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System analysis of waste heat applications with LNG regasification

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ABSTRACT

The combination of the continuously growing demand of energy in the world, the depletion of oil and its sharp price increase, as well as the urgent need for cleaner and more efficient fuels have boosted the global trade of liquefied natural gas (LNG). Nowadays, there is an increasing interest on the design philosophy of the LNG receiving terminals, due to the fact that the existing technologies either use seawater as heating source or burn part of the fuel for regasifying LNG, thus destroying the cryogenic energy of LNG and causing air pollution or harm to marine life. This investigation addresses the task of developing novel systems able to simultaneously regasify LNG and generate electric power in the most efficient and environmentally friendly way.

Existing and proposed technologies for integrated LNG regasification and power generation were identified and simple, efficient, safe and compact alternatives were selected for further analysis. A baseline scenario for integrated LNG regasification and power generation was established and simulated, consisting of a cascaded Brayton configuration with a typical small gas turbine as topping cycle and a simple closed Brayton cycle as bottoming cycle. Various novel configurations were created, modeled and compared to the baseline scenario in terms of LNG regasification rate, efficiency and power output. The novel configurations include closed Rankine and Brayton cycles for the bottoming cycle, systems for power augmentation in the gas turbine and combinations of options. A study case with a simple and compact design was selected, preliminarily designed and analyzed according to characteristics and costs provided by suppliers. The performance, costs and design challenges of the study case were then compared to the baseline case. The results show that the study case causes lower investment costs and a smaller footprint of the plant, at the same time offering a simple design solution though with substantially lower efficiencies.

Keywords: LNG, regasification, exergy, waste heat recovery, cryogenic

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NOMENCLATURE

e	change in specific exergy	[kJ·kg ⁻¹]
h	specific enthalpy	[kJ·kg ⁻¹]
LHV	lower heating value	[kJ·kg ⁻¹]
p	pressure	[bar]
P	electric power	[MW]
Q	heat	[MW]
s	specific entropy	[kJ·kg ⁻¹ ·K ⁻¹]
T	temperature	[K]
UA	thermal conductance	[kW·K ⁻¹]
$\dot{m}$	mass flow	[kg·s ⁻¹]
ξ	specific exergy	[kJ·kg ⁻¹]
ϕ	specific energy to produce and transport LNG	[kWh·kg-LNG ⁻¹]
η	energy efficiency	[dimensionless]
λ_{LNG}	efficiency of the LNG supply chain	[dimensionless]

Subscripts

a	ambient
C	compressor
CO ₂	carbon dioxide
d	delivery
e	exhaust
EM	electric motor
f	fuel
g	exhaust gas
Gen	generator
GT	gas turbine
I	referring to the first law of thermodynamics
is	isentropic
LNG	referring to LNG supply chain
LNG-HX	referring to LNG heat exchanger
Mech	mechanical
N	nitrogen
NG	natural gas
o	offload
P-LNG	referring to pumping of LNG
PP	power plant
PR-LNG	referring to liquefaction and shipping of LNG
Reg	regasification
s	stack
T	turbine
v	vaporization

Acronyms

CHP	combined heat and power
GT	gas turbine
HRS	heat recovery system
HRSG	heat recovery steam generator
HX	heat exchanger
LHV	lower heating value
LNG	liquefied natural gas
NG	natural gas
NO _x	nitrogen oxides
ORV	open rack vaporizer
PCHE	printed circuit heat exchanger
R&D	research and development
SCV	submerged combustion vaporizer

1. INTRODUCTION

1.1. Background

Liquefied natural gas (LNG) is natural gas converted from gas state into liquid state by cooling it to approximately -163°C at atmospheric pressure. Michael Faraday experimentally explored the liquefaction of natural gas already in the 19th century, though it was Karl Von Linde who built the first vapor-compression machine able to liquefy gas in 1873 [1]. The first commercial LNG liquefaction plant was built in Cleveland (U.S.) in 1939 mainly for peak-shaving reasons, to store surplus natural gas for peak demand that could not be satisfied using the pipeline capacity. In 1959, the first tanker shipment of LNG was undertaken from Lake Charles (U.S.) to Canvey Island (UK), demonstrating that it was possible to transport large LNG quantities over long ocean distances and creating a completely new system of trading energy worldwide [1].

France and the United Kingdom were the first countries to acquire LNG through trade with Algeria in 1964, which has become one of the main LNG world suppliers. Other countries interested in diversifying energy resources adopted this successful concept during the 1970's and 1980's and additional liquefaction and regasification terminals were built in the United States, Europe and Asia. However, since the world's largest reserves of natural gas are located in areas where there is no significant market or where pipelines are not available, the costs for liquefying and transporting LNG were so high that growth of LNG production and demand was very slow until the 1990's [1], [2].

Due to the growing energy demand in the world, the continuous depletion of oil and its sharp price increase, the utilization of liquefied natural gas has been rapidly spreading in recent years. Figure 1 shows a 7.4% yearly growth in LNG demand between 1995-2005. Nowadays LNG has the highest growth of traded volumes among all fuels and a strong increase is expected in the next years [2], [3], [4]. Alone in the United States LNG imports will represent 30% of its natural gas demand in 2008 (imported from Qatar) and are projected to increase fourfold from 2008 to 2030, accounting for more than 75% of U.S. net imports of natural gas in 2030 (Figure 2) [3].

Figure 1. Growth of LNG demand in the world [1], [5].

Figure 2. U.S. net imports of natural gas by source, 1990-2030 [3].

The continuously increasing gap between consumption and production of natural gas worldwide, as well as the urgent need for cleaner and more efficient fuels, have motivated the LNG industry and several governments to boost both LNG liquefaction and regasification capacities all over around the world. Figure 3 illustrates the LNG and natural gas trade movements worldwide in 2006.

For instance, since 1998 the world LNG liquefaction capacity has been doubled, growing from 100 million tons per year to more than 200 million tons [2]. Up until now 26 liquefaction facilities have been built on or offshore in 15 countries [5], [6]. More than 60 regasification terminals have been built in 18 different countries worldwide and approximately 182 LNG projects are under construction or in planning. In the United States alone approximately 113 active LNG facilities exist, of which 57 are peak-shaving plants [6].

Figure 3. LNG and natural gas trade movements in 2006 [5].

1.2. LNG supply chain

The main purpose of liquefying natural gas is to reduce the volume of the gas by approximately 600 times thus making it easier and much more cost-efficient to store and transport over long distances where pipelines are constrained technically, geopolitically or economically [1], [2]. Typically, liquefying natural gas and transporting it overseas becomes cheaper than offshore pipelines for distances larger than 1000 km and cheaper than onshore pipelines for a distance of more than 4000 km, as it is shown in Figure 4 [7].

Figure 4. Natural gas transportation costs [7].

The LNG supply chain comprises four major interdependent stages [1]:

- 1. Exploration and production:** Activities focused on searching, drilling, field development and production of natural gas resources.
- 2. Liquefaction:** LNG is produced through cryogenic refrigeration, removing contaminants to avoid freezing up and damaging equipment when the gas is cooled. Subsequently it is stored at atmospheric pressure in double-wall tanks insulated in the annular section [1]. Liquefaction is a very high energy consuming process: one ton of LNG consumes more than 850 kWh of electric energy, which may represent more than 10% of the produced fuel (including the energy consumption for shipping) [8].
- 3. Shipping:** LNG is transported in ships with double-hull that are carefully designed to avoid spills, leakages and fractures in case of an accident. There are three types of LNG tankers typically transporting about 125-140 thousand cubic meters: spherical design (44% of existing ships), membrane design (51% of existing ships) and structural prismatic design (5% of existing ships) [1].
- 4. Regasification:** The process of heating LNG to retransform it into a gaseous state is called regasification. This step also consumes electric energy, since the LNG needs to be pumped to the desired vaporization pressure passing through various receiving components for a controlled regasification. Currently and with few exceptions, most terminals worldwide use the thermal energy of seawater or other low temperature water sources to regasify LNG [9].

1.3. LNG regasification

Regasification terminals receive LNG from ships, store, pump, vaporize and send LNG into a pipeline for consumption. There are several technologies for LNG regasification which have been implemented or demonstrated in various receiving terminals worldwide:

- Open rack vaporizer (ORV).
- Submerged combustion vaporizer (SCV).
- Closed circuit vaporization.
- Ambient air LNG vaporization.
- LNG vaporization with integrated cooling systems.
- LNG vaporization with integrated power generation.

The most commonly used technology is the open rack vaporizer (ORV), which uses large amounts of seawater as heating source, has high investment costs, but it is very reliable and efficient. The second most widely used option is the submerged combustion vaporizer (SCV), which burns fuel for vaporizing LNG, reducing the send-out capacity and increasing the emissions of the plant but with lower investment costs than the ORV. To a lesser extent, closed circuit systems using suitable working fluids have been employed to vaporize LNG when a heat source is available, in order to minimize environmental impacts and footprint and maximize reliability. The cryogenic energy of LNG can also be integrated with cooling systems.

In comparison to all other methods, the combination of LNG vaporization and power generation results in a decrease in energy consumed in the LNG supply chain, an increase in the net efficiency and a significant reduction of emissions. An overall comparison of the general characteristics, advantages and disadvantages of existing LNG regasification technologies is shown in Table 1 (Chapter 2).

1.4. Motivation

As it has been shown in the previous sections, liquefied natural gas is expected to play an increasingly important role in the global energy market. However, the LNG industry faces a number of challenges that affect its profitability as well as its environmental impact. The processes required in the LNG supply chain, such as the cryogenic refrigeration, shipping, compression and regasification are very high energy consuming processes that demand more than 10% of the produced fuel and thereby reduce the send-out capacity, efficiency and profitability of the LNG [10], [11]. Furthermore, liquefaction and overseas transport significantly reduce the environmental benefits of natural gas, since the life-cycle greenhouse-gas emissions of the supply chain increase [12].

While a lot of research has focused on increasing the efficiency of the liquefaction process, the regasification process still offers a lot of potential for optimization. The vaporization methods employed at existing LNG receiving terminals have focused efforts on achieving high reliability at low investment and operational costs and have not given priority to maximizing the fuel savings and energy efficiency. In spite of being practical and effective, the existing terminals are simultaneously destroying the cryogenic energy of LNG and causing air pollution or harm to marine life by either using seawater as heating source or by burning part of the fuel.

A method for recuperating part of the energy consumed in the liquefaction process during the regasification process is the integration of power production and LNG regasification. Since the 1970's Japan has been operating LNG receiving terminals for power generation using Rankine cycles and direct expansion of natural gas, accounting for about 85 MW installed capacity (2004) with first-law electrical efficiencies between 13-23%¹ [13]. In recent years several alternative methods have been under R&D, reporting theoretical efficiencies up to 60% or more. However, system complexity and lack of commercially proven equipment have contributed to postpone any successful implementation.

Nonetheless, current predictions of the demand for LNG as well as the ever-present discussion on energy efficiency require further investigation into this topic. Identifying the most feasible technologies and configurations, as well as the potential and challenges to exploit the cryogenic energy of LNG to generate electric power, could lead to a significant improvement of the overall efficiency of the LNG supply chain and boost potential applications.

1.5. Objectives

Recognizing an increasing interest on the design philosophy of the LNG receiving terminals, this research addresses the task of developing novel systems able to perform two activities in an integrated, efficient and environmentally friendly way: regasification of LNG and electric power generation.

The main goals of this study are: i) to create, model and simulate novel configurations that integrate commercially available turbomachinery equipment with LNG regasification and compare them to a baseline scenario, ii) to identify the most promising cases, optimize the design, define process equipment and preliminary costing.

This investigation aims to maximize the energy efficiency, fuel savings and send-out capacity of the LNG supply chain, by using the cryogenic properties of LNG to generate electric power with the minimum possible environmental impacts, giving priority to simple, reliable and efficient configurations.

Specific objectives

1. Configure and simulate novel integration of commercial gas turbines, waste heat recovery systems and the regasification of liquefied natural gas (LNG).
2. Establish baseline scenario for existing technology, identify novel configurations and compare them to the baseline.
3. Identify most attractive options in terms of production levels, efficiency, reliability and safety and select one study case for a more detailed analysis.
4. Define design, process equipment, performance and preliminary costing for the selected study case.

¹ See definition of first-law electrical efficiency in section 3.4.2, Chapter 3.

1.6. Method of attack

The method of attack used as a logical framework for this investigation is shown in Figure 5. In this figure, red boxes represent the steps carried out in the investigation, while green boxes represent the criteria for selecting alternatives and the yellow boxes represent data inputs. A literature survey was performed on the existing and proposed technologies for integrated LNG regasification and power generation, in order to identify the state-of-the-art configurations and to find out the most appropriate alternatives in terms of specific criteria: environmental impact, simplicity, electrical efficiency, and compactness (see Chapter 2).

