Liquid air energy storage for combined cooling, heating and power : techno-economic performance enhancement through waste heat & cold recovery

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**LIQUID AIR ENERGY STORAGE FOR COMBINED
COOLING, HEATING AND POWER – TECHNO-
ECONOMIC PERFORMANCE ENHANCEMENT
THROUGH WASTE HEAT & COLD RECOVERY**

ALESSIO TAFONE

**Interdisciplinary Graduate School
Energy Research Institute @ NTU**

2020

**LIQUID AIR ENERGY STORAGE FOR COMBINED
COOLING, HEATING AND POWER – TECHNO-
ECONOMIC PERFORMANCE ENHANCEMENT
THROUGH WASTE HEAT & COLD RECOVERY**

ALESSIO TAFONE

INTERDISCIPLINARY GRADUATE SCHOOL

A thesis submitted to the Nanyang Technological University in
partial fulfilment of the requirement for the degree of Doctor of
Philosophy

2020

Statement of Originality

I hereby certify that the work embodied in this thesis is the result of original research, is free of plagiarised materials, and has not been submitted for a higher degree to any other University or Institution.

22nd December 2019

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Date

Alessio Tafone

.....

Alessio Tafone

Supervisor Declaration Statement

I have reviewed the content and presentation style of this thesis and declare it is free of plagiarism and of sufficient grammatical clarity to be examined. To the best of my knowledge, the research and writing are those of the candidate except as acknowledged in the Author Attribution Statement. I confirm that the investigations were conducted in accord with the ethics policies and integrity standards of Nanyang Technological University and that the research data are presented honestly and without prejudice.

22nd December 2019

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Date

A handwritten signature in blue ink, consisting of stylized initials 'AR' followed by a long, sweeping horizontal stroke.

.....

Asst. Prof. Alessandro Romagnoli

*“Finirai per trovarla
la Via...se prima hai
il coraggio di perderti”*

Tiziano Terzani

*“You will find your Way
in the end... if you are brave
enough to get lost first”*

Tiziano Terzani

*“We don't see things as they are,
we see them as we are”*

Anaïs Nin

Authorship Attribution Statement

This thesis contains material from 12 papers published in the following peer-reviewed journals and conference proceeding where I am listed as an author.

Chapter 3 is published partially as:

- 1) Borri E, Tafone A, Romagnoli A, Comodi G. *A preliminary study on the optimal configuration and operating range of a “microgrid scale” air liquefaction plant for Liquid Air Energy Storage*. Energy Convers Manag 2017;143:275–85;
- 2) Tafone A, Romagnoli A, Li Y, Borri E, Comodi G. *Techno-economic Analysis of a Liquid Air Energy Storage (LAES) for Cooling Application in Hot Climates*. Energy Procedia, vol. 105, 2017.

The contributions of the co-authors for the 1st paper 1 are as follows:

- I and Dr. Emiliano Borri wrote the drafts of the manuscript. The manuscript was revised by Prof. Alessandro Romagnoli and Prof. Gabriele Comodi;
- I designed the configurations layout and Dr. Emiliano Borri performed the steady simulations and carried out the sensitivity analysis;
- I and Dr. Emiliano Borri provided the discussion and interpretation of results.
- Prof. Gabriele Comodi, Prof. Yongliang Li and Alessandro Romagnoli provided guidance on the technical assessment.

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- I and Dr. Emiliano Borri wrote the drafts of the manuscript. The manuscript was revised by Prof. Alessandro Romagnoli and Prof. Gabriele Comodi;
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- I performed all the sensitivity analysis case studies simulations, built the parametric performance maps, performed one of the application case study and provided the discussion and interpretation of results;
- Dr. Emiliano Borri assisted on the interpretation of the results;
- Prof. Gabriele Comodi and Alessandro Romagnoli assisted with ideas for the development of the methodology.

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- Dr. Stefano Mazzoni and Mr. Sean Ooi performed all the simulations and the techno-economic assessment, designed the configurations layout, prepared and formatted all figures and provided the discussion and interpretation of results;
- I assisted on the interpretation of the results and providing the numerical results of the parametric performance maps for LAES design;
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- I wrote the drafts of the manuscript. The manuscript was revised by Prof. Alessandro Romagnoli, Prof. Gabriele Comodi, Dr. Emiliano Borri and Prof. Martjin van den Broek;
- I compounded the literature review, performed the technical assessments, designed the configurations layout, prepared and formatted all figures and provided the discussion and interpretation of results;
- Dr. Emiliano Borri assisted on the interpretation of the results;
- Prof. Gabriele Comodi, Prof. Alessandro Romagnoli and Prof. Martjin van den Broek provided guidance on the technical assessment.

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- I wrote the drafts of the manuscript. The manuscript was revised by Prof. Alessandro Romagnoli, Prof. Yulong Ding, Prof. Yongliang Li and Dr. Chunping Xue;
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- I and Dr. Marco Mengarelli wrote the drafts of the manuscript. The manuscript was

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- I performed all the steady state simulations for LAES and designed the configurations layout;
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- I compounded the literature review, performed the techno-economic and environmental assessments, designed the configurations layout and prepared and formatted all figures;
- Dr. Fabio Dal Magro provided guidance on the technical assessment.

22nd December 2019

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Date

Alessio Tafone

.....

Alessio Tafone

Abstract

Large scale or grid scale Electrical Energy Storage systems (EESs) represent one of the most viable solutions to address some of the issues related with the integration of large portion of renewables into the future grid and to facilitate their further penetration guaranteeing the required flexibility and reliability of the electrical grid. Besides mitigating grid instability, large scale EESs also allow decoupling demand and supply, hence offering the opportunity to be operated as peak-shavers during peak demand hours.

Concurrently, another global pressing issue is represented by the constant increase of cooling demand arising by the global warming and rapid development of emerging countries, usually located in the warmer areas of the world. Indeed, using conventional cooling technologies might be not sustainable putting at stake the reliability of existing electrical networks and dramatically increasing the greenhouse gas emissions. As a consequence, new thinking on how to efficiently integrate and recover cold into the wider energy system becomes necessary.

Liquid Air Energy Storage (LAES) is one of the most promising large scale energy storage concept that stores electricity in the form of liquefied air/nitrogen discharging electric power back to the grid by means of liquid air regasification and expansion in power producing devices. LAES has recently attracted significant attention in research and industry due to several advantages among which viable capital costs, high energy density and no geographical/geological constrains. In particular, due to its intrinsic thermo-mechanical nature, it is capable to be integrated with other valuable high exergy energy carriers (e.g. waste heat/cold from industrial process/ Liquefied Natural Gas regasification) and to simultaneously produce both electricity and free cooling energy being configured an ideal technology bridge between enhancement of RES exploitation and the necessity to face the booming of cooling demand. Beside those benefits, the main LAES drawback has been identified in the low value of the round-trip efficiency, estimated around 50-60 % for large scale systems, mainly due to the low exergy efficiencies during the air liquefaction and power recovery processes.

This thesis aims at contributing at the broader field of large scale energy storage by adopting a novel system perspective which puts a special focus on interactions within the system in order to seek the optimal operation conditions and the best route for performance enhancement of LAES system. In particular, the thesis proposes a novel and systematic methodology for LAES system (plant based) design in order to investigate LAES performance and identify potential areas of improvements.

To this end, a steady state model has been developed and used then to simulate the performance of different system architectures. Based on a comprehensive sensitivity analysis carried out on different LAES operative parameters, a methodology for the LAES design is progressively developed and integrated in a well defined procedure. The novel methodology incorporates new parametric performance maps as a unique and user-friendly tool for LAES design under operative parameters variation for different configurations. The optimized LAES system have been environmentally analyzed by means of LCA methodology: among three large scale EESs assessed LAES has proved to deliver the lowest environmental impact.

Once defined the main areas of opportunity based on the outcomes of the previous technical analysis, the thesis aims to develop and assess, from a techno-economic perspective, novel LAES architectures either operating in the conventional full electric configuration or providing both electricity and cooling energy in the novel polygeneration configuration. Indeed, the second part of the thesis proposes different and novel technology solutions to enhance both the thermodynamic and economic performances of LAES by a more efficient utilization of the thermal energy (heat and cold) streams during LAES operation.

Firstly, waste heat recovery concept is proposed and efficiently integrated in LAES. Indeed, the most remarkable results are achieved by LAES in polygeneration configuration where the Organic Rankine Cycle technology allows to improve the LAES round trip efficiency by 20 % decreasing at the same time the Levelized Cost of Storage by 10 %. Finally, to effectively recover the waste cold discharged by liquid air regasification, a Phase Change Material-based (PCM) High Grade Cold Storage (HGCS) is proposed. Two different

configurations (single and cascade PCM) have been modeled and compared with a baseline case configuration where Sensible Heat material is implemented. For this purpose, a numerical model of the HGCS has been developed and successfully validate against experimental data to increase the confidence on the results. The techno-economic analysis has shown that, due to its ability to act as thermal buffer, PCM implementation guarantees a decrease of LAES specific consumption up to 10 % with a remarkable payback period inferior to 5 years.

Lay Summary

The objective of this thesis is to aid in the environmental challenge posed by traditional and extensive *fossil fuel* depletion and the consequent *global warming*, namely the long-term increase of the average temperature of the Earth's environmental system. One way to face this pressing problem is by increasing the *renewable energy sources* (RESs), energy sources that are naturally replenishing but flow-limited utilization. Unfortunately, by its stochastic nature, RESs are unpredictable and intermittent not allowing to ideally match the electricity supply with the user demand imposed by the electricity grid. In order to overcome this challenge, *electrical energy storage* (EES) systems, technologies that stores energy in the form in which it will be reused to generate electricity energy whenever needed, are currently employed.

One large scale *EES* system that is used to store electrical energy from renewables is called *Liquid Air Energy Storage* (LAES) and is analyzed in this thesis. This technology is a novel *EES* system that store energy by means of liquid air or liquid nitrogen. Although *LAES* has many advantageous characteristics, one of the most limiting factors is the fact that the *LAES efficiency*, namely the ratio between the energy produced by the liquid air expansion and the energy required to liquefy the air, is relatively low compared to other large scale *EESs*. Since *EES* systems efficiency is of primary importance in technology market development, different ways to enhance the *LAES* performance are proposing in this thesis.

In particular, this thesis proposes to improve the *LAES* efficiency by means of a better thermal management of the waste heat/cold flows released during air compression and liquid air regasification, respectively. For this purpose, firstly a novel methodology for the *LAES* design is developed and presented in order to define the potential area of performance improvement. Subsequently, two different technologies are introduced to enhance *LAES* efficiency: Organic Rankine Cycle and Phase Change Material to efficiently recover the waste heat and the waste cold, respectively. The results from the implementation of the two technologies to *LAES* prove promising results for the improvement of the efficiency of *LAES* while lowering the specific costs of the system.

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“What happened in the heads of children who grow up with the impression that every problem has a solution, and that everything is at most a question of software? Singapore scared me because to a great extent it already seemed to work that way. [...]” Not only Tiziano Terzani in 1992 but, at first, even I had the same feeling in 2016 approaching the “red hot”. By the time my fear transformed in a positive will to understand another culture and another point of view enhancing my flexibility and contributing decisively to achieve this important goal in my life. For this I would like to thank firstly myself to have always willfully fought and never given up in the toughest moments.

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Nomenclature

Symbols

m	Mass [kg]
$\dot{Q}$	Thermal Power [W]
$\dot{m}$	Mass flow rate [kg/s]
A	Area [m ²]
a_s	Shape factor [m ² / m ³]
Bi	Biot number [-]
d	Particle diameter [m]
E	Energy [J]
Ex	Exergy [J]
$\dot{Ex}$	Exergy rate [W]
h	Heat transfer coefficient, [W/m ² ·K]
H	Height [m]
h	Specific enthalpy [kJ/kg]
P	Electrical Power [W]
U	Overall Heat transfer coefficient [W/m ² ·K]
v	Specific volume [m ³ /kg]
x	Recirculation Fraction [-]
y	Liquid Yield [-]
Δ	Absolute difference [-]
Δp	Pressure drop [Pa]
ε	Porosity [-]
η	Efficiency [-]
c_p	Average heat capacity of fluid, [J/kg·K]
D	Diameter [m]
N	Number of (e.g. tubes) [-]
Nu	Nusselt number [-]
Pr	Prandtl number [-]

Re	Reynolds number [-]
T	Temperature [°C]
V	Volume [m ³]
c	Specific heat capacity [J/kg·K]
k	Thermal conductivity [W/m·K]
p	Pressure [bar]
t	Time [s]
u	Velocity of fluid [m/s]
α	Thermal diffusivity [m ² /s]
β	Specific Power Consumption for ASU products [kWh _e /kg]
μ	Dynamic viscosity [Pa·s]
ρ	Density [kg/m ³]
φ	Energy Density [Wh/m ³]

Subscripts

amb	ambient
c	Cooling
ch	Charge phase
d	Discharge phase
dies	diesel
e	Electrical
exp	expansion
f	fluid phase
fp	Fluid-particle
gf	Gate fee
in	inlet
iso	isentropic
LA	Liquid Air
lat	latent
m	Mechanical
out	Outlet condition

<i>p</i>	particle phase
RF	Recirculation Fraction
s	Storage
sav	Saved
th	Thermal
to	total
u	utilized
w	wall
y	year

Acronyms/Abbreviations

ABS	Absorption Chiller
AFC	Aftercooler
ASU	Air Separation Unit
C	Compressor
CAES	Compressed Air Energy Storage
CAPEX	Capital Cost
COP	Coefficient of Performance [-]
CT	CryoTurbine
CTES	Cold Thermal energy storage
DC	District Cooling
DE	Dearman Engine
DH	District Heating
EES	Electrical Energy Storage
GN ₂	Gaseous Nitrogen
HE	Heat Exchanger
HGCS	High Grade Cold Storage
HGWS	High Grade Warm Storage
HTES	Hot Thermal energy storage
HTF	Heat Transfer Fluid
IC	Intercooler

ICE	Internal Combustion Engine
LAES	Liquid Air Energy Storage
LCA	Life Cycle Assessment
LCES	Liquid Carbon dioxide Energy Storage
LHS	Latent heat storage
LN ₂	Liquid Nitrogen
MAPE	Mean Absolute Percentage Error
NPV	Net Present Value
OEC	Oxygen Enriched Combustion
OPEX	Operational Cost
OPT	Off- Peak Electricity Tariff [\$/kWh _e]
ORC	Organic Rankine Cycle
PBP	PayBack Period
PBP	Pay Back Period
PCM	Phase change material
PHS	Pumped Hydroelectric Storage
PT	Peak Electricity Tariff [\$/kWh _e]
PTES	Pumped-Thermal Energy Storage
PV	Photovoltaic
RES	Renewable Energy Sources
ROI	Return of the Investment
SC	Specific Consumption of air liquefier [kWh _e /kg _{LA}]
SH	Sensible Heat
SMES	Smart Multi Energy System
SP	Specific Production of power recovery unit [kWh _e /kg _{LA}]
STOR	Short Term Operating Service
T	Turbine
TES	Thermal energy storage
TIT	Turbine Inlet Temperature [°C]
TOT	Turbine Outlet Temperature [°C]
TRU	Transport Refrigeration Unit

WH	Waste heat
WHR	Waste heat recovery
WtE	Waste to Energy plant
LCOS	Levelised Cost of Storage
ET	Electricity Tariff

Chapter 1

Introduction

This chapter provides a detailed description of the context in which the work has been thought and developed along with the rationale and the main objectives of the PhD project. The thesis outline structure and the contents of each chapter are also provided.

1.1 Thesis Statement

This work investigates, both from techno-economic and environmental perspectives, an integrated and novel Liquid Air Energy Storage solution providing both electricity and cooling energy for polygeneration purpose.

The main thesis of this work is that the performance of a Liquid Air Energy Storage system can be reliably predicted at the design stage by means of a customized and user-friendly tool in order to define the potential areas of LAES performance improvements. Subsequently, based on this analysis, the adoption of different proposed technology solutions can enhance both the thermodynamic and economical performances of Liquid Air Energy Storage by a more efficient utilization of the thermal energy (heat and cold) streams during LAES operation.

1.2 Background

1.2.1 Motivations for energy storage implementation

Energy in whatever form is an essential source that guarantees in the modern society high quality standards of life. Due to the concomitant effect of population growth and the rapid development of emerging countries, energy demand is dramatically increasing year by year. The year of 2018 has witnessed a remarkable trend for energy demand increasing at almost twice the average rate for 2010s decade with a dramatic fossil fuel share of 81 % [1] (Figure 1-1). Indeed, fossil fuels combustion has long been recognized as the main cause for some serious environmental issues including greenhouse effect, ozone layer depletion and acid rains [2] as well as social costs linked with combustion emissions [3]. Nevertheless, despite the commitment of many countries to reach an early peak in emissions related to fossil fuel consumption, the energy-related carbon dioxide (CO₂) emissions has reached in 2018 the highest annual increase since 2013 (+1.9%) [1]. Nowadays a key role in solving such an environmental challenge posed by traditional fossil fuel depletion is played by Renewable Energy Sources (RESs), namely energy from sources that are naturally replenishing but

limited in the amount of energy that is available per unit of time. As a consequence, the use of renewable energy systems, namely technologies harnessing the energy from RESs, has increased significantly during the early 2000s and according to the Stated Policies Scenario developed by IEA [1], due to the mix of supporting policies and rapid falling costs (-70% for solar PV and -25% for wind), in 2018 the share of renewables in global electricity generation achieved nearly 26%, with a 2040 projection up to 44% (Figure 1-2).

Figure 1-1 Growth in electricity generation and future outlook. Adapted from [1].

The rapid development and penetration of renewable energy sources in electricity grids influence the whole system reliability and stability. Unlike most conventional power plants, renewable power ones are generally smaller in size and not capable of supplying the demand at any time due to renewable energy source intermittency. In fact, highly dependent on weather conditions, most of the renewable energy sources (wind and solar in primis) cannot be dispatched: if not stored, they must be utilized as soon as it is generated.

Figure 1-2 Future outlook electricity generation share by RESs. Adapted from [1].

As a consequence, their integration into the existing grid and in stand-alone mode represents a serious challenge for grid balance in order to meet the energy supply and demand through the chain of generation, transmission, distribution and end use. Amongst all the viable solutions to deal with this issue, Electrical Energy Storage (EES) has been recognized as one of the most promising technology. EES technology refers to the process of converting energy from one form (mainly electrical energy) to a storable form and reserving it in various mediums; then the stored energy can be converted back into electrical energy when needed. Figure 1-3 illustrates some of the challenges that an EES system faces when dealing with the traditional electricity value chain and the relative benefits that is capable to offer to any of the reported links. Indeed, if future electricity systems are planned to use large proportions of intermittent energy source then an increasing scale-up of energy storage is necessary to match the supply with electricity demand profiles. Reflecting this, the International Energy Agency [4] has projected that 310 GW of additional grid-connected electricity storage capacity will be necessary in the United States, Europe, China and India.

Figure 1-3 EES benefits vs challenges imposed by the traditional electricity value chain. Adapted from [5].

1.2.2 The cold economy

Demand for cooling is constantly increasing [6] [7] and unlike electricity, transport and heating, it has received less attention in the global discussion about the different energy mix. The increased demand for cooling arises from concurrent events such as global warming [8] and rapid development of emerging markets/countries [7]— usually located in hot climate areas of the world – where the necessity to provide coolth (for residential, commercial and industrial sectors) is of vital importance. Indeed, as a consequence of the rapid economic growth of developing countries, especially remarkable in South East Asia, by 2060 worldwide the energy required for cooling purpose may overcome the energy consumption for heating [9] (Figure 1-4).

Figure 1-4 Projected global residential energy demand for heating and for air conditioning. Adapted from [9].

Nevertheless, using current technologies to meet the rapid increase of cooling demand, will introduce an additional 139 GW, putting at stake the reliability of existing electrical networks of developing countries and future smart grids and, due to the environmental impact of conventional cooling technologies, a greenhouse gas emissions increase of more than 1.5 billion tons of CO₂ per year [10]. In order to address the issues associated with cooling, the concept of cold economy was firstly introduced in previous works published by the Liquid Air Energy Network [10,11]. A cold economy involves the concepts of:

- **higher efficiency in cold generation and demand side management for cooling** [12]. Demand side management is a technique to enhance the overall efficiency of the whole electricity grid. Basically, it consists of an optimization of resources allocation by shaping the demand and limiting its peaks based on grid requirements [13];
- **recovery of waste cold energy** [14]. An important example is provided by the LNG cold-energy: a large quantity of cold is wasted in the environment during the re-gasification at import terminals; this cold energy is not recovered and it could be used for cooling/refrigeration purposes [15];
- **a new cryogenic energy vector (liquid air or nitrogen)** [16]. To make use of the waste energy discharged both by LNG regasification and by wrong time renewable source

- (e.g. wind energy) supplying both electric power and cold loads, liquid air energy storage has been investigated [17];
- **cold energy storages** [18]. In this context, cold energy storage can play an important role in shaving the peak demand from the electric grid and in developing demand side management strategies to shift the load from peak to off-peak hours, even in presence of renewable energy.

1.2.3 Research interests toward Liquid Air Energy Storage

Among large scale energy storage technologies, Liquid Air Energy Storage (LAES) has attracted significant attention in recent years due to several advantages. In fact, there have been an increasing number of studies on LAES over the past decades particularly after 2012 with a significant concentration during the last triennium, as shown in Figure 1-5.

Figure 1-5 LAES research works numerosity from 2012 to 2019.

LAES is a promising and novel long term cryogenic energy storage technology, suitable for mid to large scale applications. At the same time LAES guarantees volumetric energy density (214 Wh/kg), if compared to other energy storage systems, and no geographical constraints [19]. The system relies on well-established technologies that limits possible

development risks and ensures long life to the system (30–40 years) [20]. Due to its great flexibility under different off-design operations, the integration with other thermal processes, such as waste heat/cold recovery, enables to increase the energy storage efficiency [17]. In addition, considering a startup time within few minutes [21], a power rating above 100 MW_e and a discharge duration of several hours [20], LAES is highly applicable to energy management. The expected investment cost per installed capacity is within a range between 995 and 1774 £/kW for largescale applications [20]. One of the most interesting features of LAES technology is that it can simultaneously produce both electricity and free cooling energy from the electric generator and the liquid air regasification/expansion process, respectively [22]. Metaphorically speaking, LAES is being configured as a technological bridge between both the necessities to enhance RES exploitation and successfully tackle the motivations and the reasons behind the cold economy concept. Indeed, dealing with the compelling necessity to face the booming of cooling demand that may put at stake the reliability of the electricity grid, LAES is playing a crucial role because it has the potential to provide free cooling energy above of all during the energy demand peak period.

1.3 Objectives and Scope

This thesis investigates the technical, economic and environmental potential of the LAES concept either operated in full electric configuration, where electricity represents the main energy output, or in polygeneration configuration with a multi-energy streams output (electricity and cooling energy).

The main objective is then to enhance the LAES system performance by means of the development and the integration of novel thermodynamic cycle architectures and technologies efficiently making use of the thermal energy streams available during Liquid Air Energy Storage operations.

In order to investigate the behavior and performance of the Liquid Air Energy Storage, a steady-state thermodynamic model has been developed along with a novel systematic

methodology to produce the LAES parametric performance maps, a general and user-friendly yet reliable tool to design the Liquid Air Energy Storage system, overcoming numerical complex thermodynamic modelling. The tool can be used to individuate the components and the parameters that affect the most the technical performance of LAES and what are the potential actions to improve LAES performance.

The in-depth analysis resulting from this first stage allows to identify different areas of opportunities for LAES performance improvement. As a consequence, different technological solutions have been proposed and techno-economically investigated. In particular, Waste Heat Recovery solutions (Organic Rankine Cycle and Absorption Chiller) and Phase Change Material based High Grade Cold Storage have been integrated into LAES in order to efficiently recover the waste heat and waste cold streams discharged by the air compression and liquid air regasification processes, respectively.

1.4 Dissertation Overview

The dissertation follows a familiar structure of scientific works in order to present the research questions, the methodologies developed to provide the required answers to the research questions, a comprehensive description and discussion of the results and a closing chapter where the results are confronted to the research questions above illustrated and future potential works are highlighted.

Chapter 1 provides a rationale for the research and outlines the motivation and the main objectives of the thesis along with the related research questions.

Chapter 2 reviews the literature concerning the energy storage field with an emphasis on the Liquid Air Energy Storage system state of art. The literature review aims to highlight the research gaps and the different opportunities for improvement on each area presented.

Chapter 3 provides the methodology used to model each LAES sub-systems presenting the first preliminary technical results regarding the performance of the stand-alone system. In

order to further enhance the application of the results, a techno-economic case study is provided at the end of the chapter.

Chapter 4 investigates the possibility to provide a novel and general methodology to LAES system (plant based) design by means of dedicated performance maps. The intention of these maps allows asserting the optimum design and operating parameters for the LAES making use of a more systematic and immediate methodology.

Based on the technical assessment carried out in the Chapter 3 and Chapter 4, *Chapter 5* investigates the potential of improving the round trip efficiency of Liquid Air Energy Storage by means of different Waste Heat Recovery solutions (Organic Rankine Cycle and/or Absorption Chiller). The analysis is carried out both from technical and economical point of view.

Chapter 6 focuses on the environmental impact of LAES by means of Life Cycle Assessment (LCA) comparing the eco-friendliness of this relatively new technology, with established storage solutions such as Li-Ion Batteries and Compressed Air Energy Storage.

Chapter 7 numerically and experimentally analyzes the thermal behaviors of different novel cryogenic packed beds filled by different Phase Change Materials (PCMs) comparing their performance with that of the conventional sensible thermal energy storage. A preliminary economic analysis has been carried out in order to assess the economic feasibility of the investment in PCM.

Chapter 8 presents, in detail, a Liquid Air Economy case study referred to the possibility to use liquid nitrogen as clean energy vector to power cryogenic engine for transport application.

Chapter 9 presents the main conclusions of the thesis by contrasting the results presented with the research questions. The chapter will provide a summary of the results achieved and their possible impact, as well as the limitations and further improvements that can be

developed in the future.

1.5 Original contribution of this work

The most important research outcomes can be summarized in the following list:

1. A fundamental preliminary analysis of the LAES sub-systems and the relative impact of different factors on its performance.
2. A proposed methodology to design LAES by means of novel parametric performance maps.
3. The possibility to enhance the LAES performance by means of Waste heat Recovery solutions.
4. The possibility to enhance the LAES performance by means of PCM technology in the Waste Cold Recovery process during liquid air regasification.

Chapter 2 ¹

Literature review and research gap

This chapter provides a general and broad literature review regarding the system under investigation, namely Liquid Air Energy Storage. Once the state of the art is defined, a clear research gap can be identified in order to proceed defining the main steps of the PhD project based on the areas of opportunities identified.

¹ This section published partially as Tafone A, Romagnoli A, Borri E, Comodi G. *New parametric performance maps for a novel sizing and selection methodology of a Liquid Air Energy Storage system*. Appl Energy 2019;250:1641–56.

2.1 Energy storage overview

In engineering field, energy storage (ES) concept is based on the idea of storing energy in the form in which it will be reused to generate energy whenever needed [23]. The focus of the thesis will be mainly on Liquid Air Energy Storage (LAES) system, belonging to the category of Electrical Energy Storage (EES): a system that converts electrical energy from a power plant into a form that can be stored in order to be converted back to electrical energy for later use [5].

2.1.1 Definitions

Before to proceed in detail addressing the different EES systems and their applications, clear terminology is required to accurately describe and categorize the range of EES systems.

- 1) **Cycle.** The sequence of the three main phases performed by an EES system: charge (energy loading), storage (energy holding) and discharge (energy unloading) phases.
- 2) **Storage capacity - E [Wh].** The quantity of energy stored in the EES system after the charge phase.
- 3) **Discharge Power - P_d [W].** The peak or the average value of the electric generator in the discharge phase.
- 4) **Depth of Discharge – DoD [%].** The fraction or percentage of the storage capacity which has been discharged from the fully charged EES system.
- 5) **Discharge time – t_d [h].** The maximum discharge power duration defined as the ratio between the storage capacity and the discharge power.
- 6) **Round trip efficiency – η_{RT} [%].** The ratio between the energy released during the discharge phase and the energy stored after the charge phase.
- 7) **Energy Density – ϕ [Wh/m³].** The ratio between the energy stored in the EES system and its volume.
- 8) **Cycling capacity – N [cycles].** The maximum number of cycles an EES system is designed to guarantee before it fails to meet specified criteria.

2.1.2 Electrical Energy Storage Classification

According to the forms of energy electricity is stored in, EES systems can be classified as follows:

- 1) Mechanical Energy Storage
 - a. Kinetic energy storage (Flywheels);
 - b. Potential energy storage (Pumped Hydroelectric Storage, Compressed Air Energy Storage, Liquid Piston, Gravity).
- 2) Electrochemical Energy Storage (Rechargeable batteries and flow batteries)
- 3) Thermochemical Energy Storage (Solar fuels)
- 4) Chemical Energy Storage (Fuel cells)
- 5) Electrical Energy Storage
 - a. Electrostatic energy storage (Capacitors and Supercapacitors);
 - b. Magnetic/current energy storage (Superconducting Magnetic Energy Storage system).
- 6) Thermal Energy Storage
 - a. Sensible Heat Thermal Energy Storage;
 - b. Latent Heat Thermal Energy Storage;
 - c. Sorption Heat Thermal Energy Storage.

EES technologies can also be categorized based on the different functions and applications performed at certain discharge time scales. As shown in Figure 2-1, storage technologies are performing their services at power ratings from kW to GW and over periods from seconds to hours to months or even years.

Figure 2-1 Typical time and size scales associated with different storage technologies. Adapted from [24] and [25].

Table 2-1 Technical characteristics of LAES and large scale mature electrical energy storage systems as intended by the developers/manufacturers.

Tech.	Discharge power rating [MW]	Rated discharge duration [h]	Energy Density [kWh/m ³]	Power Capex [\$/kW]	Energy Capex [\$/kWh]	Geogr. constraints	Lifespan [years]	η_{RT} [%]	Ref.
LAES	1-300	4-24+	120 - 200	900-6000	240-640	No	30-40	45-60	[19,20,26–28]
PTES	10-150	6-20	10-50	1000-6000	100-500	No	30-40	50-65	[29,30]
CAES	1-320	1-24+	3 - 20	970-5000	4-220	Yes	20-40	38-60	[5,31,32]
PHS	100-5000	1-24+	0.5-1.5	600-2000	1000-5000	Yes	40-60	65-87	[5,32]

ESSs operating on a timescale of hours will be especially important for the large-scale integration of fluctuating renewable power sources with limited regulation capability. In the field of large-scale operation with energy storage deliverability above 100 MW_e with single unit, two mature and sustainable EES systems are represented by Pumped Hydroelectric Storage (PHS) and Compressed Air Energy Storage (CAES). Pumped-Thermal Energy Storage (PTES) and Liquid Air Energy Storage (LAES) represents two

novel large-scale EES system that can be classified as thermo-mechanical energy storage involving transformations between mechanical and thermal energy.

2.1.3 Pumped Hydroelectric Storage

PHS is a mechanical large-scale EESs that stores electrical energy by pumping water to an elevated height storing potential energy to be converted again in electricity. The water is firstly pumped, using off-peak and low-cost electricity, and stored in the upper reservoirs at high elevation. When electricity is needed, water is released to the lower reservoir in order to drive a turbo-generator producing thus electricity back to the grid. Taking into account the evaporation and conversion losses, the round-trip efficiency of PHS system is generally around 71-85% [5] [19]. PHS systems were first installed in Italy and Switzerland in the 1890s with the first large-scale commercial plant installed in 1929 in Hartford (USA). Currently, the PHS system is the most implemented large scale EES accounting for about 3 % of the worldwide generation capacity [5]. The main drawbacks of PHS lies in the geographical/geological constrain due to the shortage of available sites for large reservoirs and dams.

Figure 2-2 Aerial view of the PHS plant installed in Thuringia (Germany).

2.1.4 Compressed Air Energy Storage

CAES is another large-scale commercially available technology working on the basis of the conventional gas turbine thermodynamic cycle. That technology uses off-peak and low-cost electricity to compress ambient air being stored in pressurized tanks (40-80 bar) located in underground cavern. To extract the elastic potential energy stored in the compressed air, this energy vector is first drawn from the tanks, heated-up by means of fossil fuels combustion and finally expanded in a turbine train at high pressure and temperature generating thus electricity (Figure 2-3). Two CAES units have currently been operating in the world. Installed in Huntorf (Germany) in 1978, the first CAES plant runs on a daily cycle with 8 h of charging and can generate 321 MW for 2 h [33]. The second CAES plant, installed in McIntosh (Alabama, USA) in 1991, has a generating capacity of 110 MW and up to 26 hours working duration [33]. Similar to PHS, the main obstacle for CAES implementation is due to the geological constrains related to the requirements for the underground cavern.

Figure 2-3 CAES system process flow diagram. Adapted from [34].

A possibility to further increase the round-trip efficiency of then CAES system without utilizing fossil fuel is due to the possibility to recover the waste heat discharged by the air compression phase allowing to achieve round-trip efficiency up to 72 % (A-CAES) [35] [36].

2.1.5 Pumped-Thermal Energy Storage

Pumped-Thermal Energy Storage (PTES) system is a relatively new thermo-mechanical technology to EES that stores electricity in the form of sensible heat in insulated storage vessels containing of an appropriate storage medium, such as a packed bed of gravel or pebbles. The working principle is based on the reverse and forward Joule-Brayton thermodynamic cycle (Figure 2-4) that establishes a temperature difference between two hot/cold reservoirs kept at two different pressures. Powered by excess electricity, a high pressure-ratio heat pump is driven removing heat from the cold to the hot reservoir. During the discharge phase, the flow direction of the working fluid (Argon) is reversed within the system and the difference in temperature between the two (hot/cold) thermal stores is used to drive a Joule-Brayton heat-engine cycle in order to generate work, and thereafter electrical energy. An alternative to the conventional architecture of the PTES system is represented by the CHEST concept [37] employing a vapour compression heat pump and an Organic Rankine Cycle (ORC) for the charge and discharge, respectively.

Figure 2-4 PTES system process flow diagram. Adapted from [38].

2.2 Liquid Air Energy Storage: the concept

Already introduced in Section 1.2.3, LAES is a novel thermo-mechanical EES system that employs liquid air or liquid nitrogen as the main working fluid. Recalling the battery analogy, as depicted in Figure 2-5, LAES system operations can be divided into three phases: charge, store and discharge. During the charge phase electric work, injected into the system, is used to compress and liquefy the air. Then, the liquid air is stored at low pressure in insulated tanks. During the discharge phase the liquid air is drawn from the storage tanks and compressed by means of cryogenic pumps, regasified to ambient temperature (or even higher if waste heat is available) and expanded in power producing turbomachinery (e.g. turbines/piston engines) to generate electric work. In the following sections, a broad and detailed focus on each LAES sub-system will be provided.

Figure 2-5 LAES simplified block diagram.

2.2.1 Charge phase – Air Liquefaction process

The gas liquefaction cycles employed for scientific, commercial and industrial purposes are related with a considerable amount of different theoretical principles and technologies. Nevertheless, in the air liquefaction field, three different liquefaction processes can be identified based on the cycle configuration, systems' components and working fluids: recuperative systems, mixed refrigerant and cascade processes [39]. Table 2-2 summarizes the main families of liquefaction methods that are usually considered in the literature and are found in commercial applications.

Table 2-2 Common liquefaction methods grouped by families used in commercial application and literature.

Family	Main cycles	Ref
Recuperative systems	Linde-Hampson, Claude, Kapitza, Collins, Heylandt	[40]
Mixed refrigerant systems	Gas refrigerant supply, Liquid refrigerant supply	[39]
Cascade systems	Double, triple cascade cycle	[41]

a) Recuperative systems

An ideal recuperative system cycle consists of an isothermal compressor, a heat exchanger, a Joule-Thompson (J-T) valve and a phase separator. Depending on the complexity of the process architecture, three main recuperative cycles can be identified:

Linde-Hampson cycle. The simplest and first industrialized liquefaction process is the Linde-Hampson cycle patented in 1895 by William Hampson and Carl von Linde (Figure 2-6). The liquefaction of the air is based on the isenthalpic expansion through a Joule-Thomson valve bringing the working fluid in the two-phase zone. The not-condensed fraction of air is then recirculated through a heat exchanger (Cold Box) where is utilized to cool down the pressurized air flow while the liquid air is stored in

cryogenic vessels (LA Tank). Despite the simplicity of the cycle not comprising any rotating machinery, the exergy efficiency of the Linde-Hampson cycle is usually under 10 % due to the high exergy losses during the isenthalpic expansion process and the heat exchange process in the cold box.

