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Modelling of liquid air energy storage applied to refrigerated cold stores

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ABSTRACT

Liquid Air Energy Storage (LAES) is a promising technology for dealing with the variability in production of various concurrent Renewable Energy Sources (RES). In this context, the work presented forms part of the CryoHub project. CryoHub is an H2020 Innovation Action project to investigate and demonstrate the feasibility of using LAES in conjunction with refrigerated warehouses cooling. In this paper, multiple different configurations that could achieve the project goals have been modelled and ranked against Round Trip Efficiency (RTE). A configuration currently being deployed is presented and its merits are discussed. The effect of multiple process parameters on the RTE are assessed and discussed.

Keywords: Liquid air energy storage, cold storage, LAES, cryogenic

1. INTRODUCTION

The EU aims to achieve a 20% RES penetration in the electric energy sector by 2020. The challenge associated with grid balancing using increasing amounts of intermittent RES production has been recognized at both EU and national level. A working paper by the EU states: "European and global energy policies based simultaneously on a reduction of CO_2 emissions, a shift towards intermittent renewable power while maintaining secure energy supplies changes the ground rules for storage and calls for a new approach to storage as a key component of the future low-carbon electricity system."…" Energy storage can become an integrated part of Combined Heat and Power (CHP), solar thermal and wind energy systems to facilitate their integration in the grid." (EC, Directorate General for Energy, 2015). In this context, LAES is a relatively new technology whose applicability for large scale energy storage is not bound by geological features as it is the case for compressed air and hydro energy storage. Therefore, it could provide the flexibility needed to cope with large fluctuations in RES production at a local or national electrical grid level.

CryoHub is a European research project that aims to develop cryogenic energy storage of renewable energy to refrigerate food storage warehouses and to enhance power grid sustainability. In its entirety, the CryoHub concept would contain the following subsystems:

- Air liquefaction sub-system for storage of excess renewable energy or electricity from the grid when demand is low;
- Liquid Air (LA) storage in a pressurized cryogenic vessel;
- Discharge of LA for warehouse refrigeration and energy production when electrical demand is higher than RES production.

This paper focuses on activities performed so far as part of the CryoHub system modelling. It describes software models for liquefaction, cold-energy-storage and discharge loops, global running logic, parametric analysis and main drivers to achieve the best RTE and some operational parameters related to the reference cycle for the CryoHub demonstrator. The results are presented, and the design parameters of the demonstrator discussed.

2. CRYOHUB INTEGRATION ONSITE

The CryoHub system, as first envisaged in a research proposal funded by the European Commission in early 2016, would constitute a hub where LA would be used as an energy storage medium to be used site-wide across a range of different applications. Air liquefaction would take place at peak RES production or during off-peak times. The LA could be stored and used either directly or indirectly (via a secondary heat transfer fluid circuit), in refrigerated facilities and refrigerated transportation. The production of electrical energy from the enthalpy differential stored within the LA would take place via multiple expansion turbines generating electricity powering the site or feeding the grid at peak demand times. The large enthalpy change during evaporation of LA could be also stored and recycled to increase the liquefaction plant LA yield. Integration with onsite waste heat streams was also identified as a way to boost electricity production.

In this context, the industrial site where the CryoHub demonstrator will be implemented consists of three companies working together implementing system wide optimization, recycling and energy recovery. One of the companies produces and exports large quantities of food which arrives partially frozen to the second company which takes care of deep freezing and storage. The third company consists mainly of a joint venture between the first two companies and external stakeholders and processes the food processing waste from the first company to produce bio-methane which feeds piston generators that export around 1.6MW of electricity to the grid and reject waste heat back to the food processing plant. The site further benefits from additional local 1MW peak solar Photo-Voltaic (PV) RES production and a 5MW Combined Heat and Power (CHP) plant running with methane from the gas network and providing heat at up to 800ºC. Additionally, there are plans to increase local production from RES by installing a 2MW wind turbine. Therefore, the site would be open to the possibility to incorporate LAES as a long term solution to grid fluctuations, reducing grid reliance on an industrial scale.