A baseline scenario for existing technology was defined and its details are given in Chapter 3. Based on the scientific background of the literature review, several novel configurations have been created, modeled and simulated using a commercial software for thermodynamics and chemical processes (HYSYS), and have been compared to the baseline scenario in terms of LNG regasification rate, efficiency and power output. These novel configurations and their comparison to the baseline are also described in Chapter 3.

A study case has been selected among the different created configurations using the criteria for selecting alternatives previously adopted: environmental impact, simplicity, electrical efficiency (or power output) and compactness. This study case has been analyzed, preliminarily designed and dimensioned according to characteristics of commercial equipment provided by suppliers. The results for the study case, discussion and data analysis are explained in Chapter 4. Finally, Chapter 5 gathers the conclusions of the research and gives recommendations for further studies.

Figure 5. Method of attack.

2. LITERATURE REVIEW

This chapter presents an overview of the technologies, methods and systems used for vaporizing LNG with emphasis on the integration with power generation systems. Specific data on the most widely used technologies, the state-of-the-art systems and R&D studies from industry and academia have been collected from several types of sources (technical papers, reports, patents) and are summarized and presented. As it was mentioned in the introduction, the main technologies for LNG regasification include:

- Open rack vaporizer (ORV).
- Submerged combustion vaporizer (SCV).
- Closed circuit vaporization.
- Ambient air LNG vaporization.
- LNG vaporization with integrated cooling systems.
- LNG vaporization with integrated power generation.

The general characteristics, advantages and disadvantages of existing LNG regasification technologies are summarized and in Table 1.

Technology	Main characteristics	Advantages	Disadvantages
Open rack vaporizer (ORV)	<ul style="list-style-type: none"> • Uses seawater as heating source • Most commonly used option in the world 	<ul style="list-style-type: none"> • Very efficient, safe and reliable system • Well proven technology • Low maintenance • Low operation costs 	<ul style="list-style-type: none"> • Requires very large amounts of water • Destroys marine life • High investment costs
Submerged combustion vaporizer (SCV)	<ul style="list-style-type: none"> • Burns fuel to heat water that regasifies the LNG • Second most widely used option 	<ul style="list-style-type: none"> • Less expensive than ORV • Quick start up and respond to demand load 	<ul style="list-style-type: none"> • High operational costs and reduction of the send-out capacity • Increase in CO₂ and NO_x emissions
Closed circuit vaporization	<ul style="list-style-type: none"> • A fluid operating in a closed circuit is used to vaporize LNG 	<ul style="list-style-type: none"> • More compact and cheaper design than ORV and SCV • Minimizes environmental impact 	<ul style="list-style-type: none"> • It is constrained by the characteristics of the heat source
Ambient air LNG vaporization	<ul style="list-style-type: none"> • In warm climates ambient air can be used to regasify LNG 	<ul style="list-style-type: none"> • Minimizes environmental impact 	<ul style="list-style-type: none"> • Requires significant footprint, heat transfer area and air volumes
LNG vaporization with integrated cooling systems	<ul style="list-style-type: none"> • Use of LNG cold energy for chilling, cold storage and air conditioning 	<ul style="list-style-type: none"> • Minimizes environmental impact • Improves energy efficiency 	<ul style="list-style-type: none"> • Intermittent cooling supply and operation of the terminal
LNG vaporization with integrated power generation	<ul style="list-style-type: none"> • Use of low temperature of LNG as a cold sink for thermal cycles able to generate power 	<ul style="list-style-type: none"> • Increase in efficiency of the LNG supply chain • Minimizes environmental impact 	<ul style="list-style-type: none"> • Higher investment costs • Increased complexity

Table 1. Comparison of LNG vaporization technologies.

2.1. Open rack vaporizer (ORV)

Open rack vaporizers use ambient seawater (at approximately 15 °C) to regasify LNG from cryogenic temperatures (-160 °C) to a delivery temperature of about 10 °C. The ORV comprises a large number of vertical tubes placed side by side to form a panel through which LNG passes and is vaporized. Seawater is delivered to an overhead distributor at the top of the ORV and it flows downwards, transferring heat to the LNG that flows upwards inside the tubes (Figure 6) [14], [15]. In terms of power consumption, an ORV demands approximately 0.008 kWh/kg LNG to pump the seawater [13]. Aluminum alloy is used as construction material for the ORV since it possesses good mechanical characteristics at low temperatures, excellent thermal conductivity and workability.

In addition, chlorination of seawater is desirable to prevent bio-fouling and marine growth inside the piping, though this leads to severe destruction of marine life and pollution of seawater [14]. ORV is very practical and efficient but it also has high investment costs. Even though ORV have been widely used in Japan and Korea for base load operation, in Europe and U.S. it faces public opposition due to the serious environmental problems related. Furthermore, the large footprint and sensitivity to motions make it unsuitable for offshore environments.

Figure 6. Open rack vaporizer (Kobe Steel Ltd. [15], [16]).

2.2. Submerged combustion vaporizer (SCV)

Submerged combustion vaporizers employ a hot water bath heated by combustion of natural gas with ambient air to regasify LNG (see Figure 7). Hot flue gases are produced by burning natural gas with air and are then injected into the water bath. This water bath transfers heat to submerged tubes through which LNG flows and is vaporized [14], [17]. Stainless steel is used for the tubes and concrete (or sometimes stainless steel as well) is employed for the water bath, since it becomes acidic as the combustion products are absorbed in it [14]. The SCV is able to start up quickly as well as to respond rapidly to variable load demand. Whereas the SCV has no impact on marine life, it increases the plant emissions due to the combustion process. In general, the SCV is less expensive than the ORV, but has higher operational costs and reduces the efficiency of the LNG supply chain.

Figure 7. Submerged combustion vaporizer (Sumitomo Precision Products Co. Ltd. [16], [17]).

A variation of this technology is a combined heat and power facility with SCV (CHP-SCV), which uses the exhaust heat from a gas turbine to reduce the heating effort within the SCV. Water is injected at the top of the stack, is vaporized and subsequently condensed again at a high temperature (see Figure 8). This warm water is then sent to a SCV tank resulting in less fuel consumption by the submerged burners.

Figure 8. Combined heat and power with SCV (CHP-SCV) [18].

2.3. Closed circuit vaporization

When a heat source is available, a suitable heat transfer fluid operating in a closed circuit can be used to vaporize LNG. Commonly, these closed loops use shell and tube heat exchangers that are smaller and cheaper than ORV or SCV [14], [19]. Closed circuits working with a mixture of water and glycol have been proven to be suitable for regasifying LNG. Two terminals operating with this mixture are located in Japan [14], [16] and another terminal is located in Cove Point in the U.S. (see Figure 9). Some of the peak-shaving units operating in the U.S. also use a water-glycol loop to vaporize the LNG in a shell and tube exchanger unit, but these systems use a fired heater to heat the working fluid rather than using an external heat source such as the exhaust from a gas turbine.

Figure 9. Closed circuit working with a mixture of water and glycol (M.W. Kellogg Ltd [16]).

2.4. Ambient air LNG vaporization

In warm climate where large areas are available, an alternative is to use ambient air to regasify LNG. Companies such as Mustang (Figure 10) and Petronet (Dahej regasification terminal in India) have explored this technology [16]. However, there are several challenges to be faced, such as the large heat transfer area and air flow required, as well as the need for an intermediate fluid to transfer the heat indirectly from the air to the LNG in order to avoid any contact for safety reasons.

Figure 10. LNG Smart® air vaporization process (Mustang [20]).

2.5. LNG vaporization with integrated cooling systems

Another possible direct utilization of LNG in terms of energy production include among others: chilling for various process uses, district cooling, cold storage and air conditioning. However, direct utilization of LNG for cooling purposes entails several drawbacks, such as the intermittent operation of the LNG terminal (due to the non-continuous offloading of ships) and the problem of ensuring that cooling peak and base load demand are fully satisfied by the cryogenic energy available at the LNG terminal.

2.6. LNG vaporization with integrated power generation

As mentioned before, nowadays most of the LNG is regasified using the thermal energy of seawater in ORV vaporizers, destroying the cryogenic energy contained in the LNG. Several methods that integrate LNG regasification and electric power generation have been under R&D, in an effort to reduce the significant amount of fossil fuels, electricity and heat required within the LNG supply chain. Methods that take advantage of the cryogenic energy of LNG to generate electric power include:

- Use of LNG as working fluid in direct expansion cycles.
- Use of LNG cryogenic temperature as cold sink for closed Rankine cycles.
- Use of LNG cryogenic temperature as cold sink for open/closed Brayton cycles.
- Use of LNG cryogenic temperature as cold sink for alternative cycles or combinations.
- Use of LNG to improve the performance of conventional power plants.

LNG receiving terminals for power generation have been used in Japan since the 1970's and up to 2004 the installed electric power capacity was about 85 MW (see Table 2) [13]. The selected technologies in these plants are: i) Rankine cycles with organic or inorganic materials as working fluid and ii) direct expansion turbines with and without recuperation and reheating. Electric efficiencies for the mentioned technologies are between 13-23% [13].

Company (site)	Power output (MW)	Mass flow rate of LNG (kg/s)	Type of technology for power generation
Company 1 (A)	8	47.2	Direct expansion + Rankine
Company 1 (B)	7.5	47.2	Direct expansion
Company 1 (C)	3.3	27.3	Direct expansion + Rankine
Company 2	4	27.3	Rankine
Company 3	8.5	41.7	Direct expansion + Rankine
Company 4	6	41.7	Direct expansion + Rankine
Company 5	7.2	36.1	Direct expansion + Rankine
Company 6	5.6	47.2	Regenerative direct expansion

Table 2. LNG power generation facilities in Japan in 2004 [13].

2.6.1. Use of LNG as working fluid in direct expansion cycles

The simplest method to generate power in the LNG regasification process is the direct expansion, as shown in Figure 11 below. This cycle uses an external heat source (commonly seawater) to vaporize LNG in a heat exchanger at high pressure, and then generates electric power by expanding the regasified LNG in a direct natural gas turbine. Depending on the desired delivery temperature, the natural gas may be further heated in a second heat exchanger after the turbine in order to fulfill the temperature requirements. One technique to increase the net electrical efficiency of this system is to include a recuperator, which is a heat exchanger that transfers part of the heat of the natural gas at the turbine outlet to pre-heat the LNG before entering the seawater heat exchanger. The main advantages of expansion cycles are the simplicity, the very low operating costs and the option to generate enough power to cover the pump demand and eventually export a surplus.

Figure 11. LNG vaporization and direct expansion.

2.6.2. Use of LNG cryogenic temperature as cold sink for closed Rankine cycles

The Rankine cycle is a thermodynamic cycle that generates electric power by taking advantage of the different thermal properties of fluids in liquid and gaseous states. It comprises four components: a pump, a heater, an expansion turbine and a condenser. The working fluid in liquid state is compressed by the pump and sent to the heater, where it is converted into saturated or superheated vapor; then passes through a turbine where the fluid is expanded generating mechanical power while its temperature and pressure decrease. Subsequently the fluid at low pressure and temperature passes through the condenser where it returns into the liquid state.

The main advantages of Rankine cycles include very compact and cheap equipment due to the high densities of liquids in comparison to gases, excellent heat transfer with other liquids and gases, high reliability and simplicity. The disadvantages of Rankine cycles result from the physical limitation of liquids to be handled at cryogenic temperatures and pressures lower than atmospheric. This on one hand constricts the net expansion path of the fluid in the turbine (and therefore the electric efficiency) and on the other hand demands a careful design and a sophisticated control system to avoid solidification, leakages and contamination of the fluid.

A vast number of Rankine cycles have been proposed or implemented for LNG regasification, which can be classified into the following categories:

- Simple Rankine cycles (see Table 3).
- Recuperated Rankine cycles (see Table 4).
- Cascaded Rankine cycles (see Table 5).
- Combinations of Rankine cycles (see Table 6).

<p>Characteristics: Electrical efficiencies up to 20-25% depending on working fluid, temperature of heat source and LNG delivery pressure.</p> <p>Advantages: Simple, compact and cheap design. No environmental impact since it is a closed cycle.</p> <p>Disadvantages: Limited efficiency due to the physical restriction of liquids to be handled at cryogenic temperatures and pressures lower than atmospheric.</p> <p>References for LNG regasification:</p> <ul style="list-style-type: none"> • Seawater as heat source and propane Rankine cycle: Liu and You [8]. • Low-grade heat source and propane Rankine cycle: Quiang et Al [22]. • Heated stream of cooling water from a refinery and propane Rankine cycle: Dutt and Ramon, Patent No. WO 2007/140353 A2 [23]. • Refuse incinerator as heat source and water-ammonia Rankine cycle: Kang et Al. [24].
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Table 3. Simple Rankine cycles for LNG regasification.