Figure 2-6 Process flow diagram of Linde-Hampson cycle.

Claude cycle. In order to improve the performance of the Linde-Hampson cycle, the Claude cycle has been proposed and patented in 1902 by George Claude. As shown in Figure 2-7, a cryogenic expander is added to the liquefaction cycle combining the isenthalpic and the isentropic expansion. In fact, a fraction of the pressurized air flow is diverted from the mainstream to a cryogenic turbine or CryoTurbine (CT) and isentropically expanded. The benefit is therefore twofold: the expansion process allows to attain a lower temperature of the working fluid simultaneously producing a valuable mechanical work through the cryogenic turbine. The remaining fraction of the pressurized air flow undergoes the cooling process through the second and the third heat exchanger and finally it is expanded in the J-T valve.

Figure 2-7 Process flow diagram of Claude cycle.

Kapitza cycle. Another variant of the Linde cycle is represented by the Kapitza cycle patented by Peter Kapitza in 1939. The cycle, presented in Figure 2-8, implies the

elimination of the third low-temperature heat exchanger in the Claude system and the use of a rotary expander instead of a reciprocating expander.

Figure 2-8 Process flow diagram of Kapitza cycle.

b) Cascade cycle

The cascade refrigeration cycle employs a combination of refrigerants in order to achieve an optimized temperature profile in the cold box (Figure 2-9). Although this cycle performs better compared to the recuperative cycles, the complexity of the liquefaction process has been heavily affected: multiple sub-cycles of the cascade cooling are used to create different low-temperature levels with suitable refrigerants, providing a better matched temperature profile. The working fluids commonly employed are propane, ethane, methane and nitrogen.

Figure 2-9 Process flow diagram of a multistage cascade cycle for natural gas liquefaction.

c) Mixed Refrigerant cycle

Nowadays, over 95% of the base-load LNG plants operate on mixed refrigerant cycles, with the remaining few operating on conventional cascade processes. The mixed refrigerant cycle has a similar working principle of the cascade system, but in this case, different gases are mixed together and cooled down by a single vapor refrigeration cycle. A simple single-stage mixed refrigerant LNG liquefaction process is proposed in Figure 2-10.

Figure 2-10 Process flow diagram of a mixed refrigerant cycle for natural gas liquefaction.

2.2.2 Discharge phase – Power Recovery Process

In order to extract cryogenic energy from liquid air, nowadays different systems have been analyzed in literature and labelled based on their reference cycle. Li et al. [42] have evaluated the potential of different combination of discharge cycles using LN_2 as main working fluid. They conclude that depending on the available waste heat temperature source, the best configurations to recover cryogenic energy are the direct expansion - Brayton hybrid system and the direct expansion - Rankine hybrid system for high and low grade heat sources, respectively. Similar to this work, two cold exergy recovery cycles have been analyzed by Hamdy et al. [43] for liquid air energy extraction: direct expansion and expansion of liquid air in combination with an ORC. They concluded that the addition of

ORC helps to increase the specific power output by 24%. Although different methods for extracting energy from a cryogenic energy source are available in literature [44–46], four different processes for power generation can be identified:

a) Direct expansion cycle

The working fluid of the cycle is represented by the cryogenic fluid that, before being directly expanded in a turbine train coupled with an electric generator, is pressurized, vaporized and superheated by either ambient heat or available waste heat. The expansion stage could be also realized by a novel technology, the Dearman Engine [47], patented by Peter Dearman in 2012. The system is based on a novel piston engine powered by the vaporization and expansion of liquid air or nitrogen. A more detailed description of the concept and its application will be provided in Chapter 8.

b) Indirect Rankine cycle

The cryogenic fluid acts as a heat sink for the main power cycle. In fact, both the heat and cold sources are supplied externally to a closed cycle, which usually employs an organic working fluid that is experiencing a liquid-vapor phase change during the cycle operation. To recover both the latent cold and sensible cold released by the cryogenic source, a working fluid with a liquefaction/boiling point slightly higher than the cryogenic source would be an ideal working fluid. In order to increase the efficiency of the process, the use of cascading cycles have been proposed in literature. Different working fluids with lower and higher liquefaction/boiling point are employed in order to minimize the exergy losses during the heat transfer process between the working fluids and the cryogenic source.

c) Indirect Brayton cycle

In this process, the cryogenic source is used to cool down the inlet gaseous working fluid of the compressor train in a Brayton cycle. The working fluid is in the gaseous state throughout the cycle operation and the heat or cold transferred to the working fluid is in the form of sensible heat.

d) Mixed cycles

A combination of the above-mentioned methods (Indirect Rankine/Brayton cycles and the direct expansion process) can be employed as a more efficient approach to recover the cryogenic source.

2.2.3 Thermal Energy Storage: a thermal link between charge and discharge

A potential way to increase the round trip efficiency of LAES systems is offered by the implementation of warm and/or cold thermal energy storage technology, namely a High Grade Warm Storage (HGWS) and/or a High Grade Cold Storage (HGCS), respectively. As shown in Figure 2-11, the main purpose of both configurations is achieved by means of thermally coupling through waste heat and/or waste cold two phases (charge and discharge) operating asynchronously at two different time periods (e.g. nighttime/daytime).

Figure 2-11 Simplified block diagram of LAES process and sub-processes.

If the liquefaction process operates during wrong time renewable energy or off-peak grid tariff time, the discharge phase takes place principally to cover the peak of energy demand during day-time. Different configurations of both Thermal Energy Storage (TES) systems have been analyzed in literature.

2.2.3.1 High Grade Warm Storage

Aiming at recovering waste heat flow discharged by the compression phase, the HGWS is used to reheat the air during the discharge phase. The heat from the compression process is recovered in a similar way as in the A-CAES concept where the thermal energy generated by the compression is stored in a TES and then used to reheat the air before it is expanded again [35]. The effect of waste heat and turbine inlet temperature on the discharge phase has been tested by Morgan et al. [27] who claimed a high conversion rate of low grade waste heat source compared to other well established technologies. As already assessed by Guizzi et al. [46], although the HGWS aims to recover waste heat from charge phase, it presents significant exergy inefficiencies due to the low thermal capacity of liquid air compared to thermal oil leading to consider the implementation of other waste heat recovery solutions by means of ORC and/or absorption chillers. Until now two main technological concepts have been adopted for the HGWS: sensible and latent heat thermal energy storage.

Sensible heat thermal energy storage. The most common method to recover the waste heat at high temperature is through the use of a heat transfer fluid (thermal oil or pressurized water) that can be used directly as storage through a double-tank storage configuration. The concept, borrowed from waste heat recovery [48] and solar thermal engineering [49], comprises a TES medium that circulates between a hot and cold tank and exchanges heat to the main working fluid of the system through heat exchangers. Based on direct contact heat transfer process, packed bed is a simple technology that has been proposed in literature as a TES for both A-CAES and LAES application [36]: the container is filled by solid particles of the required TES medium and the heat transfer fluid flows directly through the packed bed both in charge and discharge phase. This technology has been implemented in the work carried out by Grazzini et al. [50] that modeled a 4.6 MWh A-CAES system using a thermal oil as heat transfer fluid for the TES. The system was able to achieve a significant round trip efficiency of 72 % without involving any combustion process. A detailed analysis of A-CAES with packed beds has been proposed by Barbour et al. [35] by developing a numerical model of the plant validated against analytical solutions.

According to the authors, compared to indirect-contact heat exchanger, packed bed guarantees higher round trip efficiency in excess of 70 %. The exergy analysis has shown that the main exergy losses occur in the compressors and expanders (accounting for nearly 20% of the work input) rather than in the packed beds. Sciacovelli et al. [51] investigated the dynamic performances of a A-CAES plant with packed bed TES. The authors claimed that a round trip efficiency in the range 60-70 % is achievable only when the packed bed system operates with a storage efficiency above 90%. Peng et al. [52] analyzed the performance of a LAES system based on a Linde cycle proposing for the HGWS system the packed bed technology filled by steatite rocks. Depending on the operative parameters, such as the charge and discharge pressure, on the round trip efficiency, the proposed LAES configuration shows a round trip efficiency of 50-62% highlighting that the highest exergy loss occurs in the cold box. Indeed, during the LAES charge phase, the temperature profile of the cold box shows that the cold energy contained in the not-condensed vapor returning from the phase separator is not fully utilized, suggesting a potential for further improvements. The concept of thermal energy storage packed bed will be further qualitatively and quantitatively addressed in Chapter 7.

Latent heat thermal energy storage. The use of phase change material in place of sensible heat material for both A-CAES and LAES has been mainly justified by the higher capacity and energy density that the latent heat TES may achieve. Peng et al. [53] studied the charging behaviour of a packed bed TES with PCM particles as the main filler for A-CAES application. The numerical model, validated against experimental results proposed in literature, has been used to evaluate the charge efficiency of the packed bed TES using a sensible heat material, a single PCM material or a PCM cascade of materials. Although highly dependent on the particle diameter, the latter configurations have shown better charge efficiency and shorten the charge time. The same concept of cascade PCMs was studied by Tessier et al. [54] who found that the implementation of the PCMs in A-CAES lead to a 15 % increase of the round trip efficiency (85 %) over current designs with sensible heat material for packed bed. Moreover, according to the authors, the melting temperature and enthalpy of PCMs could be used to further optimize the A-CAES system and improve the efficiency. The double-tank storage configuration incorporating latent heat

storage material has been studied by Bernagozzi et al. [55] who investigated different molten salts that can be applied to store the waste heat energy of a LAES. In particular, in the methodology proposed, starting from 5 baseline molten salts, 70 salt mixtures were investigated through a parametric analysis. The first screening analysis based on performance and system index (PSI) and the pseudo performance index (PPI) identified 16 potential salts where only two based on CaLiNaK salt mixtures were selected as referred candidates.

2.2.3.2 High Grade Cold Storage

According to different literature references, the HGCS in LAES round-trip efficiency has been described as a crucial component in order to achieve reasonable round trip efficiency. According to Peng et al. [56], the exergetic value contained in cold thermal energy recovered in the HGCS is considered more valuable rather than the waste heat discharged from the compression. In fact, the cold energy loss in the HGCS leads to a decrease of round trip efficiency around 7 times of the one triggered by waste heat energy loss in the HGWS system. Indeed, a 5% energy loss of high temperature thermal energy causes a negligible effect on the LAES round trip efficiency (from 59.4% to 58.1%) while the same loss on cold thermal energy can cause a 50% drop on the round trip efficiency. This outcome confirms the results achieved in a precedent work carried out by Li et al. [42]: the stored cold has been shown to be more exergetically valuable than the stored heat particularly at large temperature differences. In addition, Peng et al. [56] has shown that the analysis of the cold box temperature profile has shown that only a fraction of the cold energy of the air non-condensed flow from phase separator can be effectively utilized suggesting potential for further performance improvements. Another strong evidence of the benefit generated by the implementation of the HGCS is shown in the data released by Highview Power [57] regarding the first LAES pilot plant in the world [58]. The data shows that integrating a HGCS in the LAES, the specific consumption can be reduced by 25% (from 0.6-0.75 kWh_e/kg_{LA} to 0.45-0.55 kWh_e/kg_{LA}) with a liquid air production of 30 tons per day. In case of a commercial LAES scale, the data shows that a 50 % decrease can be ideally achieved (from 0.4 kWh_e/kg_{LA} to 0.2 kWh_e/kg_{LA}). Araki et al. [59] studied and

realized a HGCS system that consists of an insulated vessel based on an array of small diameter stainless-steel (or copper) pipes with concrete filled around as storage medium. In the work a numerical model based on one-dimensional model was developed to validate the experimental results obtained from the experimental test-rig. Morgan et al. [27] carried out a study on the pilot plant scale LAES showing that the low round trip efficiency was principally due to the fact that only 51 % of the available waste cold released by air regasification was recycled in the charge phase. In fact the design of the LAES pilot plant HGCS and the choice of the materials has been done to simplify the system and limit the investment costs. The LAES pilot plant is equipped with a HGCS packed bed technology made up of eight columns and filled with quartzite rocks that operates near ambient pressure. The modular design of the packed bed [60] allows to store the cold thermal energy in different module arranged in series or in parallel that, during the operation, can be isolated depending on the mass flow rate and the load of the system. Numerical simulations of the HGCS done from the same authors [20], shows that the efficiency of a modular packed bed, can be improved of 4.8% compared to a single module configuration. Different technological solutions for the HGCS have been proposed in literature, among which the packed bed and the two tanks are the most prevalent configurations. Li et al. [45,61] proposed a two-tanks HGCS configuration in which methane and propane are used both as cold storage medium and working fluid for the heat transfer process. Those two fluids were selected to maintain a high heat capacity within the temperature range selected. An experimental study of a packed bed HGCS for LAES applications has been reported by Chai et al. [62]. In particular, the work aims to investigate the behavior of the HGCS at different pressures (1 and 65 bar). The HGCS is a single packed bed unit with a height of 1500 mm and an inner diameter of 345 mm filled with 9mm granite pebbles as medium with an average porosity of 0.4. During the storage charge phase, liquid nitrogen is pumped by means of a cryogenic pump and enters at the bottom of the tank. The cryogenic energy is absorbed by the storage medium leading the liquid nitrogen to boil. During the discharge of the tank dried air is compressed and after being heated enters in the top of the tank. The axial temperature profiles were measured by seven thermocouples installed along the center of the column and the radial profiles were measured by means of five sensors installed in three axial position. The results show a change in the temperature profiles of

the HGCS increasing the pressure flow to critical pressure (65 bar). Indeed, in the critical point small variations of temperature and pressure results in a large change of the thermal properties. On the axial direction the thermocline thickness is greater at low pressures (under 1 bar) but decreases slightly under 65 bar. This is due to the pressure drop that affects more on low-pressure values leading more liquid nitrogen to boil along the axial direction. This effect can be reduced by increasing the packed bed radius obtaining thinner thermocline region under 1 bar. In the radial direction, the low-pressure flow is more subjected to flow instability and recirculation flow due to the higher change in density when the liquid nitrogen boils, while the piston effect and the higher density of the supercritical nitrogen reduce the temperature differences between the different radial positions. By means of a quasi-steady state modelling approach, Sciacovelli et al. [63] analyzed a stand-alone LAES plant with packed bed technology to store the cold energy released during the regasification of liquid air. The developed numerical model of the HGCS has been validated by means of the experimental results obtained from the LAES pilot plant and has been applied to a large-scale LAES system. The results show that cold recovery has a strong impact on the system performance: with a 16 % increase of the specific cold recycle, the liquid yield of the liquefaction plant can be increased by 30%. Investigating the dynamic behavior of the HGCS, the authors found out that a thermocline effect inside the packed bed has been generated increasing the specific consumption of the LAES. Indeed, the results shows that, during the LAES charge phase, the system operates under nominal conditions for the 80% of the time. For the remaining period the temperature of the intermediate heat transfer fluid (HTF) at the outlet of the packed bed increases significantly affecting the liquefaction performance. Huttermann et al. [64] analyzed the impact of different storage materials over the efficiency of a packed bed HGCS implemented in LAES to recover cold energy from liquid air regasification. In particular the authors analysed 4 metals, 1 ceramic, 2 minerals and 2 plastics. The HGCS model was based on one dimensional two-phase energy conservation equation and the performances of the different materials were evaluated considering the storage efficiency at the same boundary condition. In particular, equal values of the exergetic storage efficiency and the time-average pressure drop has been selected for the comparison. The material characterization has been simplified considering for each material the averaged volumetric

heat capacity and the heat capacity ratio that are used to generate an empirical model to estimate the storage efficiency. The analysis has shown an increase of the packed bed efficiency at decreasing volumetric heat capacities is occurring and polypropylene and high-density polyethylene are well suitable as storage materials. It worth noting that all the previous literature works have considered sensible heat material (quartzite or steatite rocks) and packed bed technology as the main filler and the main geometry, respectively, for the HGCS.

2.3 Liquid Air Energy Storage history and state of art

LAES concept has been firstly proposed by Smith [65] in 1977 who introduced a thermodynamic cycle for air liquefaction, based on adiabatic compression and expansion turbomachinery, claiming a round trip efficiency of 72 %. Another LAES concept was proposed by Mitsubishi [66] in 1997 and focuses on LAES discharge section, in particular on the design of the cryogenic pump and the power turbine. Since then, many studies have been developed on LAES focusing their attention on thermodynamic and economic analysis. Chino et al. [67] studied a method to increase the LAES efficiency proposing a system that uses the liquid air produced with off-peak power at night time to feed a combustor of a gas turbine during day time. The high round trip efficiencies achieved (73-87%) is attributed to the cold storage unit utilizing the cooling power discharged by the liquid air regasification to assist the liquefaction process.

Figure 2-12 LAES development timeline.

Ameel et al. [44] carried out a thermodynamic analysis of a LAES based on a Linde cycle coupled with a Rankine cycle, estimating a round trip efficiency as high as $\approx 43.3\%$ with no integration of warm and cold storage. Xue et al. [68] and Guizzi et al. [46] carried out a thermodynamic analysis of a LAES based on the Linde cycle integrated with a warm and cold thermal storage achieving round trip efficiencies up to $\approx 47\%$ and $\approx 50\%$, respectively. Similar work has been carried out by Dutta et al. [69] claiming a round-trip efficiency up to 47% for a Liquid-Nitrogen Energy Storage (LNES) system making use of a waste heat source at temperature higher than 500 K . Starting from the patent of Chen et al. [70] based on a Linde-Hampson Liquefaction cycle, Abdo et al. [71] evaluates other alternatives on the liquefaction based on Claude and Collins cycle. The cycles were compared in terms of an adimensional parameter based on ratio between the mass flow rate of the discharge section and the charge/liquefaction section. The results show that Collins and Claude represent the best solution to as air liquefier configuration, but in terms of cost-benefit Claude liquefaction cycle the best option due to the reduced number of components. A comparative thermodynamic analysis between LAES and CAES has been carried out by Krawczyk et al [72]. LAES has shown better performance compared to CAES with a higher round trip efficiency (55% versus 40%) and significant lower storage tank volume (5000 m^3 vs 310000 m^3). Legrand et al. [73] proposed techno-economic study of a 100 MW LAES plant integrated into the Spanish power grid. A dynamic HGCS packed bed model has been implemented and validated against experimental results. Considering different scenarios of renewables grid penetration (PV and wind power), the results suggest that the best economic scenario with a Levelised Cost of Storage as low as 50 €/MWh is realized when photovoltaic energy is stored in the day-time peak hours and released during the night-time valleys to maximize the use of storage plants.

2.3.1 LAES operating plants

A real application of LAES has been demonstrated by Highview Power [74] which developed the first pilot plant ($350\text{ kW}/2.5\text{ MWh}$) [27] (Figure 2-13) and the first grid scale Pre-Commercial Demonstrator plant ($5\text{ MW}/15\text{ MWh}$) [75] (Figure 2-14) based on the patent developed in collaboration with Chen et al. [70] in 2007. The LAES systems are

based on a Claude cycle, integrating a low pressure cold thermal energy storage enabling to achieve a round trip efficiency between 50 - 60%. In both cases, the waste heat recovery systems rely on external heat sources, namely a waste heat stream (up to 60°C) released by a biomass power plant operating in Greater London and the engine exhaust gases from a landfill gas generation plant installed in Greater Manchester, respectively.

(a)

(b)

Figure 2-13 External (a) and internal (b) views of the 300 kW_e/2.5MWh LAES pilot plant[76].

Figure 2-14 External view of the LAES grid scale demonstrator plant in Greater Manchester.

A schematic of the pilot plant is shown in Figure 2-15. The design strategies of the pilot plant were selected to both fit the small dimension of the plant and the budget accounted for the project. The first prototype was only made of a liquid nitrogen tank and a power turbine, able to process the 47% of the low-grade waste heat from the biomass plant into electrical power. The liquefaction plant, with a liquid production rate of around 1.4 ton/h, was later commissioned and supplied by Chengdu Air Separation Corporation, realizing the first LAES prototype in the world. The plant operates at the peak pressures of 12 bar

and 60 bar for charge and discharge phases, respectively. The liquefaction cycle, is based on single turbine Claude design, is assisted by a cold recycle that recovers the cold thermal energy wasted during the discharging phase of the plant. The air, used as heat transfer fluid for the cold recovery, is driven by two screw compressors able to control the mass flow rate. The cold energy released by liquid air regasification has been stored in a series of eight packed gravel beds filled with quartzite rocks inside a container insulated with perlite.

Figure 2-15 Process flow diagram of LAES pilot plant. Adapted from [27].

The cold recovery can be optimized through the use of valves that can connect the columns in series or in parallel. The concept of cold thermal storage (High Grade Cold Storage) was mainly designed to simplify the system and reduce the total cost of the plant. When the LAES is discharged, the liquid air stored is pumped by means of two reciprocating cryogenic pumps and then heated in two evaporators from the exhaust gases coming from the expander. The four-stage expansion process is based on a series of radial inlet turbines, and the air was superheated during the steps by a water-glycol heating with variable temperature circuit to simulate the use of external heat sources.

Highview Power has recently announced plans to construct the UK's first (50 MW/250

MWh) and the US's first commercial (50 MW_e/250 MWh) LAES-CRYOBatteriesTM, which will be located at a decommissioned thermal power station in northern England [77] and in northern Vermont (USA) [78], respectively.

2.3.2 LAES configurations

To offer a systematic and global approach to understand the LAES state of art, the literature works have been categorized based on the different LAES configurations proposed. In particular, the following LAES configurations can be identified (Table 2-3).

2.3.2.1 Stand-alone LAES system

A stand-alone LAES configuration refers to the system configuration not integrated with any external thermodynamic cycle and/or heat source/cold sink (see Figure 3-18). The unique and only energy input to the system is represented by the electricity required to produce liquid air. A stand-alone configuration includes:

- the charge phase based on Linde-Hampson cycle or modified Claude cycle;
- the discharge phase based on direct expansion process;
- the storage section with a pressurized liquid air tank;
- the HGCS and HGWS systems used to couple the charge and discharge phases by recovering the waste heat and waste cold flows.

The system proposed by Guizzi et al. [46], instead to rely on an external source of heat, recycle the waste heat of compression as usually adopted in a CAES system. The waste heat recovered by means of thermal oil (Essotherm 650), is stored in a hot storage section and then released in the superheaters placed before the expansion turbine of the LAES discharge section. The cold recovery section is based on the two-tank configuration proposed by Li et al. [45] that uses propane and methane as both cold storage medium and heat transfer fluid. From the reference configuration based on optimal operative parameters that maximize the roundtrip efficiency, the influence of the different design parameters has

been investigated. The results show, for the stand-alone system, a roundtrip efficiency in the range of 54-55%. A similar LAES system of Guizzi has been investigated by Xue et al. [68]. In this work, the configuration proposed differs in the number of expanders in the discharge section of the LAES and the number of compressors of the liquefier. The works mainly focus on investigating the effect of the charge and discharge pressure on the system efficiency. A thermodynamic study conducted by Ameel et al. [44] on a LAES based on Linde-Hampson liquefaction plant combined with a Rankine cycle, shows that adding an external heat source of 300 K on a stand-alone LAES, the efficiency can be improved of around 18% (from 36,8% to 43,3%). In the system studied by the authors the waste heat and cold recycle are not integrated into the configuration proposed, furthermore additional liquid air is supplied from an external source. Sciacovelli et al. [63] reports a numerical analysis of a 100MW_e/300 MWh stand-alone LAES system. The study, through a dynamic modelling of the HGCS, analyses the impact of the cold recovery on the LAES performance. The results show that the thermocline effect and the dynamic behavior of the HGCS strongly affects the LAES performance that can be decreased of 25% compared to the nominal value. A study on a stand-alone LAES focused on the HGWS and the HGCS was also conducted by Peng et al. [52]. The system, based on a Linde-Hampson liquefaction cycle, includes an HGCS with a two tanks configuration to store the cold energy at two different temperature levels. Like the solution proposed by Li et al. [33], this work uses propane and methanol as a working fluid for the HGCS and the hot thermal energy is stored in a packed bed HGWS using rocks as storage medium. The LAES studied shows in a roundtrip efficiency of 50-62% depending on the operating condition with the biggest exergy loss occurs in the cold box. Indeed, during the charge of the LAES, the temperature profile of the cold box, shows that is not possible to exploit all the cold energy contained in the cold vapor returning from the phase separator, that suggest a potential for further improvements.

2.3.2.2 Polygeneration LAES system

In order to extract most of energy stored in the form of liquid air, different authors proposed LAES as a polygeneration system that provides cooling/ heating and electric power (see

Figure 3-21). Comodi et al. [79] carried out a qualitative-quantitative analysis with different energy storages for cooling applications, including the LAES, at different scales and scenarios. et Ahmad al. [80] analyzed the potential use of liquid nitrogen produced from surplus electricity at off peak times to provide cooling and power for domestic houses. The results showed that at current liquid nitrogen price, the proposed polygenerative system is economically advantageous compared to a conventional air conditioning system. Al-Zareer et al. [81] proposed a trigenerative LAES configuration where the district heating and the adsorption cooling system harness the thermal energy recovered from the air compressors intercoolers. In general, the proposed integrated system has higher energy and exergy efficiencies than the standalone system.

2.3.2.3 Hybrid LAES system

LAES hybridization has been proposed by many authors in different configurations and concepts. Li et. al [45] proposed an integrated solution between LAES and nuclear power plant in order to perform a load-shift of the power plant. The liquid air is produced during off-peak hours and used to generate electricity during peak-hours. In that case, the heat from the nuclear plant is used to superheat the liquid air during the discharge phase of the LAES; a round trip efficiency up to 71% could be achieved. A hybrid energy storage consisting of a compressed air store at ambient temperature, and a liquid air store at ambient pressure has been proposed and thermodynamically analyzed by Kantharaj et al. [82]. The system, adopting a heat pump and a heat engine for the conversion of liquid air to compressed air and vice versa, achieves a round trip efficiency of 53%. Antonelli et al. [83] investigated the potential of different hybrid configurations based on LAES, ORC and Brayton cycle with or without the contribution of additional combusted fossil fuels. The cold Brayton cycle resulted to be the best configuration achieving round trip efficiencies higher than 80 %. A thermodynamic analysis of a hybrid system including energy storage and production based on a liquid air energy storage plant where only oxygen is liquefied using low cost energy during the hours of exceeding generation, while liquefied natural gas is used as fuel has been carried out by Barsali et al. [84]. By means of a dedicated optimization, the hybrid system is capable to reach round trip efficiencies higher than 90 %.

The technical potential of a hybrid system combining LAES and PTES has been studied by Farres-Antunez et al. [85]. The analysis has shown that the hybrid system seems to be an effective option with a round trip efficiency increase of about 10 % compared to individual cycles. The technical potential of an integrated system made of LAES and geothermal power plant has been studied by Cetin et al. [86] (Figure 2-16). The analysis has shown that LAES seems to be an effective option for load shifting of geothermal power plants with a LAES round trip efficiency and an overall integrated system efficiency of 46.7 % and 24.4 %, respectively. A further hybrid LAES system has been proposed by the Zhang et al. [87] integrating a Kalina cycle in the LAES system that uses a mixture of ammonia-water as working fluid. In this case part of the waste heat from the LAES compressor section, is used to evaporate the working fluid of the external cycle. Compared with a baseline case, the solution proposed is able to increase the roundtrip efficiency for the full-electric mode from 52.1% to 57.2%. The same authors [88] have proposed a hybrid LAES systems based on the cascaded storage and effective utilization of compression heat by means of ORC and Kalina cycle implementation. The new proposed solution has been capable to increase the round trip efficiency by 11-20 % compared to the stand-alone system.

Figure 2-16 LAES integrated with geothermal power plant. Adapted from Ref. [86].

2.3.2.4 Integrated LAES system

One of the main advantages of LAES system is represented by its thermo-mechanical nature that makes the system capable to be integrated with other external thermal processes making efficiently use of available heat sources/heat sinks (see Figure 5-5). Zhang et al. [89] proposed a novel integrated LAES system combined with ORC systems based on the utilization of liquefied natural gas (LNG) cold energy. In the charging process, the LNG helps to conditioning the inlet compressed air, reducing its temperature; concurrently the cold energy of the liquid air regasification and the waste heat discharged by the air compression phase are utilized in a two-stage ORC system to generate additional electricity during the discharging process. Compared to standalone LAES systems, the cold energy storage system is extremely simplified in the proposed system, and higher electrical storage efficiency and density are obtained. Lee et al. [90] developed a novel LAES system integrating the LNG regasification process in the LAES charge phase. With an exergy efficiency of 94.2 %, the proposed system has the unique advantage to store and release energy simultaneously by means of air liquefaction and direct expansion of LNG, respectively. The same authors [91] proposed a techno-economic analysis of a novel integrated LAES system by applying ORC technology during the LNG regasification process. The proposed LNG-ORC-LAES system was found to be both technically and economically feasible achieving the highest specific daily net power output (84.34 kJ/kg_{LNG}) among various hybrid LAES-LNG regasification systems developed by other authors.

2.3.2.5 Liquid Carbon Dioxide Energy Storage (LCES) system

A different cryogenic fluid (carbon dioxide) has been proposed as the main working fluid by some literature works. Zhang et al. [92] carried out a parametric study on a novel LCES system (Figure 2-17) proposing carbon dioxide as new working fluid achieving a maximum round trip efficiency (65.41 %) with optimal charge and discharge maximum pressure of 110 bar and 40 bar, respectively. Xu et al. [93] developed a novel LCES system with two artificial storage tanks based on Rankine cycle. A comparative study is carried out between

the LCES and the LAES to evaluate their performance. The results show that LCES has a higher round trip efficiency (45.35 % vs 37.38 %) compared to LAES, but with a significant lower energy density (18.06 kWh/m³ vs 101.6 kWh/m³).

Figure 2-17 Liquid Carbon Dioxide energy storage system schematics. Adapted from Ref. [92].

Table 2-3 Literature works on LAES categorized by configurations.

Ref.	Configuration	Liquefaction Cycle	Energy Recovery cycle	HGCS/HGWS	KPI _{max}
Ameel [44]	Stand-alone	Linde-Hampson	Direct expansion	-	$\eta_{RT} = 43 \%$
Guizzi [46]	Stand-alone	Claude	Direct expansion	Y/Y	$\eta_{RT} = 55 \%$
Xue [68]	Stand-alone	Linde-Hampson	Direct expansion	Y/Y	$\eta_{RT} = 49 \%$
Sciacovelli [63]	Stand-alone	Claude	Direct expansion	Y/Y	$\eta_{RT} = 50 \%$
Hao Peng [52]	Stand-alone	Claude	Direct expansion	Y/Y	$\eta_{RT} = 62 \%$
Comodi [79]	Polygeneration	-	-	-	$\eta_{RT} = 60 \%$
Ahmad [80]	Polygeneration	-	Direct expansion, Brayton, Rankine	-	$\eta_{th} = 74 \%$; COP = 3
Al-Zareer [81]	Polygeneration	Claude	Rankine with fuel combustion	-	$\eta_{th} = 72 \%$; $\eta_{ex} = 72 \%$
Li [45]	Hybrid	Linde-Hampson	Direct expansion with waste heat from Nuclear Power Plant	Y/Y	$\eta_{RT} = 71 \%$
Kantharaj [82]	Hybrid	Linde-Hampson	Direct expansion	-	$\eta_{RT} = 53 \%$
Antonelli [83]	Hybrid	-	Direct expansion with fuel combustion w/o ORC/Brayton	-	$\eta_{RT} > 80 \%$
Barsali [84]	Hybrid	Air Separation Unit for oxygen liquefaction	Direct expansion with fuel combustion	-	$\eta_{RT} > 90 \%$
Farres-Antunez [85]	Hybrid	Linde-Hampson	Direct expansion coupled with PTES	N/Y	$\eta_{RT} = 71 \%$

Cetin [86]	Hybrid	Linde-Hampson	Direct expansion with waste heat from geothermal power plant	Y/N	$\eta_{RT} = 47 \%$
Zhang [87]	Hybrid	Linde-Hampson	Direct expansion coupled with Kalina cycle	Y/Y	$\eta_{RT} = 57 \%$
Zhang [88]	Hybrid	Linde-Hampson	Direct expansion coupled with Kalina cycle and ORC	Y/Y	$\eta_{RT} = 57 \%$
Zhang [89]	Integrated	Linde-Hampson assisted by LNG regasification	Direct expansion coupled with ORC	Y/Y	$\eta_{RT} = 71 \%$
Kim [17]	Integrated	Linde-Hampson assisted by LNG regasification	Direct expansion with fuel combustion	Y/Y	$\eta_{RT} = 73 \%$
Lee [90]	Integrated	Linde-Hampson assisted by LNG regasification	Direct expansion with LNG expansion process	-	$\eta_{ex} = 54 \%$
Lee [91]	Integrated	Linde-Hampson assisted by LNG regasification	Direct expansion coupled with ORC	-	$\eta_{ex} = 70.3 \%$
Zhang [92]	LCES	Linde-Hampson	Direct expansion	-	$\eta_{RT} = 64 \%$
Xu [93]	LCES	Linde-Hampson	Direct expansion	Y/N	$\eta_{RT} = 45 \%$

2.3.3 Economic analysis

To date, most of the work dealing with LAES has been focused on the technical aspects of LAES system. The technical studies aim at determining which parameters and device efficiency affect the most the key performance indicator above described. Only few papers move further into fully examining the economic aspects of LAES. Georgiou et al. [25] proposed a comparative thermo-economic analysis between LAES and Pumped-Thermal Electricity Storage System (PTESS). Although PTESS is found to more economic convenient at higher electricity buying prices, LAES is estimated to have lower capital cost and levelized cost of storage. Xie et al. [94] has assessed the economic feasibility of a LAES system by means of a developed a numerical method based on a genetic algorithm to identify the optimal LAES size (50,100,150 and 200 MW_e) and the optimal operational strategy through price arbitrage and/or short term operating service (STOR). The economic profitability of the system is highly dependent on the temperature level of the waste heat recovery and size plant. Indeed, it has been found that, without using waste heat, LAES is not economically advantageous: a positive net present value (NPV) is achieved only for a waste heat of at least 150 °C for a LAES plant of 200 MW_e. The payback period could vary

from 25.7 years to 5.6 years for a 200 MW system, with the use of waste heat ranging from 0 °C to 250 °C. Confirming the results achieved by Xie et al [94], Lin et al. [95] have evaluated the economic feasibility of LAES based on price arbitrage operations in the UK real-time electricity market. Pimm et al. [96] carried out a thermo-economic analysis for an energy storage installation comprising a compressed air component supplemented with a liquid air store. The system is supposed to achieve economic profit only by means of price arbitrage: an optimization algorithm has been developed to find the maximum profits available to the hybrid energy storage plant from a given set of electricity prices. The proposed hybrid system is found to be more economical than the respective stand-alone systems, CAES and LAES, under certain conditions (storage duration longer than 36 hours). Kim et al. [17] carried out a thermo-economic and environmental analyses of a hybrid LAES combined with LNG regasification and combustion. The hybrid system technical and economical performances have been compared to the ones achieved by a diabatic compressed air-energy storage (D-CAES) systems. The proposed system achieves higher round trip efficiencies (up to 73.4 %) but with a LCOE 9.4% higher than that of CAES system. Nevertheless, considering the geographical limitations and the environmental impacts of the CAES, the authors concluded that the proposed hybrid system is an economic and viable option. Hamdy et al. [97] proposed a techno-economic analysis of seven hybrid LAES systems based on the Levelized Cost of discharged Electricity (LCOE) figure of merit. LAES commercial scales in between 50 MW_e/100 MWh and 200 MW_e/400 MWh have been taken as reference for the whole study. Based on data from literature, the economic analysis has shown that the most significant results are achieved by the diabatic LAES system integrated with combustion of natural gas and the LAES waste heat recovery system with a LCOE of 161 €/MWh_e and 171 €/MWh_e, respectively.

2.3.4 A Liquid Air Economy

With a focus on the UK energy system, a full report published in 2013 from the Centre of Low Carbon Futures (CLCF) [28] evidences the need of a new energy vector able to overcome the problem related to the intermittency of the renewables (and balancing the energy supply and demand) and transform the electricity produced with low carbon sources,

in a form suitable for transport. Furthermore, the increasing of cooling demand, mainly due to the rising number of developing countries, has to lead to consider different ways to produce and deliver cold in a sustainable way. A smarter way to produce cold is also moved by the fact that a lot of cold energy is wasted to the environment from industrial processes. Indeed, since air is composed by 72% of nitrogen, all the gases industries that separate and liquefies air products for over a century, waste a large amount of liquid nitrogen that could be exploited, for example, to fuel transports.