3. POSSIBLE CRYOHUB CONFIGURATIONS

The CryoHub system modelling activities initially considered 4 different methods that could be implemented alternatively or in parallel. These can be categorized in two separate concepts: standalone or with waste heat recovery. The LA discharge system is approximately equivalent to a Rankine cycle, with some additional features. The LA is pumped up to a set pressure or expanded from the LA storage pressure after evaporation.

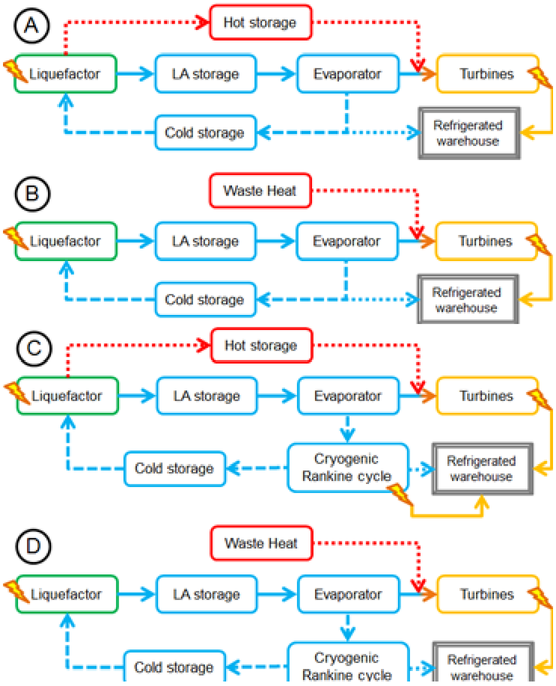

Figure 1 - CryoHub concept investigated: (A) CRYOBUB LAES with high temperature cold use in refrigerated warehouse and hot/cold energy recycle; (B) CRYOBUB LAES with high temperature cold use in refrigerated warehouse and hot/cold energy recycle; (C) CRYOBUB LAES as A) plus additional Cryogenic Rankine cycle and cold energy use in refrigerated warehouse; (D) CRYOBUB LAES as B) plus additional Cryogenic Rankine cycle and cold energy use in refrigerated warehouse

CYCLE A AND CYCLE B

In a system similar to a previous LAES demonstrator deployed by Highview and described in Morgan, *et al.* (2014) the heat rejected during LA production is stored for later use to re-heat the flow at the turbine inlet or between stages. Similarly, most of the heat of evaporation absorbed by the LA during discharge could be recycled via a cold Heat Storage Medium (HSM) and fed back to the liquefactor during LA production. However, some of the high temperature cold energy could still be used in the refrigerated warehouse for cooling. This configuration would be able to recover most of the energy output in the form of electricity production. The concept is outlined in Figure 1 (A). The use of cold energy from the LA evaporator at lower temperatures than a normal refrigerated warehouse (running for example at -20ºC) may be beneficial when considering fast freezing. The use of liquid nitrogen spray freezing of food product is carried out in some food processing plants. Cryogenic cooling provides a short term increase in freezing capacity at a fraction of the footprint and CAPital Expenditure (CAPEX) of a mechanical blast freezer. This is due to the increased heat transfer coefficient and temperature difference between the food product and its freezing medium. Therefore, if the food processing plant already uses cryogenic freezing methods switching to LA for fast freezing and additional electric energy output has little impact on the production process. Concept (A) would require the least amount of energy streams integration and would be mainly suited for deployment in the refrigerated warehouse. Along these lines, concept (B) makes use of a waste heat energy stream and avoids storing hot energy from the liquefactor, thus reducing CAPEX.