<p>Characteristics: The working fluid is pre-heated before entering the heater. Electrical efficiencies are higher than for simple Rankine cycles for the same heat source. Reduced capacity to regasify LNG for the same conditions than the simple Rankine cycle.</p> <p>Advantages: Higher electrical efficiencies and power output, smaller pressure ratio (and therefore more compact equipment) than the simple Rankine cycle.</p> <p>Disadvantages: An additional heat exchanger or recuperator is required, which increases the investment cost and complexity of the system.</p> <p>References for LNG regasification:</p> <ul style="list-style-type: none"> • Low-grade heat source and recuperated -propane Rankine cycle: Quiang et Al [22]. • Low-level waste heat recovery, recuperated propane Rankine cycle and recuperated water-ammonia Rankine cycle: Zhang and Bai [25]. • Seawater, simple and recuperated Rankine cycles (and double 3-streams LNG heat exchanger): Matsumoto et Al., Patent No. US4444015 [26]. • Low-grade heat source and recuperated refrigerant Rankine cycle (and 3 streams LNG heat exchanger): Minta and Bowen, Patent No. US6116031 [27].
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Table 4. Recuperated Rankine cycles for LNG regasification.

	<p>Characteristics: The basic concept is to use the thermal energy rejected by a topping cycle to heat a bottoming cycle. Electrical efficiencies are higher than those of simple or recuperated Rankine cycles, while the LNG regasification rate decreases.</p> <p>Advantages: Higher electrical efficiency and power output than by simple or recuperated Rankine cycles.</p> <p>Disadvantages: Additional equipment, controls and complexity in comparison to other Rankine cycles; lower reliability, higher footprint and more expensive systems. Additional safety risks.</p> <p>References for LNG regasification:</p> <ul style="list-style-type: none"> • Seawater as heat source, and cascaded propane-methane Rankine cycles: Liu and You [8]. • Exhaust gas from a GT as heat source and cascaded steam, propane, methane and LNG Rankine cycles: Bisio and Tagliafico [9]. • Seawater, simple and recuperated Rankine cycles (and double 3-streams LNG heat exchanger): Matsumoto et Al., Patent No. US4444015 [26]. • Ambient air as heat source, cascaded propane-ethane Rankine cycles: Gilmore, Patent No. US3018634 [28]. • Seawater-propane Rankine cycle: Hamworthy [29].
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Table 5. Cascaded Rankine cycles for LNG regasification.

	<p>Characteristics: Several possible combinations of simple, recuperated and cascaded Rankine cycles can be configured to increase efficiency and power output.</p> <p>Advantages: Very high electrical efficiency and improved use of cryogenic energy of LNG.</p> <p>Disadvantages: Additional equipment, controls and complexity. Additional investment and safety risks.</p> <p>References for LNG regasification:</p> <ul style="list-style-type: none"> • Low-level waste heat and combination of recuperated propane and water-ammonia Rankine cycles and direct LNG expansion: Zhang and Bai [25]. • Condenser in a steam power plant as heat source recuperated Freon and methane Rankine cycles: Hisazumi et Al [30].
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Table 6. Combinations of Rankine cycles.

	<p>Characteristics: Electrical efficiencies up to 25-30% (simple cycle) depending on the working fluid, temperature of heat source and LNG delivery pressure. Common heat source is the exhaust gas from a gas turbine. This configuration is known as a cascaded Brayton cycle with efficiencies up to 40-45%.</p> <p>Advantages: The gas used in the closed Brayton cycle can be cooled down further than any liquid, therefore the electrical efficiencies are higher than Rankine cycles for the same heat source and cold sink. It is also a simple and standard system.</p> <p>Disadvantages: Because of the low heat transfer coefficients between gas-gas and gas-liquid, very large heat transfer areas, mass flows and volumes are required, resulting in increased investment cost.</p> <p>References for LNG regasification:</p> <ul style="list-style-type: none"> • Cascaded cycle with gas turbine (topping) and closed Brayton cycle (bottoming) using nitrogen, helium or other inert gas: Desideri and Belli [31], Briesch and Feller [32], option of regasified LNG storage by Ciccarelli [33].
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Table 8. Closed simple Brayton cycles and cascaded option for LNG regasification.

	<p>Characteristics: The working fluid is pre-heated in a recuperator before entering the heat recovery system. Electrical efficiencies are higher and LNG regasification rate is lower than non-recuperated Brayton Cycles for the same heat source.</p> <p>Advantages: The compressor operates at lower pressure ratios in comparison to a simple Brayton cycle, therefore it consumes less electric power and it is smaller and cheaper. This leads to a higher net output power and electrical efficiency.</p> <p>Disadvantages: An additional heat exchanger or recuperator is required, which increases the investment cost and complexity of the system.</p>
<p>References for LNG regasification:</p> <ul style="list-style-type: none"> • Hot air from a sintering plant as heat source, and closed recuperated nitrogen Brayton cycle with intercooling: Bisio and Tagliafico [9]. • Solar air collector system as heat source and a closed recuperated nitrogen Brayton cycle: Bisio and Pisoni [34]. • Cascaded open Brayton gas turbine cycle and closed recuperated nitrogen Brayton cycle for regasifying liquid hydrogen: Bisio, Massardo and Agazzani [35]. • Low level heat source and closed recuperated air Brayton cycle with reheating and intercooling (pseudo-Ericsson cycle): Hanawa [36]. 	

Table 9. Recuperated Brayton cycles and cascaded option for LNG regasification.

2.6.4. Use of LNG cryogenic temperature as cold sink for alternative cycles or combinations

In recent years, various configurations or combinations of power cycles for regasifying LNG have been proposed with very high electrical efficiencies (claimed to exceed 60%) and very efficient use of the cryogenic energy of LNG, although up to now none of those systems have been tested, demonstrated or implemented.

Two principal categories have been identified as alternative cycles or combinations of options:

- Combination of Brayton-Rankine cycles for LNG regasification (see Table 10).
- Oxyfuel cycles for LNG regasification (see Table 11).

In particular oxyfuel cycles, which use pure oxygen or an oxygen-enriched gas as fuel oxidizer instead of air, are gaining increasing interest worldwide. The main reason is that these configurations are able to generate power very efficiently with near zero direct CO₂ emissions.

	<p>Characteristics: High complex energy systems that enable the reduction of irreversible thermal losses and maximize the effective recovery of heat from topping cycles.</p> <p>Advantages: Improved recovery of heat from topping cycles and cold energy from LNG that leads to high electrical efficiencies and power output.</p>
<p>Disadvantages: Very complex systems that require additional equipment, control and safety systems. Therefore the reliability of the plant decreases, since it depends on several variables and equipment.</p>	
<p>References for LNG regasification:</p> <ul style="list-style-type: none"> • Combined gas turbine-steam turbine, propane and methane Rankine cycles and use of additional seawater: Bisio and Tagliafico [9]. • Low level heat source, combined closed nitrogen Brayton cycle and closed ammonia-water Rankine cycle: Zhang and Bai [25]. • Combined gas turbine-steam turbine, Rankine cycle using a Freon mixture and methane Rankine cycle: Hisazumi et Al. [30]. • Combined open Brayton gas turbine with closed Brayton or closed Rankine cycle: Briesch [32]. • Topping gas turbine combined with recuperated bottoming Rankine cycle: Mak, Patent No. EP9387A1 [37]. 	

Table 10. Combination of Brayton-Rankine cycles for LNG regasification [30].

Table 11. Oxyfuel cycles for LNG regasification [38].

2.6.5. Use of LNG to improve the performance of conventional power plants

The cold energy of LNG can be used to increase power output and/or efficiency of conventional power plants. Two methods have been identified in the literature:

- Power and efficiency increase in a combined cycle by using the LNG to cool down the recycled water in the condenser (see Table 12).
- Power augmentation of gas turbines and combined cycle power plants by using LNG for inlet air cooling (see Table 13).

Table 12. Combined cycle using LNG to cool down the recycled water in the condenser [31].

	<p>Characteristics: The cold energy of LNG is used to cool down the inlet air of the gas turbine in a simple or combined cycle. Due to lower inlet temperatures the mass flow increases and likewise the compressor and electrical efficiency.</p> <p>Advantages: Higher power output and electrical efficiency (around 50%) than systems without inlet air-cooling (about 45%) [42], [43], due mainly to an increase in air mass flow.</p>
<p>Disadvantages: LNG cannot be regasified during winter due to minimum inlet air temperatures. Close attention needs to be paid to the design of the LNG-air heat exchanger in order to avoid any contact between these fluids. The use of an intermediate working fluid is advisable.</p>	
<p>References for LNG regasification:</p> <ul style="list-style-type: none"> • Methanol-water or glycol-water for inlet air cooling in a GT: Coyle [16]. • Inlet air cooling for a recuperated gas turbine: Briesch and Feller, Patent # US7299619 B2 [32]. • Inlet air cooling for the gas turbine in a combined cycle: Kim and Ro [42]. • Inlet air cooling for a gas turbine using an intermediate cycle: Brenneke et al [44]. 	

Table 13. Power augmentation of power plants using LNG for inlet air cooling [42].

2.6.6. Comparison of LNG vaporization technologies with integrated power generation

A qualitative comparison of the general layout and efficiency of the LNG vaporization technologies with integrated power generation is presented in Table 14. A more detailed quantitative comparison is shown in Table 26 (Appendix 7.1).

Technology	General layout	Efficiency (%)
LNG as working fluid in direct expansion cycles	Simple	-
Simple Rankine cycles	Simple and compact	Up to 25
Recuperated Rankine cycles	Simple and compact	Up to 30
Cascaded Rankine cycles	Complex	Up to 35
Combinations of Rankine cycles	Complex	Up to 40-45
Open Brayton cycles	Complex	Up to 50
Closed simple Brayton cycles	Simple	Up to 25-30
Closed recuperated Brayton cycles	Simple	Up to 30-35
Cascaded Brayton cycles	Simple	Up to 50-55
Combinations of Brayton-Rankine cycles	Complex	Up to 55
Oxyfuel cycles (not commercially proven)	Complex	Up to 70
Power augmentation in conventional power plants	Simple	Up to 50

Table 14. Overall comparison of LNG vaporization technologies with integrated power generation.

3. INVESTIGATION

3.1. Selection of technologies and evaluation

The selection of technologies to be analyzed in this study was based on the principle that a simple design with low environmental impact is required which preferably provides high electrical efficiency and is designed as a compact system. The goal is to identify those alternatives that are able to regasify LNG with integrated power generation without causing harm to marine life, with the minimum possible associated risks and footprint and with maximum reliability and electrical efficiency. Figure 12 explains graphically the method used to pre-select the most appropriate technologies.

Figure 12. Selection of technologies.

The preliminary findings of this selection procedure are listed as follows:

- The use of seawater as heat source was discarded since it causes severe environmental impact.
- Cascaded Rankine and Brayton cycles, combinations of Rankine and Brayton cycles, open Brayton cycles and oxyfuel cycles have not been considered for further analysis, since these alternatives were identified as too complex configurations which require additional equipment, control and safety systems and could seriously affect the reliability and operability the plant.
- Closed Rankine and Brayton cycles were judged as the most appropriate options for this application, since they are simple, efficient and compact. Direct expansion of LNG was considered a suitable technology but just to complement another basic configuration, such as a closed Rankine or Brayton cycle.
- The option of power augmentation in power plants was considered interesting and worthy for further analysis, as efficient and simple solutions are available.

3.2. Boundary conditions

The novel systems of interest in this study should be based on a cogenerative configuration aiming at vaporizing LNG while recovering part of the energy consumed in the liquefaction and shipping stage. The cogenerative configuration should consist of a topping section and a bottoming section, the former operating at higher temperatures and the latter at lower temperatures, as it is illustrated in Figure 13.

The topping cycle should comprise a gas turbine, which generates electric power and uses the exhaust gas to transfer heat indirectly to the bottoming cycle through a heat exchanger. The bottoming section is a closed cycle (for example: Brayton, Rankine or other), using a suitable working fluid, which is able to generate additional power and transfer heat to the vaporizing LNG. Other general boundary conditions are listed as follows:

- The system should be able to operate in a reliable, safe and efficient way on an offshore platform.
- The topping section is based on a typical small gas turbine.
- The system should be able to regasify at least 450 m³/h LNG (about a half of conventional LNG regasification plants).
- The intermediate working fluid used in the bottoming cycle should fulfill international safety, environmental and health standards for offshore operation.
- The temperature of the LNG at the cargo ship is -162°C at atmospheric pressure.
- Natural gas delivery conditions are 13°C and 205 bar.
- Minimum advisable temperature for gas exiting the stack: 70°C , to avoid corrosion issues by condensation.

Figure 13. General layout of a system for LNG vaporization with integrated power generation.

3.3. Characteristics of LNG

LNG is a multicomponent liquid mixture and its physical and thermodynamic properties depend on the component concentrations, pressure and temperature. All the calculations in this investigation consider the LNG as a mixture of the components and concentrations according to Table 15.