Figure 2-18 Industrial park with LAES integration. Adapted from Ref.[16].

The second major source of waste cold is the LNG regasification. CLCF estimates, that the LNG imports in the UK will rise in 30 billion cubic meters in 2030 and, if the cold wasted from the regasification were exploited in the air liquefaction, it could produce around 8 million tonnes of liquid air per year. In this case, liquid air has been considered a potential sustainable energy vector for the grid, transport, and cooling. The use of liquid air allows operating with an energy vector with a higher energy density compared, for example, with the compressed air ($150\text{--}250 \text{ Wh}_e/\text{kg}_{\text{LA}}$ vs. $30\text{--}60 \text{ Wh}_e/\text{kg}_{\text{air}}$) [5]. In an energy system based on "liquid air economy" the liquid air has the main role to satisfy at the same time more than one energy needs. This can be feasible, today with the progress on the research and development of liquid air technologies and the market evolution. In the context of the

"liquid air economy", LAES is the key technology to produce the liquid air and balance the energy supply and energy demand of a grid based on energy produced with low carbon sources. Furthermore, LAES can be used as a sink of waste cold and waste heat thermal energy. In a context of a hypothetical industrial park (Figure 2-18), the LAES can be charged with the off-peak electricity from renewables, and if located next to an LNG terminal, the specific consumption can be reduced using the waste cold energy coming from the LNG regasification. The liquid air stored in the LAES tank can be used to produce electric power at peak times or extracted for different applications such as transport and cooling. Furthermore, the small specific volume, allows the possibility to be transported and be used for different purposes in many sites.

2.4 Research gap

From the comprehensive review carried out in the previous sections, it is possible to identify some areas of interest and opportunities where research has been limited as well as the research gap that the present thesis aims to fill (Figure 2-19).

Polygeneration LAES. Until now the research on LAES has focused its primary attention mainly on the electric storage section with the principal purpose of shaping the electric energy demand without considering the possibility to partially make use of the cold energy released by LAES during the discharge process. In fact, one of the most interesting features of LAES is that, besides producing electric energy, it also provides free cooling energy as an output of the expansion/regasification process.

LAES Performance maps. From the LAES simulation and case study application studies in literature, it can be concluded that until now there is not a generalized and systematic method that has been developed for researchers or engineers in order to design and calibrate LAES system. As a consequence, recalling the close analogy with gas turbine technology, a novel and general methodology to LAES system (plant based) design by means of dedicated performance maps could be developed.

Waste heat recovery for LAES performance improvement. A bottleneck to the current development of LAES is represented by the low value of round trip efficiency principally due to the large amount of energy consumption during the charge phase. In fact, in stand-alone configuration, despite the presence of a HGWS capable to partially recover the waste heat discharged by compression phase, the major contribution to exergy losses is again represented by heat rejection after air superheaters. Therefore, a potential for LAES improvement might be represented by the employment of other waste heat recovery solutions.

LAES LCA analysis. Currently, to the best of author's knowledge, there is no work in literature involving environmental analysis on LAES in order to demonstrate the potential environmental impacts associated with the use of this system.

Latent Heat High Grade Cold Storage. HGCS optimal design has been described by many literature works as a crucial since it allows halving the specific consumption of liquefaction plant increasing in turn the round trip efficiency exponentially. Nowadays, the research has focused his attention on sensible heat storage neglecting the use of PCMs that may guarantee a considerable saving cost due to lower specific consumption of the liquefaction process guaranteeing at the same higher energy density.

Figure 2-19 Areas of improvement identified during the literature review work.

Chapter 3 ²

Methodology - Liquid Air Energy Storage Modeling

The starting point for the research work is to characterize a LAES unit by means of a steady state model that comprises both charge (air liquefaction process) and discharge phases as well as thermal energy storages to bridge both temporally decoupled sections. The methodology and the mathematical models, used to retrieve the main results, are described in detail.

² This section published partially as:

- 1) Borri E, Tafone A, Romagnoli A, Comodi G. *A preliminary study on the optimal configuration and operating range of a “microgrid scale” air liquefaction plant for Liquid Air Energy Storage*. Energy Convers Manag 2017;143:275–85;
- 2) Tafone A, Romagnoli A, Li Y, Borri E, Comodi G. *Techno-economic Analysis of a Liquid Air Energy Storage (LAES) for Cooling Application in Hot Climates*. Energy Procedia 2017;105:4450–7.

3.1 Introduction to LAES system modeling

Thermodynamic numerical models are very crucial to investigate the steady state or the dynamic behavior of thermal energy systems under different boundary and initial conditions. It is therefore essential to have reliable predictions of the performance of a system before committing any resources looking for potential solutions for system performance improvement.

The approach used to model LAES is to thermodynamically analyze the different sub-sections (charge, discharge, storage and thermal energy storages) whose LAES is composed and subsequently adopt a global system perspective which puts a special focus on system requirements and on interactions between the sub-sections.

3.1.1 Modelling language and simulation environment

The LAES models in this thesis are developed in Aspen Hysys and Engineering Equation Solver (EES) commercial software environments. Aspen Hysys is an object-oriented program, widely used as software for simulation of steady state chemical and process plants and has been used extensively both in the literature and in commercial application for steady states models. EES is a general equation-solving program that can numerically solve a set of coupled non-linear algebraic and differential equations solution by means of the Newton-Raphson method in smaller groups.

The choice of those software for the models is based on the fact that they are the most widely used language for steady-state models of thermal energy systems in the research community. Therefore, the models and simulation results of this thesis can be easily replicated and compared to other works by the scientific community on this research field.

3.2 Air liquefaction process optimization

As underlined in Chapter 2, the literature published on LAES system highlights that the low round trip efficiency of the LAES is principally due to the high specific consumption of the liquefaction process. As a consequence, the main aim of this section is to analyze and compare different thermodynamic cycles used for air liquefaction process finding an optimal configuration and operating range in order to minimize the LAES specific consumption.

The air liquefaction processes considered in this analysis are supposed to produce 10 tons/day or 0.834 t/h hypothesizing a LAES charging process of 12 hours. In order to consider only environmentally friendly processes that do not involve external fluids (such as refrigerants or hydrocarbons) in the liquefaction process, only recuperative processes have been selected as the charge phase of the LAES: Linde-Hampson, Claude and Kapitza thermodynamic cycles.

3.2.1 Simulation assumptions

In order to define an optimal configuration that minimizes the specific consumption for the liquid air production, the following assumptions have been made and will be valid throughout the whole thesis dissertation:

- cryogenic cycles are considered in steady flow conditions;
- pressure losses along the cycles have been neglected in order to have a solution which compares different cycles under the same conditions;
- before air approach the cold sections of the liquefaction process, an air purifier, based on the molecular sieves technology, removes the components from the air (H_2O , CO_2 , hydrocarbons etc.) that would interfere with the cryogenic process. The power consumption of the air purifier is considered negligible in the simulation process. Such an assumption is reasonable as the power consumption of the air liquefier mainly consists of the power requirement for the compression of the feed air [98].

3.2.2 Air liquefaction process configurations modeling

The commercial software Aspen Hysys, widely used both in the literature and in commercial application for simulation of chemical and process plants, has been used to model the different thermodynamic cycles for air liquefaction process. The pinch analysis method was employed for the heat transfer model in the heat exchangers. Pinch analysis is a methodology for minimizing energy consumption by calculating thermodynamically feasible energy targets and achieving them by optimizing heat recovery systems, energy supply methods and process operating conditions. It can be used to account the minimum temperature difference in the heat exchanger and optimize the heat exchange process. The heat transfer process is represented as a set of energy flows as a function of heat load (kW) against temperature (°C), one for the hot stream and the other for the cold stream. The pinch point is where the closest approach of the temperature between the hot and cold streams.

Linde Hampson cycle. A Linde Hampson cycle is shown in Figure 3-1 and is composed by a two-stage compression train (compressors C1 and C2), an intercooler (IC) and aftercooler (AFC), a cold box, a J-T valve, a phase separator and a liquid air tank. The dehumidified ambient air is firstly compressed by the two-stage compression train at high pressure (1-3) with an intercooler stage in between and then cooled down near ambient temperature (3-3AFC) by means of the aftercooler. The air enthalpy at each compression stage outlet is:

$$h_{i+1} = h_i + \frac{h_{i+1,iso} - h_i}{\eta_{iso,C}} \quad (1)$$

while the air enthalpy at the intercooler (2IC) and aftercooler (3AFC) outlet is derived by the constraints imposed in Table 3-1. Then, the air passes through the cold box in which it is further cooled down (3AFC-4) by the non-liquefied air cold vapor recirculating from the phase separator (5VA-6):

$$\dot{m}_{air} * (h_{3AFC} - h_4) = \dot{m}_{air} * (1 - y) * (h_6 - h_{5VA}) \quad (2)$$

where the liquid yield y is defined as the ratio of the mass flow of liquefied air ($\dot{m}_{LA}$) over the total mass flow rate ($\dot{m}_{air}$) approaching the cold box after the compression stage:

$$y = \frac{\dot{m}_{LA}}{\dot{m}_{air}} = \frac{h_6 - h_{3AFC}}{h_6 - h_{LA}} \quad (3)$$

The J-T valve then completes the liquefaction process by expanding the air down to ambient pressure ($h_4 = h_5$); this leads to a two-phase mixture which is then separated in the phase separator where the liquid yield is extracted and stored in the liquid air tank. The cold vapor (7) after passing through the Cold box is then mixed with the ambient air at inlet of the compressor.

Figure 3-1 Process Flow Diagram of Linde-Hampson cycle

Claude cycle. Unlike the Linde-Hampson cycle, the Claude cycle includes an expander denominated CryoTurbine (CT) and two more heat exchangers (HE2, HE3). After the first heat exchanger (HE1), a large fraction of the high-pressure air flow is diverted to the CryoTurbine producing the useful electric power $P_{CT,ch}$ [kW_e]:

$$P_{CT,ch} = \dot{m}_{CT}(h_5 - h_{5CT}) \quad (4)$$

The remaining fraction undergoes the heat exchange process through the second and third heat exchanger. Applying the energy balance at the control volume highlighted in Figure

3-2, the liquid yield formula is derived:

$$\dot{m}_{air} h_{3AFC} = (\dot{m}_{air} - \dot{m}_{LA}) h_{13} + \dot{m}_{LA} * h_{LA} + \dot{m}_{CT} (h_5 - h_{5CT}) \quad (5)$$

$$y = \frac{\dot{m}_{LA}}{\dot{m}_{air}} = \frac{h_{13} - h_{3AFC}}{h_{13} - h_{LA}} + (1 - x_{RF}) \frac{h_5 - h_{5CT}}{h_{13} - h_{LA}} \quad (6)$$

$$x_{RF} = \frac{\dot{m}_{air} - \dot{m}_{CT}}{\dot{m}_{air}} \quad (7)$$

where x_{RF} is defined as recirculation fraction, namely the ratio of the mass flow approaching the Joule Thomson valve (6) to the compressed mass flow (1). The cold vapor leaving the cryogenic turbine (5CT), is then mixed with the stream (10) coming from the low temperature heat exchanger (HE3) in which the air from the compression process is further cooled down. In this cycle it is important to evaluate the optimal recirculation fraction that guarantee the optimum specific consumption for each charge pressure, namely the maximum pressure of the thermodynamic cycle.

Figure 3-2 Energy balance in the Claude cycle over the green control volume.

Kapitza cycle. Kapitza thermodynamic cycle is a variant of the Claude liquefaction process. Compared to the latter one, the third heat exchanger is removed and the air flow stream 7 is directly expanded in the J-T valve. As shown in Figure 3-3, the compression phase is the same as the Linde-Hampson and the Claude cycle. After the high temperature heat exchanger (HE1) the main stream is separated in two flows (6) and (5). Unlike the

Claude cycle, the first stream (6) passes through one heat exchanger (HE2) before being expanded in the J-T valve and separated in the phase separator, whereas the second stream (5) is directly used to drive the expander. The cold air leaving the engine (5CT) is mixed directly with the vapor coming from the tank (8VA) and cools down the air in the low temperature heat exchanger (in this case HE2). Unlike the Claude cycle, the cold vapor (9) enters the heat exchanger (HE2) at higher temperature, thus affecting the final temperature of the air entering the J-T valve (7) and hence the liquid yield of the two-phase mixture entering the tank (8LA). Likewise the Claude cycle, in the Kapitza cycle is important to evaluate the optimal recirculation fraction. Applying the energy balance at the control volume highlighted in Figure 3-3, the liquid yield formula is derived:

$$y = \frac{\dot{m}_{LA}}{\dot{m}_{air}} = \frac{h_{11} - h_{3AFC}}{h_{11} - h_{LA}} + (1 - x_{RF}) \frac{h_5 - h_{5CT}}{h_{11} - h_{LA}} \quad (8)$$

Figure 3-3 Energy balance in the Kapitza cycle over the green control volume.

3.2.3 Operative parameters and key performance indicators

The optimization process consisted in a parametric analysis; in particular, two parameters were investigated: the charge pressure, namely the maximum pressure guaranteed by the air compression, and the recirculation fraction (for Claude and Kapitza cycles).

The range of charge pressures investigated during the analysis of the Claude cycle and the Kapitza cycle goes from 6 bar to 60 bar that covers both the subcritical and supercritical

conditions (critical pressure of the air is 37.7 bar). The lower value represents the typical operating pressure adopted in the air separation process industry [99], while the higher value is adopted in the high pressure gas liquefiers [98]. For the Linde cycle the range of charge pressures for the parametric analysis is different since the typical charge pressures for this cycle are around 200 bar; in the current study the pressure range considered is between 150 to 300 bar. The boundary conditions applied to each cycle are summarized in Table 3-1.

Table 3-1 Operative conditions for Linde, Claude and Kapitza cycles simulations.

Parameter	Linde	Claude	Kapitza	Unit
AFC outlet temperature, $T_{AFC,out}$	30	30	30	$^{\circ}C$
AFC pressure loss, Δp_{AFC}	0.0	0.0	0.0	<i>bar</i>
J-T Valve outlet pressure, $p_{JT,out}$	1.01	1.01	1.01	<i>bar</i>
CT outlet pressure, $p_{CT,out}$	-	1.01	1.01	<i>bar</i>
HEs pressure loss, Δp_{HE}	0.0	0.0	0.0	<i>bar</i>
C isentropic efficiency, $\eta_{ISO,C}$	85	85	85	%
CT isentropic efficiency, $\eta_{ISO,CT}$	-	70	70	%
Pinch Point Approach				
HE1	5 ± 0.5	5 ± 0.5	5 ± 0.5	$^{\circ}C$
HE2	-	5 ± 0.5	5 ± 0.5	$^{\circ}C$
HE3	-	3 ± 0.3	-	$^{\circ}C$

The main key performance indexes have been computed using both energetic and exergetic approach.

Energy analysis. In order to compare the performance of the various cryogenic cycles from energetic perspective, the Specific Consumption [kWh_e/kg_{LA}] is defined as follows:

$$SC = \frac{P_{net,ch}}{\dot{m}_{LA}} = \frac{\sum P_{C,ch} - P_{CT,ch}}{\dot{m}_{LA}} \quad (9)$$

where $P_{C,ch}$ [kW_e] is the electric power consumed by the compressors during the liquefaction process.

Exergy analysis. In order to evaluate the exergy efficiency of the different liquefaction processes investigated and study their critical components, the exergy analysis has been carried out. In general, the exergy balance equation in a closed volume control can be written as follows:

$$\sum_{in} \dot{Ex}_{stream} - \sum_{out} \dot{Ex}_{stream} + \sum_i \dot{Q}_i \left(1 - \frac{T_o}{T_i}\right) - P_{net,ch} - \dot{Ex}_{loss} = 0 \quad (10)$$

where the first two terms $\sum_{in} \dot{Ex}_{stream}$ and $\sum_{out} \dot{Ex}_{stream}$ [kW] are associated with the exergy rate of the streams entering and leaving the control volume; these can be defined as:

$$\dot{Ex}_{stream} = \dot{m}_{stream} \cdot ex_{stream} = \dot{m}_{stream} \cdot [(h - h_o) - T_o(s - s_o)] \quad (11)$$

where $\dot{m}_{stream}$ [kg/s] is the mass flow rate, ex_{stream} is the specific exergy [kJ/kg], h represents the specific enthalpy [kJ/kg] and s the specific entropy [kJ/kg K] of the inlet and outlet streams. The terms h_o , s_o and T_o are associated to the enthalpy, entropy and temperature at the reference state, that are the air thermodynamic properties at 25°C and 1 bar. The third term of Eq. (21) represents the exergy related with the heat transfer: $\dot{Q}_i$ [kW_{th}] is the thermal power, T_o is the temperature of the reference state and T_i is the temperature is the temperature at the boundary that represents, for a heat engine or refrigerator, the temperature at which the heat is absorbed. In the exergy analysis, the system is assumed to be in steady state conditions and thermal losses in the heat exchangers are neglected.

The exergy efficiency η_{ex} can be calculated for each liquefaction process as follows [39]:

$$\eta_{ex} = \frac{\dot{m}_{LA}(ex_{LA} - ex_{amb})}{P_{net}} \quad (12)$$

where ex_{liq} and ex_{amb} refers to the specific exergy related with the liquid yield and with the ambient air respectively before mixing at the inlet of the compressor. The exergy losses rate $\dot{Ex}_{loss}$ can be calculated by considering a control volume and applying the general exergy balance Eq. (11) to each component. The equations used are summarized in Table 3-2 and the subscripts refers to the stream entering and leaving each component; the exergy

rate loss in the liquid air tank and phase separator is neglected.

Table 3-2 Exergy losses equations for each component of the liquefaction process.

Component	Exergy losses balance equation
C	$Ex_{loss} = Ex_{in} - Ex_{out} + P_C$
IC/AFC	$Ex_{loss} = Ex_{in} - Ex_{out}$
CT	$Ex_{loss} = Ex_{in} - Ex_{out} - P_{CT}$
HE	$Ex_{loss} = (Ex_{in_{cold}} - Ex_{out_{cold}}) + (Ex_{in_{hot}} - Ex_{out_{hot}})$
J-T valve	$Ex_{loss} = Ex_{in} - Ex_{out}$

3.2.4 Results

This section presents the simulation results of the different configurations systems modelled as charge phase of Liquid Air Energy Storage system. The results of the different configurations have been compared each other for a daily liquid air production of 10 ton/day, considered as the reference for a “micro-grid” scale. In order to assess the influence of the main parameters affecting both the round trip efficiency and the waste heat recovery process, a comprehensive sensitivity analysis has been carried out for the charge pressure, the recirculation fraction and the storage pressure of the liquefaction process.

3.2.4.1 Effect of charge pressure and recirculation fraction on specific consumption

Figure 3-4 reports the results for the Linde-Hampson cycle in terms of specific consumption versus charge pressure. The figure shows that the charge pressure of the Linde – Hampson cycle should operate with very large pressures in order to reduce the specific consumption substantially. The main reason for the low efficiency is due to the large temperature difference between the cold vapour and the air heat exchanger (Cold box in Figure 3-1) which leads to a significant loss in the whole cycle performance and in a low liquid yield fraction that increases the specific consumption.

Figure 3-4 Specific Consumption of the Linde-Hampson cycle at different charge pressures.

Figure 3-5 shows the results for the Claude and Kapitza cycle under different recirculation fractions and charge pressures. In accordance with Barron [99], for each charge pressure, there is a value of the recirculation fraction that minimizes the specific consumption due to the non-linear relation between the two parameters. For Claude cycle the lowest specific consumption ($0.72 \text{ kWh}_e/\text{kg}_{\text{LA}}$) is achieved at supercritical pressure (refer to 40 bar pressure case) and 0.2 recirculation fraction. Indeed, the specific consumption of the cycle is affected by the heat exchangers performance that depends on the charge pressure of the cycle itself. Generally, the higher the charge pressure the higher is the heat exchangers performance; however, this is valid as long as the recirculation fraction is increased. However, for higher values of the charge pressure, the minimum specific consumption does not vary significantly as the recirculation fraction increases.

Figure 3-5 Comparison of the specific consumption of Claude and Kapitza cycle at different charge pressures.

The results for the Kapitza cycle are again showing that the specific consumption of the cycle is significantly influenced by the heat exchangers performance that in turn is mainly related with the charge pressure and the mass flow rate of the two streams: the higher the performance of the heat exchangers, the higher the yields of the liquefaction process. In order to understand the effect of the charge pressure, Figure 3-6 shows the temperature profiles of the two heat exchangers (HE1 and HE2) for the Kapitza cycle at the subcritical and supercritical operative conditions achieving the minimum specific consumptions (subcritical pressure of 10 bar and recirculation fraction of 0.1; for supercritical pressure of 40 bar and recirculation fraction of 0.2). By comparing the profile of the low temperature heat exchangers (HE2) for $p_{ch} = 40$ bar (Figure 3-6d) and the 10 bar (Figure 3-6b), it is possible to see that the curve profile of the hot stream (6-7) for $p_{ch} = 40$ bar better follows the cold stream (9-10); this leads to a lower exit temperature of the hot stream and a lower thermal power of the hot temperature heat exchanger (HE1) as shown in Figure 3-6c and Figure 3-6a.

Figure 3-6 Kapitza cycle. Plots of the heat exchange processes in HE1 (a, c) and HE2 (b, d) for $p_{ch} = 10$ bar and $x_{RF} = 0.1$ of recirculation fraction (a, b) and $p_{ch} = 40$ bar and $x_{RF} = 0.2$ (c,d).

Figure 3-7 shows the behavior of the two heat exchangers (HE1 and HE2) for a charge pressure of 40 bar but with a recirculation fraction of 0.1. This configuration allows the comparison between the subcritical Kapitza cycle operating at 10 bar and recirculation fraction of 0.1 with the supercritical Kapitza cycle operating at 40 bar with recirculation fraction of 0.2. The comparison with Figure 3-6 (a,b) and Figure 3-7 shows that with a recirculation fraction of 0.1, the two heat exchangers (HE1 and HE2) perform better for the subcritical Kapitza as confirmed by the lower specific consumption shown in Figure 8. With respect to Figure 3-6 (c, d) and Figure 3-7, it is apparent that, the 0.2 recirculation

fraction improve the overall heat exchange reducing the specific consumption.

Figure 3-7 Kapitza cycle. Plots of the heat exchange processes in HE1 (a) and HE2 (b) for $p_{ch} = 40$ bar and $x_{RF} = 0.1$.

By analyzing Figure 3-5, Claude and Kapitza cycle show similar specific consumptions trends. In order to understand the difference between the Kapitza and the Claude cycle it is necessary to investigate in depth the behaviour of the components, in particular the heat exchangers that is determinant for the cycle efficiency. With respect to the condition of minimum specific consumption presented in Figure 3-6 (charge pressure of 40 bar and recirculation fraction of 0.2), Figure 3-8 shows the curves of thermal power vs. the flow temperature of the three heat exchangers of the Claude cycle. More in detail Figure 3-8a shows the heat flow profile for the high temperature heat exchanger; Figure 3-8b shows the medium temperature heat exchanger, where is possible to notice a non-linearity of the curves due to the phase change inside the heat exchanger; Figure 3-8c shows the heat flow profiles for the low-temperature heat exchanger. In particular, Figure 3-8c shows that the thermal power exchanged is very low meaning that the low-temperature heat exchanger releases heat to a small amount of cold vapour coming from the tank. The discrepancy between the mass flow of the two stream results in a small temperature drop of the hot stream and a substantial temperature rise of the cold stream; this represents an increase of irreversibility and a decrease of the performances.

Figure 3-8 Claude cycle. Plots of the heat exchange processes in HE1 (a), HE2 (b) and HE3 (c) for $p_{ch} = 40$ bar and $x_{RF} = 0.2$.

A summary of the minimum specific consumption for each liquefaction process and an optimal range of operating condition for each cycle with two stage compression is reported in the Table 3-3.

Table 3-3 Summary of the optimal operating conditions range.

Parameters	Linde	Claude	Kapitza	Unit
Charge Pressure, p_{ch}	240-260	38-60	38-60	bar
Recirculation Fraction, x_{RF}	-	0.2-0.25	0.2-0.25	-
Specific Consumption, SC	2.5-2.6	0.72-0.73	0.71-0.72	kWh_e/kg_{LA}

Referring to the thermodynamic cycles with the optimal operating conditions achieving the minimum specific consumption, the exergy analysis comparison has been carried out conducted for each cycle and reported in Figure 3-9. Although the value of the exergy efficiency is low for all the three configurations, the Claude and Kapitza cycles give better results than the Linde cycle. Indeed, the Linde cycle, as reported in Section 3, has a higher specific consumption due to the high work of compression. The small absolute difference (0.4 %) between the Kapitza and Claude cycles can be attributed to the presence of the third heat exchanger that contributes to additional exergy loss.

Figure 3-9 Exergy efficiency of the Linde, Kapitza and Claude cycles.

In order to understand the low value of the exergy efficiency, the exergy loss in each component of the Kapitza cycle - operating at 40 bar and 0.2 of recirculation fraction - is reported in Figure 3-10. The 27.6 % exergy efficiency, reported in Figure 3-9, is due to the significant exergy losses related to the heat losses at the intercooler and aftercooler. Indeed, the heat of compression is rejected directly into the environment and it is not recovered; this suggests that if meaningfully recovered, this waste heat could improve the exergy efficiency.

Figure 3-10 Kapitza cycle. Exergy losses distribution for $p_{ch} = 40$ bar and $x_{RF} = 0.2$. The values on the top of each bar represent the absolute exergy losses rate (kW).

From Figure 3-10, independently from the way in which the air is compressed and heat is being rejected, the results show that the exergy loss in the J-T valve and the two heat exchangers (HE1) and (HE2) is relatively small as compared with the cryoturbine (CT) that results to be the critical component of the liquefaction process. The high value of exergy losses linked to the cryoturbine can be attributed to the low isentropic efficiency of 70% assumed as input parameter.

3.2.4.2 Effect of the storage pressure and the isentropic efficiencies of turbomachinery components on the specific consumption

As a last step toward the definition of the parameter that affect the performance of the liquefaction process, the effect of the storage pressure is evaluated for the reference cycle (Kaptiza) under an optimal recirculation fraction of 0.2 and supercritical pressures condition. Indeed, a LAES system is able to implement vacuum insulated storage tank [9] that can be operated at higher pressure than atmospheric. In this section, the effect of the air tank pressurization is evaluated for the Kaptiza cycle under various operating conditions. The layout of the Kapitza cycle with pressurized liquid air tank is almost the same as that already shown in Figure 3-3; the main difference lies in the compression stage, which is shown in some detail in Figure 3-11: the return air (11) mixes with the ambient air which is at different pressure; hence it is necessary to add another compressor (C3), with an aftercooler, in order to pre-compress the ambient air before mixing.

Figure 3-11 Process Flow Diagram of the Kapitza cycle with pressurized LA tank.

The parametric analysis for the cycle with pressurized LA tank was carried out by varying the charge pressure of the cycle, the air tank pressure between 1 and 12 bar and isentropic efficiencies of the turbomachinery devices. In order to reduce the specific work, the pressure ratios of the compressors (C1, C2) are assumed to be the same ($\beta_{C1} = \beta_{C2}$). The boundary conditions for the pressurized cycle are summarized in Table 3-4.

Table 3-4 Operating conditions for Kapitza pressurized cycle simulations.

Parameter	Value	Unit
AFC outlet temperature, $T_{AFC,out}$	30	$^{\circ}C$
AFC pressure loss, Δp_{AFC}	0.0	<i>bar</i>
HEs pressure loss, Δp_{HE}	0.0	<i>bar</i>
C isentropic efficiency, $\eta_{ISO,C}$	85	%
CT isentropic efficiency, $\eta_{ISO,CT}$	70	%
Minimum Approach		
HE1	5 ± 0.5	$^{\circ}C$
HE2	3 ± 0.3	$^{\circ}C$

In Figure 3-12 a performance map is plotted; the specific consumption is shown as a function of the ratio between the charging pressure and tank pressure with a range for the charging and tank pressures varying between 40 - 90 bar and 1 - 12 bar respectively. The dashed lines represent constant charge pressure curves (from 40 bar to 90 bar with constant Δp increment of 10 bar) while continuous lines represent constant tank pressure curves (1, 2, 4, 6, 8, 10, 12 bar). For each of the proposed charts, the operating conditions and efficiencies described in Section 3.2.3 have been considered; by varying some of these parameters the charts will shift.

Figure 3-12 Kapitza cycle. Combined effect of charge pressure and LA tank pressure on the Specific Consumption.

The results show that the specific consumption improves as the maximum charge pressure increases: the tendency suggests that the higher is the tank pressure the higher is the difference between the two extreme cases ($p_{ch} = 40$ bar and $p_{ch} = 90$ bar) in terms of specific consumption. Instead, as we move from 12 bar to 1 bar, the effect of the tank pressure on specific consumption is significant even though this tends to decrease as the pressure of the returning cold flow increases. In fact, the specific consumptions are sensibly reduced from 1 bar to 2 bar (blue and red continuous lines, respectively) with an average decrease of about 13%, whereas from 8 to 12 bar (orange and azure continuous lines, respectively) the reduction is less than 1%.

This twofold tendency could be explained firstly by considering that the higher the pressure of the returning cold flow the higher will be its heat capacity which means that the effectiveness of the heat exchange in the cold box will be positively affected. Secondly, the analysis of the trend associated with the relative variation of the net compression power ($\Delta \dot{W}_{net,c} / \dot{W}_{net,c}$) and that of the liquid air mass flow rate ($\Delta \dot{m}_{LA} / \dot{m}_{LA}$) vs. the tank pressure, shows that beyond 8 bar, the higher is the tank pressure the less considerable is its positive effect over the specific consumption (Figure 3-13). In fact, as demonstrated by

the Eqs. (24-26), the difference between the relative variation of net compression power and liquid air mass flow rate is proportional to the decrease in specific consumption:

$$S.C.(p_{LA} = 1 \text{ bar}) = \frac{\dot{W}_{net,c}}{\dot{m}_{LA}} \quad (13)$$

$$S.C.(p_{LA} > 1 \text{ bar}) = \frac{\dot{W}_{net,c} - \Delta\dot{W}_{net,c}}{\dot{m}_{LA} - \Delta\dot{m}_{LA}} \quad (14)$$

$$S.C.(p_{tank} = 1 \text{ bar}) > S.C.(p_{tank} > 1 \text{ bar}) \quad \text{only if} \quad \frac{\Delta\dot{W}_{net,c}}{\dot{W}_{net,c}} > \frac{\Delta\dot{m}_{LA}}{\dot{m}_{LA}} \quad (15)$$

Figure 3-13 Kapitza cycle. Relative variation of net power compression and liquid air mass flow as function of the pressure of the liquid air tank ($p_{ch} = 60$ bar).

In addition to this, in Figure 3-12 the results of a sensitivity study on the efficiencies of the main components (compressors and cryoturbine) over the specific consumption are shown; two extreme cases have been considered in which the highest and lowest possible efficiencies for these components have been assigned (90% and 65% efficiencies respectively) while still maintaining the same charge pressure and tank pressure. The figure clearly shows that as the efficiency values either decrease or increase, the chart for the specific consumption shifts towards higher and lower values respectively. Besides the

shifting of the chart, the change in the efficiency values also leads to a change in the width of the chart itself (i.e. change in the range of specific consumption); indeed, for the 90% efficiency case, the specific consumption becomes less sensitive to the variation of the charge pressure ($SC_{\min} \approx 0.31 \text{ kWh}_e/\text{kg}_{\text{LA}}$ vs $SC_{\max} \approx 0.44 \text{ kWh}_e/\text{kg}_{\text{LA}}$) whereas the opposite occurs for the 65% efficiency case ($SC_{\min} \approx 0.62 \text{ kWh}_e/\text{kg}_{\text{LA}}$ & $SC_{\max} \approx 0.98 \text{ kWh}_e/\text{kg}_{\text{LA}}$). This can be explained by considering that in the best case scenario, where the compressors and cryogenic turbine achieved an isentropic efficiency of 90 %, the positive effect of those devices performances overcomes the potential inefficiencies due to a not optimal charge pressure. Instead, for the worst case scenario, the negative effect of lower isentropic efficiency of the main devices on the specific consumption is amplified by the choice of the charge pressure. Therefore, as a general statement, the higher are the performances of compressors and cryogenic turbine, the more flexible is the operation of air liquefier in terms of charge pressure. Hence based on the sensitivity study of Figure 3-12, the best and worst specific consumption case are provided for LAES based on the Kapitza cycle.

3.2.5 Resume of the main findings

In the present optimization analysis, the optimal plant configuration is the one that minimizes the specific consumption. It has been shown that the Claude and Kapitza cycles have the lowest specific consumption and similar results in terms of operating range (charge pressure and recirculation fraction). Although this similarity is also found in terms of exergy efficiency, the results demonstrate that the third heat exchanger in the Claude cycle can be avoided and that the Kapitza cycle result to be more effective in term of minimum specific consumption. Moreover, smaller heat exchangers are required with advantage in volume and cost reduction. This led to propose the Kapitza cycle as the best configuration. A reduction of the specific consumption is also showed when the air tank of the Kapitza cycle is pressurized. Exergy analysis shows that the configuration proposed can be furtherly improved by reducing the impact of exergy losses in the aftercooler. This can be done either by recovering the waste heat at outlet of the aftercooler or by recovering waste cold energy from other processes such as the discharging phase of LAES.

3.3 Discharge process

This section presents the thermodynamic and configuration architecture assumptions related with the whole LAES comprising both the charge and the discharge processes. The discharge phase implemented makes use of a direct expansion process, previously described in Section 2.2.2, not involving any external sub-cycles and/or working fluids only leveraging the high thermal conversion rate of low grade waste heat into mechanical work as shown by Morgan et al. [27]. As previously stated in Section 2.3.2.2, LAES can be operated in two different modalities: full electric configuration, where the only output is represented by the electric energy, and polygeneration configuration capable to provide both electric and cold energy by means of power turbines and cold streams at turbine outlet, respectively. In polygeneration configuration, LAES is assumed to be thermally coupled with a water cooled chiller providing a chilled water to a hypothetical user at a temperature of 7 °C.

3.3.1 Simulation assumptions and key performance indicators

Beside the hypothesis made on the liquefaction process still valid in the following analysis, the main design features for each individual components of the discharge phase are given in Table 3-5. In addition, the following assumptions have been made throughout the thermodynamic analysis of the discharge cycle:

- all the components operate in steady state conditions;
- the electric power consumptions of the HGWS and HGCS are negligible;
- pressure losses in the components other than the expanders are negligible;
- auxiliary electrical losses are not included in the model.

Table 3-5 Operating main design parameters for LAES discharge phase components.

Parameter	Component	Value	Unit
Isentropic efficiency	Power turbines	0.8	%
	Cryopump	0.8	
Delta Temperature Pinch Point	Evaporator	10	°C
	Superheaters	10	

The following performance parameters has been introduced for the LAES discharge phase:

Round trip efficiency [%], defined as the ratio of the net energy recovered during discharge over the net compression energy during charging. In case LAES system is operated in a steady-state regime, the round-trip efficiency can be defined as the ratio of the net power of the discharge phase $P_{net,d}$ [kW_e] to the net power of charge phase $P_{net,ch}$ [kW_e] of the LAES multiplied by the ratio of the charging time (τ_{ch}) to the discharging time (τ_d):

$$\eta_{RT} = \frac{E_{net,d}}{E_{net,ch}} = \frac{P_{net,d}}{P_{net,ch} \cdot (\tau_{ch}/\tau_d)} = \frac{\sum P_{T,d} - P_{CP,d}}{(\sum P_{C,ch} - \sum P_{CT,ch})} \quad (16)$$

Overall energy storage efficiency [%], that takes into account the cooling load $\dot{Q}_c$ provided in poly-generation configuration (converted into electricity by means of the COP of the chiller technology):

$$\eta_o = \frac{\sum P_{P,d} - P_{CP,d} + \frac{\dot{Q}_c}{COP_{chillers}}}{(\sum P_{C,ch} - \sum P_{CT,ch}) \cdot (\tau_{ch}/\tau_d)} \quad (17)$$

Exergy efficiency [%] of charge and discharge of the system (i.e. liquefaction and regasification respectively):

$$\eta_{ex,ch} = \frac{\dot{E}x_{LA} + \dot{E}x_{HTF}}{\sum P_{C,ch} - \sum P_{CT,ch} + \dot{E}x_{HGCS}} \quad (18)$$

$$\eta_{ex,d} = \frac{\sum P_{T,d} - P_{CP,d} + \frac{\dot{Q}_c}{COP_{PL,phase}} + \dot{E}x_{HGCS}}{\dot{E}x_{LA} + \dot{E}x_{HTF}} \quad (19)$$

where $\sum P_{T,d}$ is the power produced by the power turbines during the discharge phase; $P_{CP,d}$ is the power input for the cryogenic pump; $\sum P_{C,ch}$ is the power input for the

compressors; $\sum P_{CT, ch}$ is the power produced by the liquefier Kapitza turbine; $\dot{m}_{LA}$ is the mass flow of liquefied air coming from the storage tank; $\dot{Ex}_{LA}$ represents the exergy flow rate of liquid air produced during the charging phase; $\dot{Ex}_{HTF}$ and $\dot{Ex}_{HGCS}$ are the exergy flow rate associated with the hot thermal oil and the High Grade Cold Storage, respectively; $COP_{chillers}$ is the average COP of chillers taken as reference.