3.1. CYCLE C AND CYCLE D

In (C) and (D) the LA evaporator could be used as a condenser for an additional Cryogenic Rankine cycle using the enthalpy of evaporation of LA at a fixed low temperature point, while the refrigerated warehouse thermal mass provides the temperature point at which heat is extracted. Cycle (C) and (D) will generate electrical energy while transferring cold energy from the LA to the refrigerated warehouse. Some of the cold energy could also be recycled back to the liquefactor plant after storage in a cold HSM. However, the smaller amount of cold energy being recycled to the liquefaction plant means a smaller LA yield and, in turn, higher energy input to the liquefactor. In turn, the increased complexity makes more cooling power available to the refrigerated warehouse and some relatively small increase in electricity production. In parallel with Cycle (B), concept (D) makes use of a waste heat energy stream and avoids storing hot energy from the liquefactor.

4. MODELLING APPROACH

Three configurations (A, B and D) were modelled by using NIST REFPROP (NIST, 2017) connected to Microsoft Excel and by using basic thermodynamic modelling methods. Once built, the liquefaction part of the model was cross checked against and independent Aspen Hysys model by Air Liquide - Centre de Recherche de Paris Saclay (AL-CRPS). The results of the comparison for a reference case have been reported in Table 2 and Figure 2. The turbomachinery components efficiencies are assumed to be "state-of-the-art" as if the system was designed and run at multi MW/MWh scale and are reported in Table 1.

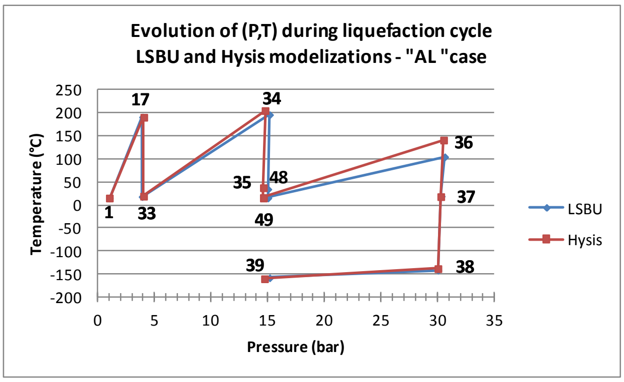

Figure 2 - AL-CRPS checks of Cryohub Liquefaction model by LSBU

Table 1 - turbomachinery and heat exchangers key performance parameters	
Turbomachinery parameters	
Component	Isentropic efficiency
Top-up compressor	85.0%
Liquefaction Compressor	85.0%
Liquefaction Turbine	75.0%
Liquid Air Pump	50.0%
Rankine Cycle Pump	50.0%
Liquid Air Discharge Turbines	87.5%
Rankine Cycle Turbine	85.0%
Heat exchangers parameters	
Pressure drop in heat exchangers	1.0%
Pinch point in heat exchangers	4.0 K

Table 2 – LAES liquefaction loop checks with Aspen Hysys

#	Component	LSBU results		AL-CRPS results		%error pressure	% error temperature
		P [bara]	T [K]	P [bara]	T [K]		
1	Inlet	1.013	288.15	1.013	288.15	0%	0%
17	LP Compressor-out	3.94	464.25	4.05	463.65	3%	0%
33	HXc1-out	3.9	291.15	4.03	292.15	3%	0%
34	HP Compressor-out	15.15	469.25	14.75	478.15	-3%	2%
35	E-104-out	15	307.45	14.6	310.55	-3%	1%
48	REC-LP-out	15	288.15	14.6	288.15	-3%	0%
49	CL-Comp-in	15	290.15	14.6	289.15	-3%	0%
36	CL-Comp-Out	30.6	377.65	30.5	414.15	0%	10%
37	Aftercooler-Out	30.3	292.15	30.25	291.15	0%	0%
38	REC-HP-out	30	131.45	30	134.65	0%	2%
38	EXP-out	15.15	114.75	14.7	112.65	-3%	-2%

An interesting modelling problem was the calculation of the thermal-energy-storage efficiencies. The HSM was chosen to be gravel. Thermal properties as a first order approximation could be considered similar to quartzite and were taken from (E.D. Marquardt , et al., 2001). A Heat Exchanger (HE) with the air being charged / discharged transfers the cold energy being stored. Both the cold and hot stores suffer losses due to heat leaks to/from ambient. The recirculation of the secondary Heat Transfer Fluid (HTF) will warm up both the stores with a detrimental effect on the "cold energy quality", since an increase in temperature would decrease the cold energy available. An increase in temperature of the hot store can however be considered beneficial.

