	2153mins	99.7%

Table 3 Results of STOR trials

Discharge turbine mass flow rate	233kg/s
RAC mass flow rate	103kg/s
Cold Store volume	5800m^3
Warm Store mass	1763 tonnes
Liquid Air tank mass	5032 tonnes

Table 4 Parameters for a 100MW / 600MWhr LAES system