3.3.2 Effect of the number of expansion stages

The simplest LAES configuration proposed is the one that makes use only of the thermal energy from the environment (Figure 3-14).

Figure 3-14 Process flow diagram of LAES system with 4 reheating stages during expansion and ambient air as heat source.

The main parameter under investigation is represented by the maximum discharge pressure imposed by the CryoPump. Figure 3-15 reports the round trip efficiency as a function of the discharge pressure under different number of reheating stages for a charge pressure of 60 bar. In fact, in order to increase the round trip efficiency, the air undergoes different expansion stages: after each expansion the air is reheated by the ambient environment and

further expanded in the next power turbine; in order to maximize the total power extracted through the expanders, each expansion is characterized by the same pressure ratio ($\beta_{PT1} = \beta_{T2} = \dots = \beta_{Ti}$).

Figure 3-15 Round trip efficiency as a function of maximum discharge pressure under different reheating stages for $p_{ch} = 60$ bar.

The curves of Figure 3-15 show that the optimum value of the round trip efficiency is achieved for different discharge pressures as more reheating stages are included. It is worth noting that the relative difference in term of round trip efficiency between the 3 stages and 4 stages tends to be quite negligible and therefore a further complication of the system with the introduction of more turbines might be not justified by the performance improvement.

3.3.3 Effect of the High Grade Cold Storage

Another option to further improve the key performance indicators of the LAES is represented by the introduction of a high grade cold storage, which involves by means of a “cold recycle” capturing and storing the cold thermal energy released during liquid air regasification and using it to reduce the work required for the liquefaction process. In fact, the component HGCS is used as cold thermal energy storage when the liquefaction and the discharge processes operate at different times. In the configuration proposed, the Heat

Transfer Fluid used to charge and discharge the HGCS is dehumidified ambient air. When the LAES discharge phase is operative, the HGCS charge takes place (continuous blue line in Figure 3-16). Conversely when LAES is in charge phase, the HTF is circulating between the HGCS and the cold box (dashed blue line in Figure 3-16) in order to thermodynamically assist the liquefaction process. The HGCS is numerically modeled by means of its thermal efficiency, namely the ratio between the useful thermal power used during the liquefaction process in the cold box and the total available thermal power discharged by the regasification of liquid air:

$$\eta_{HGCS} = \frac{\dot{Q}_{u,HGCS}}{\dot{Q}_{tot,HGCS}} \quad (20)$$

Figure 3-16 Process flow diagram of LAES system with HGCS implementation.

Taking into account the mentioned assumptions, the specific consumption is computed and plotted in Figure 3-17 for different HGCS efficiency for a discharge pressure of 100 bar. The results show that the effect of HGCS is significant leading to a substantial decrease of specific consumption ($0.23 \text{ kWh}_e/\text{kg}_{LA}$) as compared to the results achieved considering only air liquefaction plant operating in stand-alone mode ($0.48 \text{ kWh}_e/\text{kg}_{LA}$).

Figure 3-17 Specific consumption as a function of available recycled cold flow for $p_{ch} = 60$ bar and $p_d = 100$ bar.

3.3.4 Effect of the High Grade Warm Storage

As previously shown in Section 2.2.3, the recovery of the waste heat discharged by the charge phase during air compression process plays a significant role in achieving higher round trip efficiencies. In order to further improve the performance of the LAES, the impact of waste heat recovery has been assessed as shown in Figure 3-18. The heat from the compression process is recovered in a similar way as in the Adiabatic Compressed Air Energy Storage concept where the thermal energy generated by the compression is stored in a packed bed thermal energy storage and then used to reheat the air before it is expanded in the power turbine. In the present work, the HGWS has been modelled by considering an intermediate circuit charged with thermal oil (Therminol 66) used as HTF.

Figure 3-18 Process flow diagram of stand-alone LAES cycle with HGCS and HGWS implementation.

The combined effect of the HWCS and HGCS is examined in Figure 3-19 where the maxima of the round trip efficiency is plotted as the pressure of the discharge process varies. The HGCS configuration has been implemented with an ideal exploitation of 100 % of the available cold flow. The plot shows that the presence of both thermal energy storages shifts the maximum of the round trip efficiency towards lower pressure. In fact, in accordance with [46], the benefit of the discharge pressure increase is reduced as it exceeds 160-170 bar. In fact, increasing the discharge pressure lowers the waste cold to be recycled due to the increase of the liquid air temperature at the outlet of the cryogenic pump caused by the pumping work. As a consequence, the positive effect on the higher inlet enthalpy values for the power turbine offsets the negative impact of the higher specific consumption.

Figure 3-19 Round trip efficiency as a function of discharge pressure ($p_{ch} = 60$ bar).

3.4 Thermal demand side management: techno-economic case study

This case scenario investigates the technical and economic feasibility of a LAES system for building demand management applications. The quantitative analysis has been carried out for a daily cooling energy demand of an existing office building, located in Singapore, locality characterized by a typical hot climate. The School of Art, Design and Media (ADM), located within the Nanyang Technological University (NTU) campus in Singapore, has been taken as reference case. For additional details on the implemented methodology and the key performance indicators assessed, the reader can refer to Tafone et al.[22].

3.4.1 Energy Cooling Demand Data

The case study covered in this work is for a building located in Singapore; air conditioning for the building is provided by water cooled chillers: the chiller plant is fitted with three water cooled chillers. Chillers (CH) A and B are fitted with centrifugal compressor, using R-123 refrigerant, having a cooling capacity of 1582 kW_c each. Chiller C is fitted with a screw compressor, using R-134a refrigerant, having a cooling capacity of 1055 kW_c. Chiller B usually provides the cooling energy demand exceeding the capacity of chiller C.

Chiller A is usually used as backup unit. Usually, the building is closed on Sundays and Public Holidays (PH) and therefore none of the chillers operates during these periods.

The energy audit of the building and the analysis of both cooling load and COP of the cooling system have been already assessed by a previous study [12] that has utilized real data obtained by monitoring the chiller system over 4 months. Since in Singapore there is no real alternation in climate between summer and winter, the measured cooling load is almost steady throughout the year, thus, based on the behavior of the building over 4 months, a representative cooling load profile for a typical working day has been provided, as shown in Figure 3-20. Three different operating phases can be identified for the cooling system: a peak-load phase in the morning between 07:00 and 09:00; a maintaining phase, between 09:00 and 19:00, that covers most of the day when the cooling load ranges between 1000-1200 kWc; a partial load phase, between 19:00 and 23:00, where the reduction of cooling demand is due to lower occupancy of the building. The average COP of the chiller system is 5.343 (during office hours between 08.30 and 17:30). The energy audit and the analysis of both cooling load and COP of the cooling system has underlined potential for further improvement of its techno-economic performance.

The purpose of the present analysis is to assess the economic viability of using LAES to implement demand side management strategies in order to exploit the price arbitrage potential due to the difference between peak and off-peak electric tariffs in Singapore. In particular, LAES is used to replace the less efficient chillers (chillers A and B) between 09:00 and 19:00. The proposed strategy would be that of running the LAES together with chiller C (more efficient than chillers A and B) in order to reduce the peak load during the maintain phase (yellow area in Figure 3-20) so that only the most performing chiller (chiller C) needs to operate; the other two serve as backup units. In this case, the LAES would be charged during night time with consequent economic benefit due to price arbitrage.

Figure 3-20 Cooling load profile for a typical normal operative day [12].

3.4.2 LAES polygeneration configuration design

One of the most interesting features of LAES, is that besides producing electric energy it also provides heat and cool as by-product of the charge and discharge phase respectively; hence the LAES can also be considered as a poly-generation system capable to be integrated with an air conditioning system, in order to supply a well-defined cooling load, and with a heat exchanger network, in order to be used in industrial settings and/or space heating/domestic hot water.

In this case scenario, in order to fulfil the cold energy demand required by the building, the air at turbine outlet is thermally coupled with the water cooling circuit, by means of three heat exchangers (AC1, AC2, AC3), as shown in the process flow diagram in Figure 3-21.

Figure 3-21 Simplified schematic of the LAES discharge phase integrated with a district cooling system.

3.4.3 Exergy analysis

Based on the assumptions of Section 3.4.1, the simulations showed that, under an optimal charge and discharge pressures of 80 bar and 124 bar, respectively, a specific consumption of 0.226 kWh/kg_{LA} and an overall round trip efficiency of 45 % could be achieved. It was assumed that the new cooling system, integrating the cold storage and the existing chillers, had to satisfy a daily average cooling energy demand of 12,872 kWh_c, considering 275 operative days per year. The main performance parameters are specified in Table 3-6.

Table 3-6 Thermodynamic results.

Parameter	Value	Unit
Cooling Demand, Q_c	731	kWh _{th}
LAES Power Rating, $P_{net,d}$	982	kW _e
Round trip efficiency, η_{RT}	45	%
Specific consumption, SC	0.20	kWh _e /kg _{LA}
Exergy efficiency liquefaction, $\eta_{ex,ch}$	84	%
Exergy efficiency discharge, $\eta_{ex,d}$	67	%

As illustrated by Figure 3-22 and Figure 3-23, the results achieved with the exergy analysis show that the analyzed configuration achieves high level of exergy efficiency for the charging phase, thanks to the presence of the HGCS, while exergy efficiency is sensibly lower for discharge process. Strictly in accordance with [42], the method employed to extract the cold exergy from the cryogen is the direct expansion method, a simple but also inefficient discharge process. In fact, it does not fully use the cold energy of the cryogen since the cold energy discharged by liquid air in HGCS is recycled in order to decrease the specific consumption of the charging process. Moreover, since the cold energy is provided to the building by means of the flow at turbine outlet, the maximum air temperature at turbine inlet is limited by the chilled water supply temperature. Therefore, as emphasized by the notable dissipation of exergy flow linked with the waste heat recovery system, due to energy cooling demand, the system is not able to fully exploit the WHR provided during the compression phase.

Figure 3-22 Irreversibility distribution for liquefaction process.

Figure 3-23 Irreversibility distribution for discharge phase.

3.4.4 Economic analysis

The scenario analyzed is meant to exploit the price arbitrage potential due to the difference between peak and off-peak electricity tariffs in Singapore, shifting the daily average surplus due to cooling peak (731 kWh) in the average working day. In fact, during off-peak hours LAES is charged while from 09:00 to 19:00 chiller C supplies the cooling energy required to the building at its maximum capacity: whenever the energy demand exceeds this limit, the cold storage provides the surplus energy required. As shown by Table 3-7 the economic investment is not economic viable due to current low round trip efficiency (45 %),

the actual PT/OPT of Singapore and the high COP of the chillers. It is worthwhile noting that Singapore represents the worst case scenario for the present study since the nominal COP of the chiller is sensibly higher compared to European standard characterized by lower COP values ($\approx 3.5-4$).

Table 3-7 Economic results.

Ref.Case	Round trip Efficiency [%]	PT [USD/kWh _e]	OPT [USD/kWh _e]	CAPEX [MUSD]	Annual Savings [MUSD]
	45	0.138	0.09	5.4	Negative

In order to investigate the combined effect of PT/OPT and round trip efficiency over the economic feasibility of the LAES coupled with the chillers, a sensitivity analysis has been carried out.

Each curve presented in Figure 3-24 and Figure 3-25, for a defined value of overall round trip efficiency, shows the annual savings and the payback period of the system function of the OPT, expressed as a percentage of PT. It points out that for the reference case the break-even point of the investment is achieved only if the OPT is about 45 % of PT. Moreover, the annual savings are always positive just for value of round trip efficiency above 70%. As highlighted by Figure 3-25, at high round trip efficiency ($>60\%$), the payback period tends to be economically remarkable (< 20 years) only if the off-peak energy tariff is about 20 % of the peak one. It is worthwhile noting that such an analysis does not take into account the operative costs associated with LAES that may put at stake the economic feasibility of the investment or in alternative make it particularly profitable. As a final remark, it is worthwhile pointing out that since the technology considered in this study has not achieved the market maturity: the figures for prices provided by literature can be considered more as estimates than actual market prices.

Figure 3-24 Annual Savings function of OPT and η_{RT} .

Figure 3-25 Payback period function of OPT and η_{RT} .

3.5 Summary

The present chapter contributes to provide a preliminary analysis for the estimation of the LAES performance and to suggest an optimal configuration and an optimal operating range to be used as a guideline for future researches on LAES applications.

A preliminary analysis is conducted first comparing different liquefaction processes. From the simulation results, Kapitza cycle is proposed as the best configurations among the ones addressed that guarantees the lowest specific consumption with the range of operating conditions proposed in Table 3-8. Indeed, a final optimal configuration for the liquefaction process can be considered a Kapitza cycle with an operating pressure in the range of 40-60 bar and a storage pressure of 8 bar, due to the positive effect of the liquid air tank pressurization on the specific consumption.

Table 3-8 Optimal operating parameters for the Kapitza cycle.

Parameters	Value	Unit
Cycle	Kapitza	-
p_{ch}	40-60	<i>bar</i>
x_{RF}	0.15-0.3	-
p_s	8 bar	<i>bar</i>
SC	0.48-0.52	<i>kWh_e/kg_{LA}</i>

The exergy analysis has shown that the highest exergy losses occur during the aftercooling/intercooling process due to the waste heat discharged to the environment: as a consequence waste heat recovery process can be configured as a potential method to further improve air liquefaction efficiency.

The direct expansion process has been chosen for the model of the LAES discharge phase. Interheating process has been implemented in order to increase the power output and a 4 stages expansion configuration seems to be a good compromise between high level of performance and plant complexity. Both High Grade Cold Storage and High Grade Warm storage are crucial components to increase the round trip efficiency from about 15 % to 48 %.

Finally, a thermodynamic and economic case study carried out on LAES for building demand management in Singapore was analyzed assessing the capability of LAES to exploit the difference between peak and off-peak electricity rate, thus leveraging on price arbitrage strategy in order to reduce peak loads. The resulted value of round trip efficiency in cooperation with the high COP of chillers and the PT/OPT of Singapore does not allow to achieve the economic feasibility of the investment (negative annual savings). Analyzing

the effect of PT/OPT and round trip efficiency over the economic key performance indices, for the reference case the break- even point is achieved only if OPT is approximately 45 % or below of the PT value. Even though the annual savings are always positive when the round trip efficiency of LAES is increased to higher level ($>70\%$), the sensitivity analysis on payback period confirms that only at low OPT percentage of PT the investment may be attractive with a payback period inferior to 20 years. Nevertheless, both the remarkable uncertainty over the capital costs figures and the fact that the analysis does not take into account the operation costs associated with LAES, may significantly affect the economic feasibility of the new configuration.

Chapter 4 ³

New parametric performance maps for a novel sizing and selection methodology of a Liquid Air Energy Storage system

Considering the complexity of the Liquid Air Energy Storage system, composed by three different phases (charge, discharge and storage), thermodynamic modelling could be a challenging undertaking. Making use of the strong similitude with gas turbine technology, this chapter aims to deliver new generalized performance maps for Liquid Air Energy Storage system. The performance maps, validated against the experimental results of Highview Power pilot plant, have been modelled by means of a comprehensive sensitivity analysis carried out considering three macro-scenarios imposing the storage pressures and the turbomachinery performance (design/off-design conditions). By means of the performance maps, the impact of the main LAES operative parameters, as well as the effect of the cold/warm thermal energy storage utilization factor, over the key performance indicators has been assessed and analysed.

³ This section published substantially as:

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- 2) Mazzoni S, Ooi S, Tafone A, Borri E, Comodi G, Romagnoli A. *Liquid Air Energy Storage as a polygeneration system to solve the unit commitment and economic dispatch problems in micro-grids applications*. Energy Procedia 2019;158:5026–33.

4.1 Introduction

From the literature review reported in Chapter 2, the previous works on LAES mainly focus on thermodynamic analysis and optimization based on complex numerical models and algorithms. Based on those, LAES has been designed and its main key performance indices, such as the liquefaction specific consumption and the round trip efficiency, derived. To the authors' best knowledge, there is not a generalized and systematic method that has been developed for researchers or engineers in order to design and calibrate LAES system.

By developing a LAES plant model by means of Aspen Hysys, the current study aims to propose a novel and general methodology to LAES system (plant based) design by means of dedicated performance maps. The intention of these maps allows asserting the optimum design and operating parameters for the LAES making use of a more systematic and immediate methodology. Each map is generated conducting a focused sensitivity analysis carried out on the main operative parameters (charge and discharge pressure, storage pressure, turbomachinery isentropic efficiencies, waste heat and cold potential) in order to produce a relevant amount of data encompassing a wider range of LAES real operation. The above-mentioned maps could be a helpful user-friendly tool for handling LAES design and operation calculations - easy to be used - and addressed to energy storage experts, who can simply look up the maps to design and calibrate the size of LAES system and operational conditions without entering in the more complex approach based on detailed modeling and computing.

4.2 LAES model implemented

This section describes the modeling of the LAES system utilized to obtain the data to develop the parametric performance maps. The charge and discharge section of the system are described and, successively, the simulation assumptions are presented including the operative range of the system used for the parametric analysis.

4.2.1 Charge and discharge phase

Figure 4-1 shows the process flow diagram of the LAES system implemented in the software for the numerical simulation. The adopted configuration is the same described in Chapter 3 based on the Kapitza thermodynamic cycle and direct expansion process for charge and discharge phases, respectively. The Kapitza cycle consists of two stages of compression, aftercooling, one intercooling stage (IC1), a recuperative heat exchanger (Cold box), an expander (CryoTurbine), a J-T valve, a phase separator and liquid reservoir (LA tank). The system operates as follows: air to be liquefied is firstly compressed in two stages; the high pressure air is then cooled down in the recuperative heat transfer device by two different flows: the former is the return low pressure air vapor stream and it is expanded in the J-T valve; the latter is the heat transfer fluid used in the HGCS. A fraction of the high pressure air stream is split before the cold box outlet through the CryoTurbine (CT) and sent to the phase separator. In that way, the expansion process leads to a large temperature reduction of the air stream. The liquid and vapour phases are separated in the phase separator: the not-liquefied air is used to cool down the high pressure stream in the recuperative process while the liquid air is stored in the liquid reservoir. The waste heat released by the compression phase is stored in the HGWS in order to make the waste heat available for the discharge phase for later use.

In order to extract cryogenic energy from liquid air a direct expansion process, not involving any external sub-cycles and/or working fluids, has been implemented. During the discharge phase, liquid air from the LA tank is pumped to high pressure through a cryogenic pump (CryoPump) and regasified to ambient temperature; the cold energy released during the regasification process is stored in the HGCS in order to make the waste cold available for the charge phase. The high pressure air will then be further heated up at the superheaters (SHs) by means of the waste heat stored in the HGWS. The high pressure and high temperature gaseous air is then re-heated in a 4 stages expansion process to achieve a quasi-isothermal expansion. The operating parameters of the LAES used for the sensitivity analysis (i.e. pressures of the charge, discharge and air storage and isentropic efficiencies of the main turbomachinery) are reported in Table 4-1.

Figure 4-1 Process flow diagram of the LAES implemented in the simulation.

4.2.2 Thermal energy storages: High Grade Cold-Warm Storages

In order to increase the round trip efficiency of the LAES system, configurations comprising both/either HGCS and/or HGWS have been analyzed. In the case of HGCS, the cycle efficiency is improved through “cold recycle”, an intermediate circuit that captures and stores the cold thermal energy released during the discharge phase in order to reduce the specific consumption of the liquefaction process. Aiming at recovering the waste heat flow discharged by the compression phase, the HGWS is used to reheat the air during the discharge phase in order to increase the turbine inlet temperature of the gaseous air and in turn the specific production of the whole LAES system.

In the present chapter, the thermal energy storages have been modeled by means of their efficiency and their utilization factor to figure in both the thermal performance and the presence of a potential external final user that requires a specific thermal load. As a consequence, eight different utilization factors (from 10% to 100%), namely the ratio between the effective thermal power recovered and the maximum available thermal power,

have been considered for LAES sensitivity analysis. The working fluids selected to transport waste heat and cold energy are Therminol 66 and air, respectively.

4.2.3 Operative parameters and Key Performance Indicators

The results of the simulations are presented in the next section with reference to the following operative parameters whose process flows are highlighted in the process flow diagrams (Figure 4-1) for charge and discharge phases:

- Charge pressure, p_{ch} [bar]: air pressure achieved in charge phase (3) immediately after the last stage of compression (C3);
- Recirculation fraction x_{RF} [-]: ratio of the mass flow elaborated by the Joule-Thomson valve ($\dot{m}_{JT}$ at point 5) and the mass flow entering the cold box ($\dot{m}_{air}$ at point 1);
- Storage pressure, p_S [bar]: pressure of liquid air inside the liquid air tank (point LA);
- Discharge pressure, p_d [bar]: liquid air pressure achieved in discharge phase immediately after the CryoPump (point 8);
- Turbine Inlet Temperature, TIT [°C]: temperature of air immediately after the superheating process (points 8SH-9SH-10SH-11SH);
- Utilization factors of thermal energy storages HGCS/HGWS [%]:

$$\eta_{HGCS} = \frac{\dot{Q}_{u,HGCS}}{\dot{Q}_{tot,HGCS}} \quad (21)$$

$$\eta_{HGWS} = \frac{\dot{Q}_{u,HGWS}}{\dot{Q}_{tot,HGWS}} \quad (22)$$

where $\dot{Q}_{u,HGCS}$ and $\dot{Q}_{u,HGWS}$ [kW_{th}] are the waste cold and waste heat power effectively utilized, respectively;

- Turbomachinery (compressors, CryoTurbine, cryoPump, expanders) isentropic efficiency η_{ISO} [%];

In order to provide the performance maps the following key performance parameters are defined:

Specific electric power output, SP [kW_e/kg_{LA}]:

$$SP = \frac{P_{net,d}}{\dot{m}_{LA}} = \frac{\sum_i^{n_{exp}} P_{i,d} - P_{CP,d}}{\dot{m}_{LA}} \quad (23)$$

Liquefaction specific consumption, SC [kWh_e/kg_{LA}]:

$$SC = \frac{P_{net,ch}}{\dot{m}_{LA}} = \frac{\sum_i^{n_c} P_{i,ch} - P_{CT,ch}}{\dot{m}_{LA}} \quad (24)$$

Round trip efficiency, η_{RT} [%]:

$$\eta_{RT} = \frac{E_{net,d}}{E_{net,ch}} = \frac{P_{net,d}}{P_{net,ch} \cdot (\tau_{ch}/\tau_d)} = \frac{\sum_i^{n_{exp}} P_{i,d} - P_{CP,d}}{(\sum_i^{n_c} P_{i,ch} - P_{CT,ch}) \cdot (\tau_{ch}/\tau_d)} \quad (25)$$

where $P_{net,ch}$ [kW_e] is the net electric power consumed during the LAES charge phase, n_c and n_{exp} are the number of compression and expansion stages, respectively, $P_{CT,ch}$ [kW_e] is the electric power produced by the CryoTurbine, $\dot{m}_{LA}$ [kg/h] is the liquid air production at the end of charge phase, $P_{net,d}$ [kW_e] is the net electric power produced by the power turbines, $P_{CP,d}$ [kW_e] is the electric power consumed by the CryoPump, $\dot{Q}_{tot,HGCS}$ [kW_{th}] and $\dot{Q}_{tot,HWCS}$ [kW_{th}] are the thermal power available at the inlet of HGCS and HGWS, respectively.

In Table 4-1, the main performance parameters are summarized in order to visualize the number of runs required to acquire the simulation data on which the performance maps are built up. Basically, three macro-scenarios are considered imposing the storage pressures (p_s) and the turbomachinery performance (design/off-design conditions):

1. design/ $p_s = 8$ bar: design conditions have been selected for the turbomachinery components and the storage pressure has been set to 8 bar;
2. design/ $p_s = 1.5$ bar: design conditions have been selected for the turbomachinery components and the storage pressure has been set to 1.5 bar;
3. off-design/ $p_s = 8$ bar: off-design conditions for the main turbomachinery components have been selected and the storage pressure has been set to 8 bar.

Table 4-1 Process parameters and their operative range for the LAES system under study.

Parameters	Value-Range	Unit	References
T_{amb} , Air inlet temperature	25	$^{\circ}C$	-
p_{ch} , Charge pressure	40-90	bar	[21,48]
x_{RF} , Recirculation fraction	0.10-0.55	-	[48]
p_d , Discharge pressure	60-160	bar	[14,44]
η_{HGCS} , HGCS utilization factor	10-100	%	-
η_{HGWS} , HGWS utilization factor	10-100	%	-
p_s , Storage pressure	1.5/8	bar	
ΔT_{CB} , Cold Box pinch point	3	$^{\circ}C$	-
ΔT_{IC} , Intercoolers pinch point	5	$^{\circ}C$	-
ΔT_{AFC} , Aftercoolers pinch point	5	$^{\circ}C$	-
ΔT_{SH} , Superheater hot end temperature approach	10	$^{\circ}C$	-
$\eta_{iso,c}$, Isentropic efficiency of compressors	85/68	%	[8,21]
$\eta_{iso,CT}$, Isentropic efficiency of CryoTurbine	70/56	%	[8,21]
$\eta_{iso,CP}$, Isentropic efficiency of CryoPump	80/64	%	[8,21]
$\eta_{iso,T}$, Isentropic efficiency of power Turbines	80/64	%	[8,21]

The off-design isentropic efficiencies are obtained lowering the design values by 20 %. Then, eight different levels of thermal energy storage utilization factor are analyzed and for each of those, eleven discharge pressures are considered. Finally, for each discharge pressure, eleven charge pressures are considered; in addition, for each charge pressure, in order to identify the value of recirculation fraction that minimize the specific consumption, 5 different recirculation fraction x_{RF} have been employed for a total of 14520 runs. Once the data have been acquired, the performance maps of LAES have been elaborated by means of Matlab Curve Fitting Tool (cftool) [100]. The Curve Fitting app provides a flexible interface which allows of interactively fitting curves and surfaces to data and view plots; the linear interpolation approach has been selected.

Figure 4-2 The flow chart of the methodology procedure applied for the performance maps elaboration.

4.3 Performance maps elaboration and validation

This section presents the simulation results of the sensitivity analysis carried out for the LAES system modelled in this analysis. As stated in previous sections, the results are shown by means of different performance maps in order to visualize the effects of the main operative parameters over the key performance indicators; a total liquid air production of 1 ton/h has been considered as the reference for LAES. The intention of these maps is to allow identifying the optimum design and operating parameters for the LAES in a more systematic and immediate way. For each of the proposed charts, the operating conditions described in Section 4.2.3 have been considered; by varying some of these parameters the charts are shifted and this is also to be discussed in the work. Along with the thermodynamic analysis, the analytic equations associated with the thermodynamic processes in both charge and discharge phases have been developed in order to provide an alternative and simplified way to achieve and validate the results obtained by means of Aspen Hysys simulations. A model validation has been carried out against experimental results achieved at the LAES pilot plant located at the University of Birmingham and finally the tool

potential has been shown by means of real case scenarios in order to immediately show its applicability. The maps have been also showed in their original format in Appendix B in order to be easily consulted and utilized.

4.3.1 Effect of charge pressure and waste cold power on the liquefaction specific consumption

The twofold effect of the charge pressure and waste cold recovery over the specific consumption is introduced with the first performance map (Figure 4-3). The dashed lines represent constant and optimum values of the recirculation fraction while continuous lines represent constant specific consumption curves. The graph shows that the higher is the value of the HGCS utilization factor, namely the waste cold thermal power recycled, the lower is the positive impact of charge pressure over the specific consumption. For high value of cold thermal energy storage efficiency, an optimum value of the charge pressure is approximately in the range of 65-85 bar. Indeed, the map confirms what is already stated in literature and reported in Chapter 2: an optimally designed cold thermal energy storage is fundamental for ensuring the lowest values of specific consumption and the highest round trip efficiency, leading the LAES to be a viable techno-economic solution for electric energy storage.

Figure 4-3 Effect of charge pressure and waste cold recovery efficiency on specific consumption for different optimum values of recirculation fraction (design $-p_s = 8$ bar).

By adopting the energy conservation equation on the control volume, defined by the cold box and the liquid air storage (Figure 4-4) and recalling the formula in Eq. (24), the specific consumption can be expressed as a function of both charge pressure and recirculation fraction:

$$SC = \frac{y \cdot c_{p,ave,1} \cdot T_{1S} \cdot \left(\alpha_{c1}^{\frac{1}{\eta_{pol,c}}} - 1 \right) + 2 \cdot c_{p,ave,2} \cdot T_1 \cdot \left(\alpha_{c2}^{\frac{1}{\eta_{pol,c}}} - 1 \right) - (1 - x_{RF}) \cdot (h_4 - h_{4CT})}{y} \quad (26)$$

$$\alpha_{c1} = (\beta_1)^{\frac{k-1}{k}} \quad (27)$$

$$\beta_{c1} = \left(\frac{p_s}{p_{amb}} \right) \quad (28)$$

$$\alpha_{c2} = (\beta_{c2})^{\frac{k-1}{k}} \quad (29)$$

$$\beta_{c2} = \left(\frac{p_{ch}}{p_s} \right)^{\frac{1}{n_c}} \quad (30)$$

where $\eta_{pol,c}$ is the polytropic efficiency of the compression process, β_c is the compression ratio, $c_{p,ave}$ [kJ/kgK] is the average isobaric specific heat of air and k is the average specific heat ratio of air.

Figure 4-4 Energy balance in the charge phase over the green control volume.

The liquid yield y and the relation between the isentropic efficiency and the polytropic efficiency can be expressed by the following formulae:

$$y = \frac{\dot{m}_{LA}}{\dot{m}_{3AFC}} = \frac{h_7 - h_{3AFC} + \left(\frac{\dot{Q}_{tot,HGCS}}{\dot{m}_{LA}} \right) \cdot \eta_{HGCS}}{h_7 - h_{LA}} \quad (31)$$

$$\eta_{iso,c} = \frac{\alpha_c - 1}{\frac{1}{\alpha_c \eta_{poli,c}} - 1} \quad (32)$$

The effect of discharge pressure over the maximum available cold thermal power available at the inlet of HGCS is shown in Figure 4-5. The graph confirms the linear dependence between those variables expressed by:

$$\Delta h_{tot,HGCS} = \frac{\dot{Q}_{tot,HGCS}}{\dot{m}_{LA}} = (h_9 - h_{LA} - \Delta h_{CP}) \quad (33)$$

$$\Delta h_{CP} = \frac{(p_8 - p_{LA}) \cdot v_{LA}}{\eta_{iso,CP}} \quad (34)$$

where v_{LA} [m³/kg] is the specific volume of liquid air at storage pressure at point LA.

Figure 4-5 Maximum available cold thermal power as a function of discharge pressure (design $-p_s = 8$ bar).

4.3.2 Charge pressure-TIT relation

Figure 4-6 shows the effect of the charge pressure on the TIT for different HGWS utilization factors whose constant values are represented by continuous lines. At low η_{HGWS} values, the TIT variation is limited for the range of charge pressure considered, while at higher HGWS efficiency the TIT spans over a wide range values as the charge pressure changes. A maximum TIT of 187 °C, directly available during the discharge phase at the inlet of any power turbines, is achieved at the upper end of both operational parameters, namely a charge pressure of 90 bar and an HGCS utilization factor equal to 100 %. The analytic relation between the charge pressure and the TIT is provided as follows:

$$TIT = T_{8H} + \frac{\dot{Q}_{tot,HGWS} \cdot \eta_{HGWS}}{\dot{m}_{DOWQ} \cdot c_{DOWQ}} - \Delta T_{SH} \quad (35)$$

$$\dot{Q}_{tot,HGWS} = \sum_i^{n_c} \dot{Q}_{i,HGWS} = \sum_i^{n_c} \dot{m}_{in,i} \cdot c_{p,ave,i} \cdot \left[\left(T_{in,i} \cdot \alpha_i^{\frac{1}{\eta_{pol,i,i}}} \right) - T_{IC,i} \right] \quad (36)$$

where $\dot{m}_{in}$ and T_{in} are the mass flow rate and the temperature of air at the inlet of the i -th compression stage, T_{IC} is the temperature at the outlet of the intercooling/aftercooling process (points 3S, 2IC and 3AFC), $\dot{m}_{DOWQ}$ and c_{DOWQ} are the mass flow rate and the specific heat capacity of Dowtherm Q, respectively.

Figure 4-6 Effect of charge pressure and waste heat recovery on the turbine inlet temperature (design $-p_s = 8$ bar).

4.3.3 Effect of Turbine Inlet Temperature on Specific Electric Power output

The combined effect of the discharge pressure and TIT over the specific electric power output of LAES is presented in Figure 4-7. An increase of both parameters positively contributes to the specific electric power output increase due to the following well-established correlations based on gas turbines technology:

$$P_{net,d} = n_{exp} \cdot \dot{m}_{LA} \cdot cp_{ave,air} \cdot TIT \cdot \left(1 - \frac{1}{\alpha_{ex}^{eta_{poli,exp}}} \right) \quad (37)$$

$$\eta_{iso,exp} = \frac{\frac{1}{\alpha_{ex}^{\eta_{poli,exp}}} - 1}{\frac{1}{\alpha_{exp}} - 1} \quad (38)$$

$$\alpha_{exp} = (\beta_{exp})^{\frac{k-1}{k}} \quad (39)$$

$$\beta_{exp} = \left(\frac{p_d}{p_{amb}} \right)^{\frac{1}{n_{exp}}} \quad (40)$$

where $\eta_{poli,exp}$ is the polytropic efficiency of the expansion process, β_{exp} is the expansion ratio, n_{exp} is the number of the expansion processes (in this specific case corresponding to 4) and p_{amb} [bar] is the ambient pressure. It is worth noting that the assumptions of constant expansion ratio for all the stages and polytropic expansion hold.

Figure 4-7 Effect of discharge pressure and Turbine Inlet Temperature on the specific electric power output for different storage pressures and isentropic efficiencies (design $-p_s = 8$ bar).

4.3.4 Effect of the isentropic efficiencies of the main turbomachinery

In Figure 4-8-Figure 4-10 the performance maps of LAES are plotted for off-design condition of the main turbomachinery (compressors, CryoTurbine, CryoPump, power turbines) to analyze the effect of this parameter over the main performance indicators. The isentropic efficiencies of those components have been lowered by 20 % of their design value.

Figure 4-8 Effect of charge pressure and waste cold recovery efficiency on specific consumption for different optimum values of recirculation fraction (off-design $-p_s = 8$ bar).

Figure 4-9 Effect of charge pressure and waste heat recovery on the turbine inlet temperature (off-design $-p_s = 8$ bar).

Figure 4-10 Effect of discharge pressure and Turbine Inlet Temperature on specific electric power output for different storage pressures and isentropic efficiencies (off-design $-p_s = 8$ bar).

Figure 4-8 shows that as the isentropic efficiency values decrease, the map for the specific consumption shifts towards higher values. Besides the shifting of the map, the change in the specific consumption values are quite significant as the charge pressure and the HGCS utilization factor vary; indeed for the 90% HGCS utilization factor case, the specific

consumption becomes less sensitive to the variation of the charge pressure (approximately constant specific consumption at 0.25 kWh_e/kg_{LA} and 0.33 kWh_e/kg_{LA} for design and off-design condition, respectively). The opposite trend occurs for the 10% HGCS utilization factor case with a SC in the range between 0.45 and 0.47 kWh_e/kg_{LA} for design conditions between 0.7 and 0.78 kWh_e/kg_{LA}. This can be explained by considering that in the design scenario, in which the compressors and the CryoTurbine achieve an isentropic efficiency of 85% and 70%, respectively, the positive impact of those turbomachinery performances overcomes the potential inefficiencies due to a not optimal charge pressure. Instead, for the off-design scenario, the negative effect of lower isentropic efficiency of the main turbomachinery on the SC is amplified by the choice of the charge pressure. The higher the performances of compressors and cryogenic turbine are, the more flexible the LAES operation in terms of charge pressure is.