The thermocline is the portion of the store within the thermal gradient. The energy needed to change the temperature between the upper and lower thermal store temperatures is degraded in "quality" as it changes in temperature in the heating and cooling of the thermal storage material. Its embodied energy therefore becomes less and less “re-cyclable” in the LAES system at a later stage. In fact, it is “crystalized” in the thermal gradient at variable temperatures despite being supplied at constant temperature. The recycling of the hot/cold energy stored at a variable temperature is difficult and of little practical use in both the liquefaction and energy recovery process. Therefore, all the energy stored within the thermocline has conservatively been considered lost.

To account for the cold energy production, a Coefficient Of Performance (COP) was calculated as described in (Cleland, 1994) for each of the cold energy streams being produced. These cold energy streams are recovered from the LA evaporator and downstream of the turbines whenever possible. In fact, for high discharge pressures and/or low temperature energy recycling, the turbines outlet temperature may go below the ambient temperature and provide some useful cold energy output that can be made available to the refrigerated warehouse. All energies have been measured in kWh of electricity equivalent. Therefore, the RTE has been calculated as an equivalent electrical energy output / the electrical energy input. Overall, the formula used was as follows:

$$\boldsymbol{RTE} = \frac{\sum \boldsymbol{El.Energy\ Out} + \sum \boldsymbol{Cold\ Energy}/\mathrm{COP}}{\sum \boldsymbol{El.Energy\ In}} \qquad \text{Eq.(1)}$$

Where:

$$El.\,Energy\ Out = \ Electrical\ Energy\ Output$$
$$Cold\ Energy = Cold\ Energy\ Output$$
$$COP = \ Coefficient\ Of\ Performance$$
$$El.\,Energy\ In = Electrical\ Energy\ Input$$

The charging and discharging time was considered constant at 10 h and 2.5 h respectively. This approximates the current electricity tariffs bands in Europe. Different countries, even within the EU may have different tariffs and energy policies, but, broadly, it is a reasonable assumption.

5. SYSTEM WIDE PARAMETRIC ANALYSIS

The CryoHub system models were used to run a series of parametric analyses aimed at providing the driving parameters to achieve a high RTE with currently available turbomachinery and heat exchangers. A parametric analysis was done considering:

- 3 different cycles for a CryoHub system: Cycles (A) , (B) and (D)
- Variable LA storage pressure between 2 and 36 bara
- Variable liquefaction cycle maximum pressure before expansion takes place and the cold stream reaches the LA storage pressure. The maximum pressure of the liquefaction system was set to 180 bara and the maximum pressure ratio was set to 30:1; while the minimum pressure ratio was set at 2:1.
- Variable discharge pressure between 100 bara and the LA storage pressure (avoiding having a LA pump)

The results of the parametric analysis are reported in Figure 3 to Figure 8. There is a general benefit in aiming for higher pressures at the inlet of the liquefaction turbine as it increases the PR, the temperature drop and the LA yield.

The RTE strongly depends on the pressure ratio in the liquefaction cycle considered. But, technical talks with Air Liquide Centre de Recherche de Paris Saclay (AL-CRPS) stressed the fact that most heat exchangers in the cryogenic industry operate at relatively low pressures. Therefore, the maximum pressure was fixed at 180 bara and the maximum pressure ratio was fixed at 30:1 in order to match what was “readily available” on the market. The pressure ratio across the liquefaction turbine was consequently a function of the LA storage pressure with an upper limit set at 30:1.

