Simulation and Design of a Boil-Off Gas Re-liquefaction System in a Small-Scale LNG Supply Chain

Case Study of Trafaria

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Abstract

Liquefied Natural Gas (LNG) supply chains have gained a relevant position in the energy market during the last 20 years, and economic projections indicate it will continue to grow. The Port of Lisbon offers geographically worthy conditions to develop a local small-scale supply chain, and the present work analyses the feasibility of the Trafaria fuels terminal to house a small-scale LNG (SSLNG) storage facility, mainly dedicated to LNG ships bunkering. Thus, a preliminary study of the main characteristics of an SSLNG terminal is made, including its BOG (Boil-off Gas) management system as a major concern within the operation of an LNG terminal. The location and realistic dimensions of the chosen in-ground tank are selected, and heat ingress into the tank is simulated through the definition of the dimensions and insulation of the tank, respecting the maximum BOG rate defined, allowing also the calculation of energy use and possible operating costs. As a method of managing the generated BOG, a re-liquefaction system based on a Nitrogen Turbo-Expander cycle is studied. The system is simulated for two different configurations, allowing to determine the most efficient one. Also, a cogeneration plant is considered as a potential solution to manage the generated BOG; both solutions are simulated for different BOG rates, climate conditions and modes of operation of the terminal. The cogeneration plant is found to be an interesting alternative to manage the BOG together with a re-liquefaction system as backup, capable of processing all the BOG in the most demanding operation scenario.

Keywords: Liquefied Natural Gas, Small-scale supply chain, Boil-off gas, BOG Re-Liquefaction, Single Nitrogen Expansion.

1. Introduction

The world primary energy demand is anticipated to grow 1.5% annually from 2012 to 2035, while natural gas consumption is expected to grow 1.9% per annum. With the increase of the global gas trade, projections indicate that LNG (Liquefied Natural Gas) will account for 15% of the global gas consumption, with an annual growth of 3.9% in LNG trade, being expected that it will play a large role on the energy economy [1].

Whilst HFO (Heavy Fuel Oil) is the most used fuel by merchant ships nowadays, LNG has been earning a much stronger presence worldwide on the last two decades [2], being actually widely used and with a growing evolution in the area of shipping [3], as per the different risks it presents while being handled, as for its regulation-compliant pollutant characteristic when comparing with conventional maritime fuels such as HFO.

Being so, LNG has become one of the pronounced answers to comply with MARPOL regulations [4]. Due to its chemical and combustion properties, the use of LNG allows for a significant reduction of PM, SO_X, NO_X and CO_2, mostly when compared with HFO: thanks to lean combustion that occurs on dual-fuel internal combustion engines, NO_X emissions are reduced by 80-85%; as LNG does not contain precursors of SO_X, those emissions are almost inexistent; CO_2 emissions decrease 20-30%, particularly when comparing to HFO/MDO (Marine Diesel Oil), due to a higher hydrogen content in molecules.

Given the crescent regulatory constraints related with emissions control that influence ship owners, and that these owners search for cleaner fuels to head to northern Europe, an LNG terminal capable of supplying these vessels might be a great contribution for the local and national economy as increasingly more ships would be expected to enter the Port of Lisbon and use its facilities, representing a promising affluence to an LNG terminal to be built on this zone.

Accompanying the market trends and the increasingly stricter regulatory demands, Portugal gathers the conditions to develop a local LNG supply chain in the Port of Lisbon, as it is a waypoint for ships travelling to and from ECA and SECA zones, areas where LNG is largely used as fuel due to emissions restrictions. This work results from the cooperation of the author with Tecnoveritas, a Portuguese engineering company, on the initial study and development of a small LNG supply chain for OZ Energia, in the south bank of river Tagus, focusing on the energy usage during the energy conversion associated to the normal operation of a small-scale LNG terminal.

The LNG Value Chain and BOG

The LNG production process starts with the exploration and production of the natural gas wells, being then transported via pipeline to the liquefaction plant, where it is cooled to achieve -161ºC at atmospheric pressure, and becoming 1/600 of its original volume at gaseous state. The LNG is then loaded into tankers to be transported by sea to the receiving countries, being then unloaded and stored as liquid or regasified to be fed into the connecting pipelines to the natural gas grid of the accepting country. Due to the low temperatures at which LNG is stored, one major terminal operation concern is the occurrence of boil-off gas (BOG) due to heat ingress in the LNG storage facilities. This event requires simultaneously very good insulation and efficient methods to manage BOG.

2. The Case-study

The present work was developed in order to simulate and study the feasibility of a Small-Scale LNG (SSLNG) terminal in Trafaria, and assess its energy behaviour, addressing mainly the heat ingress in the LNG system and the BOG management issue that consequently arises, studying the hypothesis of re-liquefaction and other available alternatives.