Comparing the map reported in Figure 4-9 with that obtained for the design scenario (Figure 4-6), for any values of HGWS utilization factor a higher TIT is obtained with a maximum achieved at 225°C. According to Eq. (41), the lower efficiency of the compression phase leads to higher waste heat temperatures and to a significant increase of the TIT:

$$T_{out,i} = \left(T_{in,i} \cdot \alpha_i^{\frac{1}{\eta_{poli,c,i}}} \right) \quad (41)$$

where T_{out} is the temperature of the fluid at the outlet of the i -th compression stage. As a consequence, the negative impact of the higher specific consumption, due to lower isentropic efficiency of compressors, partially offsets the positive effect on the higher inlet enthalpy values for the power turbine. Nevertheless, similar to Figure 4-8, as the isentropic efficiency of the power turbines decreases, the map shown in Figure 4-10 shifts towards lower values of the specific electric power output due to the dominant effect of the lower power turbines isentropic efficiency.

4.3.5 Effect of storage pressure on specific consumption

In Figure 4-11 the performance map of LAES related to the effect of charge pressure and HGCS utilization factor over the SC is plotted for a different storage pressure (1.5 bar) keeping a design value of the isentropic efficiencies of compressors, CryoTurbine, CryoPump and power turbines.

Figure 4-11 Effect of storage pressure on liquefaction specific consumption (design - $p_s = 1.5$ bar).

As already analyzed in Chapter 2, the storage pressure has a significant effect on the specific consumption. This trend could be explained by considering that the higher the pressure of the returning cold flow the higher is the heat capacity. This effect is beneficial for the effectiveness of the heat exchange in the Cold box. Comparing Figure 4-11 and Figure 4-3, the magnitude seems to be dependent on the different levels of HGCS utilization factor: at low η_{HGCS} (10%), the specific consumption can be reduced by 26 % while at higher η_{HGCS} the relative percentage decreases until reaching its minimum at $\eta_{HGCS} = 100$ % (9 %). As for the charge pressure and turbomachinery isentropic efficiencies, the higher the HGCS utilization factor is, the lower the positive impact of the storage pressure over the specific consumption is.

4.3.6 Round trip efficiency evaluation

Once the charge and discharge pressure and the utilization factor of HGWS and HGCS are defined, the TIT, the SC and the specific electric power output have been extrapolated from the performance maps shown in the previous sections. As a consequence, the round trip efficiency is computed as a function of both the specific consumption and the specific electric power output:

$$SP = \eta_{RT} \cdot SC \quad (42)$$

Such a relation, graphically represented in Figure 4-12, allows to finally evaluating the potential of LAES in terms of round trip efficiency.

Figure 4-12 Round trip efficiency as a function of specific electric power output and liquefaction specific consumption.

4.3.7 Maps validation

The availability of experimental data for LAES plant is only restricted to those obtained at the pilot plant operated in Slough (U.K.) by Highview Power but actually located at University of Birmingham. According to Morgan et al. [27], the main parameters of LAES

pilot plant are summarized in Table 4-2. Mainly due to the lower quantity of the maximum cold recycled (50%), the round trip efficiency drops to 8 % with a specific consumption higher than 0.6 kWh_e/kg_{LA}.

By adopting the operative parameters of the pilot plant in our model, the following outcomes are presented and underlined:

- since the maximum charge pressure value (12 bar) is outside the optimal boundaries studied, based on an HGCS utilization factor of 50 %, the specific consumption is computed extrapolating the curves in Figure 4-6 beyond the limit of 40 bar. By means of such a method, the calculated specific consumption is equal to 0.64 kWh_e/kg_{LA};
- based on a TIT of 64 °C, the calculated SP is equal to 0.071 kWh_e/kg_{LA};
- combining the previous results, the calculated round trip efficiency is about 10.5 % with a relative percentage difference compared to the efficiency of the pilot plant equal to 23 %.

Table 4-2 Process and performance parameters for LAES pilot plant.

Parameters	Value-Range	Unit
Charge pressure	12	<i>bar</i>
Discharge pressure	56	<i>bar</i>
η_{HGCS}	50	%
Storage pressure	8-10 bar	<i>bar</i>
TIT	64	°C
Isentropic efficiency of axial compressors	89	%
Isentropic efficiency of CryoTurbine	70	%
Isentropic efficiency of CryoPump	80	%
Isentropic efficiency of radial power turbines	90	%
SC	0.60	-
SP	0.05	-
η_{RT}	8.3	%

The percentage deviation among the calculated and the experimental data are mainly due to the following differences among the model proposed and the pilot plant:

- waste heat is provided by an external heat source at 60 °C;
- maximum charge pressure is subcritical (12 bar);
- maximum discharge pressure is set at 56 bar;

- pressure losses lower the inlet pressure of the first stage of power turbine leading in turn to a smaller enthalpy drop and a consequent lower specific electric power output.

Considering the low values of both estimated and experimental round trip efficiencies and the approximations due to the model developed in Aspen Hysys and the interpolation of the results, it can be inferred that the proposed methodology offers a valid option for preliminary selection of LAES systems.

4.4 Application of the results

4.4.1 Full electric and polygeneration LAES configurations

In order to highlight the immediate applicability of the results, two different case studies corresponding to two different LAES configurations have been provided and assessed.

The first case study is related to the full electric configuration, namely when LAES is operated only for electric power production to the electric grid. A round trip efficiency of 40 % and a liquefaction specific consumption of 0.25 kWh_e/kg_{LA} have been assumed as the main outputs requested by a potential customer. As shown in Figure 4-13a, once the LAES round trip efficiency and specific consumption SC are defined, a SP of 0.09 kWh_e/kg_{LA} is computed. Assuming a thermal efficiency of 87 % and 90 % for the HGCS and the HGWS, respectively, both charge and discharge pressure are derived from Figure 4-13b and Figure 4-13d with a TIT of 152 °C.

The second case has addressed the potential of LAES operating in polygeneration configuration; both an electric power output and a cooling power are available for the electric grid and district cooling system, respectively. The cooling output is provided by the direct expansion of gaseous air; as a consequence, the turbine inlet temperature of gaseous air is constrained (90 °C) by a defined turbine outlet temperature (5°C) which is required by the district cooling system. Assuming a lower round trip efficiency (30 %) and an slightly higher specific consumption (0.3 kWh_e/kg_{LA}), as shown in Figure 4-14, the same

procedure applied to full electric configuration could be followed for the cogenerative configuration in order to derive the main operative parameters.

It needs to be remarked that the maps based method proposed here, could easily be bypassed since the proposed work, also offers all the key-analytical correlations which have been used to generate the maps. Hence this means that an end-user of such a methodology, could directly calculate the performance parameters by applying the design input values/constraints. The advantage of the proposed maps is that they offer the possibility to assess different options and operating conditions depending on how some key input performance parameters (e.g. amount of waste heat/cold recovered, turbine inlet temperature and so on) vary.

Figure 4-13 Full electric configuration: graphical method to derive the main operating parameters using the performance maps.

Figure 4-14 Polygeneration configuration: graphical method to derive the main operative parameters using the performance maps.

4.4.2 LAES as a polygeneration system to solve the economic dispatch problems in micro-grids applications

This section illustrates another potential use of the performance maps that have been used to design LAES into micro-grid context for economic dispatch purpose. Specifically, this case scenario compares the adoption of an Electrochemical Energy Storage (Li-Ion batteries) and LAES as part of a polygeneration system, which includes a cogeneration plant (reciprocating internal combustion engine and absorption chiller), solar PVs and vapour compression chillers, aimed at satisfying the cooling and electrical load of an industrial building located in Singapore. Due to the hot and humid climate, there is no demand/need for heating and the focus is mainly focused on the cooling side. The Smart Multi Energy System (SMES) project national Singaporean project for demonstrating the capabilities of Unit Commitment Problem (UCP) and Optimal Dispatch Problem (ODP) solving, a well-referenced building estate has been taken into consideration. The demonstration test case refers to the Clean Tech Park (CTP), in the west district of Singapore; the CTP consists of three buildings for office use. The CTP primary energy consumption is set to satisfy the electricity demands for lighting, chillers and other building requirements. The electricity and cooling demands of the CleanTech One (CTO), CTP main building, are known and have been taken as a reference case for this study.

Figure 4-15 Proposed arrangement for the polygeneration plant equipped with energy storage

In order to satisfy the cooling and electric load (i.e. chilled water and electric work) during a typical working day (24hours, 48 intervals of 30 minutes each), in the present work the configuration shown in Figure 4-15 has been proposed. On the electricity bus, a gas engine, solar PVs (which size is related to the rooftop surface availability), Li-Ion battery or LAES have been considered to satisfy the generation and demands of electric power. On the cooling bus, by means of the proposed gas engine cogeneration arrangement, the cooling demand is satisfied by an absorption chiller, a vapour compression chiller and, when considered, by the cold energy made available by the LAES discharging process. The primary energy sources for running the whole system are highlighted by red arrows representing the fuel mass flow rate, the electric power purchased from the grid and the power consumed for running the vapour compression chiller. Details on the modelling approach coupled with the proposed solution strategy are given in the following section.

4.4.2.1 Modelling and Optimization Method

For solving the UCP and ODP, the modelling of the polygeneration plant is based on a modular approach that consists in matching the elementary components (i.e. gas engine, solar PVs, Li-Ion, LAES and vapour compression chillers) for achieving the whole polygeneration plant simulator. The modelling approach takes into account steady state 0-D component models. For each component the conservation equations of mass, momentum, energy and entropy, constitutive and the auxiliary equations are stated.

The component models are based on lumped performance feature discretization approach, in which boundary surfaces and central nodes are adopted, as described in [101]. The quadratic programming technique has been adopted and coupled with a mixed integer solver (compared with the adoption of genetic algorithms) for ensuring reduced computational costs and robustness of the solution. The adopted approach has been presented by many authors who have proven the benefit of the proposed MIQP programming technique [102]. In this analysis, details on the modelling of the LAES will be provided in the next section together with the optimization procedure for solving the ODP. The other modelled component models such as gas engine, PV, chillers are modelled taking into account off-design maps that correlate the load at which each component is

operated in respect of the nominal values and the performance of the component itself (i.e. efficiency of the gas engine and coefficient of performance of the chillers).

Liquid Air Energy Storage model. The LAES component model has been developed taking into consideration the operating parameters such as the Round Trip Efficiency, the liquefaction Specific Consumption (SC) and the cold energy utilization ratio. Such parameters are helpful because they allow to characterize the LAES performance for the given power/energy of charge (P_L^-). The work carried out previously on LAES modeling by parametric performance maps allows to compute by means of global correlations the global quantities such as recoverable cooling (CP_{LAES}) and heating power ($\dot{Q}_{H,LAES}$), LAES storage capacity (V_{LAES}) and generated power. The LAES capacity (expressed by its storage volume) is established by Eq. (43), with ρ_{air} as the liquid air density.

$$V_{LAES} = \frac{P_{LAES} \cdot \Delta t}{SC \cdot \rho_{air}} \quad (43)$$

By means of energy conservation equations it is possible to establish the amount of cold energy generated and the electric work produced by the turbine during the discharge phase as expressed by the functional correlation given in Eq. (44), being COP_{ABS}^{LT} the low temperature loop absorption chiller coefficient of performance.

$$f(P_L^-, P_L^+, \eta_{RT}, SC, CP_{LAES}, \dot{Q}_{H,LAES}, V_{LAES}, COP_{ABS}^{LT}) \quad (44)$$

Adoption of inequality constraints for checking that the LAES volume – at the instant $t+1$ - is in the range between the minimum volume and the maximum volume has been introduced Eq. (45).

$$V_{LAES}^{min} \leq V_{LAES}(t+1) \leq V_{LAES}^{MAX} \quad (45)$$

Taking the LAES capacity expressed in kWh or in m^3 into account, during the ODP solving the evaluation of the LAES capacity has to be performed for ensuring the feasibility of the

numerical solution and the capability of the system of storage energy during the off-peak operations and release it during peak ones. The LAES capacity (similar approach of the state of charge in Li-Ion) at the next interval ($t+1$) is established by Eq. (46) and it is done during the 48 intervals of the day operation.

$$V_{LAES}(t+1) = V_{LAES}(t) + \frac{P_L^- \cdot \Delta t}{SC \cdot \rho_{air}} - \frac{P_L^+ \cdot \Delta t}{\eta_{RT} \cdot SC \cdot \rho_{air}} \quad (46)$$

The binary variable (the mixed integer one 1/0), representing a logic operator, ensures that during each Δt interval the LAES system can only be in charge, storage or discharge mode.

Objective function definition and constraints structure. The solution of an ODP consists of two main steps such as the minimization or maximization of the objective function (ObF) and the satisfaction of the equality constraints, namely power flows (electricity and cooling bus load demands). From a numerical perspective, the adopted solver is based on simultaneous solutions; this means that concurrently to the equality constraints satisfactory also the ObF is optimized. In the current work, the ObF to be maximized has been set to be the Net Present Value (NPV), expressed by Eq. (47).

$$ObF - Search \ MAX \ of : \ NPV = \sum_{k=1}^N \frac{CF}{(1+i)^k} - CAPEX \quad (47)$$

being CAPEX the overall polygeneration plant capital expenditure, expressed as the sum of the various components investment costs, and the Cash Flow (CF) defined as the difference - integrated over the year - between the cost of the generation of the proposed polygeneration system (cost of fuel plus electricity) versus the cost of the generation in the case that all the electricity (also used for feeding the vapour compression chiller for cooling power generation) is purchased at the CTO contracted electric price p_G from the Singaporean national grid, as given in Eq. (48).

$$CF = \int m_F(t) \cdot p_F(t) \cdot dt + \int P_G(t) \cdot p_G(t) \cdot dt - \int P_G^R(t) \cdot p_G(t) \cdot dt \quad (48)$$

where m_F [kg] and p_F [S\$/kg] are the mass and the price of the fuel consumed, respectively.

The satisfaction of the energy flows (operational constraints) for the ODP both on the electric and cooling buses is expressed by Eqs. (49-50), respectively.

$$P_{EL} \cdot \Delta t = P_{ENG} \cdot \Delta t + P_{PV} \cdot \Delta t + P_G^+ \cdot \Delta t + P_L^+ \cdot \Delta t - P_L^- \cdot \Delta t \quad (49)$$

$$CP \cdot \Delta t = CP_{ABS} \cdot \Delta t + CP_{VCCH} \cdot \Delta t + CP_{LAES} \cdot \Delta t \quad (50)$$

Under this conditions the ODP has been fully stated and in the next section the test case and the analysis of the results is presented.

4.4.2.2 Results and discussion

The capability of the proposed polygeneration system has been explored for both the cases in which either LAES or Li-Ion is being considered. As stated in the previous sections, the electric and cooling power profile have been set as constraints to be satisfied, with the CTO consumptions known. The assessment has been carried out taking into consideration the following component specifications, in terms of sizes and costs. The internal combustion engine is a 1MW_e gas engine; the waste heat is recovered through an absorption chiller generating 1.2 MW_e of cooling power. The capex of the cogeneration plant has been established by factorized methods and estimated to be equal to 1.6MS\$. The solar PV surface is of 2000 m² and the corresponding nominal power is of 230 kW_e for an investment cost of 0.6 MS\$. The vapour compression chiller shows a nominal plate cooling power of 2000 kW_e and a capital cost has not been included because it is already installed at CTO. The sensitivity analysis has been performed varying the LAES and Li-Ion capacity in the range 300 kWh to 2000kWh. For these storage capacities, the specific cost of LAES is of 320S\$/kWh [79] while for Li-Ion is 560 S\$/kWh [103]. The evaluations have been performed taking a typical day profile of electricity consumption into consideration (Singapore does not have huge seasonal weather changes) and assuming an interest rate i

of 6% and a lifetime of the power plant of 20 years. The price of the fuel, on the basis of the natural gas trade price, has been assumed to be 0.45 S\$/kg for a low heating value of 48MJ/kg.

The optimized solution of the ODP is reported in Figure 4-16. The electric load (continuous red line on the left) and the cooling load (dashed blue lines on the right) are data available and they have been set as constraints for solving the ODP. The black line - in both the charts - represents the price of the electricity purchased from the national grid. With the fuel cost assumed to be constant (as in the typical case of take or pay contract for the natural gas), the p_G plays a key role in the optimal control strategy definition, as clearly shown in the Figure 4-16. Accordingly, during night hours, when the price of the electricity is minimum, the gas engine is turned off and the whole amount of the electric load is satisfied by the electric work purchased by the grid (red area on the left). As a consequence, the cooling load demand is fully satisfied by running the vapour compression chiller, being the absorption chiller turned off as well. The control strategy of the whole systems manages to avoid purchasing the electricity from the grid during the peak hours (13.00 to 15.00) thus the LAES is charged in the first hours of the morning, when also the solar PVs contribute to generate electric work (yellow area) for matching the electricity demand. Indeed, it could be seen that the green bars (representing the level of the LAES storage capacity, similarly to the state of charge of the Li-Ion) increase during the relatively low p_G hours and decrease until the minimum allowed value Eq. (45) in which the electricity demand is high. It should be remarked that the LAES is charged before the spike on the cooling demands that takes place between 08.30 and 10.00 in the morning. A small contribution to the cooling bus is given by the LAES (orange area on the right) during the discharge phase, when some cold energy is released. In Figure 4-17, the ODP solved for the polygeneration system equipped with Li-Ion is presented. As a consequence of the Li-Ion having a higher round trip efficiency than the LAES, the Li-Ion are used also during the night hours. Li-Ion is fully charged by the cogeneration till 01.30 and then, it is used for reducing the amount of electric work purchased by the grid during the night hours, when the cogeneration is turned off because of the reduction of the cooling demand. Also during day time (16.00-17.30), the adoption of Li-Ion allows to reduce the electric work purchased from the grid, as shown by

the comparison of the Figure 4-16 and Figure 4-17 (left charts). In both the scenario, with Li-Ion and LAES, the control strategy substantially searches for the minimum operational cost along the reference period of operation.

For understanding the techno-economic feasibility of the two solutions, the cost benefit analysis of the proposed layout, both equipped with Li-Ion (continuous line) and LAES (dashed line), has been carried out. Results of this analysis are summarized in Figure 4-18. The NPV and ROI for the two energy storage technologies and for two different storage capacities 300kWh (red lines) and 2000kWh (blue lines) have been presented. For the smaller capacity, the ROI of the two solution is practically the same, about 7 years. For the 300kWh storage capacity, a NPV of 2.3 MS\$ is achieved. Increasing the capacity of the storage systems, the weight of CAPEX of the Li-Ion becomes more significant in the overall capital expenditure of the polygeneration plant if compared with the LAES plant layout. As a consequence, the ROI of 2000kWh Li-Ion is about 11 years, while the LAES ROI is of 9 years. NPVs in this configuration are lower. It is worthy of note that typical lifetime of Li-Ion is less than 10 years. It means that in the 300 kWh case, after 20 years, the Li-Ion NPV should be lower because of the need to replace the battery component in the 10th year.

Figure 4-16 300kWh LAES arrangement - Optimal Dispatch (electric Load – Left) – (Cooling Load – Right)

Figure 4-17 300kWh Li-Ion arrangement - Optimal Dispatch (electric Load – Left) – (Cooling Load – Right)

Figure 4-18 Net Present Values and ROI for Li-Ion EES and LAES capacities of 300kWh and 2000kWh.

4.5 Summary

In this chapter, a comprehensive analysis of the main operative variables on the performance of LAES with steady state simulations has been carried out. The motivation behind the proposed study is due to the lack of systematic and methodologic analysis of LAES. As a general observation, LAES performance maps serve as a user-friendly and unique reference tool to select different operative parameters to achieve a desired level of LAES performance in term of specific consumption, specific electric power output and round trip efficiency.

More in particular, the following conclusions can be drawn from the analysis of the main results:

- by means of the novel approach proposed, eight different LAES performance maps have been built up and analyzed. Each map presents the combined effect of the main operative parameters over the defined key performance indicators;
- as a general observation, the higher is the HGCS utilization factor, the lower will be the effect of the charge pressure over the liquefaction specific consumption. Indeed, it has been shown that, for the thermodynamic process modeled, the charge pressure plays a negligible impact on specific consumption compared to the amount of waste cold recovered during liquid air regasification;
- lowering the isentropic efficiencies values of the main turbomachinery produces a general shift of the performance maps towards higher values of liquefaction specific consumption and therefore lower round trip efficiency. As long as the HGWS utilization factor is kept at higher values, the round trip efficiency decrease is partially offset by the higher Turbine Inlet Temperature available for the expansion process of the discharge phase. However, the higher are the performances of compressors and cryogenic turbine, the more flexible is the LAES operation;
- comparing the round trip efficiency of the LAES pilot plant operated in Slough (UK) by Highview Power with the calculated value obtained by the performance maps, a percentage relative difference of 25 % is revealed. The relatively large gap between the computed and the experimental data is principally due to the suboptimal charging and discharging pressure used for the pilot plant operation, beyond the range assumed in the present work;
- the liquefaction specific consumption is significantly affected by the storage pressure with a decrease up to 26 %. As the HGCS utilization factor increases, the advantage of higher storage pressure is sensibly lower with a relative decrease of 9 % for full exploitation of the waste cold discharged by the liquid air regasification ($\eta_{\text{HGCS}} = 100 \%$);

- the maps represent unique guidelines for LAES design under operative parameters variation and serves as a systematic tool for the design of LAES operating in different configurations (full electric and polygeneration);
- by adopting the maps as a tool for LAES design in polygeneration configuration, an economic case study has shown that the adoption of a 300 kWh capacity LAES for the economic dispatch of an Eco-building in Singapore produces a higher Net Present Value after 20 years and a shorter time period to obtain the Return of Investment compared to that of Li-ion battery.

4.5.1 LAES Performance maps limitations

Although the performance maps show a good agreement with the experimental data of the LAES pilot plant and are capable to predict reliably the performance of the system, the following limitations should be underlined in order to further improve the quality of the maps.

Novel thermodynamic cycles. The novel parametric maps refers to a well-defined thermodynamic architecture for both charge (Kapitza cycle) and discharge phases (direct expansion process). In order to further develop the approach, different thermodynamic cycles may be considered.

Integration of external waste heat/cold source. Due to its thermo-mechanical nature, LAES is capable to be integrated with other valuable high exergy energy carriers (e.g. waste heat/cold from industrial process/ Liquefied Natural Gas regasification). This ability might be captured by a further development of the performance maps taking into account external waste heat/cold sources.

Methodology refinement. The parametric maps have been provided for specific values of the storage pressures (1.5/ 8 bar) and isentropic efficiencies of the turbomachinery components (design/ off-design conditions). As a consequence, new values of those parameters can be taken into account along with a necessary refinement of the methodology

by the implementation of the design of experiments technique.

Chapter 5 ⁴

Techno-economic study of Liquid Air Energy Storage integrated with Waste Heat Recovery Solutions

In this chapter, the potential of improving the round trip efficiency of Liquid Air Energy Storage was investigated through modelling and simulations using the numerical software EES-Engineering Equation Solver. As already shown in Chapter 3 and Chapter 4, Liquid Air Energy Storage performance is actually limited both by the inefficiencies of the charging (liquefaction cycle) and discharging (regasification and expansion) leading to a low value of round trip efficiency when compared to other energy storage solutions. In order to further improve the round trip efficiency, the opportunity to recover the waste heat released during the compression has been considered in this work. Different integrated energy systems consisting Organic Rankine Cycle and/or Absorption Chiller were compared against a stand-alone Liquid Air Energy Storage used as a baseline. The integrated systems are compared in terms of different performance indices both from technical and economic point of views.

⁴ This section published substantially as

1) Tafone A, Borri E, Comodi G, Broek M Van Den. *Liquid Air Energy Storage performance enhancement by means of Organic Rankine Cycle and Absorption Chiller*. Appl Energy 2018;228:1810–21.

2) Tafone A, Ding Y, Li Y, Chunping X, Romagnoli A. *Levelised Cost of Storage (LCOS) analysis of liquid air energy storage system integrated with Organic Rankine Cycle*. Energy, 2020, 117275.

5.1 Introduction

The main bottleneck to the deployment of LAES is represented by its low value of round trip efficiency which is mainly due to the large amount of energy consumption during the liquefaction cycle (charge phase) in which the air needs to be compressed at relatively high pressures in order to achieve adequate liquid air yield. By analyzing the performance of a microgrid scale air liquefaction plant for LAES, Borri et al. [104] linked the low exergy efficiency value achieved by the system with the total heat rejected to the environment during the charge phase. In their thermodynamic analysis of LAES, both Guizzi et al. [46] and Tafone et al. [105] have highlighted that, despite the presence of a heat storage section capable to partially recover the waste heat discharged during the compression phase, the major contribution to exergy losses is represented by heat rejection after air superheaters at the discharge phase of LAES.

Among the waste heat recovery solutions currently under analysis in heat-to-power conversion processes, ORC is a well-established technology with good reliability and relatively high efficiency as opposed to other solutions proposed in the literature [106]. In recent years, commercial application of ORCs has risen up worldwide with a total installed capacity of 376 MW_e [107]. Besides heat-to-power, another important application for waste heat can be found in heat-to-cool conversion processes in which absorption heat pumps are used in applications where heating and cooling is required [108]. In heat-to-cool applications, absorption chiller (ABS) is a refrigerator device that provides cooling power by means of a closed solution cycle where the main working fluid is absorbed into the solvent at evaporative pressure and desorbed at condenser pressure [109]; the most common working fluids are water/aqueous lithium bromide solution (LiBr) and water/ammonia [110]. Compared with other technologies (mechanically driven heat pumps), absorption chillers have shown significantly higher Coefficient of Performance (COP) but at the expense of compactness and the simplicity of the whole plant [108].

One of the most interesting features of LAES, is that besides producing electric energy it also provides heat and cool as by-product of the charge and discharge phase respectively;

hence the LAES can also be considered as a polygeneration system capable to be integrated with an air conditioning system, in order to supply a well-defined cooling load, and with a heat exchanger network, in order to be used in industrial settings and/or space heating/domestic hot water.

Based on these considerations, the present analysis aims to propose first a thermodynamic comparison between two waste heat recovery technologies, ORC and ABS, when applied to LAES in full electric (in which the only output is the electric power released by the LAES) and in trigenerative⁵ configurations (in which the electrical, cooling and heating power are produced at the same time). Indeed, this work tries to propose an innovative energy storage solution that is based on the integration of LAES with well-established waste heat recovery solutions (ORC and/or ABS). The comparative analysis aims to highlight whether and how much the integrated systems are thermodynamically superior over the stand-alone LAES system. An economic analysis will also be performed on the integrated LAES&ORC system comparing the results with the ones achieved by a stand-alone LAES and Li-ion batteries. The Levelised Cost of Storage (LCOS) method has been employed in order to evaluate the economic viability of the investment using data obtained during the development and the installation of the LAES pilot plant and the LAES grid scale demonstrator plant.

5.2 Models description

5.2.1 Systems boundary conditions

The systems under investigation are supposed to operate in a hot and humid environment such that of Singapore. Singapore lies just north of the Equator (near Latitude 1.5 deg N and Longitude 104 deg E) and due to its geographical location and maritime exposure, the

⁵ In the next paragraphs, the following terminology will be used: “trigenerative” or “cogenerative” LAES configuration (depending on the different nature of the thermal co-products) for the polygeneration LAES.

Singapore climate is characterized by a uniform temperature and pressure, high humidity. There is not a distinct wet or dry season: rainfall maximum occurs in December and April while the drier months are usually in February and July. Daily temperatures usually range from a minimum of 23–26 °C to a maximum of 31–34 °C, with extremes of minimum ≈ 19.4 °C and maximum ≈ 36 °C.

Each system is supposed to meet the baseload of electricity and/or cooling demand of a potential user so that, it can operate at all time in quasi steady state conditions (except during the plant startup). Moreover, since the tropical climate of Singapore is almost constant all over the year, the duty cycle of the energy storage systems analyzed can be considered representative of a whole year of operation. For illustrative purpose, Figure 5-1 shows the cooling load profile of the ADM building located in NTU campus for a typical normal operative day. The potential area of intervention is represented by the blue area in Figure 5-1: LAES operates in charging mode overnight during a low cooling demand period while energy is supplied by the LAES during day-time for the electricity/cooling demand matching.

Figure 5-1 Battery analogy scheme.

5.2.2 Stand-alone LAES (baseline case)

A 100 MW_e/400 MWh_e commercial size LAES system has been taken as a reference for

this study and have been modeled in EES software environment. The process flow diagrams of the full electric and trigenerative LAES plant configurations are shown in Figure 5-2 and Figure 5-3, in which the charge and discharge phases of the LAES are shown respectively. In the discharge phase, the cogenerative (heating) and trigenerative (cooling) subsections are enclosed in the small boxes (red and blue for cogenerative and trigenerative sections, respectively). As already mentioned earlier, the LAES can be separated into three sub-processes: charge, store and discharge. During the charge phase, the gaseous air is compressed in a 4 stage compression process with intercooling and aftercooling and turned into liquid air after passing through a throttle valve (J-T valve) and phase separator; the liquid air is then stored in a low pressure cryogenic tank (LA Tank). During the discharge phase, the liquid air is pumped to high pressure by means of a cryogenic pump and regasified; the excess cold released during the regasification is stored in a High Grade Cold Storage (HGCS) which serves as a cryogenic thermal energy storage.

Figure 5-2 Stand-alone LAES charge phase.

Figure 5-3 Stand-alone LAES discharge phase – Full electric and trigenerative configurations.

The high pressure air gas will then be further heated up at the superheaters (SHs) by means of the heat coming from a so called High Grade Warm Storage (HGWS) which stores the heat of compression released during the charge phase. The high pressure and high temperature gaseous air is then re-heated in a 4 stages expansion process to achieve a quasi-isothermal expansion. The main function of the two thermal storage subsystems, the HGWS and the HGCS, is that of linking the charge and discharge phases in order to increase the round trip efficiency; the working fluids used to transport the heat and cold are Therminol 66 and air respectively.

The operating parameters of the LAES (i.e. temperatures and pressures of the charge, discharge and liquid air storage) have been obtained from a thermodynamic optimization of the round trip efficiency carried out by means of EES; the process parameters for the LAES system are reported in Table 5-1.

Table 5-1 Process parameters for the LAES system under study.

Parameters	Value	Unit
Air inlet temperature	25	°C
Charge pressure	110	bar
Discharge pressure	180	bar
Liquid Air storage pressure	8	bar
Cold Box pinch point ΔT	3	°C
ICs pinch point ΔT	5	°C
Hot end temperature approach SHs	10	°C
Isentropic efficiency of compressors	85	%
Isentropic efficiency of cryoturbine	75	%
Isentropic efficiency of pump	80	%
Isentropic efficiency of power turbines	80	%

In full electric configuration, the hot end temperature approach at the superheaters ($\Delta T = T_{\text{HTF, hot}} - T_{\text{Air, hot}} = 10^{\circ}\text{C}$) constraints the turbine inlet temperature (TIT in Figure 5-3) of the air. Conversely, in trigenerative configuration, since the cooling load is provided by the direct expansion of gaseous air, the turbine inlet temperature of gaseous air is constrained by a defined turbine outlet temperature ($T_{\text{OT}} = 5^{\circ}\text{C}$ in Figure 5-3) which is required by the air conditioning system (i.e. water cooled chiller with an average COP of 5). Hence, as shown in Figure 5-3, the LAES is thermally coupled with the air conditioning system by means of air cooled heat exchangers. The remaining waste heat required for industrial uses and/or space heating/domestic hot water is computed by taking into account the temperature and mass flow of the heat transfer fluid immediately before the HGWS cold tank.

5.2.3 Integrated systems LAES-ORC

As already stated in Section 4.1, the most significant exergy loss in the LAES takes place during the charge phase: in a typical compression operation, approximately 90% of the electrical input is lost as heat [111]. In order to further improve the round trip efficiency of LAES, an ORC is coupled with the LAES in order to partially recover the large amount of exergy lost during the compression phase.

Figure 5-4 LAORC-1 integrated system.

Figure 5-5 LAORC-2 integrated system.

The waste heat discharged during the compression phase is used in the ORC evaporator to heat up the high pressure organic fluid that is expanded in the ORC-turbine; the organic working fluid is then condensed (ORC condenser) and pumped to high pressure and the cycle starts again. Two integrated systems LAES-ORC (LAORC) are proposed and shown in Figure 5-4 and Figure 5-5, respectively. The first LAES-ORC system (LAORC-1) exploits the waste heat downstream of the superheating process of the gaseous air. The heat source temperature is therefore linked with the superheating process of gaseous air. In alternative, as shown in Figure 5-5, an additional integrated system is introduced (LAORC-2) which harnesses the waste heat by means of a mass flow derivation of Therminol 66 from the HGWS. In this case, the thermal power available at ORC evaporator will be lower than the integrated system (LAORC-1) but with a higher heat source temperature. Due to the different heat source temperatures available for the ORC (T_{WH} in Figure 5-4 and Figure 5-5) [28], R134a and R245fa have been considered as the ORC working fluids for the first and second integrated LAES-ORC systems respectively; the parameters used to model the ORCs are summarized in Table 5-2.

Table 5-2 Process parameters for the ORC plant.

Parameters	Value	Unit
ORC Eva pinch point, ΔT	5	°C
Condensation temperature, T_k	25	°C
Isentropic efficiency of pump, $\eta_{iso,ORC,p}$	80	%
Isentropic efficiency of turbine, $\eta_{iso,ORC,T}$	80	%

5.2.4 Integrated system LAES-ABS

As already highlighted in Section 3.2, the charge phase of LAES has the largest impact on the performance of the entire cycle due to the high specific consumption to produce liquid air. Hence, one option to reduce the specific consumption of the liquefaction cycle by means of waste heat recovery, is that of coupling the LAES system with an absorption chiller (LAABS). More in particular, the waste heat released during the charge phase is used to drive a single stage water-LiBr absorption chiller (ABS) in which the resulting cooling power is used to sustain the liquefaction cycle, thus reducing the specific consumption. It is worth pointing out that the waste heat required to drive the ABS is only

a fraction of the total amount of waste heat released during the charge phase. Indeed, the size of the ABS is based on the cooling power necessary to decrease the temperature at the inlet of the second compressor and of the cold box. Due to the smaller contribution to the specific consumption of the first phase of compression (C1 and C2), the cooling power of the chiller is not used to reduce the specific work of the first two compressors. As a result, only the waste heat released by the second and the fourth compressor will be recovered. The remaining waste heat will be reused to increase the turbine inlet temperature as described in Section 5.2.2.

Figure 5-6 LAABS integrated system.

As shown in Figure 5-6, the waste heat is not immediately recovered after the compressors due to the temperature level of the heat source ($170\text{ }^{\circ}\text{C}$), too high to drive a single stage water-LiBr ABS [112]. Therefore, the air temperature is decreased through the AFCs to the temperature level ($\approx 98\text{ }^{\circ}\text{C}$) required by the ABS to operate.

5.2.4.1 Absorption chiller modelling and sizing

The ABS thermodynamic modelling results in a complex set of non-linear equations with a large number of input parameters. In this work, the single stage water-LiBr ABS is modelled with the characteristic equation method proposed by Kühn et al. [113]. This approach consists in a set of simple equations used to fit the technical data of a commercial ABS. The single stage water-LiBr ABS is considered as a system made up of the three major components: generator, evaporator and absorber-condenser. With the characteristic equation method, the thermal power ($\dot{Q}_k$) of each k-component is calculated by a linear

correlation with an arbitrary characteristic temperature function ($\Delta\Delta T'$) defined as:

$$Q_k = s' \cdot \Delta\Delta T' + r \quad (51)$$

$$\Delta\Delta T' = T_{gen} - a \cdot T_{ac} + e \cdot T_{evap} \quad (52)$$

where T [$^{\circ}\text{C}$] represents the average temperature of the medium fluids of each component of the chiller and the four parameters (a , s' , r , e) of Eqs. (51-52) are the constant parameters estimated by multiple regression from the technical data of the chiller selected. The performance of the chiller is then evaluated with the coefficient of performance (COP) calculated as:

$$COP = \frac{\dot{Q}_{evap}}{\dot{Q}_{gen}} \quad (53)$$

Fig. 7 shows the results of the application of the method proposed by Kühn et al. [113] for a single stage water-LiBr ABS with a cooling capacity of 767 kW_c (Figure 5-7a) and 2558 kW_c (Figure 5-7b). These two commercial sizes are used as a reference for the cycle described in Section 5.2.4 and Section 5.2.5, respectively. In particular, the figures show the cooling power, the heat input and the condenser power related with the characteristic temperature difference. The figures report the value of the four coefficients used for Eqs. (51-52) to fit the technical data of the selected ABSs and the R-squared values; the results of the linear correlation show a good agreement with the technical data, accounting for $\approx 76\%$ of the variance from the technical data. Therefore, the proposed correlations can be considered a valid solution for the ABS model.