5.1. Results

Comparing Figure 3 and Figure 6 for a standalone system [Cycle (A)], there was a significant RTE penalty in applying a low LA storage pressure. This was due to the combined effect of the increase in the liquefaction temperature and hot energy storage temperature that takes place with increasing LA storage pressure. Additionally, the COP of the liquefaction system increased in parallel with a decrease in enthalpy change needed for liquefaction to take place at higher pressures.

In Figure 3 and Figure 6 the effect of increasing the LA discharge pressure had a generally positive effect on the RTE in a standalone configuration [Cycle (A)]. This was true up to the point where the increase in LA discharge pressure did not achieve a net positive energy yield in the discharge turbines and cold energy recovery circuit. As the discharge pressure increased, the enthalpy change the LA decreased dramatically, with a direct reduction of the cold energy recovered. This effect is presented in Figure 9 [Cycle (A)] and Figure 10 [Cycle (B)] at 36 and 2 bara LA storage pressure respectively. The evaporator, the cold energy recovery and the liquefaction plant of cycle (A) and (B) are identical. Therefore, the two plots are directly comparable.

In Figure 4 and Figure 7, in cycle (B) the supply of high grade waste heat at 800 ºC achieved a higher energy output from the turbines, especially at high LA discharge pressures. The amount of specific cold energy recycled was constant compared to cycle (A) while the energy output and, conversely, the RTE is higher. This was true across the entire range of LA storage pressures. However, this was particularly noticeable at low LA storage pressures and low pressure ratios across the liquefaction turbine. One would expect that a lower pressure ratio and LA yield would impact the RTE significantly. Nevertheless, the increase in RTE was due to the constant LA specific cold energy recycle and the reduced energy input into the liquefaction compressor and top-up compressor. Furthermore, the increase in specific cold energy recycling without the penalty of a not having enough hot energy recycling from the liquefaction compressors stages, “artificially” raised the RTE of the LAES system.

Thermodynamically, reducing the specific cold energy recycling increased the energy input in the liquefaction plant and therefore tended to reduce the RTE. This was true in general but in particular when considering Cycle (C). Therefore, the integration of the waste heat stream in Cycle (D) was considered. This choice was justified in light of the previous results achieving a better RTE in Cycle (B) than in Cycle (A). Analysing Cycle (D) performances, in Figure 5 and Figure 8 there was a much lower RTE for equal LA discharge pressures when compared to Cycle (B). The effect was exacerbated at low LA storage pressures. The reason for this was that the Cryogenic-Rankine cycle energy output did not make up for the loss in high grade cold energy recycled back to the liquefaction plant. At the same time, the increase in LA storage pressure was found to be beneficial in terms of the liquefaction system COP. Since Cycle (C) is thermodynamically bound to achieve lower RTE than cycle (D), further modeling into Cycle (C) was not justified for the purpose of this paper.

For some of the lowest pressure ratios across the turbine in the liquefaction loop, the processed mass-flow was not able to be liquefied. Therefore, since no convergence was reached, those data points were omitted from this discussion.

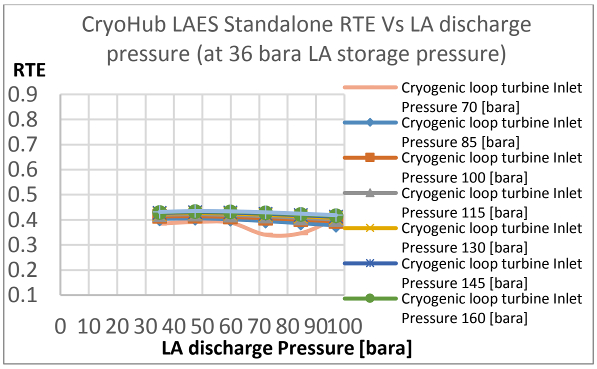

Figure 3 - Standalone CryoHub RTE performance at 36 bara LA storage pressure and varying LA discharge pressure