Components	Mole fraction
Methane	0.953
Ethane	0.037
Propane	0.005
Nitrogen	0.004
i-Butane	0.001

Table 15. LNG composition.

However, more than 90% of LNG is methane in liquid state. Therefore, it is useful to examine the properties of pure methane for visualizing the overall behavior of LNG. Table 16 presents some properties of pure methane such as the triple point and critical condition, as well as the specific gravity and viscosity in the liquid state. Figure 14 shows a temperature-entropy diagram and a pressure-enthalpy diagram for pure methane.

Critical temperature (°C)	-82.6
Critical pressure (bar)	46
Triple point - temperature (°C)	-182.5
Triple point - pressure (bar)	0.117
Specific gravity @ -162 °C, 1.2 bar	0.424
Viscosity (cP) @ -162 °C, 1.2 bar	0.113

Table 16. Main properties of pure methane.

Figure 14. Thermodynamic properties of pure methane.

The software HYSYS was used to calculate the energy required for vaporizing LNG at different temperatures and pressures (see Figure 15). From this figure, it can be noticed that the thermal energy required for vaporizing LNG decreases either by increasing the vaporization pressure of LNG or by decreasing its delivery temperature.

This calculation agrees with results reported in other studies [8], [31], and shows that by pumping the LNG at supercritical pressure (> 46 bar) it is possible to reduce energy losses in the heat transfer process from the bottoming cycle to the natural gas. For the delivery conditions adopted in this investigation ($13\text{ }^{\circ}\text{C}$, 205 bar) the specific energy required to vaporize LNG is 592 kJ/kg .

Figure 15. Energy required to vaporize LNG at different vaporization pressures and temperatures.

3.4. Baseline scenario

The thermodynamic cycle selected for the baseline scenario is a cascaded Brayton cycle, as shown in Table 17. This configuration takes advantage of the high temperatures of the exhaust gas exiting a gas turbine (around $400\text{--}500\text{ }^{\circ}\text{C}$) and the very low temperatures of LNG (-140 to $-160\text{ }^{\circ}\text{C}$). A closed Brayton cycle using nitrogen as a working fluid serves as a bottoming cycle that simultaneously regasifies LNG and generates electric power.

Table 17. Layout and characteristics of the baseline scenario.

3.4.1. Data input

The set of data input employed to create the thermodynamic model of the baseline scenario include: software and modeling parameters, overall technical assumptions and information on the characteristics of the components and equipment.

Software and modeling parameters:

The design point and the off-design performance of the gas turbine were calculated using gas turbine performance tools. Aspen HYSYS was employed to model the bottoming nitrogen cycle. HYSYS is a software for simulation of chemical and energy processes. It includes tools to estimate the physical properties of fluids and the heat and material balance for a wide range of fluids as well as a simulation of various types of equipment. In this study, the Peng-Robinson fluid property package was used to simulate the processes.

Overall technical assumptions:

The technical assumptions include: the composition assumed for the liquefied natural gas (see Table 15) and for the exhaust gas exiting the gas turbine (see Table 18).

Components of exhaust gas	Mole fraction
Nitrogen	0.7485
Oxygen	0.1371
Water	0.0729
Carbon dioxide	0.0325
Argon	0.0089

Table 18. Composition of exhaust gas.

Information on the characteristics of the components and equipment:

The various parameters assumed for the components of the baseline scenario are summarized in Table 19.

Components	Information
Gas turbine	Ambient conditions: 15 °C, 80% relative humidity.
	Rated power: 43 MW, assumed generator efficiency: 98%, net power: 42 MW, electrical efficiency: 41%.
	Exhaust gas conditions: 125 kg/s, 450 °C.
Heat recovery system (HRS)	Pressure of exhaust gas: 1.01 bar, estimated pressure drop on nitrogen side: 3%.
	Stack temperature: 76 °C, pinch temperature: 27 °C.
Nitrogen compressor	Assumed isentropic efficiency: 80%, mechanical efficiency: 98%.
Nitrogen turbine	Assumed isentropic efficiency: 85%, generator efficiency: 98%.
LNG heat exchanger	Pinch temperature: 5.5 °C.
	Assumed pressure drop on nitrogen side: 8.5%, Assumed pressure drop on LNG side: 1.5%.
LNG pump	Assumed efficiency: 70%, driven by electric motors with assumed 96% electrical efficiency.

Table 19. Information on components and equipment.

3.4.2. Thermodynamics of baseline scenario

The topping cycle of the baseline scenario comprises a gas turbine, which compresses air at ambient conditions and then mixes it with natural gas. The pressurized air and fuel mixture is ignited in a combustion chamber achieving very high temperatures then the combustion products expand through a turbine. The design and operation of the gas turbine were not subject to changes for the baseline scenario². The ambient conditions for operation were defined at 15 °C and 80% relative humidity. The rated power generated by the gas turbine ($[P_{GT}]_{Rated}$) was calculated using gas turbine performance tools. The net power of the topping cycle ($P_{Topping}$) is defined as the rated power generated by the gas turbine multiplied by the generator efficiency (η_{Gen}), as it is shown in Eq. 1.

$$\text{Eq. 1} \quad P_{Topping} = \eta_{Gen} \cdot [P_{GT}]_{Rated} = 42 \text{ MW}$$

The energy consumed by the GT is defined as the product of the fuel mass flow ($\dot{m}_f$) and the lower heating value of the natural gas (LHV_f). The first law efficiency of the gas turbine (η_{GT}) can then be calculated as the ratio between the net power of the topping cycle and the energy consumed in the process, as expressed in Eq. 2.

$$\text{Eq. 2} \quad \eta_{GT} = \frac{\text{Power}_{Topping}}{\text{Fuel Energy}} = \frac{P_{Topping}}{\dot{m}_f \cdot LHV_f}$$

After the turbine, the combustion products pass through a heat exchanger (Heat Recovery System = HRS) transferring heat indirectly to the nitrogen, the working fluid of the bottoming cycle. The heat available in the exhaust gas is inversely proportional to the stack temperature³ (T_{Stack}), which is limited to 76 °C to avoid potential corrosion issues. Assuming no heat losses, the heat transferred from the exhaust gas to the nitrogen (Q_g) is then a fixed parameter defined as the product of the gas flow ($\dot{m}_g$) and its change in specific enthalpy between the gas turbine outlet and the stack (Δh_g), see Eq. 3.

$$\text{Eq. 3} \quad Q_g = \dot{m}_g \cdot (h_s - h_e) = \dot{m}_g \cdot \Delta h_g = 52 \text{ MW}$$

This process can be also expressed in terms of the heat absorbed by the nitrogen, as the product of the mass flow of nitrogen ($\dot{m}_N$) and its change in specific enthalpy between inlet and outlet of the HRS ($h_3 - h_2$), as shown in Eq. 4.

$$\text{Eq. 4} \quad Q_g = \dot{m}_N \cdot (h_3 - h_2)$$

² To see the performance of the gas turbine at variable ambient conditions refer to section 7.2.

³ For more information on the heat available in the exhaust gas, see Figure 47.

From Eq. 4 the mass flow of nitrogen follows

$$\text{Eq. 5} \quad \dot{m}_N = \frac{Q_g}{h_3 - h_2}.$$

The maximum temperature and pressure of the nitrogen cycle (T_3 , p_3) are assumed to be fixed parameters: $T_3 = 430$ °C, $p_3 = 130$ bar. Thus, the specific enthalpy of nitrogen at the turbine inlet (h_3) can be easily determined. The heated nitrogen expands through the turbine and subsequently is sent to a heat exchanger, which transfers heat indirectly to a stream of LNG, converting it from liquid into gaseous state for delivery. At the same time, the nitrogen is cooled down at a low pressure (p_1) and is then pressurized and sent back to the heat recovery system completing the cycle. Since the stack temperature is known, the temperature of nitrogen at inlet of the heat recovery system (T_2) is determined by the pinch of the heat recovery system (ΔT_{HRS}), see Eq. 6.

$$\text{Eq. 6} \quad T_2 = T_{Stack} - \Delta T_{HRS}$$

The pressure of the nitrogen at inlet of the heat recovery system (p_2) depends on the maximum pressure of the nitrogen (p_3) and the pressure losses in the heat recovery system (Δp_{HRS}), according to Eq. 7.

$$\text{Eq. 7} \quad p_2 = p_3 - \Delta p_{HRS}$$

As Eq. 6 and Eq. 7 determine the temperature and pressure of nitrogen, then the specific enthalpy of nitrogen at the inlet of the heat recovery system (h_2) can be calculated. Note that all the parameters on the right side of Eq. 5 are known and therefore it is possible to derive the mass flow of nitrogen, which for the given assumptions is 120 kg/s.

The thermodynamic power of the turbine is defined as the product of the mass flow of nitrogen and its change in specific enthalpy in the turbine ($h_3 - h_4$), as described in Eq. 8. Note that the change in specific enthalpy of nitrogen in the turbine is defined as the product of the isentropic efficiency of the turbine (η_T) and the reversible change in specific enthalpy ($h_3 - h_{4-is}$).

$$\text{Eq. 8} \quad P_T = \dot{m}_N \cdot (h_3 - h_4) = \dot{m}_N \cdot \eta_T \cdot (h_3 - h_{4-is})$$

Further, the compressor power is defined as the product of the mass flow of nitrogen and its change in specific enthalpy in the compressor ($h_2 - h_1$), as presented in Eq. 9. The change in specific enthalpy in the compressor is calculated as the ratio between the reversible change in specific enthalpy ($h_{2-is} - h_1$) and the isentropic efficiency of the compressor (η_C).

$$\text{Eq. 9} \quad P_C = \dot{m}_N \cdot (h_2 - h_1) = \dot{m}_N \cdot \left(\frac{h_{2-is} - h_1}{\eta_C} \right)$$

Eq. 10 determines the thermodynamic power required for pumping the LNG (P_{P-LNG}), $\dot{m}_{LNG}$ being the mass flow of LNG and ω_{LNG} being the specific work for pumping LNG⁴. The LNG pumps are driven by electric motors, which have a corresponding electrical efficiency (η_{EM}).

$$\text{Eq. 10} \quad P_{P-LNG} = \omega_{LNG} \cdot \dot{m}_{LNG}$$

The compressor and turbine are mounted on the same shaft coupled to an electric generator. A mechanical efficiency (η_{Mech}) and a generator efficiency (η_{Gen}) were assumed in order to take into account the power losses due to mechanical friction and electric generation. Thus, the net power generated by the bottoming cycle including the power consumption by the LNG pumps, can be expressed as

$$\text{Eq. 11} \quad P_{Bottoming} = [\eta_{Gen} \cdot \eta_{Mech} \cdot (P_T - P_C)] - \left(\frac{P_{P-LNG}}{\eta_{EM}} \right).$$

Combining Eq. 9, Eq. 10 and Eq. 11 to form Eq. 12, the net power generation of the bottoming cycle can be redefined as

$$\text{Eq. 12} \quad P_{Bottoming} = \eta_{Gen} \cdot \eta_{Mech} \cdot \dot{m}_N \cdot \left[\eta_T \cdot (h_3 - h_{4-is}) - \left(\frac{h_{2-is} - h_1}{\eta_C} \right) \right] - \left(\frac{\omega_{LNG} \cdot \dot{m}_{LNG}}{\eta_{EM}} \right).$$

The heat transferred from the nitrogen cycle to the LNG for vaporization is defined as the product of the mass flow of nitrogen and its change in specific enthalpy between inlet and outlet of the LNG heat exchanger ($h_4 - h_1$) see Eq. 13. The heat of vaporization can be also expressed as the product of the mass flow of LNG and the specific energy required for vaporizing LNG ($h_d - h_v$), which as it was shown in Figure 15 is a function of the LNG vaporization pressure and delivery temperature.

$$\text{Eq. 13} \quad Q_{Vaporization} = \dot{m}_N \cdot (h_4 - h_1) = \dot{m}_{LNG} \cdot (h_d - h_v)$$

From Eq. 13, the mass flow of LNG follows

$$\text{Eq. 14} \quad \dot{m}_{LNG} = \dot{m}_N \cdot \left(\frac{h_4 - h_1}{h_d - h_v} \right).$$

⁴ For more information on the specific work for pumping LNG see Figure 50 in Appendix 7.4.