(a)

(b)

Figure 5-7 LAABS integrated system. ABS cooling capacity of 767 kW_c (a) and 2558 kW_c (b)

5.2.5 Integrated system LAES-ABS-ORC

The most integrated option assessed in this work combines the LAES, ORC and ABS systems operating only in trigenerative mode. In this particular case the ABS has not been designed to assist the liquefaction phase of the LAES: for trigenerative purposes, the ABS will generate chilled water at a temperature of 6°C to be used for air conditioning applications. As a consequence, no constrain has been imposed on the turbine inlet temperature of the power turbine, meaning that the only cooling power source will be that provided by the ABS (this is because the exit temperature of the power turbine will not be forced to any specific level). As illustrated in Fig. 8, the ORC will exploit the waste heat

by means of a mass flow derivation from the HGWS, as already described in Section 2.3. The ABS will be then driven by the waste heat discharged by the ORC at a temperature of 105 °C, therefore limiting the size of the ORC evaporator in order to provide the required temperature to the ABS. The low-grade temperature waste heat ($\approx 90^\circ \text{C}$) at the outlet of ABS will be reused by the ORC thanks to a dedicated preheater (PH). The low-grade temperature waste heat ($\approx 90^\circ \text{C}$) at the outlet of ABS will be reused by the ORC thanks to a dedicated preheater (PH).

Figure 5-8 LAABS-ORC integrated system.

5.2.6 Technical Key Performance Indicators

The results of the simulations regarding the thermodynamic analysis will be presented in the next section with reference to the following performance parameters:

Round trip efficiency of the systems [%]:

$$\eta_{RT} = \frac{E_{net,d}}{E_{net,ch}} = \frac{P_{net,d,LAES} + P_{net,ORC}}{P_{net,ch,LAES} \cdot (\tau_{ch}/\tau_d)} \quad (54)$$

Overall efficiency of the systems [%]:

$$\eta_o = \frac{P_{net,d,LAES} + P_{net,ORC} + \phi_H * \frac{\dot{Q}_{H,LAES}}{COP_H} + \phi_C * \frac{\dot{Q}_{C,LAES}}{COP_{AC}}}{P_{net,ch,LAES} * (\tau_{ch}/\tau_d)} \quad (55)$$

Liquefaction Specific consumption [kWh_e/kg_{LA}]:

$$SC = \frac{P_{net,ch,LAES}}{\dot{m}_{LA}} \quad (56)$$

ORC efficiency [%]:

$$\eta_{ORC} = \frac{P_{net,ORC}}{\dot{Q}_{WH,ORC}} \quad (57)$$

Electric power output of the systems [MW_e]:

$$P_{e,tot} = P_{net,d,LAES} + P_{net,ORC} \quad (58)$$

Utilization factor of waste heat recovery systems [%]:

$$\eta_{WHRS} = \frac{\dot{Q}_{WH,u}}{\dot{Q}_{WH,tot}} = \frac{\dot{Q}_{WH,LAES,SH} + \dot{Q}_{WH,ABS} + \dot{Q}_{WH,ORC}}{\dot{Q}_{WH,tot}} \quad (59)$$

where $P_{net,d,LAES}$ [MW_e] is the net electric power produced by the discharge phase of the LAES, $P_{net,ORC}$ [MW_e] is the net electric power produced by the ORC plant, $P_{net,ch,LAES}$ [MW_e] is the electric power consumed during the charge phase of the LAES, $\dot{Q}_{WH,LAES,SH}$ [MW_{th}] is the thermal power utilized by the LAES superheaters, $\dot{Q}_{WH,ORC}$ [MW_{th}] is the thermal power utilized by the ORC plant, $\dot{Q}_{WH,ABS}$ is the thermal power utilized by the ABS plant, $\dot{Q}_{C,LAES}$ [MW_c] and $\dot{Q}_{H,LAES}$ [MW_{th}] are the cooling power and the heating power

discharged by the LAES respectively, $\dot{Q}_{WH,u}$ [MW_{th}] is the thermal power effectively utilized by either the LAES superheaters, the ORC or ABS, $\dot{Q}_{WH,tot}$ [MW_{th}] is the total thermal power discharged by the charge phase of the LAES, Φ_C and Φ_H are the utilization factors of the cooling and heating power available at the discharge section of the LAES (at a temperature of T_{CS} and T_{HS} , respectively).

It is worth nothing that by introducing the COP of both vapor compression chiller ($COP_{AC} = 5$ [12]) and heat pump ($COP_{HP} = 3.5$ [114]), the reference baseline is constituted by two different systems that are producing cooling and heating power by means of electricity. By means of such approach, already applied by Li et al. [115], the contributions of cooling and heating power are homogenously converted in an electrical equivalent form.

5.2.7 Levelised Cost of Storage (LCOS) analysis

Recently a new metric, Levelised Cost of Storage (LCOS), directly comparable to Levelised Cost of Energy (LCOE) for generation technologies [116], has been introduced as a valid tool for cost comparison of electricity storage technologies [117]. The LCOS quantifies the discounted cost per unit of discharged electricity for a specific storage technology and application. The metric therefore accounts for all technical and economic parameters affecting the lifetime cost of discharging stored electricity [118]. Julch [119] and Smallbone et al. [120] based on that metric their economic comparative analysis of different electricity storage technologies: PHS, CAES, PTES, various battery technologies and power-to-gas storage. Likewise, Schmidt [118] shows LCOS of energy storage technologies including PHS, CAES and battery energy storage systems. It can be seen that the economic evaluation has predominantly been based on the deployment of well-known technologies including batteries, CAES and Power-to-Gas Solution. In addition, a detailed costing exercise comparing LAES and batteries systems in these configurations, in particular based on LCOS methodology, is currently lacking.

5.2.7.1 LCOS Methodology

In order to reflect in a simple metric all of the cost factors for energy storage technologies, a constant or levelised cost per kWh_e over the storage system lifetime is introduced. The key input numerical data for LAES and ORC for LCOS calculation are summarised in Table 5-3 along with the assumed references.

LCOS [€/kWh_e] can be mathematically described as the total lifetime cost of the investment in an electricity storage technology divided by its cumulative delivered electricity estimated at each *n* step [years] over the total storage lifetime *N* [years] discounted with the interest rate *i* (%):

$$LCOS = \frac{CAPEX + \sum_n^N \frac{OPEX}{(1+i)^n} + \sum_n^N \frac{EC}{(1+i)^n} + \sum_n^N \frac{IC}{(1+i)^n}}{\sum_n^N \frac{E_d}{(1+i)^n}} \quad (60)$$

In the current LCOS formula it has been assumed that the residual value for the system components at the end of LAES lifetime is neglected and the financial lifetime *N* of the plants is equal to the lifetime of the storage capacity. The key parameters for the economic analysis are defined for a LAES commercial plant as follows.

CAPEX [€] is the capital cost of the investment computed as:

$$CAPEX = CP_{ch} * P_{net,ch} + CP_{PT} * P_{net,d} + CE_{LA\&HGS} * C_{rate} + CE_{HGWS} * C_{WH} + CP_{ORC} * P_{ORC} \quad (61)$$

where CP_{ch} [€/kWh_e] is the specific CAPEX power based per charging of power unit, CP_{PT} [€/kWh_e] is the specific CAPEX power based per discharging of power unit, CE_{tanks} [€/kWh] is the specific CAPEX energy based for the liquid air storage tank and the HCGS, C_{rate} [kWh] is the rated capacity of the plant, CE_{HGWS} [€/kWh] is the specific CAPEX energy based for the HGWS, C_{WH} [kWh] is the thermal capacity of the HGWS and CP_{ORC} [€/kWh_e] is the specific CAPEX per power unit of ORC.

OPEX [€/year] accounts for the power (OPEX_P [€/kW_e/year]) and energy specific (OPEX_E [€/kWh_e/year]) operation and maintenance costs related to the nominal power capacity and annual charged electricity:

$$OPEX = OPEX_P * P_{net,d} + OPEX_E * E_d * n_{cycles} + OPEX_{ORC} \quad (62)$$

where E_d [kWh_e] is the electricity discharged in one operation cycle, n_{cycles} [cycle/year] is the number of cycle per year and OPEX_{ORC} [€/year] is the operational cost for ORC estimated as a fraction of the total ORC capital cost.

Table 5-3 Summary of the input data for the LCOS calculation.

Parameter	Value	Unit	Reference
Storage lifetime	30	year	[94]
Self discharge rate	1	%	[120]
CP_{ch}	480.2	€/kW _e	[57]
CP_{PT}	162.6	€/kW _e	[57]
$CE_{LA\&HGCS}$	27.8	€/kWh	[57]
CE_{HGWS}	15	€/kWh	[120]
CP_{ORC}	2200	€/kW _e	[121]
OPEX _P	11.2	[€/kW _e /year]	[120]
OPEX _E	0.00264	[€/kWh _e /year]	[120]
OPEX _{ORC}	2.5	% of CP_{ORC}	[122]
i	8	%	[120]
IC	0.5	% of CAPEX	[119]

EC [€/year] are the annual electricity charging costs, namely the cost of purchasing charging electricity at a certain electricity tariff ET [€/kWh_e]. Mathematically manipulating LCOS definition, EC can be alternatively expressed as a function of the LAES round trip efficiency and the electricity tariff:

$$\frac{\sum_n^N \frac{EC}{(1+i)^n}}{\sum_n^N \frac{E_d}{(1+i)^n}} = \frac{ET}{\eta_{RT}} \quad (63)$$

where IC [€/year] is the insurance cost estimated as a fraction of the capital cost CAPEX.

5.3 Results

Table 5-4 presents the simulation results of the different LAES systems modelled in this work. As already stated, a basic differentiation between the full electric and trigenerative LAES configuration has been considered. The results of the integrated systems have been compared against a stand-alone LAES which has been used as a baseline; a total electric power production of 100 MW_e has been considered as the reference commercial size. In order to assess the influence of the main parameters affecting both the round trip efficiency and the waste heat recovery process, a comprehensive sensitivity analysis has been carried out for the charge and discharge pressure, the isentropic efficiency of the compression section and the evaporation pressure of the ORC. The economic comparative analysis between stand-alone LAES and LAORC integrated system is presented in Figure 5-14- Figure 5-17. A global sensitivity analysis has been carried out in order to evaluate the influence of the main parameters affecting LCOS analysis. Finally, the LCOS of both stand-alone and integrated LAES has been compared to Li-ion battery previously analyzed by Julch [119].

5.3.1 Energy analysis – Full electric configuration

Based on the thermodynamic assumptions made in Section 5.2, the simulations show that for the baseline LAES, round trip efficiency of 48.22 % is achieved under full electric configuration with a specific power consumption of 0.243 kWh_e/kg_{LA}. Such a low value of round trip efficiency is mainly due to the fact that a significant fraction of the total waste heat available from the compression phase is discharged to the environment immediately after the superheating process ($\eta_{\text{WHRS}} = 54.51\%$).

The introduction of an ORC in the LAES plant ensures an additional electric power output by means of the recovery of the low-grade waste heat. As a consequence, the round trip efficiency of the integrated system (LAORC) is improved due to a better exploitation of the waste heat that leads to higher values of the utilization factor of waste heat recovery system. Comparing the two different LAORC integrated systems, the electric power output

of LAORC-1 and the consequent improvement of round trip efficiency are smaller than those associated with LAORC-2 (50.51 % vs 52.89 %). The reason is mainly related to the heat source temperature (T_{WH}) available for the different LAORC integrated systems: the lowest T_{WH} (97 °C) strongly limits the ORC efficiency to low level (≈ 4.5 %). Indeed, despite the larger amount of waste heat available for LAORC-1 (104.14 MW_{th} vs. 84.70 MW_{th}), it was found that the electric power production is approximately 50% lower than the LAORC-2 (4.74 MW_e vs 9.68 MW_e): the best utilization factor of waste heat recovery systems is therefore achieved by the LAORC-2 integrated system ($\eta_{WHRS} = 84.55$ %).

Table 5-4 Simulation results for LAES_{ELE} and LAES_{TRIGE} configurations with $\Phi_C = 1$ and $\Phi_H = 0.5$.

Performance parameters	LAES _{ELE}				LAES _{TRIGE}				
	LAES	LAABS	LAORC-1	LAORC-2	LAES	LAABS	LAORC-1	LAORC-2	LAABS-ORC
η_{RT} [%]	48.22	48.21	50.51	52.89	40.13	42.60	44.16	47.51	52.55
η_o [%]	48.22	48.21	50.51	52.89	49.96	44.93	47.73	49.70	55.72
SC [kWh/kg _{LA}]	0.243	0.232	0.243	0.243	0.243	0.232	0.243	0.243	0.243
TIT [°C]	162.2	141.8	162.2	162.2	97	97	97	97	162.2
$\dot{Q}_{WH,ORC}$ [MW_{th}]	-	-	104.14	84.70	-	-	159.92	160.92	84.70
η_{WHRS} [%]	54.51	38.44	69.05	84.55	41.87	33.20	81.58	90.30	83.94
T_{WH} [°C]	-	-	97.0	172.2	-	-	115.4	172.2	172.2
η_{ORC} [%]	-	-	4.56	11.43	-	-	6.28	11.43	10.59
$P_{e,tot}$ [MW_e]	100	100	104.74	109.68	100	100	110.04	118.40	108.97
$\dot{Q}_{H,LAES}$ [MW_{th}]	-	-	-	-	159.92	26.76	50.67	26.70	36.77
T_{HS} [°C]	-	-	-	-	115.4	56.5	55.5	41.3	51.7
$\dot{Q}_{c,LAES}$ [MW_c]	-	-	-	-	8.24	8.24	8.24	8.24	6.68
T_{CS} [°C]	-	-	-	-	8	8	8	8	6

As described in Section 5.2, another waste heat recovery option taken into account is represented by the replacement of the ORC with an absorption chiller (LAABS) in order to decrease the inlet temperature during the compression phase. Thanks to the additional cooling power produced by the ABS, the specific consumption is 0.232 kWh/kg_{LA}, which is 4.52 % lower than that achieved by the stand-alone LAES. However, such a reduction in specific consumption is not followed by an increment in round trip efficiency (which stays almost unaltered). Indeed, the introduction of the ABS has a negative impact on the discharge phase of LAES since the turbine inlet temperature (TIT in Table 5-4) decreases from 162.2°C to 141.8°C due to the lower air temperature at the inlet of the LAES

compression phase. As a consequence, the positive effect on specific consumption reduction offsets the negative impact of the lower inlet enthalpy values for the power turbine.

5.3.1.1 Effect of charge pressure on round trip efficiency

Figure 5-9 shows the effect of the charge pressure (p_{ch}) on the performance of stand-alone LAES and the integrated system LAORC-2. It was found that, under the hypothesis of the current analysis, the round trip efficiency improvement due to the ORC is as high as 8-12 % when compared to the stand-alone LAES. In addition to this, Figure 5-9 shows that the theoretical maxima of the round trip efficiency are achieved for different charge pressures. As already analyzed by Guizzi et al. [46], the stand-alone LAES achieves its maximum round trip efficiency at approximately 150-160 bar; beyond such a value any further increase in charge pressure does not produce any benefit because the corresponding increase in compressors power consumption is not balanced by any significant increase in liquid air production in the charge phase. Conversely, the integrated system LAORC-2 overcomes such an issue allowing to further increase the round trip efficiency for higher pressure (around 180 bar) until the benefit of the additional electric power output is balanced by the compressors power consumption.

Figure 5-9 Round trip efficiency of stand-alone LAES and LAORC-2 as a function of charge pressure ($p_d = 180$ bar).

5.3.1.2 Effect of discharge pressure on round trip efficiency

Figure 5-10 shows the effect of the discharge pressure (p_d) on the round trip efficiency of the stand-alone LAES and the integrated system LAORC-2. Both the systems achieve their round trip efficiency maxima at approximately 180 bar; beyond such a value any further increase in discharge pressure does not produce any benefit because the corresponding increase in electric power output is offset by the consequent less waste cold discharged by the liquid air in the HGCS. In fact, increasing the discharge pressure lowers the waste cold to be recycled due to the increase of the liquid air temperature at the outlet of the cryogenic pump caused by the pumping work. As a consequence, the positive effect on the higher inlet enthalpy values for the power turbine offsets the negative impact of the higher specific consumption. In addition, it was found that the round trip efficiency improvement due to the ORC is as high as 9 % when compared to the stand-alone LAES.

Figure 5-10 Round trip efficiency of stand-alone LAES and LAORC-2 as a function of discharge pressure ($p_{ch} = 110$ bar).

5.3.1.3 Effect of compression isentropic efficiency and ORC evaporation pressure on round trip efficiency

Figure 5-11 (a) shows the effect of the compression isentropic efficiency ($\eta_{\text{iso,ch}}$) on the performance of stand-alone LAES and the integrated system LAORC-2. The figure shows that, for lower values of the isentropic compression efficiency, the gap between the two performance curves tends to increase, as shown by the green curve representing the round trip efficiency improvement due to the ORC (14 % at $\eta_{\text{iso,ch}} = 70$ %). In fact, with its additional electric power production, the LAORC-2 balances the performance degradation of the compression section, limiting the negative effect of the specific consumption increase (as shown in Figure 5-11 (b) by the blue curve) on the round trip efficiency. Conversely, at higher values of the isentropic compression efficiency, the round trip efficiency improvement of the LAORC-2 over the baseline case tends to decrease, until its minimum at 7 %, due to the lower waste heat temperature (T_{WH}) available for the discharge section of the LAES and the ORC, as shown in Figure 5-11 (b) by the orange curve.

Figure 5-11 (a) Round trip efficiency of stand-alone LAES and LAORC-2 as a function of compression isentropic efficiency. (b) Effect of compression isentropic efficiency on the specific consumption and waste heat temperature ($p_{\text{ch}} = 110$ bar; $p_{\text{d}} = 180$ bar).

Finally, Figure 5-12 shows the effect of the ORC evaporation pressure ($p_{\text{eva,ORC}}$) on the performance of the integrated system LAORC-2. R245fa has shown an optimal evaporation pressure between 17 and 18 bar in which the ORC efficiency achieves its maximum of

11.4 %. As expected, the LAORC-2 round trip efficiency follows the same trend of the ORC efficiency curve, with a quadratic dependence on the ORC evaporation pressure (as already shown by Quoilin et al. [123]) achieving its maximum at 52.9 %.

Figure 5-12 Round trip efficiency of LAORC-2 and ORC efficiency as a function of the ORC evaporation pressure.

5.3.2 Energy analysis – Trigenerative configuration

The potential efficiency improvement of integrated systems over the stand-alone LAES have also been analyzed for a LAES operating in trigenerative configuration, where the heating and cooling power are discharged by the HGWS and the direct expansion process respectively. As expected, the round trip efficiency of $LAES_{TRIGE}$ is sensibly lower than that associated with $LAES_{ELE}$. For example, the stand-alone LAES in full electric configuration achieves a round trip efficiency of 48.22% compared to 40.13 % performed by the stand-alone LAES in trigenerative configuration. In fact, the round trip efficiency is negatively affected by the cooling load provided by the air at the outlet of the power turbine, as explained in Section 5.2. The temperature constrain at turbine outlet ($TOT = 5\text{ }^{\circ}\text{C}$) leads to a lower turbine inlet temperature ($TIT = 97\text{ }^{\circ}\text{C}$) compared to the full electric configuration: as a consequence of this, a lower enthalpy drop in turbine expansion and in utilization factor of the waste heat recovery system ($\eta_{WHRS} = 41.87\text{ }%$) occur. On the other

hand, the higher thermal power ($\dot{Q}_{WH,ORC}$) discharged by the LAES_{TRIGE} leads to a higher waste heat recovery potential compared to the LAESELE configuration. Due to such a reason, the low efficiency of ORC in the LAORC-1 integrated system in full electric configuration is partially reduced: the increase in the heat source temperature ($T_{WH} = 115.4 \text{ }^{\circ}\text{C}$) produces a slight improvement of the ORC efficiency ($\eta_{ORC} = 6.28 \%$). Nevertheless, the LAORC-2 still represents the most energy efficient integrated system among the ORC solutions, improving the round trip efficiency of the stand-alone LAES by 18% (from 40.13% to 47.51%); indeed, the integrated LAORC-2 system is able to achieve a round trip efficiency value comparable to that of the stand-alone LAES in full electric configuration. By analyzing the performance of LAABS_{TRIGE} integrated system, it is worth nothing that compared to full electric configuration a slight improvement of round trip efficiency occurs. In fact, since the turbine inlet temperature is the same as that of the stand-alone LAES_{TRIGE} ($\approx 97 \text{ }^{\circ}\text{C}$), the reduction of the specific consumption produced by the introduction of ABS leads to a 5 % improvement of the round trip efficiency ($\eta_{RT} = 42.60 \%$). In order to mitigate the drawbacks related with the trigenerative configuration negatively affecting the turbine inlet temperature, another potential integrated system is introduced: the LAABS-ORC. Such an integrated system, providing the required cooling load by means of the ABS and efficiently exploiting the waste heat by means of the ORC, does not present any constrain at the turbine inlet temperature ($T_{WH} = 162.2 \text{ }^{\circ}\text{C}$). In fact, it was found to provide the lowest thermal power available for the ORC ($\dot{Q}_{WH,ORC} = 84.7 \text{ MW}_{th}$) with one of the highest utilization factor of the waste heat recovery systems ($\eta_{WHRS} = 83.94 \%$). Due to such reasons, LAABS-ORC_{TRIGE} has shown the best key performance indices among the other integrated systems achieving a round trip efficiency 30 % higher than that obtained by the stand-alone LAES_{TRIGE} (from 40.13% to 52.55%).

Taking into account the exergetic value of useful cooling and heating power by means of the overall efficiency (η_o), the stand-alone LAES_{TRIGE} is able to achieve significant level of overall efficiency (49.96 % in case of full - $\Phi_C = 1$ - and partial - $\Phi_H = 0.5$ - exploitation of the cooling and heating power respectively). This is principally due to the large amount of heating power (159.92 MW_{th}), available at the superheaters outlet at a temperature of $115.4 \text{ }^{\circ}\text{C}$, which a potential industrial final user or a district heating system may benefit.

Confirming the result based on round trip efficiency evaluation, the integrated system LAABS-ORC allows achieving the best overall efficiency (55.72 %) among the integrated system simulated.

5.3.2.1 Effect of the utilization factors of heating and cooling power on the performance of LAABS-ORC integrated system

Figure 5-13 shows the effect of the utilization factors of heating and cooling power, Φ_H and Φ_C , namely the ratio between the effective demand of heating and cooling load by a potential final user and the availability of both quantities at LAES discharge, on the system overall efficiency. The integrated system achieving the best performance indices (LAABS-ORC) has been taken as a reference for the sensitivity analysis carried out on Φ_H and Φ_C . Figure 5-13 reports the results of the sensitivity analysis: the overall efficiency has been plotted as a function of Φ_C for three different values of Φ_H (0.8, 0.4, 0.2). As expected, the linear dependence of the overall efficiency over the utilization factors produces the maximum values at the right edge of the curves: the largest value achieved is 55.6 % with a 6 % improvement over the baseline case in which no heating and cooling demand is required, namely the round trip efficiency value (52.6 %) shown in Table 5-4.

Figure 5-13 Overall efficiency of LAABS-ORC as function of the utilization factors Φ_C and Φ_H .

5.3.3 Energy analysis – Application of the results

LAES is a relatively novel technology whose application for full electric configuration has been recently shown by Highview Power through the development and the installation of a pilot plant [27] and a grid scale demonstrator plant [75]. In both cases, the waste heat recovery systems rely on external heat sources, namely a waste heat stream (up to 60°C) released by a biomass power plant operating in Greater London and the engine exhaust gases from a landfill gas generation plant installed in Greater Manchester, respectively. Taking into account the possibility to exploit the waste heat discharged by the LAES charge section by means of the waste heat recovery system introduced in the present work, it might be feasible to guarantee approximately the same level of round trip efficiency claimed by Highview Power for the grid scale demonstrator plant. Beside full electric configuration of LAES, another real case study involving polygeneration configuration has been previously assessed in Section 3.4. This work has introduced a poly-generation LAES in order to fulfill the peak cooling demand imposed by a building located within the Nanyang Technological University (NTU) campus in Singapore. Applying to the mentioned work the waste heat recovery technologies introduced in the current work (LAABS-ORC), it has been roughly estimated that the round trip efficiency of LAES increases from 45 % to 60 %, potentially improving the techno-economic feasibility of the whole LAES plant.

5.3.4 Economic analysis

Due to the promising technical results shown for the LAES coupled with Organic Rankine Cycle, an economic comparative analysis of the stand-alone LAES and the integrated system LAORC⁶ is presented by means of LCOS methodology. In addition, only two different LAES configurations have been considered: full electric and cogenerative, where without considering the waste heat available for cogeneration, the main LAES energy

⁶ In the next paragraphs, the following terminology will be used: LAORC for LAORC-2 integrated system and LAES_{COGE} for the cogenerative (electricity&cooling co-production) LAES configuration.

vectors outputs are represented by the electricity and cooling energy.

5.3.5 LCOS comparison: stand-alone LAES vs LAORC

Figure 5-14 shows the LCOS of the stand-alone LAES and LAORC systems considered in the economic analysis. The commercial size LAES (100 MW_e/ 400 MWh) is supposed to operate for 365 cycles per year. As already stated, two different LAES configurations (full electric and cogenerative) have been considered. The average electricity tariff of Singapore (0.15 €/kWh_e) [124] has been taken as reference for the whole economic analysis. Nevertheless, this value can be representative of any other case scenario and country.

The economic analysis confirms the technical outcomes discussed in the previous paragraph, namely the LAORC integrated system shows better economic performance compared to the stand-alone LAES: lower LCOS is achieved at 0.385 and 0.437 €/kWh_e for the electric and the cogenerative configurations, respectively. The inhomogeneous distribution of the share of the main cost components within the LCOS of each system provides an explanation for the economic performance of the LAORC integrated system. With an average share around 77 % the electricity charging cost is predominant compared to the other components: as a consequence, the round trip efficiency and the electricity tariff have a significant impact on the LCOS value. Due to this reason, the additional capital and operational cost introduced with the ORC is balanced by the increase in round trip efficiency that allows to decrease the share of the electricity charging cost. Although the LCOS of the LAES_{COGE} is higher than the LAES_{ELE} due to the lower round trip efficiency, the most significant results are achieved in cogenerative configuration where the LAORC integrated system is found to decrease the LCOS by 10%. The share of charging, discharging and storage units within the CAPEX is almost uniform among the different systems analyzed with the highest impact of the liquefaction plant capital cost over the discharge phase due to the relatively low round trip efficiency especially in cogenerative configurations. Another significant impact on the CAPEX share is represented by the storage units due to the presence of two thermal energy storages (HGCS and HGWS) that are thermally coupling the charge and discharge phase. It is worth noting that the share of

$CAPEX_{storages}$ is higher for full electric configuration compared to cogenerative configuration due to the higher η_{HGWS} (98 % vs 50 %). As a consequence, the HGWS of the LAES operating in cogenerative mode can be downscaled with a resulting lower CAPEX.

Figure 5-14 Cost components of the LCOS for electric and cogenerative configurations at 365 cycles per year and an electricity price of 0.15 €/kWh_e.

5.3.6 LCOS sensitivity analysis

Figure 5-15 reports the results of the sensitivity analysis carried out in order to assess the influence of the electricity tariff and the number of cycles per year over the LCOS. The cogenerative system has been taken as a reference for the sensitivity analysis. Four different electricity tariffs have been considered, from the scenario when electricity for charging is free or entirely provided by a renewable energy source ($ET = 0$ €/kWh_e) up to the reference scenario ($ET = 0.15$ €/kWh_e). According to an approximately inverse relation, the LCOS decreases as the number of cycles per year, and therefore the total amount of energy discharged, increase. In fact, by increasing the amount of energy discharged per year, the

LCOS decreases significantly due to the fact that the same CAPEX and OPEX costs are distributed over a larger amount of energy discharged. In addition, Figure 5-15 provides a further explanation on how the round trip efficiency strongly affects the LCOS. Excluding the cost of electricity for charging the LAES, the LCOS curves of both systems show better performance of the stand-alone LAES especially at low number of cycles per year. As long as the number of cycles is below a certain threshold value for every electricity tariff scenario this trend is almost identical. Nevertheless, the higher is the electricity tariff, the lower will be the threshold value of the number of cycles per year beyond which the LCOS of the LAORC integrated system becomes lower than the one of the stand-alone LAES. In fact, the gap between the two curves becomes significant as the electricity tariff increases up to the reference value of 0.15 €/kWh_e with a LCOS decrease as high as 14 % at n higher than 700 cycles per year. Further manipulating the data obtained, the so called “turning points” curve shown in Figure 5-16 has been created in order to immediately correlate the number of cycles that guarantee, at a fixed electricity tariff, the same LCOS between LAORC integrated system and stand-alone LAES.

Figure 5-15 LCOS depending on the cycles per year at different electricity tariffs for LAES cogenerative configuration.

Figure 5-16 Turning points curve between LAORC and LAES systems for cogenerative configuration.

Figure 5-17 shows the global sensitivity analysis for the LCOS of the LAORC integrated system for full electric configuration. The analysis has been carried out by fixing a reference case scenario and varying the considered parameters (round trip efficiency, electricity tariff, number of cycles, specific CAPEX power based per charging of power unit, per discharging of power unit and per power unit of ORC, interest rate and total lifetime) by $\pm 30\%$. A linear proportional dependency can be seen between LCOS and the specific CAPEX figures, the electricity tariff and the interest rate while, as already shown in the previous sections, the round trip efficiency, the number of cycles and the total lifetime have an inverse and non-linear relation to the LCOS. Both the round trip efficiency and the electricity tariff have the most significant impact on LCOS due to the relatively high electricity tariff taken into account for the reference case. In fact, confirming the results of Figure 5-14, the higher is the electricity tariff, the more significant will be the impact of the round trip efficiency over the LCOS. Other main impacting parameters are represented by the number of cycles and the discount rate of which variation by $\pm 30\%$ leads to a LCOS change up to 10% and 4%, respectively. Among the specific CAPEX figures, the cost of liquefaction plant has the strongest influence on LCOS with a change up to 3 %.

Figure 5-17 LCOS sensitivity analysis for LAORC full electric configuration. Reference case at 365 cycles per year and 0.15 €/kWh_e electricity tariff.

5.3.7 LCOS comparison: LAES vs Li-ion battery

The LCOS of the LAORC integrated system in full electric configuration has been compared with Li-ion battery technology. In particular, Figure 5-18 and Figure 5-19 report the results of the analysis carried out by Julch [119] that has computed the LCOS of five energy storage technologies applying the same methodology employed in this paper. In order to fairly compare the results of the two analysis, a LAORC in full electric configuration has been taken as a reference and the cost of electricity for charging has been considered equal to 0 €/kWh_e and 0.03 €/kWh_e for Figure 5-18 and Figure 5-19, respectively. Figure 5-18 shows that LAORC integrated system generally achieves the lowest LCOS with 0.16 vs 0.34 €/kWh_e for Li-ion battery at 365 cycles per year. The share of the main cost components within the LCOS of each system for 365 cycles per year and an ET equal to 0.03 €/kWh_e is reported in Figure 5-19. It clearly shows that LAORC has a high share of electricity cost component while Li-ion battery is dominated by the CAPEX, in which the storage unit has the highest cost share.

Figure 5-18 LCOS depending on the cycles per year not including electricity costs for LAORC integrated system in full electric configuration and Li-ion battery technology.

Figure 5-19 Cost components of the LCOS for LAORC integrated system in full electric configuration and Li-ion battery technology at 365 cycles per year and an electricity price of 0.03 €/kWh_e.

5.4 Summary

In this chapter, the techno-economic feasibility analysis of the Organic Rankine Cycle and absorption chiller integration for waste heat recovery purpose in Liquid Air Energy Storage has been carried out under full electric and trigenerative configurations. The motivation behind the proposed study is due to the inefficient exploitation of the heat discharged during the compression phase of a stand-alone LAES. As a general observation, the study showed that the utilization of the low-grade waste heat from the compression phase of a Liquid Air Energy Storage seems to be technologically viable and capable to significantly improve the round trip efficiency of the system by producing additional electrical power output and/or decreasing the specific consumption. However, the level of efficiency improvement depends significantly on both the configuration (full electric or trigenerative) and the waste heat recovery system introduced in the Liquid Air Energy Storage system. From the economic perspective, the study showed that the implementation of an Organic Rankine Cycle to recover the low-grade waste heat discharged by the Liquid Air Energy Storage charge phase seems to be economically viable and capable to significantly decrease the levelized cost of storage of the plant under opportune conditions. In fact, the economic benefit due to Organic Rankine Cycle integration depends significantly on both the configuration (full electric or cogenerative) and the related round trip efficiency, the electricity tariff and the number of cycles per year strictly related to the amount of energy discharged per year. More in particular, the following conclusions can be drawn from the analysis of the main results:

- among the waste heat recovery systems investigated, the Organic Rankine Cycle seems to be the best candidate system to recover the low-grade waste heat, increasing both the nominal electric power output (109.68 MW_e and 118.40 MW_e in case of full electric and trigenerative configurations) and the round trip efficiency due to a more efficient exploitation of the waste heat (as shown by the highest utilization factors of the waste heat recovery systems). Hence Organic Rankine Cycle represents the best option to recover waste heat from Liquid Air Energy Storage where the need for both electric, cooling and heating power is required;

- although absorption chiller is able to decrease by 5% the specific consumption of the charge phase of the Liquid Air Energy Storage, the round trip efficiency slightly decreases compared to the stand alone Liquid Air Energy Storage due to the lower quality of the waste heat available at the Liquid Air Energy Storage superheaters. This effect is partially mitigated in trigenerative configuration due to the constrain at the turbine inlet temperature imposed by the cooling load;
- the most remarkable results are achieved in trigenerative configuration where the LAORC-2 and the LAABS-ORC integrated systems were found to improve the round trip efficiency by 20 % and 30 %, respectively;
- the possibility to integrate both Organic Rankine Cycle and absorption chiller in Liquid Air Energy Storage has been assessed in trigenerative configuration. The LAABS-ORC integrated system has shown promising results achieving the best overall efficiency (55.72 %) among the other cases. Nevertheless, as a future work, an economic analysis requires to be carried out in order to check the economic feasibility of the integrated plants LAORC and LAABS-ORC that may put at stake the technological viability of the integrated systems addressed;
- the most significant economic results are achieved by the cogenerative configuration where the LAORC integrated system, compensating the large amount of waste heat discharged to the environment in stand-alone LAES, was found to decrease LCOS by 10 % considering the same electricity tariff applied in Singapore;
- similar to the other energy storage technologies, the LCOS of LAES is very sensitive to the operation of the plant, namely an increasing of the number of cycles per year produces a significant LCOS decrease;
- for every electricity tariff a threshold value of the number of cycles per year beyond which the LCOS of the LAORC integrated system is lower than the one for stand-alone LAES has been identified;
- the annual electricity charging costs are the predominant component in LCOS cost structure for LAES: the higher is the electricity tariff, the more economically profitable will be the LAORC integrated system compared to stand-alone LAES due to the higher economical valorization of the additional electricity output produced by the ORC;
- neglecting the annual electricity charging costs, an economic comparison carried out

with Li-ion battery showed that LAORC integrated system has a comparatively lower LCOS. Indeed, since LAES is currently in development stage, a larger potential for cost reduction is expected in the future.

Chapter 6 ⁷

Environmental performance of Liquid Air Energy Storage: a Life Cycle Assessment based comparison

The focus of this chapter is to compare the eco-friendliness of a relatively new technology, namely Liquid Air Energy Storage with established storage solutions such as Li-Ion Batteries and Compressed Air Energy Storage. The comparison is carried out through Life Cycle Assessment whose aim is to measure the environmental impact from cradle to gate, excluding plant decommissioning. The study applies to the unit of electric power stored. The “flexibility” of Liquid Air Storage, which is able to produce cooling power as a co-product, designates this technology as the most environmentally competitive. However, further investigations regarding the use phase must be implemented as it plays a relevant role in this context.