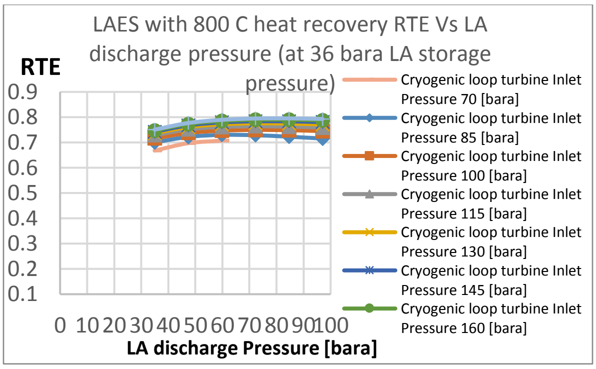

Figure 4 - Heat recovery CryoHub RTE performance at 36 bara LA storage pressure and varying LA discharge pressure

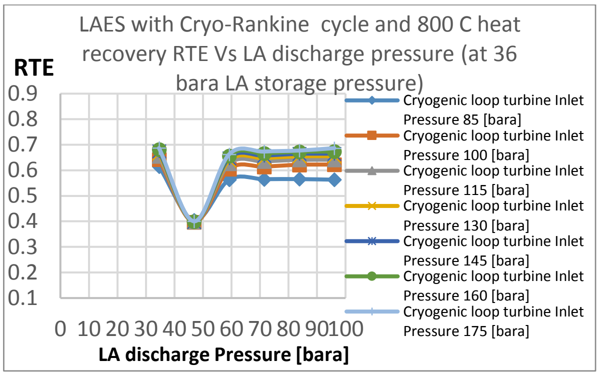

Figure 5 – Heat recovery and Cryo-Rankine cycle CryoHub RTE performance at 36 bara LA storage pressure and varying LA discharge pressure

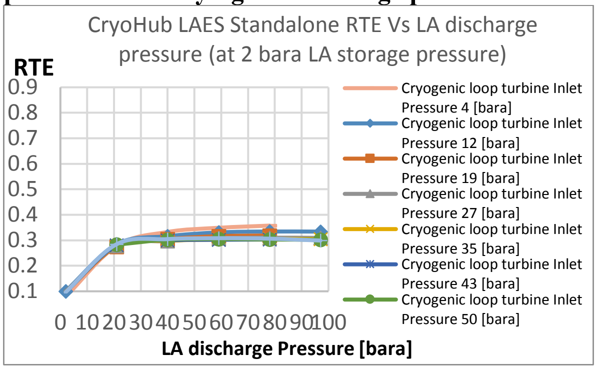

Figure 6 - Standalone CryoHub RTE performance at 2 bara LA storage pressure and varying LA discharge pressure

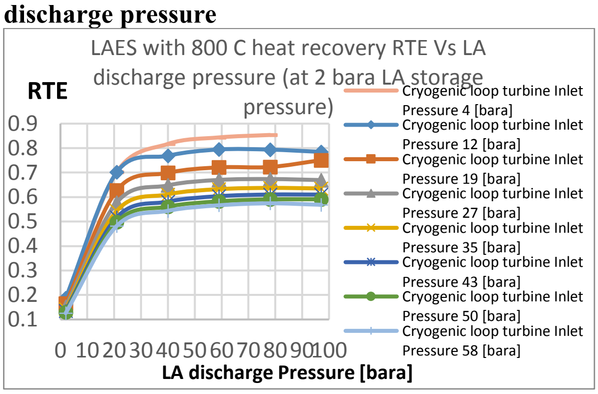

Figure 7 - Heat recovery CryoHub RTE performance at 2 bara LA storage pressure and varying LA discharge pressure