The specific enthalpy of nitrogen at the turbine outlet (h_4) depends on the minimum pressure of the cycle (p_1), which in turn is a function of the coldest temperature of nitrogen (T_1) at the outlet of the LNG heat exchanger. The coldest temperature of the nitrogen (T_1) is defined as the sum of the LNG temperature entering the heat exchanger (T_v) and the pinch temperature of the heat exchanger (ΔT_{LNG-HX}), see Eq. 15.

$$\text{Eq. 15} \quad T_1 = T_v + \Delta T_{LNG-HX}$$

The pumping of LNG in liquid state results in an increase in temperature proportional to the pressure. Therefore the LNG vaporization pressure determines the actual temperature of the LNG entering into the heat exchanger for vaporization and therefore according to Eq. 15 the coldest temperature of the nitrogen in the bottoming cycle⁵. The temperature of the LNG entering the heat exchanger is defined as the sum of the temperature of LNG that is being offloaded from the cargo ship ($T_o = -162 \text{ }^\circ\text{C}$) and the increase in temperature due to pumping (ΔT_{P-LNG}), as expressed in Eq. 16.

$$\text{Eq. 16} \quad T_v = T_o + \Delta T_{P-LNG}$$

Combining Eq. 15 and Eq. 16 to form Eq. 17, the coldest temperature within the nitrogen cycle can be redefined as:

$$\text{Eq. 17} \quad T_1 = T_o + \Delta T_{P-LNG} + \Delta T_{LNG-HX}.$$

Since both the temperature of LNG being offloaded from the cargo ship and the pinch temperature of the LNG heat exchanger are fixed parameters, the temperature of the nitrogen at the outlet of the heat exchanger depends only on the LNG vaporization pressure (p_v). Therefore, the change in specific enthalpy of the nitrogen in the LNG heat exchanger ($h_4 - h_1$) also depends on the LNG vaporization pressure.

From these observations, it can be derived that the ratio $(h_4 - h_1) / (h_d - h_v)$ in Eq. 14 depends only on the vaporization pressure and delivery temperature of the LNG. As the mass flow of nitrogen is known because it was previously calculated, it can be concluded that the mass flow of LNG for vaporization is influenced merely by the LNG vaporization conditions, as plotted in Figure 16. This figure shows that the mass flow of LNG for vaporization rises by increasing the vaporization pressure or by decreasing the LNG delivery temperature.

The net power output generated in the bottoming cycle is also a function of the LNG vaporization pressure and temperature, as can be seen in Figure 17. It can be observed that the net power of the bottoming cycle decreases sharply by increasing the vaporization pressure, being negative for pressures higher than 400 bar. Further, the net power of the bottoming cycle can rise by increasing the delivery temperature of the regasified LNG, although the effect is smaller than that produced by the vaporization pressure.

⁵ For more information on the influence of the vaporization pressure on the LNG temperature, see Figure 51 in Appendix 7.4.

Figure 16. Influence of the LNG vaporization conditions on its mass flow.

Figure 17. Influence of the LNG vaporization conditions on the net power generated in the bottoming cycle.

The definition of energy efficiency of LNG vaporization systems with integrated power generation is debatable and should therefore be carefully defined in order to properly compare the performances of the various systems. Some studies use the first law of thermodynamics to formulate the efficiency (η_1) as the ratio between the total power output ($P_{Topping} + P_{Bottoming}$) and the energy consumed by the gas turbine, see Eq. 18.

$$\text{Eq. 18} \quad \eta_1 = \frac{\text{Power}_{\text{Total}}}{\text{Fuel Energy}} = \frac{P_{Topping} + P_{Bottoming}}{\dot{m}_{fuel} \cdot LHV_f}$$

However, this definition does not take into account that these systems are generating power by using the LNG as a cold sink and therefore the efficiency may in certain cases be higher than 100%. A more suitable criterion for evaluating the efficiency of these systems, which has been used in various studies on LNG regasification with integrated power generation [9], [24], [39], is the ratio between obtained and consumed exergy (see Eq. 19). Exergy is a recognized term for the quasi-thermodynamic property $\xi = h - T_a \cdot s$, which relates to the available energy of a system based on the conditions at a reference state [45], [46]. When the pressure and temperature of the system are equal to those of the reference state, a thermal and mechanical equilibrium is reached. Commonly the reference state is the ambient condition (T_a). Exergy is considered a quasi-thermodynamic property because it depends on both the state of the system and the specific reference conditions that are employed. In Eq. 19 the obtained exergy is the net power output of the cycle and the consumed exergy is the sum of the energy content of the fuel and the cryogenic exergy of the LNG. The latter one is calculated as the product of the LNG mass flow rate to be vaporized and e_{LNG} , the change in specific exergy between the initial and final states of the LNG evaporation process⁶ as described by Eq. 20.

$$\text{Eq. 19} \quad \eta_{\text{Exergy}} = \frac{\text{Obtained Exergy}}{\text{Consumed Exergy}} = \frac{P_{\text{Topping}} + P_{\text{Bottoming}}}{\left(\dot{m}_{\text{fuel}} \cdot LHV_f \right) + \left(\dot{m}_{\text{LNG}} \cdot e_{\text{LNG}} \right)}$$

$$\text{Eq. 20} \quad e_{\text{LNG}} = (h_o - h_d) - T_a \cdot (s_o - s_d)$$

In Eq. 20, T_a is the ambient temperature: 15 °C, $h_o - h_d$ is the change in specific enthalpy of LNG between offloading and delivery and $s_o - s_d$ is the change in specific entropy of LNG between offloading and delivery. The first law efficiency and the exergy efficiency of the baseline scenario for variable LNG vaporization conditions are shown in Figure 18 and Figure 19 respectively. It is clear from these figures that both the first law efficiency and the exergy efficiency decrease by either increasing the LNG vaporization pressure or decreasing the delivery temperature.

Figure 18. First law efficiency of the baseline scenario for variable LNG vaporization conditions.

⁶ For more information see Appendix 7.5.

Figure 19. Exergy-efficiency of the baseline scenario for variable LNG vaporization conditions.

3.4.3. Performance of baseline scenario

The main performance characteristics of the baseline scenario based on the technical assumptions and LNG vaporization conditions adopted are summarized in Table 20. The temperature versus specific entropy diagram of the bottoming cycle is presented in Figure 20.

Characteristics	Value
Net power - bottoming cycle (MW)	12
Total power (MW)	54
Mass flow of nitrogen (kg/s)	120
LNG regasification rate (kg/s)	60
LNG regasification rate (m ³ /h)	490
First law efficiency (%)	53
Exergy efficiency (%)	46
Generation capacity (kWh/kg-LNG)	0.25
Ratio (fuel consumed) / (LNG regasified)	0.04
Maximum volumetric flow – turbine outlet (m ³ /h)	48600

Table 20. Main performance characteristics of baseline scenario.

Figure 20. Temperature versus specific entropy diagram for the baseline scenario.

3.5. Proposed configurations

Various new configurations were examined in order to improve the performance characteristics in comparison to the baseline scenario. The design philosophy employed to identify possible alternatives is shown in Figure 21, and is based on the requirement of a simple and compact design with low environmental impact and preferably high electrical efficiency. The thermodynamic analysis of the proposed configurations followed guidelines similar to the method used to study the performance of the baseline scenario.

Figure 21. Design philosophy to identify potentially attractive alternatives.

The proposed configurations are the following, which are explained in the next sections⁷:

Improvements or modifications to the baseline scenario:

1. Inlet air cooling for the gas turbine.
2. Inlet air cooling for the gas turbine and supercharging.

Alternative configurations:

3. Rankine cycle.
4. High-efficiency cycle 1.
5. High-efficiency cycle 2.
6. Thermal oil loop (no power generation).
7. Combinations of options.

3.5.1. Inlet air cooling for the gas turbine

Characteristics:

This configuration consists of a gas turbine as a topping cycle and a simple closed nitrogen Brayton cycle as a bottoming cycle, just as for the baseline configuration. The difference of this configuration is that apart from being used as a cold sink for the bottoming cycle the LNG is at the same time used for cooling down the inlet air entering the gas turbine. A special heat exchanger is required, which operates with three streams (LNG, nitrogen and air) as shown in Figure 22. In addition, it is necessary to extract any condensed water after the cooling to avoid damage to the compressor blades in the gas turbine. Therefore a demister should be installed between the heat exchanger and the gas turbine in order to remove water droplets.

Assumptions:

- Annual ambient conditions of an average Mediterranean location (see Figure 48, Appendix 7.3).
- Lower limit of cooled air: 8°C.
- Operating point for the gas turbine: 8°C, 100% relative humidity (since the water contained in the air condenses after the cooling process).

Advantages:

- Increase in power output, electrical efficiency and exhaust gas flow of the gas turbine.
- Increase in power generation in the nitrogen cycle, due to the increase in exhaust gas flow from the gas turbine.
- Increase in LNG regasification rate.

Related issues compared to the baseline:

- Higher complexity, footprint and investment costs.
- More sophisticated design of the LNG heat exchanger to avoid any contact between the air and the natural gas.
- Higher nitrogen mass flow, thermal conductance (UA) and duty of the heat recovery system.

⁷ Various configurations were created, modeled and simulated, but presented lower performance than the baseline scenario. For the sake of brevity, these configurations were not included in the present report.

Figure 22. Inlet air cooling for the gas turbine.

3.5.2. Inlet air cooling for the gas turbine and supercharging

Characteristics:

This configuration is based on the basic inlet air cooling for the gas turbine extended by the option of supercharging, which means increasing the pressure of the ambient air before the cooling process by using a fan (see Figure 23). The effect of increasing the pressure and reducing the temperature of the inlet air is similar to having an additional intercooled compressor, which increases the net power output of the gas turbine [48].

Assumptions:

- Lower limit of cooled air: 8°C.
- Operating point for the gas turbine: 8°C, 100% relative humidity (since the water contained in the air condenses after the cooling process).
- Air pressure after the fan: 1.05 bar.

Advantages:

- Increase in power output and exhaust gas flow of the GT.
- Increase in power generation in the nitrogen cycle, due to increase in exhaust gas flow from the gas turbine.
- Increase in LNG regasification rate.

Related issues compared to the baseline:

- Increased fuel consumption and decrease in GT efficiency.
- Required fan leads to increase in the total investment cost.
- Higher complexity, footprint and investment costs.
- Higher nitrogen mass flow, thermal conductance (UA) and duty of the heat recovery system.

Figure 23. Inlet air cooling for the gas turbine and supercharging.

3.5.3. High-efficiency cycle 1 and 2

Characteristics:

The design of the high efficiency cycles may not be disclosed within this document. Results on the performance will nonetheless be included in the analysis.

Advantages:

- Increase in overall efficiency and power output.

Related issues compared to the baseline:

- Decrease in the LNG regasification rate.
- Higher investment costs, footprint and complexity.

3.5.4. Rankine cycle

As mentioned in Chapter 2, the main advantages of the Rankine cycle include compact and cheap equipment due to the high densities of liquids in comparison to gases, very good heat transfer, high reliability and simplicity. For the special application of LNG regasification located offshore, a careful selection of the fluid operating the Rankine cycle is essential. The working fluid for the Rankine cycle should be able to operate in a broad temperature range, with maximum temperatures at around 400 – 500 °C and minimum temperatures around -150 to -160 °C. As well, this working fluid should not be flammable, toxic or corrosive and it should not pose any danger to the environment. A wide number of working fluids are available for operating Rankine cycles and the technologies are mature, reliable and able to satisfy the current quality and safety standards. However, the majority of these fluids involve different drawbacks that make them unsuitable for offshore LNG regasification (see Table 21).

According to Table 21, the substance with the lowest associated risks to the environment, to the safety plant and human health is carbon dioxide (CO₂). Therefore, carbon dioxide was selected as the working fluid to operate a closed bottoming Rankine cycle. The details of this CO₂ Rankine cycle are presented in the following.

Available fluids	Characteristics
Refrigerant HFC-134a	Non-flammable, asphyxiant, high contribution to global warming (GWP = 1200), it will be banned in the EU from 2011 in all new cars.
Refrigerant R143a	Highly flammable, asphyxiant, high contribution to global warming (GWP = 3800).
Refrigerant R152a	Highly flammable, asphyxiant.
Ammonia	Corrosive, toxic, hazardous, flammable at high temperatures.
Cyclohexane	Highly flammable, irritant, very toxic to aquatic organisms.
Other hydrocarbons	Highly flammable.
Carbon dioxide (CO ₂)	Non-flammable, non-corrosive, non-toxic (in low concentrations).

Table 21. Characteristics of available fluids to operate Rankine cycles [49].

Characteristics:

This configuration consists of a gas turbine as topping cycle and a simple closed Rankine cycle using carbon dioxide as bottoming cycle (see Figure 24). As for the baseline scenario, the heat rejected by the bottoming cycle is used to regasify LNG. The main restriction of the Rankine cycle using carbon dioxide is its triple point (-58 °C), which limits the minimum temperature and pressure of the cycle and thereby the overall efficiency.

Assumptions:

- Isentropic efficiency of the CO₂ pump: 50%.
- Isentropic efficiency of the CO₂ turbine: 75%.
- Pressure loss of the CO₂ in the heat recovery system: 3%.
- Pressure loss of the CO₂ in the LNG heat exchanger: 8.5%.

Advantages:

- Higher LNG regasification rate.
- Lower mass and volumetric flow, smaller equipment (rotary systems, heat exchangers, piping) and therefore cheaper plant.