⁷ This section published substantially as Mengarelli, M., Tafone, A., Romagnoli, R. (2017). *Environmental performance of electric energy storage systems: a Life Cycle Assessment based comparison between Li-Ion batteries, Compressed and Liquid Air Energy Storage system*. In Proceedings of Ecos 2017: 30th International Conference on Efficiency, Cost, Optimization, Simulation and Environmental Impact of Energy Systems. 2nd –6th July 2017. San Diego, USA.

6.1 Introduction

Likewise any other systems, EESs require a certain energy input for the use of material for the components' manufacturing, construction of infrastructures and facilities, maintenance during operation and disassembly occurring at the decommissioning stage. Therefore, in order to maintain a low environmental impact profile, it is very important to assess the energy consumption and emissions generated by EESs at a life cycle level [125].

This chapter focuses on the environmental impact of EESs by means of Life Cycle Assessment (LCA). The LCA metric has been selected as it represents the most widely recognised methodology to evaluate environmental impact of product systems and services [126]. A comparison between two established energy storage technologies and a relatively novel technology has been carried out. The already technically developed and commercially available technologies are CAES and Li-Ion battery (Li-Ion). These two energy storage systems are considered as the most performing energy storage technologies from different angles. CAES is mainly recommended for large energy management applications and can reach a roundtrip efficiency beyond 60% [5]. Among different battery types, Li-Ion is the leading option in terms of energy density, lifetime expectancy and the use of less environmentally intensive materials [127]; in addition to this, Li-Ion withstand higher depth of discharge and can reach up to 90% of roundtrip efficiency [5,128–130]. The novel technology considered is represented by LAES system whose full potential has not been explored yet as LAES main advantage is to simultaneously provide electricity and cooling power from the same energy mean, whereas other technologies require additional machines (e.g. chillers) to supply cooling power. The scientific literature lacks robust and realistic case studies regarding this capability. Therefore, the aim of the current analysis is to evaluate the intrinsic flexibility of LAES in analogy with the reliability and robustness of Li-Ion and CAES by adopting a life cycle approach. LCA metrics for Li-Ion and CAESs are retrieved from the literature whereas those for the LAES have been calculated as part of the current work.

6.2 The battery analogy

The battery analogy represents the methodology that has been adopted in order to carry out the comparison between the technologies described earlier. The comparison model is of fundamental importance in order to not bias the comparison. In fact, as previously mentioned, LAES is characterised by a dual output: electric and cooling power and therefore the three different phases namely charge, storage and discharge are explained for each technology (refer to Figure 6-1); the layout of CAES and LAES are simplified since the aim is to display the key components and processes without going into much details.

Figure 6-1 Battery analogy scheme.

At first, it is possible to notice that the storage and the discharge phase are quite similar for both LAES and CAES. This is due to the fact that they use the same energy medium (air) but at different physical status. The main difference between LAES and CAES is that the

former, after compression, requires few more steps in order to liquefy the air. In addition to this it is worth noting that all of the ESSs considered in this study use the same source of energy to be charged (in Figure 6-1 indicated as M “motor”); hence this will not be included in the comparison. The same occurs for the discharge phase in which the output is assumed to be provided by a “generator”.

6.3 Life cycle assessment (LCA) Methodology

LCA is a step by step methodology which involves four main stages: goal and scope definition, life cycle inventory, life cycle impact assessment and interpretation of results [131,132]. Each stage will be examined in the following paragraphs.

6.3.1 Goal and scope definition

The goal of the study is to evaluate the environmental performance of three technologies - LAES, CAES, Li-Ion - used to store and deliver electric energy. The current study has been carried out only for LAES, while data have extracted from the literature for CAES and Li-Ion.

6.3.2 Functions and functional units

In this study, the ESSs are used to store electric energy during off-peak hours and deliver it during peak load demand. Therefore, all the ESSs are assumed to be fed by the same energy source and connected to the grid. They are designed to ensure a certain electric power for a finite amount of time to the grid or to the utilities.

Table 6-1 Different configuration scenario for LAES

Scenario	Description
Scenario 0	This configuration aims at maximising the roundtrip efficiency: the HGCS is entirely utilized to reduce the specific consumption of the air liquefier (charge phase) while the HGWS is used to heat up the air during the discharge phase; the only output is the electric power.
Scenario 1	This configuration aims at producing both electric and cooling power. A commercial user (e.g. warehouse) that needs a cold temperature source of -20 °C is used for this scenario. The cooling load is provided by the HGCS whose cold energy is also partially utilized to reduce the specific consumption of the air liquefier. Approximately 25% of HGCS available is provided to the cooling load while the remaining 75% is exploited by the air liquefier. The cooling/electric power ratio is 0.20.
Scenario 2	This configuration aims at producing both electric and cooling power. A commercial user (e.g. warehouse) that needs a cold temperature source of -20 °C is used for this scenario. In order to fulfil the cold energy demand required, the air at the expander outlet is thermally coupled with the commercial user. The cooling/electric power ratio is 0.63.
Scenario 3	This configuration aims at maximising the cooling power discharged by the LAES as well as producing electric power. A commercial user (e.g. warehouse) that needs a cold temperature source of -20 °C is used for this scenario. The cooling load is provided both by the HGCS (as in Scenario 1) and the air at outlet of the expander (as in Scenario 2). The cooling/electric power ratio is 0.90.

The functional unit is then defined as the delivery of 1 MWh_e of electric. Such function should be accomplished over a lifetime of 10 years, meaning that the ESSs must guarantee the selected function for the entire lifetime, which, in some cases, would require the replacement and maintenance of some components/parts. The geographical context for the development of the function is Singapore; the reference year for this study is 2016.

As previously mentioned, LAES is actually able to provide cooling power beside electric power, which respect to the abovementioned function is considered as a co-product. The actual share of electric and cooling power can be controlled depending on the needs. This aspect has been taken into consideration by including different operating scenarios for the LAES, summarized in Table 6-1.

6.3.3 System boundaries definition

The present study is from cradle to gate, namely raw material extraction, plant

manufacturing and the use phase are included in the analysis. The plant decommissioning is left outside the system boundaries; this has not been considered since for LAESs there is lack of real data due to the few pilot plants available which have not reached their End of Life (EoL) yet. However, it is worth noting that LAES plants consist of existing components (e.g. compressors, heat exchangers, turbines, etc.) and therefore their EoL scenario could easily be predicted; this is also valid for the CAES. The electricity supply to charge the ESSs has not been included in the LCA since the same source will be used for each technology.

Figure 6-2 System boundaries.

The electricity dispatch is not considered for the same reasons of the energy supply, since the same utility can be applied to all technologies. Due to the extremely large number of components that characterize LAES, a cut-off threshold of 5% in mass share has been set. This means that those components or materials which do not reach 5% in mass share with respect to the entire plant are not considered in the inventory. A summary of the system boundaries is shown in Figure 6-2.

6.3.4 Data requirement and quality

The life cycle data related to LAES can be considered as primary data since the equipment, the components and the mass flows are based on the pilot plant already existing [27] and further elaborated within the computational analysis model.

For what concern the Li-Ion pack and the CAES, life cycle data have been obtained from the literature since there is a wide and consistent list of LCA studies carried out on both technologies [125,128,130].

6.3.5 Life cycle inventory

In this stage, all the life cycle data included in the goal and scope are collected and inserted in the LCA modelling tool which in this case is *SimaPro 8* with the *Ecoinvent 3.1* database. For reasons of brevity and space constraint, the inventory will not be displayed in this analysis; however, the approximations and modelling approach will be explained.

The “Allocation, recycled content” system model has been used since it does not take into account any benefit related with the recycling of a material. In this model, recyclable materials are available burden-free to recycling processes, which means that secondary (recycled) materials only bear the impacts of the recycling processes. Moreover, the model does not give any credit to producers of waste for the recycling or re-use of products from any waste treatment. In this case the “Recycled content” system model is preferable since the EoL is outside of the boundary conditions, therefore neither credits nor burdens should be included.

The “Market Processes” dataset has been used wherever no modifications to the original dataset have been made. “Transformation Processes” dataset has been used in those datasets that have been modified by adding or removing materials or energy processes. Correction factors have been used in order to adapt the size of the machine dataset (e.g. turbines, compressor, heat exchangers, etc.) with that of the plant size under study.

Regarding the use phase, only the energy used during operation has been taken into account. The amount of energy has been calculated in Eq. (64) as the losses of energy that must be replaced due to inefficiencies. It is assumed that the plant operates with a capacity utilization factor of 0.4. This is due to the fact that it basically operates only during night hours (approximately 9-10 hours). Such assumption takes into consideration ordinary and

extraordinary maintenance of the plant.

$$En_{losses} = \left(\frac{1}{\eta_{RT}} - 1 \right) * k_u * h_y * N_y \quad (64)$$

where η_{RT} is the roundtrip efficiency of the energy storage system, k_u is the capacity utilization factor, h_y [h/year] are the operation hours per day and N_{year} [year] is the number of years.

A sensitivity analysis regarding the consumption of energy during operation has been carried out since the use phase might play a key role. Two different energy mix dataset have been used to model the use phase. In the first one, the energy production mix of Singapore has been used, in order to simulate the connection of the plant to the national grid.

Figure 6-3 Singapore energy mix for electricity generation [133].

As shown in Figure 6-3, the major energy source in Singapore are fossil fuels. Therefore, in order to emphasize the environmental impact share related to the consumption of energy the second scenario embodies 100% renewable energy. The selected dataset describes the construction and the utilization of a Photovoltaic plant (open ground installation) with a capacity of 570 kW_p. Practically, it is assumed that the energy needed to operate is drawn only from a renewable energy source. It is obvious that this condition would not be physically feasible since photovoltaic energy is captured during day time while the LAES is assumed to be charged during night time. However, it is scientifically relevant as it foresees the effect of the energy source for such long lasting and massive technologies. In the Impact Assessment phase, the label Photovoltaic is used to represent the renewable

variant of the relative scenario. Among the all scenario described in Table 6-1, only the most performing would be included in the sensitivity analysis.

The production of cooling power (valid for all scenario except for Scenario 0) has been credited as avoided production of cooling power coming from a Water-cooled Vapour Compression chiller with an average Coefficient of Performance (COP) of 4.5-5. Such technology has been chosen as nowadays it represents one of the most efficient options available. By using the most performing machine, the avoided consumption of electric energy would be brought to the minimum, thus leading to the worst case scenario. Any less efficient chiller would consume more electricity for the same amount of cooling power, giving higher credit as avoided impact.

6.3.6 Life Cycle Impact Categories

In the comparative analysis, the Cumulative Energy Demand (CED) and the Global Warming Potential (GWP), defined according to [134], have been assessed as the main Life Cycle Impact (LCI) categories for the three energy storage technologies. Those categories have been characterized and computed by the SimaPro software by means of its environmental impacts database. For a mathematical visualization of the method to effectively compute the LCI categories, the reader can refer to the work carried out by Heijungs and Suh [135].

6.4 Results

6.4.1 LAES Life Cycle impact assessment

This section illustrates the environmental impact of the LAES plant calculated according to the following Life Cycle Impact Assessment (LCIA) methods:

- ReCiPe mid-point - Hierarchist (H) version - World [136];

- Cumulative Energy Demand LCA food V1.02 / Cumulative energy demand (CED) [137].

ReCiPe is a widely established impact assessment method which embrace both mid-point and end-point indicators [136]. It comprises two sets of impact categories with associated sets of characterisation factors. In this analysis, a sub set of categories have been used at mid-point level. The authors try to provide a broad spectrum of indicators accounting for damage to the ecosystem (e.g. Terrestrial acidification, Photochemical oxidant formation), damage to human health (e.g. Ozone depletion, Particulate matter formation, Climate change) and damage as consumption of resources (e.g. Metal depletion, Fossil depletion). CED instead has been selected as it focusses more on the energy related impact as it calculates the embodied energy for a definite product/plant/service. CED has been considered as a valid index as it is indicative of many environmental problems, such as global warming, acidification, eutrophication and photochemical ozone formation especially when a large consumption of energy occurs. CED can also be used as screening indicator for environmental performance in the absence of specific data [138]. Results are shown both at characterisation at normalisation level. Characterised results allow comparison within the same impact category, whereas normalised results, which are dimensionless, are useful to compare different scenarios by summing up all the categories in a single column.

Figure 6-4 Characterised results from ReCiPe Midpoint (H) V1.12 / World Recipe H for LAES

Figure 6-5 Normalised results from ReCiPe Midpoint (H) V1.12 / World Recipe H for LAES.

Figure 6-6 Characterised results from "Cumulative Energy Demand LCA food V1.02" for LAES

Figure 6-7 Characterised results from ReCiPe Midpoint (H) V1.12 / World Recipe H, of the different life stages for Scenario 2 (Photovoltaic) for LAES.

In Figure 6-4, Figure 6-5 and Figure 6-6 two more scenarios are added, namely Scenario 0 (Photovoltaic) and Scenario 2 (Photovoltaic). The term Photovoltaic indicates that the dataset used to model the consumption of electric energy during the use phase refers to the construction and use of a solar power plant. This is part of the sensitivity analysis regarding the use phase. Such analysis targets only these two scenarios (i.e. Scenario 0 and Scenario 2) since they are the most environmentally performing plant configurations.

Looking at Figure 6-4, characterized results show a smooth equilibrium for the Ecosystem damage related categories and for the Particulate matter formation category. The column of Metal Depletion is mainly “occupied” by the two solar powered scenarios due to the large deployment of natural resources to build the photovoltaic plant. On the contrary, for Climate change, Fossil depletion and Ozone depletion, the column share of these two scenarios is practically absent. Thus, the adaptation of renewable energy resources leads to a lower damage to the human health.

From Figure 6-5, which allows comparison among the different scenarios, it can be inferred that the most eco-friendly scenario is Scenario 2 (Photovoltaic). This scenario represents the best compromise between the maximization of electric power and the maximization of cooling power. In fact, the cooling load is provided by the HGCS whose cold recycle is partially utilized to reduce the specific consumption of the air liquefier. However, it is interesting to notice that if the Singaporean energy production mix is used for the use phase, Scenario 1 results as the least impactful scenario, followed by Scenario 2. This is due to the highest roundtrip efficiency obtained in such configuration. On the contrary, if the LAES draws electricity from a photovoltaic plant, the opposite occurs. This means that the magnitude of impact related to the production of electricity together with the cooling power produced as co-product (avoided impact) governs the environmental profile of this technology. In other words, if the energy source is not “clean”, then the roundtrip efficiency can be considered as the driving parameter, otherwise, if the plant is connected to a renewable source, then, a balanced compromise between electricity and cooling would represent the best solution.

In Figure 6-6 it is shown how the use phase massively deviates the overall impact. Both photovoltaics' scenarios are much different from the others, meaning that the energy embodied in the use phase is larger than in the other phases. It is interesting to observe that the trend is similar to Figure 6-5, however, the most performing scenario is Scenario 0 (Photovoltaic). This is a confirmation of the importance of the use phase. Despite Scenario 2 (Photovoltaic) benefits from the credits of the cooling power production, it is still more impactful than Scenario 0 (Photovoltaic), meaning that in this case, the avoided production of cooling power is not enough to compensate the reduction of roundtrip efficiency. For this impact category, the highest roundtrip efficiency represents the driving factor, which “decides” which scenario is the most eco-friendly.

Figure 6-7 analyzes impacts from different life stages within Scenario 2 (Photovoltaic). The chart shows that “raw material extraction + plant manufacturing” carry negligible impacts in most categories except for Metal Depletion. From a design point of view, it means that despite a large investment in terms of material resources such as the deployment of metals to build compressors and turbines as well as coating materials for the storage tanks, the most impactful phase remains the use phase. In human health damage categories, the cooling power avoid impact almost equals the impact from the electricity consumption. For ecosystem damage and for Particulate matter formation instead, the electricity consumption from the use phase overcome avoided impact of one order of magnitude, meaning that “shifting” to a clean energy source favour human health and resource consumption, at the expense of the ecosystem.

6.4.2 Comparison between Energy Storage Systems

In this section, the comparison between the different ESSs is reported. It is important to remark that the environmental quantities chosen from the literature for the CAES and Li-Ion might not completely overlap with those obtained in the LAES analysis described earlier. For most data available in the literature, neither the calculation behind the results was clearly stated nor the system boundaries, whereas more information was made available about the assumptions and cut-off. Thus, the proposed comparison might be

affected by these inconsistencies. The use phase has not been included in the comparison since the use of different dataset for energy consumption would heavily affect the results leading to a not reliable comparison. Therefore, it is not relevant which Scenario is used, since they all refer to the same raw material extraction and plant manufacturing datasets.

All the numbers found in the literature have been adapted to the reference impact categories and relative unit, namely:

- GWP [$\text{kgCO}_2\text{eq/ MW}$];
- CED [MJ / MW].

Figure 6-8 GWP (a) and CED (b) comparison results among Li-Ion, CAES and LAES.

From Figure 6-8 it is possible to notice that LAES presents the lowest environmental impact from the point of view of the raw material extraction and the manufacturing phase. In agreement with the literature, CAES has lower impact than Li-Ion even though this might not be necessarily true if the use phase is also taken into account. From a material perspective, despite being the most benign among all the battery technologies, electrochemical storages still involve high embodied energy resources. In fact, as previously mentioned, looking at the use phase, Li-Ion praise the best roundtrip efficiency among all technologies. More in particular, they are almost 30% more efficient than LAES (0.9 for Li-Ion against 0.6 for LAES). On the contrary, the EoL stage is still fairly unexplored and thus, it may again shift the Li-Ion environmental impact to the highest

value. The deployment of rare metals and their assembly status make their disassembly procedure still challenging from an environmental point of view, and more important from an economic point of view, leading to lower recycling rates.

In addition, as mentioned before, the dataset used to model the avoided production of cooling power is relative to a Water-cooled Vapour Compression chiller. Depending on the specific application, and on the specific technology, the generation of cooling power is usually more energy consuming than the electricity generation. Thus, from the opposite perspective, the avoided impact of cooling power could be more effective than the avoided production of electric energy. This is not always true, however, the advantage of LAES is that the ratio between the two outputs namely, electric and cooling energy, can be adjusted according to the specific needs and applications. Such flexibility might declare LAES as the “greenest” technology, however given all the assumptions made in the background, there is a strong motivation in undertaking further studies where the use phase as well as the EoL are included in the comparison.

Such a low environmental impact for LAES can be justified by considering that LAES is realised with well-developed and established components such as compressors, heat-exchangers, expanders and tanks which mainly involve the most common metals. This is also verified in the case of CAES. However, a big difference might be played by the storage cavern which in LAES is replaced by the cryogenic container. The realization of the cavern to store the compressed air might involve consistent deployment of resources as well as production of emissions. Similarly, the cryogenic tank should be properly designed in order to minimize thermal losses. Particularly, for the current study, the cryogenic tank has been modelled with calculated data, as it has been considered as a characterising component. Unfortunately, only generic information regarding the inventory have been found in the literature, therefore it might not be so obvious to define the origin of such gap as in Figure 6-8.

6.5 Summary

The current chapter analysed the environmental performance of LAES by means of LCA. At first, different technical configurations have been compared in order to find out which structure would deliver the lowest environmental impact. Second, such technology has been compared with current competitors namely Li-Ion and CAES. The comparison highlights the massive role played by the use phase: such systems are characterised by a relatively long use phase which annihilate impacts related to the manufacturing of the plant itself. However, to some extent, roundtrip efficiency is not the only driven parameter since the study demonstrates that a well-balanced production of electricity and cooling power can lead to lower impacts.

The analysis demonstrates that despite being still a relatively new technology, LAES has proved to be environmentally the most competitive among the three technologies analysed. A key role is played by the production of cooling power, which in some categories, such as Climate Change, is able to mirror the impact related to the electricity consumption, leading to drastic difference compared to CAES and Li-Ion where cooling power cannot be considered as co-product.

As a final remark, such comparison would need more investigations in terms of available LCA study as well as data from the field. The best case scenario would be to develop a detailed LCA analysis for the three different technologies in order to be aligned at every step. Moreover, a deeper analysis should focus on the use phase as it has been demonstrated that it plays a key role. As previously mentioned, the current study refers to a period of 10 years in which all technologies are supposed to charge during night time and deliver energy at day time. Such scenario should be enlarged by including a longer time span as well as different use configurations.

Chapter 7

Experimental and numerical investigation of a novel High Grade Cold Storage for Liquid Air Energy Storage

As demonstrated in the previous chapters, Liquid Air Energy Storage system efficiency is largely affected by the thermal performance of the sub-thermal energy storages, among which the High Grade Cold Storage is by far the most important one. The objective of the present work is to numerically investigate and compare the thermal behaviors of different novel cryogenic packed beds filled by different Phase Change Materials (PCMs). The performance of the investigated configurations is compared with that of the conventional sensible heat thermal energy storage (SH). For this purpose, a simplified transient one-dimensional numerical model to simulate both the charge and discharge phases of the HGCS system has been developed and validated against experimental results provided by an experimental campaign carried out on a lab scale HGCS at TESLAB@NTU. In addition, a preliminary economic evaluation has been performed in order to assess whether the technical advantage achieved by the introduction of the PCMs is likewise economically feasible.

7.1 Introduction

As concluded in Chapter 4, the performance of the High Grade Cold Storage (HGCS) recovering the cryogenic energy from the evaporation of liquid air is of primary importance for lowering the specific consumption of the liquefaction process and in turn improving the round trip efficiency of the whole LAES system. Figure 7-1, already shown in Chapter 4, highlights this concept showing that, for a defined charge pressure, a decrease of HGCS efficiency from 97% to 10 % leads to $\approx 100\%$ increase of the specific consumption.

Figure 7-1 LAES Performance Map case study. Specific consumption increase by HGCS efficiency degradation.

Chapter 8 ⁸

Liquid Air economy case study – A Dearman Engine application

As mentioned in Chapter 2, one major advantage of nitrogen, as a potential new energy vector, is that, globally, the industrial gases industry has a substantial surplus of production capacity due to less demand for it commercially. This surplus, estimated around 8500 tons/day in the solely UK, could be potentially used to fuel millions of cars daily. At the same time, oxygen production could be used to enhance the efficiency and limit the environmental impact of a Waste-to-Energy plant by means of oxygen enriched combustion. Nevertheless, the electricity required by the Air Separation Unit to generate the oxygen, leads to a penalty in energy efficiency that puts at stake its economic feasibility. In order to overcome that criticality, a further economic revenue opportunity is offered by the possibility to exploit one of the main by-products of the Air Separation Units (i.e. nitrogen) by means of a high efficiency open Rankine-cycle expander, namely the Dearman Engine. The proposed research investigates the feasibility of an integrated system - Waste-to-Energy plant, Air Separation Unit and Dearman Engine - in terms of technical, economic and environmental performance indices.

⁸ This section published substantially as Tafone A, Dal Magro F, Romagnoli A. *Integrating an oxygen enriched waste to energy plant with cryogenic engines and Air Separation Unit: Technical, economic and environmental analysis*. Appl Energy 2018;231:423–32.

8.1 Introduction

The constantly growing worldwide population is leading to a constant increment of waste production [158]. In most developing and developed countries an ongoing challenge is that to collect, recycle, treat and dispose significant quantities of solid waste [158,159]. In this context, Waste-to-Energy (WtE) plants play a crucial role as they convert waste into energy. Among the different technical issues, such as temperature fluctuations of the flue gas [160,161] and high temperature corrosion [162,163], emissions represent one of the main concerns due to the stringent emissions level enforced on WtE plants and to the global trend which focusses on minimizing pollutant emissions [164,165].

A potential solution to reduce emissions consists of adopting a well-established technology in combustion processes: Oxygen Enriched Combustion (OEC). Nowadays, such a technique is mainly adopted in industrial production processes where an oxidant containing higher molar concentration of oxygen than that present in the air, is used to improve the combustion process [166]. The wider adoption of OEC over the last decades is due to several advantages:

- **increase in thermal efficiency:** the losses at the stack are reduced because the mass flow rate of the flue gas decreases as the oxygen molar concentration in the combustion air increases: instead of heating up inert nitrogen, more steam is produced in the coupled Rankine cycle of the WtE plant [166];
- **lower emissions:** OEC generates lower levels of pollutants (e.g. nitrogen oxide) and of products derived from incomplete combustion (e.g. carbon monoxide, aromatic polycyclic hydrocarbons and chlorinated organic compounds) [167,168];
- **improve temperature stability and heat transfer:** increasing the oxygen content allows more stable combustion and higher combustion temperatures that can lead to better heat transfer within the load [166,169];
- **increase productivity:** by means of oxygen enrichment of the oxidant gas, the throughput of the plant can be increased for the same fuel input because of higher flame temperature, increased heat transfer to the load and reduced flue gas [170].

OEC is actually considered one of the most potential technologies for CO₂ capture in power plants. Yin et al. [171] have reviewed pulverized fuels oxy-fuel combustion fundamentals and their recent development with a focus on CFD modeling and systems performance. Hanak et al. [172] have evaluated the techno-economic performance of cryogenic O₂ storage implemented in an oxy-combustion coal-fired power plant as a means of energy storage. The proposed system compensates the average daily efficiency penalty of the system with higher daily profit by 3.8–4.1% only if the carbon tax is higher than 29.1–29.2 €/tCO₂. Xiang et al. [173] have proposed an integrated system of Natural gas combined cycle and oxy-fuel combustion finding a significant increase of the power generation efficiency. Pettinau et al. [174] have compared three different power generation technologies for CO₂-free power generation from coal finding that, although not enough mature for commercial-scale applications, oxy-coal combustion has a relevant future potential due to its relatively low levelized cost of electricity (62.8 USD/MWh_e).

From an industrial perspective, OEC is a well-established practice in the glass [175], steel, iron [176,177] and cement industries [178]; however this is not the case in WtE plants in which the economic penalties associated with the production of oxygen used to enrich the combustion process, overcome the operative and environmental benefits [179]. Even though oxygen enriched combustion leads to higher thermal efficiencies (and hence to higher electricity generation) of the WtE plants, the electricity required by the Air Separation Unit (ASU) to produce oxygen is more than the extra energy produced by the WtE plant, thus resulting in an overall reduction in power supply capacity [180,181]. According to Mathieu [182], an oxy-fuel combustion process applied to power generation systems leads to a penalty in energy efficiency equal to 10-14%. The economic penalty introduced by oxygen enriched combustion in WtE plants has been evaluated by Verdone et al. [183] who computed a net disadvantage in the range of 0.016 - 0.035 €/kg_{waste}, an increase in specific treatment cost of waste mainly caused by the oxygen production cost.

A possible way to enhance the economic feasibility of an integrated plant composed by a WtE plant and an ASU is offered by the opportunity to use the by-products coming from the ASU (mainly nitrogen streams in gaseous or liquid form) [76]. A promising technical

solution is represented by a high efficiency open Rankine-cycle expander, the Dearman Engine [184], which uses liquid air (or liquid nitrogen, LN_2) as main energy vector. The introduction of cryogenic engines running on liquid air could produce substantial economic and environmental benefits to the integrated plant WtE-ASU since it allows monetizing the by-products from the ASU. Indeed the Dearman Engine (DE) could be used in a number of configurations [184]: as the ‘prime mover’ or principal engine of a zero emissions vehicle; combined with an internal combustion engine (ICE) to form a ‘heat hybrid’ engine that converts waste heat from the ICE; or as a ‘power and cooling’ refrigeration unit (TRU). This work tries to propose an innovative integrated system that is based on the integration of Waste-to-Energy plant with Air Separation Unit and cryogenic engines. The comparative analysis aims to highlight whether and how much the integrated systems are technically, economically and environmentally superior over the baseline case study. Two configurations for two different commercial sectors have been analyzed: 1) a cold and power refrigeration unit (DE-TRU) for the transport of frozen goods and 2) a waste heat recovery/ air conditioning unit employed in the public transport (DE-Bus). The analysis has been carried out to assess the technical, economic and environmental feasibility of the two selected configurations (DE-TRU & DE-Bus) coupled with the integrated plant (WtE-ASU). Real data provided by Dearman Engine have been implemented in our model in order to further enhance the real applicability of the results.

8.2 Methodology and approach

8.2.1 The baseline case study: Waste-to-Energy plant and diesel engines

In order to compare and assess the possible benefits introduced with the integrated system WtE-ASU-DE⁹, two different baseline case studies have been considered as described below.

⁹ In the work, the following terminology will be used: ‘integrated plant’ for the WtE-ASU and ‘integrated system’ for the WtE-ASU-DE.

Table 8-1 Assumptions for WtE plant.

Parameters	Values	Unit
Air inlet composition	21% O ₂ , 79 % N ₂	% molar
Waste mass flow	11.6	kg/s
Yearly hours of operation	8000	h/year
Net Power	30.16	MW _e
Thermal efficiency	25.6	%

A model of a WtE plant has been developed by using Aspen Hysys [185]. The main inputs are summarized in Table 8-1: the air is assumed to be composed by a fixed molar composition of oxygen and nitrogen, (21% and 79 % respectively), neglecting the presence of other components (Argon, CO₂, etc.) in minor concentrations. Two configurations of diesel engines have been considered in the study: an auxiliary diesel engine (~19 kW) to power a TRU for frozen good transport and a EURO VI (~200 kW) diesel engine for City-Buses. The air conditioning system of the City-Buses - a water cooled condenser chiller with COP ~ 2 [186] - consumes ~ 25% of the total energy produced by the engine [187]¹⁰. While the 200 kW engine is fuelled with road diesel (or white diesel), the auxiliary engine for TRUs runs on *red diesel* which is a cheaper but sootier combustible. From an environmental point of view, TRUs belong to non-road engine types which emit more air pollution (NO_x and PM) than a modern Euro VI diesel engine since TRUs emissions are effectively unregulated compared to road diesel engine [186].

8.2.2 Description of the integrated system: WtE plant - ASU - DE

In order to increase the oxygen enrichment of the air fed into the WtE plant, a cryogenic ASU has been considered. Depending on the enrichment required by the WtE plant and on the required purity of the ASU products, two different output streams have been considered: a liquid nitrogen and a gaseous oxygen stream. Taking into account the specific consumption of a cryogenic ASU generating liquid nitrogen and gaseous oxygen [28, 29], the values considered in this work have been set to 0.549 kWh_e/kg_{LN2} and 0.37 kWh_e/kg_{O2}

¹⁰ These assumptions have been considered throughout the analysis of the DE-Bus configuration.

respectively. The first value has been evaluated by the European Industrial Gases Association assuming to use the best available Air Separation technology; the specific consumption of gaseous oxygen has been computed by means of Air Liquide technical brochure for a standard air separation unit with a gaseous oxygen capacity of 200 tons per day. The technical assumptions regarding the ASU are summarized in Table 8-2.

Table 8-2 Assumptions for cryogenic Air Separation Unit.

Parameters	Values	Unit	Ref.
Air inlet composition	21% O ₂ , 79 % N ₂	% molar	-
Nitrogen specific consumption	0.549	kWh/kg _{LN₂}	[32]
Oxygen specific consumption	0.37	kWh/kg _{O₂}	[33]
Output stream 1 (Gaseous O ₂ concentration)	99.5 %	% molar	-
Output stream 2 (Liquid N ₂ concentration)	99.999%	% molar	-

As highlighted in the layout of the integrated system proposed in Figure 8-1, once the gaseous oxygen and liquid nitrogen are produced, the oxygen stream is supplied to the WtE plant and mixed with the main air flow whereas the liquid nitrogen is used to run the DE.

Figure 8-1 Layout of the integrated system WtE-ASU-DE.

The DE is a novel cryogenic engine concept driven by the vaporisation and expansion of liquid air or LN₂ to produce high pressure gas that can generate clean cold and power [47]. In fact, besides the mechanical work produced at the shaft, as the liquid air or LN₂ regasifies,

they also give off large amounts of valuable cold, which can be used to provide “free” refrigeration or air conditioning¹¹. The DE cycle (Figure 8-2) requires the use of a heat transfer fluid inside the cylinder of the engine as a source of heat in order to augment the efficiency of the expansion of the liquid air or LN₂ by resembling a nearly isothermal expansion. Ambient or low-grade waste heat is used as an additional energy source for the liquid air or LN₂ in order to enhance the system efficiency [47].

Figure 8-2 Dearman engine process phases [186].

Figure 8-3a shows the DE-TRU configuration, while Figure 8-3b shows the DE-Bus configuration. The DE-TRU configuration provides refrigeration for the frozen goods by two means:

- the latent heat of vaporisation of the LN₂ extracted from the refrigerated compartment (corresponding to approximately 0.101 kWh_c/kg_{LN2} of cooling energy);
- the mechanical work produced by the DE, which is partially used to drive a vapour refrigeration cycle to provide approximately 0.080 kWh_c/kg_{LN2} of cooling energy.

The DE-Bus configuration instead, provides refrigeration only from the regasification of the LN₂ employing the mechanical work produced to partially power the main diesel engine.

¹¹ Nitrogen liquefies at -196°C at atmospheric pressure; during regasification a large amount of cold energy is released which could potentially be used for cooling and/or refrigeration purposes.

Figure 8-3 a) DE-TRU and b) DE-Bus configurations.

8.2.3 Key performance indicators and assumptions

In order to carry out a comparative analysis between the baseline case study and the integrated system, it is necessary to define different performance indices. A deterministic model was developed to simulate the behaviour of the systems. The main purpose of the model is to evaluate the amount of electrical energy produced by the integrated plant, the liquid nitrogen production, the total amount of diesel saved and the economic feasibility of both ASU and DE investments. The first step consists of setting the oxygen enrichment required by the WtE plant. Based on that parameter, it is possible to evaluate both the mass flows of pure gaseous oxygen and liquid nitrogen and as a consequence the electric power input required for the ASU (P_{ASU}) [MW_e]:

$$P_{ASU} = \dot{m}_{O_2} * \beta_{O_2} + \dot{m}_{LN_2} * \beta_{LN_2} \quad (65)$$

where β_{O_2} and β_{N_2} represent the specific power consumption [kWh/kg] for producing gaseous oxygen and liquid nitrogen, respectively, and $\dot{m}$ [ton/h] is the mass flow rate of each product, O_2 and LN_2 .

Therefore, the Net electric power output ($P_{WtE-ASU}$) [MW_e] from the integrated WtE-ASU plant is calculated as the difference between the electric power output of the WtE plant (P_{WtE}) and the electric power input required for the ASU (P_{ASU}):

$$P_{WtE-ASU} = P_{WtE} - P_{ASU} \quad (66)$$

Once the net electric power and the available liquid nitrogen mass flow are computed, the annual net electricity production ($E_{WtE-ASU}-E_{WtE}$) [kWh], and liquid nitrogen production (m_{LN2}) [$kg_{LN2}/year$] can be evaluated considering a WtE plant operation period of 8000 h/year, as specified in Table 8-1.

Number of Dearman Engine Units (N_{DE}) can be computed by dividing the liquid nitrogen daily production ($m_{LN2,daily}$) [kg_{LN2}/day] for the average daily consumption of liquid nitrogen for each DE application (β_{DE}) [$kg_{LN2}/day/unit_{DE}$]:

$$N_{DE} = m_{LN2,daily} / \beta_{DE} \quad (67)$$

Diesel saved ($m_{sav,dies}$) [$kg_{dies}/year$] represents the total amount of diesel saved computed on yearly basis. It is the sum of two components, related to cooling energy ($m_{sav,dies,c}$) and mechanical work ($m_{sav,dies,m}$) produced by the engine:

$$m_{sav,dies} = m_{sav,dies,c} + m_{sav,dies,m} \quad (68)$$

$$m_{sav,dies,c} = Q_c / \beta_{c,dies} \quad (69)$$

$$m_{sav,dies,m} = W_m / \beta_{m,dies} \quad (70)$$

where:

- $Q_c = m_{LN2} * \beta_{c,DE}$: annual cooling energy production [kWh_c] expressed as a function of the liquid nitrogen specific consumption for cooling production ($\beta_{c,DE}$) [kWh_c/kg_{LN2}];
- $W_m = m_{LN2} * \beta_{m,DE}$: annual mechanical work production [kWh_m] expressed as a function of the liquid nitrogen specific consumption for mechanical work production

$(\beta_{m,DE}) [kWh_m/kg_{LN_2}]$;

- $\beta_{c,dies}$ and $\beta_{m,dies}$: diesel specific consumption for cooling production $[kWh_c/kg_{dies}]$ and diesel specific consumption for mechanical work production $[kWh_m/kg_{dies}]$, respectively.