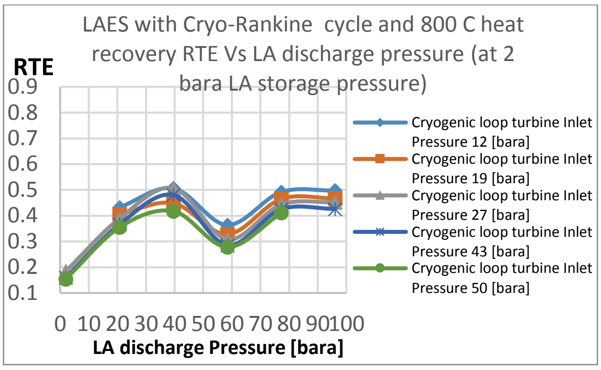

Figure 8 – Heat recovery and Cryo-Rankine cycle CryoHub RTE performance at 2 bara LA storage pressure and varying LA discharge pressure

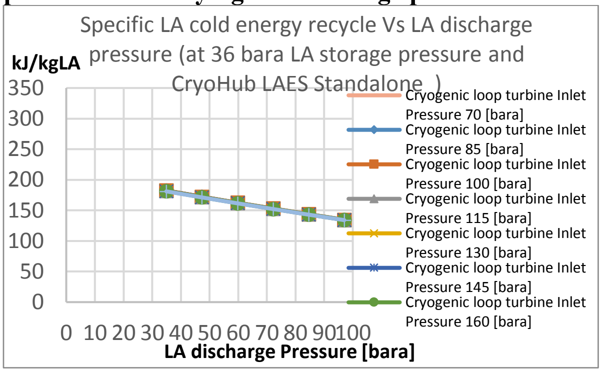

Figure 9 - CryoHub LAES Cycle (A) specific LA energy recycle performance at 36 bara LA storage pressure and varying LA discharge pressure

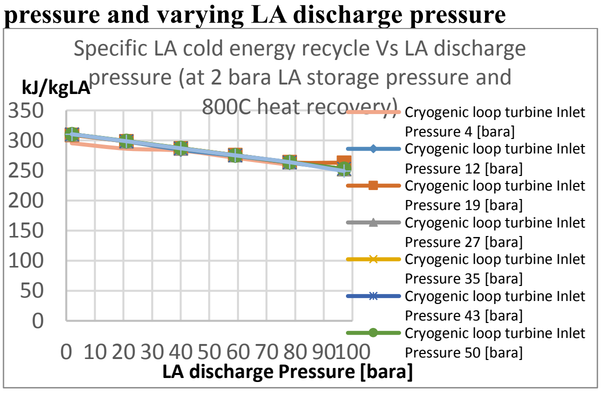

Figure 10 - CryoHub LAES Cycle (B) specific LA energy recycle performance at 2 bara LA storage pressure and varying LA discharge pressure

6. DEMONSTRATOR DESIGN OUTLINE AND PREDICTED PERFORMANCES

The CryoHub demonstrator has been subject to a series of technical and budgetary constraints that somewhat reduced the scope of the demonstrator.

The main constraint was the cost of procuring a liquefaction plant to be coupled with the LA discharge cycle. This was found to be too expensive and was a significant factor affecting design choices downstream. The hot storage could not be implemented and therefore, the aim changed to demonstrate as much cooling capacity as could be delivered. Instead of liquefaction, the cryogen will be transported on-site when needed.

A second constraint consisted of a reduced discharge pressure compared to what the model showed would be ideal. This was mainly due to the use of existing hardware designed for pressures up to 20 bara. The maximum discharge pressure was set to 15 bara, with the option of using a pressurized tank that avoided the need for a cryogenic pump to increase the LA pressure.

A third constraint came from the turbine supplier and consisted of a minimum working fluid temperature of -50 ºC. This set the minimum number of stages to 3 with inter-stage heating to -7.5ºC. The ideal scenario would see an increased number of stages, but budgetary constraints again limited the possible design choices.