Related issues compared to the baseline:

- Lower electric power generation.
- Lower electrical efficiency as the minimum temperature is limited by the triple point of CO₂.

Figure 24. Rankine cycle using carbon dioxide as working fluid.

Figure 25. Temperature versus entropy diagram for the Rankine cycle using carbon dioxide.

3.5.5. Thermal oil loop (no power generation)

For the sake of comparison, a configuration using a thermal oil loop was assessed as well. In this system thermal oil absorbs the heat from the gas turbine exhaust and transfers it to the LNG without producing power (see Figure 26). This configuration is able to transfer almost the complete energy available in the exhaust gas to the LNG and therefore very high regasification rates can be achieved.

Other features of this option are listed below:

- Non-flammable, non-corrosive and non-toxic (within a certain temperature and pressure ranges) working fluid.
- Very high regasification rates.
- Simple and compact design, as well as low investment costs.
- Drawback: due to higher regasification rates, the power for pumping the LNG increases.

Figure 26. Thermal oil loop.

3.5.6. Combinations of options

The following combinations of options were also created, simulated and compared to the baseline scenario:

- Combination of high-efficiency cycle 1 and 2.
- Inter air cooling for the gas turbine and simple nitrogen cycle (with supercharging option).
- Inlet air cooling for the gas turbine and high-efficiency cycle 1.
- Inlet air cooling for the gas turbine with supercharging and high-efficiency cycle 1.
- Inlet air cooling for the gas turbine and high-efficiency cycle 2.
- Inlet air cooling for the gas turbine with supercharging and high-efficiency cycle 2.

3.6. Efficiency of the LNG supply chain

As mentioned in previous chapters, part of the produced natural gas is used to supply the energy required in the liquefaction and shipping processes. The specific energy to produce and transport LNG (ϕ_{LNG}) is defined by Eq. 21 [8].

$$\text{Eq. 21} \quad \phi_{LNG} = \frac{\text{Power}_{\text{Liquefaction + Shipping}}}{\text{LNG mass flow}} = \frac{P_{PR-LNG}}{\dot{m}_{LNG}} = 0.85 \frac{\text{kWh}}{\text{kg} - \text{LNG}}$$

The regasification process also consumes energy depending on the LNG vaporization conditions. Thus, the efficiency of the LNG supply chain (λ_{LNG}) is defined as the ratio between the delivered rate of regasified LNG ($\dot{m}_d$) and the natural gas extracted from the field ($\dot{m}_{NG}$), as determined by Eq. 22.

$$\text{Eq. 22} \quad \lambda_{LNG} = \frac{\text{Delivered rate of regasified LNG}}{\text{Produced natural gas}} = \frac{\dot{m}_d}{\dot{m}_{NG}}$$

3.6.1. Efficiency of the LNG supply chain using an open rack vaporizer (ORV) system

For the LNG supply chain using an ORV system, the energy flow is shown in Figure 27. Assuming that the specific work (ω_{LNG}) required to pump the LNG up to 205 bar is 67 kJ/kg (see Figure 50), that the LHV of natural gas is 44194 kJ/kg and that the average efficiency to convert natural gas into electricity (η_{PP}) is 41%, it is possible to derive Eq. 23.

In this equation, $\dot{m}_{PP}$ is the mass flow of natural gas feeding the power plant and $\dot{m}_{LNG}$ is the mass flow of natural gas that is liquefied.

Figure 27. Energy flow of the LNG supply chain using an ORV system.

$$\text{Eq. 23} \quad \phi_{LNG} = \frac{P_{PR-LNG}}{\dot{m}_{LNG}} = 0.85 \frac{kWh}{kg-LNG} \Rightarrow \frac{\dot{m}_{PP} \cdot LHV \cdot \eta_{PP}}{\dot{m}_{LNG}} = 0.85 \frac{kWh}{kg-LNG}.$$

Eq. 24 defines the mass flow of natural gas extracted from the field.

$$\text{Eq. 24} \quad \dot{m}_{NG} = \dot{m}_{LNG} + \dot{m}_{PP}$$

Combining Eq. 23 and Eq. 24, it is possible to derive Eq. 25.

$$\text{Eq. 25} \quad \frac{\dot{m}_{PP}}{\dot{m}_{LNG}} = 0.1678 \Rightarrow \frac{\dot{m}_{LNG}}{\dot{m}_{NG}} = 0.8563$$

Open rack vaporizers (ORV) use seawater to regasify LNG, and therefore require energy only to pump the LNG up to the desired vaporization pressure. The Eq. 26 defines the power for pumping LNG (P_{P-LNG}) as the product of the specific work for pumping (ω_{LNG}) and the mass flow of natural gas that is liquefied. There is a second power plant in the regasification unit, able to convert natural gas into electricity for pumping the LNG with an assumed efficiency (η_{PP}) of 41%. Thus, the power for pumping LNG can also be defined as the product of the efficiency of the second power plant, the mass flow of regasified LNG used to feed the second power plant ($\dot{m}_{Reg}$) and the LHV of the natural gas, as shows Eq. 26

$$\text{Eq. 26} \quad P_{P-LNG} = \omega_{LNG} \cdot \dot{m}_{LNG} = \eta_{PP} \cdot \dot{m}_{Reg} \cdot LHV$$

Substituting the previously mentioned parameters into Eq. 26, it is possible to calculate the ratio ($\dot{m}_{Reg} / \dot{m}_{LNG}$), as shown in Eq. 27. Eq. 28 defines the delivered rate of regasified LNG.

$$\text{Eq. 27} \quad \frac{\dot{m}_{Reg}}{\dot{m}_{LNG}} = \frac{\omega_{LNG}}{\eta_{PP} \cdot LHV} = 0.0037$$

$$\text{Eq. 28} \quad \dot{m}_d = \dot{m}_{NG} - \dot{m}_{PP} - \dot{m}_{Reg}$$

Therefore the efficiency of the supply chain using an ORV system is calculated as:

$$\lambda_{LNG} = \frac{\text{Delivered rate of regasified LNG}}{\text{Produced natural gas}} = \frac{\dot{m}_d}{\dot{m}_{NG}} = \frac{\dot{m}_{NG} - \dot{m}_{PP} - \dot{m}_{Reg}}{\dot{m}_{NG}} = 0.8531.$$

3.6.2. Efficiency of the LNG supply chain using a submerged combustion vaporizer (SCV) system

For the LNG supply chain using a SCV system, the energy flow is shown in Figure 28.

Figure 28. Energy flow of the LNG supply chain using a SCV system.

Assuming the same values as adopted for the LNG supply chain using an ORV system, it is possible to derive Eq. 29 and Eq. 30.

$$\text{Eq. 29} \quad \frac{\dot{m}_{PP} \cdot LHV \cdot \eta_{PP}}{\dot{m}_{LNG}} = 0.85 \frac{kWh}{kg-LNG} \Rightarrow \frac{\dot{m}_{PP}}{\dot{m}_{LNG}} = 0.1678 \Rightarrow \frac{\dot{m}_{LNG}}{\dot{m}_{NG}} = 0.8563$$

$$\text{Eq. 30} \quad P_{P-LNG} = \omega_{LNG} \cdot \dot{m}_{LNG} = \eta_{PP} \cdot \dot{m}_{Reg} \cdot LHV \Rightarrow \frac{\dot{m}_{Reg}}{\dot{m}_{LNG}} = 0.0037$$

SCV systems employ a hot water bath heated by combustion of natural gas with ambient air to regasify LNG. Taking into account that the specific heat required to vaporize LNG at 13 °C and 205 bar (Δh_{LNG}) is 592 kJ/kg, it is possible to calculate the heat required to vaporize the LNG (Q_V), as expressed by Eq. 31.

$$\text{Eq. 31} \quad Q_V = \Delta h_{LNG} \cdot \dot{m}_{LNG} = \dot{m}_V \cdot LHV \Rightarrow \frac{\dot{m}_V}{\dot{m}_{LNG}} = 0.0134$$

Eq. 32 defines the delivered rate of regasified LNG.

$$\text{Eq. 32} \quad \dot{m}_d = \dot{m}_{NG} - \dot{m}_{PP} - \dot{m}_{Reg} - \dot{m}_V$$

Thus, the efficiency of the supply chain using a SCV system is calculated as:

$$\lambda_{LNG} = \frac{\text{Delivered rate of regasified LNG}}{\text{Produced natural gas}} = \frac{\dot{m}_d}{\dot{m}_{NG}} = \frac{\dot{m}_{NG} - \dot{m}_{pp} - \dot{m}_{Reg} - \dot{m}_V}{\dot{m}_{NG}} = 0.8416.$$

3.6.3. Efficiency of the LNG supply chain for the baseline scenario

Figure 29 illustrates the energy flow of the LNG supply chain using the regasification technologies established in the baseline scenario.

Figure 29. Energy flow of the LNG supply chain using the baseline scenario.

As explained in section 3.4.2, the baseline case supplies both the power to the LNG pumps and the heat required for the LNG vaporization. Assuming that there is an electricity grid for trading power, then it would be possible to transmit the excess power generated by the regasification plant to the liquefaction plant. Since the power plant of the regasification terminal operates with a much higher efficiency than the power plant for liquefaction the overall consumption of natural gas within the LNG supply chain is reduced.

Eq. 33 defines the power required for liquefaction and shipping of LNG (P_{PR-LNG}) as the sum of the power generated by the power plant (P_{PP}) and the surplus power generated by the regasification plant ($P_{Surplus}$). It is assumed that the complete excess of power generated by the baseline scenario can be transmitted to the liquefaction plant.

$$\text{Eq. 33} \quad P_{PR-LNG} = P_{PP} + P_{Surplus} = \left(\eta_{PP} \cdot \dot{m}_{PP} \cdot LHV \right) + \left(\eta_{Baseline} \cdot \dot{m}_{Reg} \cdot LHV \right)$$

Dividing Eq. 33 by the mass flow of LNG it is possible to obtain the specific energy to produce LNG (ϕ_{LNG}), see Eq. 34.

$$\text{Eq. 34} \quad \phi_{LNG} = \frac{P_{PR-LNG}}{\dot{m}_{LNG}} = \eta_{PP} \cdot LHV \left(\frac{\dot{m}_{PP}}{\dot{m}_{LNG}} \right) + \eta_{Baseline} \cdot LHV \left(\frac{\dot{m}_{Reg}}{\dot{m}_{LNG}} \right)$$

In Eq. 34, the term $\dot{m}_{\text{Reg}}/\dot{m}_{\text{LNG}}$ for this case is 4%, as shown in Table 20. Therefore it is possible to derive the following equation:

$$\text{Eq. 35} \quad \frac{\dot{m}_{PP}}{\dot{m}_{LNG}} = \frac{\phi_{LNG} - \eta_{\text{Baseline}} \cdot LHV \left(\frac{\dot{m}_{\text{Reg}}}{\dot{m}_{LNG}} \right)}{\eta_{PP} \cdot LHV} = 0.118 \Rightarrow \dot{m}_{NG} = 1.118 \cdot \dot{m}_{LNG}$$

Finally, the efficiency of the LNG supply chain using the baseline scenario in the regasification process is calculated as

$$\text{Eq. 36} \quad \lambda_{LNG} = \frac{\text{Delivered Rate of Regasified LNG}}{\text{Produced Natural Gas}} = \frac{\dot{m}_d}{\dot{m}_{NG}} = \frac{\dot{m}_{NG} - \dot{m}_{pp} - \dot{m}_{\text{Reg}}}{\dot{m}_{NG}} = 0.86.$$

3.6.4. Efficiency of the LNG supply chain for the proposed novel configurations

The efficiency of the LNG supply chain for the various proposed configurations is calculated in the same way than for the baseline scenario, assuming as well that the excess power generated by the regasification plant can be transmitted to the liquefaction plant. The overall energy flow of the LNG supply chain for the novel configurations is illustrated in Figure 30. The results of this analysis are presented in section 3.7.

Figure 30. Overall energy flow of the LNG supply chain for the new configurations.

3.7. Comparison of alternatives

Two comparative analyses were conducted to evaluate the performance of the various alternatives with regard to the baseline scenario. A first overall analysis evaluated the efficiency of the LNG supply chain as well as the first- law efficiency, the exergy efficiency and the LNG regasification capacity of the different alternatives and compared them to the baseline scenario. The most attractive options were pre-selected for a more detailed analysis. An optimization analysis evaluated the performance of these pre-selected options for different stack temperatures, maximum temperatures and pressures of the working fluid operating the bottoming cycle.

3.7.1. First overall comparative analysis

Following the guidelines of the thermodynamic analysis made for the baseline scenario in section 3.4.2, the following performance parameters were evaluated for the proposed configurations: first law efficiency (see Figure 31), exergy efficiency (see Figure 32) and total power output (see Figure 33).