The economic analysis is performed by analysing the economic gap between the baseline case study and the integrated system scenario. Annual incremental income (ΔI_y) [$MUSD$] between the WtE plant and the integrated WtE-ASU plant is calculated as the sum of the following parameters:

$$\Delta I_y = \Delta I_e + \Delta I_{gf} + \Delta I_{LN_2} \quad (71)$$

where the three main economic components that contribute positively (+) or negatively (-) to the annual incremental income over the baseline case study are:

- $\Delta I_e = (E_{WtE-ASU} - E_{WtE}) * ET$: annual incremental income due to electric energy (E) sold to the grid (-)¹² at the current electric tariff (ET);
- ΔI_{gf} : annual incremental income due to the gate fee paid by local authority per ton of waste¹³ (+);
- ΔI_{LN_2} : annual incremental income due to the tons of liquid nitrogen sold to the DE operator (+).

Annual economic savings ($S_{ec,y}$) [$MUSD$] for the company that operates the refrigerated trucks/bus fleet is calculated as the difference between the annual operative costs before and after the introduction of the DE:

¹² The negative impact is due to the ASU that consumes some of the electric energy produced by the WtE plant.

¹³ Gate fee is the payment that the landfills or WtE plants operators receive per ton of waste coming from the local government; the source of this part of subsidy mainly comes from the government and the waste disposal fee charged to local residents [37].

$$S_{ec,y} = C_{dies} * m_{sav,dies} - C_{LN_2} * m_{LN_2} \quad (72)$$

where C_{dies} [USD/kg_{dies}] and C_{LN_2} [USD/kg_{LN_2}] are the costs of diesel and liquid nitrogen respectively.

Capital cost of the cryogenic ASU ($CAPEX_{ASU}$) [$MUSD$] is function of the cost per power unit (CP_{ASU}) [$MUSD/MW_e$] and the electric power input required for the ASU:

$$CAPEX_{ASU} = CP_{ASU} * P_{ASU} \quad (73)$$

while the Incremental Capital cost of the DE fleet ($\Delta CAPEX_{DE}$) [$MUSD$] can be computed as the product of:

$$\Delta CAPEX = \Delta CP_{DE} * N_{DE} \quad (74)$$

where ΔCP_{DE} [$USD/unit_{DE}$] and N_{DE} are the incremental costs per each refrigerated truck/bus and the number of DE required.

The Payback period [$years$] for the WtE-ASU plant is evaluated taking into account the CAPEX index of Eq. (97) and the annual incremental income of Eq. (95) while the Payback period [$years$] for the DE fleet is computed considering the incremental CAPEX of Eq. (98) and the annual economic savings of Eq. (96).

From the environmental perspective, Annual total emissions savings ($S_{em,i,y}$) [$ton/year$] is computed as the difference between the annual emissions of the baseline case study ($Em_{i,base}$) and the integrated system ($Em_{i,int}$) for each specific emission of the i -th pollutant species:

$$S_{em,i,y} = Em_{i,base} - Em_{i,int} \quad (75)$$

In addition to the parameters described earlier, the following assumptions have been made throughout the whole analysis:

- No additional fuel is introduced in the combustion chamber of the WtE plant under OEC operations;
- The economic analysis neglects the social costs linked with CO₂, NO_x and PM emissions [163];
- The CO₂ emissions related with the WtE plant have been considered to be constant for each level of oxygen enrichment¹⁴ ;
- The electric energy required by the ASU must not exceed the net electric energy produced by the WtE plant;
- The red diesel price is assumed to be half the price of the road diesel [76].

8.3 Results and discussion

8.3.1 Technical analysis

In order to analyze the impact of oxygen enrichment on the WtE plant performance, a standard WtE plant has been modeled in Aspen Hysys. The WtE plant model consists of a conversion reactor (i.e. a combustion chamber) coupled with a classic dual pressure Rankine cycle designed for a steam pressure and temperature of 35 bar and 370°C respectively, and a condensing pressure of 0.1 bar. From the technical perspective, the results showed that the integrated plant (WtE-ASU) leads to an overall penalty in terms of net electric power output. In Figure 8-4 the net electric power output has been computed for four different oxygen concentrations: 21%, 23%, 25% and 27%. The first and the last

¹⁴ Since no additional fuel is added in the combustion chamber for waste processing, no fuel savings occurs and therefore any significant reduction of CO₂ does not takes place in our analysis [9].

oxygen concentrations represent the extreme scenarios of the analysis: the baseline case study, in which there is no oxygen enrichment in the combustion process (in other words this corresponds to the case in which there is no ASU) and the quasi-zero net electric power case in which all the net electric energy produced by the WtE plant is almost completely consumed to operate the ASU.

Figure 8-4 Net electric power production of WtE-ASU as a function of oxygen molar concentration.

Another relevant effect linked with oxygen enrichment is represented by the increase of the throughput of the WtE plant. In fact, as mentioned earlier, the higher temperature associated with OEC enhances the heat transfer to the load (i.e. to the waste being incinerated) thus increasing the waste processing rate through the combustion chamber. Since the WtE plant model developed is not able to capture the increase of mass flow with the oxygen enrichment, the increased capacity has been estimated taking as reference the work carried out by Melo et al. [181], in which it was estimated that the rate of waste being incinerated can be increased up to 60% for an oxygen concentration of 30%. Assuming a linear dependence between rate of waste and the oxygen enrichment, the increase of the waste being incinerated has been computed by means of a linear interpolation and the results are given in Figure 8-5.

Figure 8-5 Rate of waste being incinerated as a function of oxygen molar concentration.

As per the by-product of the ASU (i.e. the LN_2), this is used to satisfy the demand of refrigerated trucks or buses. The type of vehicles and the parameters used in the energy analysis are summarized in Table 8-3. Assuming a utilization factor of 100% for the liquid nitrogen produced by the ASU, the number of vehicles powered by the DE and the tons of diesel annually saved have been computed and reported in Figure 8-6 for three different oxygen molar concentrations of the oxidant.

Table 8-3 Assumptions for the energy analysis.

Vehicle	Fuel	Specific consumption		LN_2 consumption [184,186]
		Cooling [184,186,187]	Mechanical work [184,186]	
City Bus	Road diesel	2.27 kWh/kg _{diesel}	3.02 kWh _m /kg _{diesel}	-
TRU 40 ft trailer	Red diesel	2.17 kWh/kg _{diesel}	1.09 kWh _m /kg _{diesel}	-
DE-Bus	Road diesel/ LN_2	0.101 kWh/kg _{LN_2}	0.08 kWh _m /kg LN_2	0.185 ton/day
DE-TRU 40 ft trailer	Red diesel/ LN_2	0.182 kWh/kg LN_2	0.02 kWh _m /kg LN_2	0.275 ton/day

Figure 8-6 Comparison of Dearman engine applied to a) City-Bus and b) 40 ft refrigerated trailer in term of number of units and tons of diesel saved for different oxygen molar concentration.

8.3.2 Economic analysis

In order to carry out the economic feasibility of the integrated system over the baseline case study, the following assumptions have been made (Table 8-4)

Table 8-4 Nominal assumptions for the economic analysis.

Parameter	Value	Unit	Reference
Electric tariff- ET	0.102	USD/kWh _e	[188]
Gate fee	20	USD/ton	[189]
LN ₂ price	0.07	USD /kg	[184]
CAPEX _{ASU}	0.35	MUSD /MW _e	[190]
Red Diesel price	0.7959	USD /kg	[184]
Road Diesel price	1.5918	USD /kg	[191]
ΔCAPEX _{DE- Bus}	7677	USD /unit	[184]
ΔCAPEX _{DE-TRU}	5117	USD /unit	[186]

Figure 8-7 shows the impact of the oxygen molar concentration over the total annual incremental income disaggregated into its various economic components. As we move along the x-axis, for a decrease of the income from the electricity sold to the grid (due to the higher electric energy consumed by the ASU), the integrated plant increases its annual incremental income due to the increased processing capacity (i.e. increase of the income

from the gate fee) and the sale of the liquid nitrogen.

Figure 8-7 WtE-ASU annual incremental income components as a function of oxygen molar concentration.

In order to assess the influence of the ET (USD/kWh_e) and the price of liquid nitrogen (USD/kgLN₂) over the economic feasibility of the integrated plant, a sensitivity analysis has been carried out for both these parameters as illustrated in Figure 8-8. From the figure is apparent that the economic investment is more convenient if the electric energy tariff decreases from its nominal value (0.102 USD/kWh_e) to lower values. In fact, since the annual incremental income depends on three factors¹⁵ - refer to Eq. (95) - as the electric tariff decreases, the annual incremental income due to the electricity sold to the grid increases. The break-even point (i.e. threshold values for the integrated plant economic feasibility) is achieved in the range of 0.11 - 0.13 USD/kWh_e and 0.05 - 0.063 USD/kgLN₂.

¹⁵ In this specific case, the gate fee and the income from LN₂ sale are fixed while the electric tariff is the only variable.

Figure 8-8 WtE-ASU annual savings for $x_{O_2} = 0.25$ as a function of: a) LN₂ price for a defined ET (0.102 USD /kWh_e) and for different gate fees; b) ET for a defined LN₂ (0.07 USD /kg_{LN2}) and for different gate fees.

Another factor that affects the economic feasibility of the integrated WtE-ASU plant is represented by the so-called liquid nitrogen utilization factor, defined as the ratio between the tons of liquid nitrogen required by a potential fleet of DE over the total produced by the ASU. Figure 8-9 highlights that the integrated plant produces positive annual incremental income only for a utilization factor higher than approximately 83%; however, in order to obtain an economically viable investment the utilization factor should be at least higher than 87% thus allowing to achieve a payback period below 10 years.

Figure 8-9 WtE-ASU incremental annual savings and payback period as a function of LN₂ utilization factor for different oxygen molar concentrations for a defined gate fee of 20 USD/ton.

Finally, under the hypothesis of 100% LN₂ utilization factor, the economic feasibility of the combined investment (ASU + DE) has been investigated. In Figure 8-10 and Figure 8-11 the annual economic savings, incremental income and the payback period of the investment is compared for each configuration (either DE-TRU or DE-Bus) in order to evaluate the threshold values of LN₂ and diesel price. The LN₂ price has two opposite effects as shown by the trend of the dashed lines in Figure 8-10; indeed, high values of LN₂ may guarantee positive annual incremental income for the integrated plant but at the same time it may limit the opportunity to consider the DE. Figure 8-9 shows that the minimum price of LN₂ which guarantees positive incremental income for the integrated plant is around 0.06 USD /kg_{LN2} (if a gate fee of 20 USD/ton is considered, as indicated in Table 8-4. Figure 8-10 shows that an optimum LN₂ price range which allows payback periods below 5 years for both the DE configurations and the integrated plant, falls between 0.065-0.070 USD /kg_{LN2}. Nevertheless, taking into account the (likely) longer life of ASU compared to DE [192,193], it would be advisable to operate with LN₂ price closer to the lower end of the price range (~ 0.065 USD /kg_{LN2}) in order to guarantee an economic advantage towards the DE investment. In fact, without the economic feasibility of the DE,

the WtE-ASU is deprived of its commercial counterpart to which to sell the LN_2 by-product. In addition to this, due to the lower price of red diesel (half price of road diesel), the DE-TRU configuration is the most sensitive to the increase of LN_2 price variation: at 0.08 USD / kg_{LN_2} the annual savings approach the zero value.

Figure 8-10 WtE-ASU, DE-TRU and DE-Bus incremental annual savings and payback period as a function of LN_2 price for $x_{\text{O}_2}=0.25$ for a defined gate fee of 20 USD/ton and diesel price of 1.5918 USD/ kg_{dies} .

Finally, the combined effect of the diesel and the LN_2 prices on the DE annual economic savings and payback period is shown in Figure 8-11, where three different prices for road diesel [194] (e.g. Singapore, USA and EU) are marked with red and blue dots. For the assumed nominal liquid nitrogen cost (0.07 USD/ kg_{LN_2}), the DE-Bus configuration shows that very low road diesel price (≈ 1.08 USD / kg_{dies}) leads to longer (>20 years) payback period as compared with the higher fuel price (larger than 1.3 USD / kg_{dies}). As a result, the European market seems to be the most favorable for the DE penetration due to the highest red and road diesel price that leads to attractive payback period inferior to 5 years. Nevertheless, considering the steep gradient of the payback period curves, especially remarkable for DE-Bus, a 10% increase of road diesel cost, leads to significantly higher

annual economic savings with payback period below 10 years. The payback periods of both DE configurations have also been compared with those computed in the business case study carried out by Strahan [184]. In fact, by taking into account the red and road diesel prices of 0.90 USD / kg_{dies} and 1.80 USD / kg_{dies} as well as a LN₂ price of 0.73 USD/ kg_{dies} [184], a good agreement could be achieved between the work done by Strahan [184] and the current work, which predicts a slight overestimation of the payback period referred to the DE-TRU.

Figure 8-11 DE-TRU and DE-Bus payback period as a function of Diesel price for $x_{O_2}=0.25$ and for LN₂ price of 0.07 USD / kg_{LN2}.

8.3.3 Environmental analysis

In integrated plants, despite the penalty efficiency introduced due to the ASU, the OEC could lead to several environmental benefits. Amongst the others the increase of incineration capacity, the removal of pollutants caused by a more complete reactive combustion in presence of high oxygen concentration and the decrease of the flow rate of the flue gas that in turn leads to lower specific emissions. Besides the pollutant reduction related with OEC, a substantial environmental benefit is achieved with the avoided diesel

consumption substituted by the LN_2 .

Although the possible presence of hazardous pollutants such as polychlorinated biphenyls (PCBs) and principal organic hazardous constituents (POHCs) in the emitted flue gas of the waste incinerator, the present analysis will focus only on three pollutant species, CO_2 , NO_x and PM.

The emissions factor [g/kgwaste] for the different oxygen enrichments are computed by interpolating the results obtained by Verdone et al. [183] for the extreme cases of 0 and 100 % oxygen concentrations; the assumptions on the Diesel engines emissions are based on the works carried out in [184,186]. The comparison between the results related with the emissions reduction of the pollutants and the baseline case study are shown in Figure 8-12 for both the DE configurations.

Figure 8-12 Integrated system annual emissions reduction for different oxygen molar concentrations and DE configurations allocated for the subsystem analysed.

The introduction of DE is the main driver of the integrated plant emissions reduction. Although both DE configurations allow achieving a substantial limitation of the emissions, in this context the emissions reduction associated with the DE-TRU is predominant due to

the less severe regulations for *non-road mobile machinery* as compared with the normative involving road diesel engines. Indeed, the lack of an effective regulation for TRU allows to save as high as 140 ton/year of NO_x and 17 ton/year of PM. In addition to this, it is worthwhile noting that in the case of the DE-TRU, the CO₂ emissions reduction (ranging from 9 to 23 ktonCO₂/year) is due to both the reduction of diesel consumption and leaks of HFC refrigerant gases employed for the TRU cooling unit; these account approximately for 80% and 20% of the total greenhouse emissions of the vehicle, respectively.

8.4 Summary

In this work, an energetic, economic and environmental analysis of an integrated system (WtE-ASU-DE) was studied. The baseline case study (WtE - Diesel engines) and the integrated system have been described in detail and the assumptions adopted have been highlighted as well as the main key performance indices. The energy analysis confirms that the integrated plant (WtE-ASU) solution leads to a penalty in thermal efficiency due to the Air Separation Unit power consumption; on the other hand, if the by-product of the Air Separation Unit is entirely sold to a commercial company operating the Dearman Engines, the analysis shows the possibility to daily save a substantial quantity of diesel as high as 34 kton_{dies}/year. From the economical perspective, the disadvantage introduced with the Air Separation Unit may be compensated by the income coming from the sale of liquid nitrogen. More in particular, the analysis showed that the integrated plant produces positive annual incremental income only for a utilization factor higher than approximately 83%. Analysing the diesel price influence on the Dearman Engine economic feasibility, the analysis showed that very low diesel prices (≈ 1.08 USD/kg_{dies}) lead non-positive annual economic savings. Nevertheless, the sensitivity analysis has shown that even a small decrease ($\approx 10\%$) of liquid nitrogen and/or diesel prices may lead to attractive economic scenarios leading to payback periods below 10 years. Finally, substantial environmental benefits are achieved by means of Air Separation Unit and Dearman Engine implementation, especially remarkable for the DE-TRU configuration where the less severe regulations for non-road mobile machinery allows to save several tons per year of NO_x and PM.

Chapter 9

Conclusions and future perspectives

In the conclusion chapter, the main outcomes achieved during the thesis dissertation are resumed along with the research questions stated in Chapter 1. In addition, the main impact and the main limitations of the project will be highlighted as well as the potential future works that can be developed in order to further enhance the research on Liquid Air Energy Storage.

9.1 Summary of the main works

The main goal of this PhD project is the enhancement of LAES system performance by means of the development of different integrated thermodynamic cycle architectures and novel technologies applied to the LAES system providing both electricity and cooling energy for polygeneration purpose. The main driver and focus of this project have been that of working on system level optimization developing a detailed thermo-economic and environmental model of the system in order to identify measures for thermodynamic performance enhancement, cost and environmental impact reduction.

Although research outcomes and steps are by nature not deterministic, it is very important to ensure that meaningful, sound and comprehensive answers are provided to the research questions related to the research gaps stated at the early stage of the research. Those research gaps will be recalled again in this Section with what has been filled by the PhD project highlighting both the methodology behind the research work and, in Section 9.2, what is left to be explored and investigated in potential future works.

In the first stage of the PhD project, in order to deeply comprehend the components and the parameters that affect the most the technical performance of LAES and what the potential actions to improve LAES performance, a thermodynamic analysis of LAES has been carried out by means the development of a steady state model.

First focusing on the charge phase, different liquefaction processes involving recuperative thermodynamic cycles have been compared: Kapitza cycle, with an operating pressure in the range of 40-60 bar and a storage pressure of 8 bar, is proposed as the best configurations among the ones addressed guaranteeing the lowest specific consumption while keeping the complexity of the plant simple and the size small. A potential method to further improve air liquefaction specific consumption have been identified in the recovery of the waste heat discharged to the environment at the intercoolers/aftercoolers during the air compression process.

A good compromise between high level of performance and plant complexity for the design of the discharge phase have been identified in the direct expansion process involving 4 stages expansion with interheating. Both High Grade Cold Storage and High Grade Warm Storage have been identified as crucial components to increase the round trip efficiency from about 15 % to 48 %. In particular, the implementation of the High Grade Cold Storage has demonstrated to sensibly decrease the energy input to the air liquefier with a maximum of ≈ 100 % reduction of the specific consumption, which in turn produces positive effect for the overall round trip efficiency.

In conclusion, further improvement of LAES efficiency could be achieved by means of a combination of more performing components (compressors, expanders and High Grade Cold Storage) and the implementation of a waste heat recovery system to the discharge phase able to increase the overall round trip efficiency. Obviously, the selected areas of LAES development will show to have a considerable impact on the economics of the whole LAES system as well.

This thesis not only has identified an optimal design for LAES, its components and the parameters heavily impacting on LAES technical performance, but it has also presented a systematic methodology to design the system and subsequently make use of it.

The methodology starts by defining the optimized cycle architecture (refer to Chapter 3) and the boundary conditions (hypothesis & constrains) of LAES systems. Subsequently, three macro-scenarios imposing the storage pressures (p_s) and the turbomachinery performances (design/off-design conditions) have been identified.

Based on literature references and the results achieved in Chapter 3, the next step is to identify the variation range of the main operational parameters. These include the charge and discharge pressure, the recirculation fraction of the liquefaction process, the High Grade Warm/Cold Storages factors, namely the ratio between the waste heat/waste cold effectively utilized and the maximum hypothetical waste heat/waste cold discharged during air compression and liquid air regasification. Once the constrained requirements are

defined, parametric performance maps can be elaborated and used to quantify the main LAES key performance indicators (round trip efficiency, liquefaction specific consumption and specific electric power output) as a function of the operational parameters above illustrated. Notably, the maps represent unique guidelines for LAES design under operative parameters variation and serves as a systematic tool for the design of LAES operating in different configurations (full electric and polygeneration). For more information on the parametric performance maps elaboration refer to Sections 4.2-4.3 and Appendix B. In parallel, in order to prove the necessary confidence on the maps, the parametric performance maps have been successfully validate by comparing the performance predictions against the experimental results of the LAES pilot plant operated in Slough (UK) by Highview Power.

As a final step, in order to highlight the immediate applicability of the results and to show how to effectively utilized the maps, those ones have been used to design the LAES for different configurations (full electric and polygeneration) imposing the level of performances required by a potential customer. In addition, in Section 4.4.2, by adopting the maps as a tool for LAES design in polygeneration configuration, an economic case study has been carried out for the economic dispatch of an Eco-building in Singapore. The results has shown the adoption of a 300 kWh capacity LAES produces a higher Net Present Value after 20 years and a shorter time period to obtain the Return of Investment compared to that of Li-ion battery.

As underlined by the thermodynamic analysis carried out in the first stage of the PhD project, the main bottleneck to the deployment of LAES is currently represented by its low value of round trip efficiency which is mainly due to the large amount of energy consumption during the liquefaction process (charge phase) and the inefficient exploitation of the heat discharged during the air compression phase. In order to overcome such an issue, Chapter 5 proposes an innovative LAES solution that is based on the integration of LAES with well-established waste heat recovery solutions (ORC and/or ABS).

The study showed that the utilization of the low-grade waste heat from the compression

phase of a LAES seems to be technologically viable and capable to significantly improve the round trip efficiency of the system by producing additional electrical power output. However, the level of efficiency improvement depends significantly on both the LAES configuration and the waste heat recovery system introduced in the Liquid Air Energy Storage system. Indeed, the most significant results are achieved when LAES is operated in polygeneration configuration adopting both ORC and ABS with a round trip efficiency improvement of 30 %.

From an economic perspective, this results in a significant decrease (up to 10 %) of the Levelized Cost of Storage of the LAES system under opportune conditions. Again, the economic benefit due to the waste heat recovery system integration depends significantly on both the configuration (full electric or polygeneration) and the related round trip efficiency as well as the electricity tariff LAES purchases charging electricity and the number of cycles per year. It is worth noting that for each electricity tariff there exist a threshold value of the number of cycles beyond which the Levelized Cost of Storage of the integrated LAES-Waste Heat Recovery system is lower than the one computed for the stand-alone LAES.

As a last step, the economic viability of the investment has been assessed by also comparing the integrated system with Li-ion batteries. Neglecting the annual electricity charging costs, the analysis has showed that the integrated system has a comparatively lower LCOS (0.16 vs 0.34 €/kWh_e) and, considering potential development for a novel technology as LAES, a larger potential for LCOS reduction is expected in the future.

In order to assess if LAES can be considered an environmentally friendly solution compared to other large-scale energy storage solutions, an environmental comparative performance analysis of LAES, CAES and Li-Ion batteries by means of Life Cycle Assessment has been carried out in Chapter 6 in order to assess which systems would deliver the lowest environmental impact.

The use phase of the EESs has been identified as the main key actor to establish whether

or not those technologies are environmentally friendly. In fact, the relatively long use phase may offsets the impacts related to the manufacturing of the plant itself. However, to some extent, round trip efficiency is not the only driven parameter since the study demonstrates that a well-balanced production of electricity and cooling power can lead to lower impacts.

Focusing the attention on LAES, the analysis demonstrates that despite being still a relatively new technology, it has proved to be environmentally the most competitive among the three technologies analyzed. Notably, the polygeneration scenario is the most interesting one. Indeed, a key role is played by the production of cooling power, which is capable to annihilate the environmental impact related to the lower round trip efficiency and the higher electricity consumption. This unique characteristic of LAES leads to a significant difference compared to CAES and Li-Ion batteries where cooling power cannot be considered as co-product of the discharge phase.

As shown by the novel performance maps elaborated in Chapter 4, Thermal Energy Storages implementation in LAES has shown to have by far the most significant impact on the LAES performance. Indeed, High Grade Cold Storage efficiency impacts by lowering up to $\approx 100\%$ on liquefaction specific consumption. Therefore, innovative HGCS systems based on Phase Change Materials implementation (single and cascade PCMs) is proposed and techno-economically compared to the baseline case configuration (HGCS Sensible Heat material). A mathematical model of the different HGCS configurations describing the heat transfer process between the heat transfer fluid and the storage materials has been developed in Matlab. The models have been validated against experimental results both retrieved from literature review and produced by an experimental campaign carried out on a test rig installed at TESLAB@NTU. The validation has shown that the model can accurately predict both quantitatively and qualitatively the dynamic heat transfer behavior of the High Grade Cold Storage.

The techno economic analysis has shown that the PCM utilization in the HGCS leads to a decrease of the time average specific consumptions with a notable payback period inferior to 5 years. Indeed, the most significant results are achieved by the cascade 2PCM

configuration where the thermal buffer of both PCMs allows to decrease the time average specific consumption of SH configuration by 10 % compensating thus the higher PCM capital costs with the annual savings produced by a lower electricity consumption.

As a final step of the PhD project, the real techno-economic potential of Liquid Air/Nitrogen as a valid energy vector to provide clean cold and power has been thermodynamically, economically and environmentally analyzed.

The large amounts of cold thermal energy wasted (from spare liquid nitrogen and LNG regasification) and the pressing problem of cooling demand increase have led to analyze the potential of cryogenics on grid, transport and cooling applications defining a possible "liquid air economy". For this purpose, an interesting case study involving an integrated system based on the utilization of liquid nitrogen as the main energy vector was studied from techno-economic and environmental perspectives in Chapter 8. An innovative integrated system that is based on the integration of Waste-to-Energy plant with Air Separation Unit and novel cryogenic engines (Dearman Engines) fueled by liquid nitrogen is proposed and technically, economically and environmentally compared to the baseline case study (Waste-to-Energy plant - Diesel engines).

Despite the integrated plant (Waste-to-Energy plant – Air Separation Unit) solution leads to a penalty in thermal efficiency due to the Air Separation Unit power consumption, a significant daily save of diesel as high as 34 kton/year might be achievable if the by-product of Air Separation Unit is entirely sold to a commercial company operating the Dearman Engines. In particular, this results in an economical advantage for the integrated plant compensating the economic losses due to the decrease of net electric power output only for a liquid nitrogen utilization factor higher than approximately 83%. In this context, diesel and liquid nitrogen prices are also key actors in order to establish the economic feasibility of the investment: despite very low diesel prices (≈ 1.08 USD/kg) lead to non-positive annual economic savings, a slight decrease ($\approx 10\%$) of liquid nitrogen and/or diesel prices may lead to attractive economic scenarios leading to payback periods below 10 years. From an environmental perspective, the introduction of Dearman Engine and the linked

significant quantity of diesel saved allow to achieve a substantial reduction of the emissions, especially remarkable for the Transport Refrigeration Unit configuration where the less severe regulations for non-road mobile machinery allows to save several tons per year of NO_x and PM.

9.2 Limitations and future works

Any technical analysis has limitations and well-defined boundaries of applicability. Therefore, it is of primary importance identifying the main restrictions of the PhD project in order to consciously better understand both the methodology and the outcomes of the thesis as well as identify potential areas for further improvements. The main limitations and the potential future research directions, to undertake accordingly, are summarized as follows.

LAES modeling. The most significant limitation of this work stem from the steady-state thermodynamic models implemented throughout the whole analysis. In fact, due to both the intermittent nature of renewable energy sources and the significant temporal variation of electricity demand, LAES should behaves dynamically during most of the operation time. Consequently, a dynamic modeling should be necessary to correctly predict the transient phenomena that should significantly affect the LAES performance.

Environmental analysis. The focus of the environmental LCA-based analysis is to compare the eco-friendliness of a relatively new technology, namely Liquid Air Energy Storage with established storage solutions such as Li-Ion Batteries and Compressed Air Energy Storage. Such comparison would need more investigations in terms of available LCA study as well as data from the field. The best case scenario would be to develop a detailed LCA analysis for the three different technologies in order to be aligned at every step. Moreover, a deeper analysis should focus on the use phase as it has been demonstrated that it plays a key role. As previously mentioned, the current study refers to a period of 10 years in which all technologies are supposed to charge during night time and deliver energy at day time. Such scenario should be enlarged by including a longer time span as well as

different use configurations.

Economic analysis refinement. The economic analysis carried out in order to assess the economic feasibility of LAES integrated systems is based on data published by Highview Power shortly after the LAES pilot plant commissioning. Since these data might represent only an estimation of the capital cost and operative costs due to the novel nature of LAES system, a more precise determination of these parameters should be required. Regarding the preliminary economic analysis of the High Grade Cold Storage with Phase Change Material implementation, reliable specific costs PCM materials for cryogenic application are still needed. In fact, these cost uncertainties may significantly affect the economic feasibility of the proposed configuration. As a final remark, this work assumes energy arbitrage as the only service provided to the grid in order to obtain revenue. An additional revenue opportunity might be represented by the ancillary services, such as Fast Reserve (FR) and Short Term Operating Reserve, to be provided upon the request of the grid operator. The provision of those services might be economically analyzed in future works.

LAES Performance Maps. The current methodology of performance maps analysis can be further refined by applying the design of experiment technique. In addition, it can be easily extended to other types of charge and discharge thermodynamic processes, different storage scales and waste heat recovery solutions applied to the LAES. In addition, taking into account both the dynamic behavior of the system and introducing new economic parameters (such as specific capital cost figures and LCOS), new parametric techno-economic performance maps can be developed.

Experimental investigation. Due to the intrinsic novelty of PCM implementation in cryogenic application, the main limitation is of technical nature. Indeed, currently there is no PCMs with a phase change temperature close to the liquid air storage temperature. Therefore, the lowest phase change temperature has been offered by ‘customized’ PCM formulated and prepared at TESLAB@NTU. Potential improvements in material formulation with lower phase change temperatures may allow to improve the current performance of the PCM-based HGCS. In addition, other experimental studies on some

LAES components, such as the CryoTurbine, which are not commercially available and may need new design and more research and development, might be required.

Numerical High Grade Cold Storage modeling. Although the one dimensional model shows a good agreement with the experimental measures, the following improvements can be suggested:

- *Availability of PCM capsules temperature measures.* The experimental set-up aiming at reproducing the thermal behavior of the PCM High Cold Storage does not allow to measure the temperature inside the PCM capsules enabling to validate the numerical model only by means of the HTF temperature;
- *Multi-cascade PCM High Grade Cold Storage.* The HGCS performance in terms of LAES Specific Consumption might be improved by the implementation of a multiple cascade PCM HGCS in order to enhance the “thermal buffer” effect triggered by PCMs;
- *PCM selection based on optimization algorithm.* The selection of PCMs is crucial in the design of any Thermal Energy Storages. In the present work, the selection has been carried out by simply selecting the PCM. Therefore, a systematic selection procedure of PCMs for HGCS cryogenic application, based on the methodology and the optimization algorithm developed by Xu et al. [195], could be applied.

“Island grids” or “Remote area” LAES case study application. The main idea is that, once a dynamic characterization of LAES has been performed, the model could be applied to remote or isolated areas in tropical region whose cooling and electrical needs could be satisfied for instance by the tight integration of a renewable energy system, diesel engine generator and LAES. Moreover, LAES application to renewable energy systems (CSP or wind farm) seems to be promising in energy storage field due to: 1) the possibility to exploit the wrong time electricity production of wind turbine for liquefaction plant electric consumption 2) the tight integration, both electrical and thermal, that may guarantee LAES with a CSP power plant. Nevertheless, publications tend to concentrate on integration with conventional plant or industrial processes rather than focusing on this topic. Therefore, from our point of view, there is room to study a possible and necessary coupling between renewables and LAES.

APPENDIX A

Publications & Awards

Awards

- 1) Best Paper Award at ECOS 2017. A. Tafone, F. Dal Magro, A. Romagnoli, *Energetic, Economic and Environmental Analysis of an Integrated Waste-to-Energy Cryogenic Air Separation plant*, 30th International Conference on Efficiency, Cost, Optimisation, Simulation and Environmental Impact of Energy Systems, San Diego, California, USA, 2017.

Journal Papers

- 1) Borri E, Sze JY, Tafone A, Romagnoli A, Li Y, Comodi G. LF. *Experimental and numerical characterization of sub-zero phase change materials for cold thermal energy storage*. Appl Energy 2020.
- 2) Borri E, Tafone A, Zsembinszki G, Comodi G, Romagnoli A, Cabeza LF. *Recent Trends on Liquid Air Energy Storage: A Bibliometric Analysis*. Appl Sci 2020;10:2773.
- 3) Tafone A, Ding Y, Li Y, Xie C, Romagnoli A. *Levelised Cost of Storage (LCOS) analysis of liquid air energy storage system integrated with Organic Rankine Cycle*. Energy 2020;198:117275.
- 4) Tafone A, Romagnoli A, Borri E, Comodi G. *New parametric performance maps for a novel sizing and selection methodology of a Liquid Air Energy Storage system*. Appl Energy 2019;250:1641–56.
- 5) Tafone A, Dal Magro F, Romagnoli A. *Integrating an oxygen enriched waste to energy plant with cryogenic engines and Air Separation Unit: Technical, economic and environmental analysis*. Appl Energy 2018;231:423–32.
- 6) Tafone A, Borri E, Comodi G, van den Broek M, Romagnoli A. *Liquid Air Energy Storage performance enhancement by means of Organic Rankine Cycle and Absorption*

Chiller. Appl Energy 2018;228.

- 7) Borri E, Tafone A, Romagnoli A, Comodi G. *A preliminary study on the optimal configuration and operating range of a “microgrid scale” air liquefaction plant for Liquid Air Energy Storage*. Energy Convers Manag 2017;143:275–85.

Conference Papers

- 1) Tafone A, Romagnoli A, Li Y, Chunping X. *Techno-economic study of Liquid Air Energy Storage integrated with Waste Heat Recovery Solutions*. Sustainable Thermal Energy Management International Conference (SUSTEM2019), Hangzhou, China.
- 2) Tafone A, Borri E, Comodi G, Romagnoli A. *Parametric performance maps for design and selection of Liquid Air Energy Storage system for mini to micro-grid scale applications*. Energy Procedia 2019;158:5053–60.
- 3) Mazzoni S, Ooi S, Tafone A, Borri E, Comodi G, Romagnoli A. *Liquid Air Energy Storage as a polygeneration system to solve the unit commitment and economic dispatch problems in micro-grids applications*. Energy Procedia 2019;158:5026–33.
- 4) Borri E, Sze JY, Tafone A, Romagnoli A, Li Y, Comodi G. *An experimental and numerical method for thermal characterization of phase change materials for cold thermal energy storage*. Energy Procedia 2019;158:5041–6.
- 5) Tafone A, Borri E, Comodi G, Van Den Broek M, Romagnoli A. *Preliminary assessment of waste heat recovery solution (ORC) to enhance the performance of Liquid Air Energy Storage system*. Energy Procedia, vol. 142, 2017.
- 6) Borri E, Tafone A, Comodi G, Romagnoli A. *Improving liquefaction process of microgrid scale Liquid Air Energy Storage (LAES) through waste heat recovery (WHR) and absorption chiller*. Energy Procedia, vol. 143, 2017.
- 7) Mengarelli M, Tafone A, Romagnoli A. *Environmental performance of electric energy storage systems: A life cycle assessment based comparison between Li-Ion batteries, compressed and liquid air energy storage systems*. 30th Int. Conf. Effic. Cost, Optim. Simul. Environ. Impact Energy Syst. ECOS 2017, 2017.
- 8) Tafone A, Romagnoli A, Li Y, Borri E, Comodi G. *Techno-economic Analysis of a Liquid Air Energy Storage (LAES) for Cooling Application in Hot Climates*. Energy

Procedia, vol. 105, 2017.

APPENDIX B

LAES parametric performance Maps

Effect of charge pressure and waste cold recovery efficiency on specific consumption for different optimum values of recirculation fraction (design $-p_s = 8$ bar)

**Effect of charge pressure and waste heat recovery on the turbine inlet temperature
(design - $p_s = 8$ bar)**

Effect of discharge pressure and Turbine Inlet Temperature on the specific electric power output (design $p_s = 8$ bar)

Effect of charge pressure and waste cold recovery efficiency on specific consumption for different optimum values of recirculation fraction (off-design - $p_s = 8$ bar)

**Effect of charge pressure and waste heat recovery on the turbine inlet temperature
(off-design - $p_s = 8$ bar)**

Effect of discharge pressure and Turbine Inlet Temperature on specific electric power output (off-design - $p_s = 8$ bar).

Effect of storage pressure on liquefaction specific consumption (design $-p_s = 1.5$ bar)

Round trip efficiency as a function of specific electric power output and liquefaction specific consumption

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