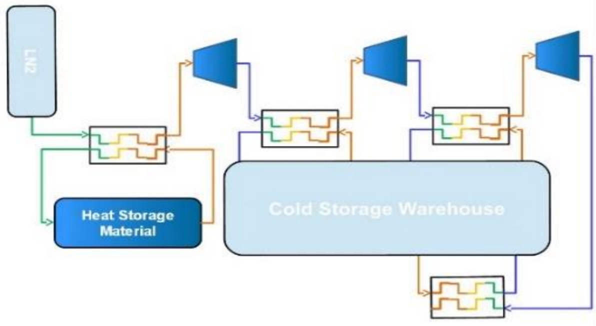

Figure 11 - CryoHub demonstrator P&ID

The LAES system being deployed is schematically shown in Figure 11. The Liquid Nitrogen (LN2) tank will feed the discharge and power generating cycle. The LN2 will be evaporated and the cryogenic energy either stored or used in the warehouse either directly (via evaporators), or indirectly (via a secondary coolant loop). The turbines will be fed at -7.5ºC and additional cold energy will be recovered from the outlet flow after each stage. Electrical energy output from the turbines will be transformed to standard main frequency and fed to the refrigerated warehouse to be used onsite.

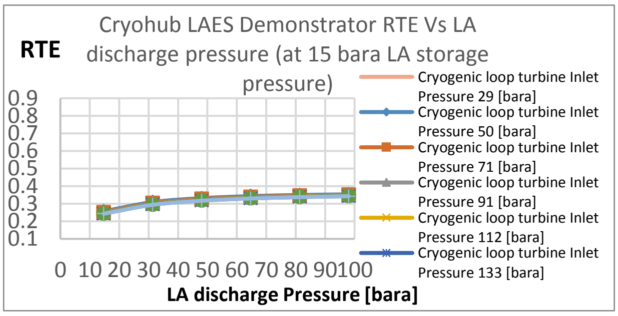

Figure 12 - CryoHub LAES demonstrator RTE performance at 15 bara LA storage pressure and varying LA discharge pressure

Figure 12 shows the results of the parametric analysis run on the CryoHub configuration being deployed. The system will lack a liquefactor. But if it were available, the demonstrated RTE would be between 20% and 26% depending on the LN2 storage pressure and discharge turbines performance; while the thermal energy store efficiency has been predicted to be around or above 88%. Therefore, the equivalent RTE performance calculated for the CryoHub standalone LAES as it is being deployed is predicted to be above previous RTEs reported in Morgan, *et al.* (2014).

A previous LAES demonstrator by Highview claimed a cold-energy-storage efficiency of around 50% Morgan, *et al.* (2014). However, during the CryoHub project, the thermal energy storage has been the result of very detailed design and development activities. New IP has been created and will be distributed after extensive field testing and model validation. Being able to recover close or above 90% of the cryogenic energy stored has been recognized to be a key enabler for LAES systems by Morgan, *et al.* (2014). Therefore, the thermal energy store testing in a real-life industrial environment is likely to provide new and innovative insights on how the challenges associated with LAES feasibility could be tackled.

7. CONCLUSION

In conclusion, the current paper has presented the results of various parametric analyses run as part of the CryoHub system development. The outline and predicted performance of the demonstrator have been presented and assessed against key performance indicators. The RTE of the demonstrator has been predicted to be above previous LAES systems already developed. The equivalent RTE performance calculated for a CryoHub standalone LAES is predicted to fall within the 20-26% range. The cold energy recovery sub-system is predicted to achieve efficiency above 88%. This will constitute a key technological enabler for any LAES technology to be commercially viable in the future.

8. ACKNOWLEDGMENTS

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9. ACRONYMS AND NOMENCLATURE

AL-CRPS : Air Liquide - Centre de Recherche de Paris Saclay
CAPEX : CAPital EXpenditure
CES : Cold Energy Storage
CHP : Combined Heat and Power
COP : Coefficient Of Performance
CSW: Cold Storage Warehouse
HTC : Heat Transfer Coefficient
HTF : Heat Transfer Fluid
HSM : Heat Storage Medium
JT : Joule-Thomson
LA : Liquid Air
LAES : Liquid Air Energy Storage
LN2 : Liquid Nitrogen
PR : Pressure Ratio
RES: Renewable Energy Sources
RTE : Round Trip Efficiency
TRL : Technology Readiness Level

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