Figure 31. First law efficiency for the proposed configurations compared to the baseline scenario.

Figure 32. Exergy efficiency for the proposed configurations compared to the baseline scenario.

Figure 33. Total power output (GT + bottoming cycle) of the proposed configurations compared to the baseline scenario.

The plot of the maximum volumetric flow of the working fluid of the bottoming cycle versus the power generation in the bottoming cycle is shown in Figure 34. The maximum volumetric flow of the working fluid occurs at the turbine outlet and it is an important parameter, since it will affect the total volume of the regasification facility. The influence of the specific power generation on the efficiency of the LNG supply chain for the different configurations is presented in Figure 35. A detail of the upper area of Figure 35 is shown in Figure 36.

Figure 34. Effect of the maximum volumetric flow of the bottoming cycle on its net power generation.

Figure 35. Specific power generation versus efficiency of LNG supply chain for different configurations.

Figure 36. Specific power generation versus efficiency of LNG supply chain (detail).

The alternatives using inlet air cooling and supercharging were considered less attractive for several reasons. On one hand, the efficiency of the LNG supply, the first-law and exergy efficiencies are lower than the efficiencies of the baseline scenario and the other novel configurations. Further, these configurations add complexity to the regasification system and modify the performance of the gas turbine in the topping cycle. Additionally, a special LNG heat exchanger is required, in order to ensure absolutely no contact between LNG and air to avoid explosions in case of material failure. By using supercharging it is possible to increase the power generation, but both first-law and exergy efficiencies decrease and more fuel is consumed (see Appendix 7.6 for a comparison of the inlet air cooling and supercharging options to the baseline scenario during the year). In addition, the supercharging option requires the use of an expensive fan that increases the investment costs and reduces the reliability of the system.

The high efficiency cycles 1 and 2 were considered the most attractive in terms of electricity production. These alternatives were pre-selected for further analysis, since they can both raise the efficiency of the LNG supply chain as well as the exergy efficiency and the power output of the system without considerably increasing the complexity, footprint or cost of the plant. The CO₂ Rankine cycle and the thermal oil loop were considered the most attractive in terms of LNG regasification and were also selected for further analysis, since these options are relatively simple, compact, cheap and can vaporize up to 50% more LNG than the baseline scenario.

3.7.2. Optimization analysis

The performances of the pre-selected alternatives were analyzed in detail by varying different parameters such as:

- i) Stack temperatures: 40 °C, 76 °C and 90 °C.
- ii) Maximum temperature in the bottoming cycle: 350 °C, 400 °C and 430 °C.
- iii) Maximum pressure in the bottoming cycle: 40 bar, 130 bar.

Since none of these parameters affect the topping cycle, only the performance of the bottoming cycle is further analyzed for the different pre-selected alternatives. The results for the various options at a maximum pressure of 130 bar in the bottoming cycle are shown in Figure 37, the results for a maximum pressure of 40 bar are shown in Figure 38.

Figure 37. Results for the pre-selected alternatives at a maximum cycle pressure of 130 bar.

Figure 38. Results for the pre-selected alternatives at a maximum cycle pressure of 40 bar.

3.7.3. Discussion

Figure 31 and Figure 32 show the tradeoff diagrams of first-law and exergy efficiencies versus regasification rate for the proposed configurations compared to the baseline scenario. Both efficiencies are inversely proportional to the LNG regasification rate and thus the most efficient options have the lowest capacity to regasify LNG. In the same manner, Figure 33 shows that for all the proposed configurations, the total power output (GT + bottoming cycle) is inversely proportional to the LNG regasification rate.

From the optimization analysis it can be concluded that generally for a given stack temperature, there is a performance line that links the point of maximum LNG regasification rate and minimum power with the point of minimum LNG regasification rate and maximum power. All the different alternatives lie along this line. By varying the stack temperature, it is possible to vary the heat absorbed by the bottoming cycle and therefore the LNG regasification rate and power generation. By decreasing the stack temperature, there is an increase in mass flow of the bottoming cycle fluid and in the LNG regasification rate. The power generation in the bottoming cycle is proportional to the maximum pressure and temperature of the working fluid. At the same time, the LNG regasification rate is inversely proportional to these parameters.

The cycle alternatives that modify the operating conditions of the gas turbine in the topping cycle, such as the inlet air cooling and supercharging, may increase the total power output of the plant but also involve a decrease in the first-law efficiency, exergy efficiency and efficiency of LNG supply chain. This is due to the fact that by cooling the inlet air of the GT, the air mass flow increases and more fuel is required to ensure a proper mixture in the combustion chamber, reducing thereby the efficiency of the LNG supply chain. The efficiency of the supercharging option is lower than that of the inlet air cooling, since the former consumes additional power to drive the fan.

The options with an alternative bottoming cycle, such as the high-efficiency cycle 1 and 2, are able to increase the total power output of the plant. At the same time the energy efficiencies (see Figure 31 and Figure 32) and the efficiency of the LNG supply chain (Figure 35 and Figure 36) are higher for these cycles.

The net power output of the CO₂ Rankine cycle is positive, though significantly lower than for the options working with a Brayton cycle. Additionally, the CO₂ Rankine cycle involves lower supply chain and energy efficiencies than the other proposed configurations. However, among all the studied alternatives, the CO₂ Rankine cycle is the option that can regasify LNG and generate power with the lowest mass and volumetric flow (see Figure 34). Since simplicity and compactness are priority parameters for offshore applications, the CO₂ Rankine cycle was considered an interesting option for further investigation.

The regasification system using a thermal oil loop was analyzed in order to visualize the performance of a regasification plant that is not able to generate electric power. As expected the regasification rate for this system is very high, while the energy efficiencies and efficiency of the LNG supply chain are lower than the baseline scenario and the other proposed configurations.

4. STUDY CASE

After recognizing the performance of the various proposed configurations compared to the baseline scenario, the following step in this investigation was to select a study case and perform a preliminary engineering and cost analysis. For this study case all plant components were analyzed and the most relevant parameters were identified. Different vendors were contacted to investigate: the feasibility of the proposed system, the possible challenges, limitations and constraints associated with the design and operation parameters and the costs of the components and requirements for auxiliary or special equipment. General assumptions, technical data and limitations provided by vendors were used to refine the energy model of the study case made in HYSYS. The results of this refined model are presented and the impact and implications are discussed. The refined energy model and the cost data provided by vendors were used to elaborate a preliminary performance, cost and footprint analysis for the study case compared to the baseline scenario.

4.1. Selection of the study case

As study case the closed Rankine cycle using carbon dioxide as working fluid was selected. Among all the studied alternatives it was the option that could regasify LNG and generate power with the lowest mass and volumetric flow in the bottoming cycle, therefore making it the most compact alternative. In addition, the design is simple and should provide a relatively cheap option compared to the baseline scenario and other proposed configurations. Furthermore carbon dioxide is not flammable, toxic or corrosive and produces the minimum possible danger to the environment.

4.2. Analysis of the closed Rankine cycle using carbon dioxide

As mentioned before, the CO₂ Rankine cycle is the system that can regasify LNG and generate power with the lowest mass and volumetric flow among all the studied configurations (see Figure 34). It was considered for further investigation because its simple and compact design could lead to other advantages such as reduction of costs and footprint. Since carbon dioxide is not commonly employed in Rankine cycles and there are very few examples in the literature [50]-[52], challenges are expected for all the components of the cycle. However, limitations in the design and therefore to the cycle are mainly expected from the turbomachinery.

In order to compare the study case with the baseline case in a suitable fashion a consistent reference frame was chosen. For the baseline case within the chosen boundary conditions (see section 3.4, Table 19) equipment performance and cost data is available. Therefore it was intended to apply the boundary conditions used for the baseline scenario also to the study case. These boundary conditions include the maximum pressure and temperature of the bottoming cycle, the stack temperature, as well as the LNG vaporization and delivery conditions. As it is shown in the next sections, due to limited equipment performance, a higher stack temperature had to be chosen resulting in less energy being transferred to the bottoming cycle. Therefore the study case and the baseline scenario lie on different performance lines in the power versus regasification rate tradeoff diagram (see Figure 44 in section 4.3).

4.2.1. Carbon dioxide expander

Manufacturers of expanders were contacted to investigate the feasibility of a carbon dioxide device. According to vendors, no information on expanders working with this fluid is currently available, as usually these expanders operate with hydrocarbons. Common inlet temperatures for expanders working with hydrocarbons are between 400 and 500 °C. However, due to lack of experience on this special application, vendors recommend a limited range of inlet temperatures between 250 and 350 °C. Following a conservative approach it was decided to limit the inlet temperature of the CO₂ to 300 °C, which is substantially lower than the 430 °C used for the nitrogen within the baseline scenario.

The following values were chosen:

- The maximum pressure at the expander inlet is limited to 130 bar.
- The minimum pressure at the expander outlet is limited to 21 bar.
- The isentropic efficiency of the turbine for these conditions is 80%.
- The mass flow of CO₂ is 72.5 kg/s.

In order to keep the energy absorbed by the bottoming cycle the same, the mass flow of CO₂ would have to increase. At this moment in time no information was available on CO₂ expanders for higher mass flows. As a result of the lower temperature but constant mass flow, less energy is transferred from the exhaust gas to the CO₂ stream, leading to a higher stack temperature. This also means that this configuration will lie on a different performance line than the baseline configuration (see Figure 44 in section 4.3). Using HYSYS to refine the model of the expander, the recalculated volumetric flow at the expander outlet is 9600 m³/h and the power generation of the expander is 9 MW.

4.2.2. Liquid CO₂ pump

After rejecting heat in the LNG heat exchanger, the liquid CO₂ is pumped from 21 bar to 130 bar. Several vendors of cryogenic pumps were contacted, as well as manufacturers of CO₂ pumps for special applications. However, it seems that none of the CO₂ pumps currently available on the market are able to satisfy the desired operating conditions of this application. In particular, none of the commercial CO₂ pumps were designed for mass flows higher than 3 kg/s.

At this point in time, the solution would be a customized pump designed for this particular application, affecting strongly the equipment cost and its reliability. For the sake of comparison, the equipment cost was assumed to be the same as the price of a cryogenic LNG pump for the same size. The same assumption was done for the footprint. On the other hand, the CO₂ pump is mounted on the same shaft with the expander, which is coupled to an electric generator. The isentropic efficiency of the CO₂ pump was conservatively assumed to be 50%, the mechanical efficiency 98% and the generator efficiency 98%.

4.2.3. LNG heat exchanger

Different types of heat exchangers operating at cryogenic temperatures were studied in order to find the most suitable alternative for regasifying LNG from -162°C to 13°C and 205 bar:

- Plate fin heat exchanger.
- Coil wound heat exchanger.
- Printed circuit heat exchanger (PCHE).

The plate fin heat exchanger is not appropriate for regasifying LNG at the high pressure adopted in this investigation (205 bar), since it is able to operate only up to 160 bar. On the other hand, the coil wound heat exchanger is a sophisticated but very expensive option that requires a large volume. Therefore the printed circuit heat exchanger (PCHE) was considered the most suitable solution as it is a very compact heat transfer device that can work up to 620 bar and within a wide range of operating temperatures (-170°C to 815°C).

The CO_2 enters the LNG heat exchanger at about 150°C transferring heat to the LNG and condensing at a saturation temperature of about -20°C . The LNG enters the heat exchanger in liquid state at about -150°C and 208 bar and it is vaporized up to 13°C . The diagram of heat flow versus temperature for the PCHE of the CO_2 Rankine cycle is presented in Figure 39. Other information provided by vendors of the PCHE used in this investigation is shown in Table 22.

Figure 39. Diagram of heat flow versus temperature of the PCHE for the CO_2 Rankine cycle.

Duty (MW)	33.5
Weight (Ton)	4
Volume (m^3)	4

Table 22. Information from vendors of the PCHE for the CO_2 Rankine cycle.

4.2.4. Heat recovery system

The heat recovery system employed in this configuration is a standard finned tubes heat exchanger for liquid-gas heat transfer, similar to the heat recovery steam generators (HRSG) used in combined cycles. The main operating conditions of the HRS are summarized in Table 23. The main parameters of the heat exchanger provided by vendors are summarized in Table 24. The diagram of heat flow versus temperature for the HRS according to the main operating conditions and data provided by vendors is shown in Figure 40.

Pinch ($^{\circ}\text{C}$)	27
Stack temperature – calculated with HYSYS ($^{\circ}\text{C}$)	160
Maximum temperature of CO_2 ($^{\circ}\text{C}$)	300
Maximum pressure of CO_2 (bar)	130

Table 23. Main operating conditions of the HRS for the CO_2 Rankine cycle.

Duty (MW)	42
Weight (Ton)	770
Volume (m^3)	7255

Table 24. Information from vendors of the HRS for the CO_2 Rankine cycle.

Figure 40. Diagram of heat flow versus temperature for the HRS in the CO_2 Rankine cycle.

The use of CO_2 leads to a range of challenges within the HRS. If steel is exposed to CO_2 at a temperature range of 350 - 450 $^{\circ}\text{C}$ a thin protective layer of magnetite is formed, which can increase corrosion. If free water is present, carbonic acid is formed which is corrosive to carbon steels. On the other hand, supercritical CO_2 has a very low viscosity. This could introduce a leaking problem with valves because there is no liquid with a surface tension on the sealing surface. When leakage occurs, the CO_2 gas will reach its atmospheric boiling point, which can cause embitterment of materials, especially ferritic steels.

4.2.5. LNG pumps

Suction vessel mounted centrifugal pumps operating at cryogenic conditions are used to increase the pressure of LNG from atmospheric pressure up to the vaporization pressure. According to vendors, two pumps in series are required for this process, which receives the LNG from the cargo ship at -162°C and pumps it up to 208 bar. The flow can be adjusted between 450-500 m^3/h . The rated efficiency of these pumps is 80%, which are driven by electric motors with 90% electrical efficiency. The price for the two pumps includes the electric motors.

4.2.6. Performance of the system

Information provided by vendors and the assumptions mentioned in the previous sections were used to refine the HYSYS energy model of the CO_2 Rankine cycle. A diagram of temperature versus entropy is presented in Figure 41. The performance results of the refined calculations for this configuration are compared to the baseline scenario in Table 25.

Characteristics	Baseline scenario	CO_2 Rankine cycle
Net power - bottoming cycle (MW)	12	3
Total power (MW)	54	45.5
Mass flow – bottoming cycle (kg/s)	120	72.5
LNG regasification rate (kg/s)	60	56
LNG regasification rate (m^3/h)	490	460
First law efficiency (%)	53	44.3
Exergy efficiency (%)	46	39
Efficiency of the LNG supply chain (%)	86	85.3
Generation capacity (kWh/kg-LNG)	0.25	0.22
Ratio (Fuel consumed) / (LNG regasified)	0.04	0.04
Maximum volumetric flow – turbine outlet (m^3/h)	48600	9600
Ratio (Power bottoming) / (Max. volumetric flow) (kWh/ m^3 -fluid)	0.25	0.32

Table 25. Performance results for the CO_2 Rankine cycle compared to the baseline scenario.

Figure 41. Temperature versus entropy diagram of the refined CO₂ Rankine cycle.

4.2.7. Cost

Equipment cost data provided by vendors and economic information available were collected for the CO₂ Rankine cycle and the baseline scenario. The equipment cost comparison of these systems is presented in Figure 42.

Figure 42. Equipment cost comparison of the CO₂ Rankine cycle and the baseline scenario.

4.2.8. Footprint

Figure 43 shows the comparison of approximate footprint for the different components of the two compared configurations, as provided by vendors. These values do not include piping, connections and auxiliary equipment.

Figure 43. Comparison of footprint of the CO₂ Rankine cycle and the baseline scenario.

4.3. Discussion

As mentioned in section 4.2.1 the two compared configurations lie on different performance lines within the power and regasification rate tradeoff diagram (see Figure 44). This is a result of the availability of performance and cost data for key equipment.

Figure 44. Power and regasification rate tradeoff diagram for the baseline scenario and the study case.

As can be seen in Figure 44 the regasification rate of the compared configurations is roughly the same and within the limits that were set in section 3.2. This means that the CO₂ Rankine cycle as dimensioned in this chapter is able to regasify close to the same amount of LNG as the baseline scenario.

Since the balance between regasification rate and power leans towards a high regasification rate for the CO₂ Rankine cycle, the power production by this cycle is substantially lower than for the baseline case. Therefore the efficiency of the LNG supply chain as well as the first-law and exergy efficiencies are also lower than for the baseline case.

While the power production is reduced and the LNG regasification rate is roughly the same for both cases, the cost and the footprint of the CO₂ Rankine cycle are smaller. The simple design of the CO₂ Rankine cycle presents 20% lower investment costs compared to the baseline case as well as a footprint reduced by nearly 20%.

A positive side effect of the temperature limitation of the expander (300 °C) is the dramatic reduction of corrosion problems in the heat recovery system. The main obstacle of the CO₂ Rankine cycle is the lack of a commercially proven CO₂ pump for the given mass flow, which could affect the technical feasibility and reliability of this alternative.

5. CONCLUSIONS AND RECOMMENDATIONS

The increasing concern on reducing the environmental impact and energy demand of the LNG supply chain has motivated worldwide efforts to optimize the design of LNG liquefaction and regasification plants. Developing novel systems able to regasify LNG and generate electric power in an efficient and environmentally friendly way was the main focus of this investigation.

Existing and novel technologies for integrated LNG regasification and power generation were identified and the most appropriate alternatives were selected for further analysis. Priority was given to those technologies that could operate without causing harm to marine life, with the minimum associated risks and footprint and with maximum reliability and electrical efficiency. The selected options include: i) closed Rankine and Brayton cycles, ii) systems for power augmentation in power plants and iii) combination of options.

A baseline scenario for integrated LNG regasification and power generation was established and simulated, consisting of a cascaded Brayton configuration with a gas turbine as topping cycle and a simple closed Brayton cycle as bottoming cycle. A detailed thermodynamic analysis was performed for the baseline scenario, in order to investigate the influence of various parameters on the overall performance of the cycle.

Various new configurations were studied and compared to the baseline scenario:

Improvements or modifications to the baseline:

1. Inlet air cooling for the gas turbine.
2. Inlet air cooling for the gas turbine and supercharging.

Alternative configurations:

3. Rankine cycle.
4. High-efficiency cycle 1.
5. High-efficiency cycle 2.
6. Thermal oil loop (no power generation).
7. Combinations of options.

A first analysis was conducted presenting a general evaluation of the different alternatives in comparison to the baseline scenario. Parameters examined within this comparison were the efficiency of the LNG supply chain, the first-law and exergy efficiencies and the LNG regasification rate. A second more detailed analysis focused on a range of proposed configurations that only implied modifications to the bottoming cycle. For this analysis the cycles were optimized and compared for varying operating conditions such as stack temperatures, maximum temperatures and pressures of the bottoming cycle.

From the optimization analysis a study case was selected and a preliminary engineering and cost analysis was carried out. As the study case a closed Rankine cycle using carbon dioxide as the working fluid was selected, since among all the studied alternatives it is the option that can regasify LNG and generate power with the lowest mass and volumetric flow in the bottoming cycle. The energy model of the study case was adapted and recalculated for the particular characteristics provided by suppliers of each component.

The major conclusions of this study are as follows:

1. The tradeoff between power production and LNG regasification subject to the cycle boundary conditions was examined. Generally the LNG regasification rate is inversely proportional to the power generated in the bottoming cycle. This is due to the fact that by increasing the power production less heat is transferred from the bottoming cycle to the LNG. By decreasing the stack temperature, there is an increase in heat absorbed by the bottoming cycle and therefore the LNG regasification rate and power generation (or the sum of the two) increase as well. Furthermore, the power generation in the bottoming cycle is proportional to the maximum pressure and temperature of the working fluid, whereas the LNG regasification rate is inversely proportional to these parameters. The net power, as well as the first-law and exergy efficiencies decrease sharply by increasing the vaporization pressure and can be raised slightly by increasing the delivery temperature of the regasified LNG.
2. A new method was suggested to evaluate the efficiency of the LNG supply chain depending on the vaporization technology employed. Assuming a fixed energy requirement for the liquefaction and shipping, it was found that the LNG supply chain using an open rack vaporizer (ORV) was able to deliver about 85% of the natural gas extracted from the field, whereas the supply chain using an submerged combustion vaporizer (SCV) could deliver 84% of the extracted natural gas. The LNG supply chain using the technologies established in the baseline scenario improved this value to 86%.
3. The simple design of the CO₂ Rankine cycle presents 20% lower investment costs compared to the baseline as well as a footprint reduced by nearly 20%. At the same time, the LNG regasification rate for the two compared configurations is roughly the same. However, the efficiency of the LNG supply chain as well as the first-law and exergy efficiencies are much lower for the CO₂ Rankine cycle than for the baseline case. This is due to the fact that the temperatures and pressures of the cycle are conditioned by the limited characteristics of the expander, therefore leading to less heat recovery from the exhaust gas stream. A positive side effect of the temperature limitation of the expander (300 °C) is the strong reduction of corrosion problems in the heat recovery system. The main obstacle of the CO₂ Rankine cycle is the lack of a commercially proven CO₂ pump for the given mass flow, which could affect the technical feasibility and reliability of this alternative.

This investigation was limited to create, model and simulate novel cycles operating at steady state and full load, at the design points of the components and with constant ambient conditions (with the exception of the inlet air cooling and supercharging for a simple Brayton cycle). A detailed off-design analysis including part load operation and dynamic performance of the components as well as variable ambient conditions would be required to evaluate the technical and economic feasibility of the study case more accurately. Additionally a more detailed engineering study of the components of the regasification system including dimensioning, material selection, piping and connections, electrical and auxiliary equipment as well as control systems should also be performed.

Special attention should be paid to the different challenges associated with the critical components of the study case for future work. It would be a priority to find a CO₂ pump or design a new model capable of handling very large mass flows. Further, it would be worthy to extend the operating range of the CO₂ expander to higher temperatures and in this manner improve the efficiency of the cycle. This would also involve a redesign of the heat recovery system to avoid corrosion problems.

Finally, an analysis of the life-cycle greenhouse emissions for the whole supply chain of LNG using the technologies proposed in this investigation would be of great interest. Such an analysis could help to assess in detail the additional environmental benefits of LNG vaporization with integrated power generation.

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7. APPENDICES

7.1. Comparison of LNG vaporization technologies for power generation

Technology	Electrical/ exergy efficiency (%)	Power output (MW)	LNG regas. rate (kg/s)	Net generation capacity (kWh/kg- LNG)	Heat source	Ref.
Simple and recuperated hydrocarbon Rankine cycles	-	6.8	27.7	0.068	Seawater	[26]
Cascaded propane-methane Rankine cycles	-	11.7	41.7	0.078	Seawater	[8]
Water-ammonia Rankine cycle	29	19.3	51.1	0.105	Refuse incinerator as heat source	[24]
Subcritical recuperated CO ₂ Rankine cycle-oxyfuel for LNG regasification	71	41.3	100	0.115	Combustion of natural gas in presence of oxygen and CO ₂	[38]
Recuperated nitrogen Brayton + direct expansion	46	47	107.6	0.121	Hot air from a sintering plant	[9]
Closed recuperated air Brayton cycle with reheating and intercooling (pseudo-Ericsson cycle)	37	15.9	31.6	0.139	Exhaust gas from a GT / low level heat source	[36]
Recuperated nitrogen Brayton cycle	18.5	100	77.5	0.358	Solar collector	[34]
Supercritical recuperated CO ₂ Rankine/Brayton cycle-oxyfuel	65	71.4	54.7	0.362	Combustion of natural gas in presence of oxygen and CO ₂	[40]
Steam Rankine cycle + direct expansion	52.2	107.2	80	0.372	Exhaust gas from a GT	[31]
Combined gas turbine-steam turbine, Rankine cycle using a Freon mixture and methane Rankine cycle	53 (HHV)	240	166.7	0.400	Exhaust gas from a GT	[30]
Supercritical recuperated CO ₂ Rankine/Brayton cycle-oxyfuel with reheat	67.5	79.4	54.9	0.402	Combustion of natural gas in presence of oxygen and CO ₂	[40]
Cascaded open gas turbine (topping) and closed Brayton cycle (bottoming) using nitrogen	51.9	133	80	0.462	Exhaust gas from a GT	[31]

Table 26. Comparison of LNG vaporization technologies for power generation.

7.2. Characteristics of the gas turbine

Figure 45. Performance of the GT versus ambient temperature.

Figure 46. Exhaust gas flow and temperature versus ambient temperature for the GT.

Figure 47. Specific heat available in exhaust gas depending on the stack temperature.

7.3. Climate conditions and performance of the GT

Figure 48. Monthly ambient conditions in an average Mediterranean location [47].

Figure 49. Performance of the GT in an average Mediterranean location over the course of one year.

7.4. Variation of LNG properties in the pumping process

Figure 50. Specific work for pumping LNG.

Figure 51. Variation of LNG temperature in the pumping process.

7.5. Change in specific exergy for LNG during regasification for variable vaporization conditions

Figure 52. Change in specific exergy of LNG during regasification for variable vaporization conditions.

7.6. Comparison of the performance of the baseline scenario, the inlet air cooling and supercharging option over the course of one year

Figure 53. Comparison of the relative power of inlet air cooling and supercharging options to the maximum power of the baseline scenario over the course of one year.

Figure 54. Comparison of the relative efficiency of inlet air cooling and supercharging options to the maximum efficiency of the baseline scenario over the course of one year.