Trafaria Terminal, an already existing fuels terminal located on the south bank of River Tejo, Murcafém, comprises a total area of 79,825 m^2. The possibility of retrofitting the terminal also to an LNG receiving terminal was assessed and started by describing the terminal and its adjacent areas. There are three sensitive areas around the terminal: the river Tagus, a National Ecological Reserve and a Historical Centre on the vicinity. Being on the bank of the river Tagus, the terminal possesses a jetty with 170 metres long and the bathymetric on the riverbed allows it to support ships that have up to 9 metres of sea-gauge, which means that the maximum loaded draft of the incoming ships cannot exceed 8.5 metres as, otherwise, safety of both the vessels, the jetty and the personnel might be at risk [5].

The chosen location to place the LNG storage tank was the one that presents the most adequate safety distance from other tanks and from the National Ecological Reserve and the larger available area; this allows the construction of adjacent facilities such as the BOG management system. In addition, the chosen location allows the construction of an in-ground tank, which will grant two significant advantages – effective land use and structural safety [6] – which meets the requirements of a terminal located in a hillside with such a steep slope.

The capacity of the storage tank was chosen to be 30,000m^3, and the following work was developed in order to address the BOG management, and simulate its production through the calculation of the heat ingress in the tank. One hypothesis is to manage BOG through its re-liquefaction, and the available technologies were explored so to choose the most adequate.

3. NG Liquefaction Technologies

In order to better evaluate and quantify the performance of the liquefaction cycle chosen to recover BOG, one of the assessment factors relied on the performance achieved by each technology. The Coefficient of Performance (COP), equation (5), was used to assess the performance of the liquefaction system chosen. Nevertheless, particular attention should and was paid to the expected investment on equipment, especially due to the small-scale dimension of the terminal, which implies that not all systems are feasible nor economically viable.

The liquefaction processes used on LNG plants can be classified in three main groups of processes: cascade liquefaction, mixed refrigerant and expansion-based processes.

Cascade Liquefaction Cycles

Liquefaction is achieved by operating two or more liquefaction cycles in series, named cascade liquefaction cycles [7], and attaining progressively lower temperatures with moderate pressure ratios. Cascade cycles are based on the use of a number of refrigerants that present different, yet constant, boiling temperatures, thus reducing irreversible heat exchange losses [8] by guaranteeing a minimum area in the heat exchangers. It is common to utilize pure refrigerants such as methane, ethylene and propane. Each refrigerant can be controlled separately, ensuring flexibility of operation to the liquefaction system as function of the natural gas flow rate. However, these systems are very sensitive to the natural gas composition, exhibiting problems in adapting to composition fluctuation.

Single Mixed Refrigerant (SMR) and Dual Mixed Refrigerant (DMR) Cycles

Based on the reverse Rankine cycle, SMR systems are based on cooling and liquefaction in a single heat exchanger [8]. SMR cycles often present low efficiency when compared with other options, therefore being more common to find these systems in midsize to small-scale LNG plants. The DMR processes achieve the intended liquefaction by using two different mixed refrigerant cycles. A broad utilization of this cycle is the C3-MR technology which employs as first refrigerant a single propane refrigerant, and a mixed refrigerant on the second refrigerant stream [8]. The disadvantage of this system resides in a higher complexity of the process as well as higher inherent cost in equipment.

Gas Expander Cycles

Liquefaction using a gas expander cycle is based on the circulation of a fluid, achieved by means of a turbo-expander, compressing and expanding it in order to generate the required refrigeration. The fluid that is commonly used as refrigerant is nitrogen (or methane, in alternative), due to the characteristic low temperature it can achieve. This process is usually suitable for small-scale facilities as it is of simple operation, low-maintenance and presents a fast start up response [9]. The simplicity is related to the fact that, by using pure components as refrigerant, it is not necessary to adjust its composition according to the NG composition. Plus, the temperature control is facilitated, resulting in a more stable cycle over a range of conditions of liquefaction [10]. Having a considerably lower investment associated when compared with other technologies available, the gas expander cycles have the disadvantage of presenting a low-efficiency operation when compared with cascade of mixed refrigerant liquefaction systems. The fact that these systems require lower investments makes gas expander cycles more suitable for small-scale LNG plants, peak shaving plants or BOG recovery and re-liquefaction systems, where lower natural gas flow rates need to be processed, not being appropriate for load-base plants [8]. In addition, Contrary to cascade and mixed refrigerant cycles, expansion-based liquefaction systems using nitrogen have the advantage of being inherently safe, as there is no hydrocarbon liquid inventory on the installation [8]. Also, emergency venting of refrigerant in gas expander systems does not represent a problem for the operator, as the released component is inert and does not comprise an environmental issue to be dealt with. Table 1 systematizes the different liquefaction technologies and their key characteristics.

Table 1 - Comparison of Different LNG Liquefaction Technologies [8].

Cycle	Cascade / C3-MRC	MR	Expander
Efficiency	High	Moderate / High	Low
Complexity / Cost	High	Moderate	Low
Heat Exchanger Area	Low	High	Low
Flexibility	High	Moderate	High

Taking into account all the presented pros and cons, and presenting low complexity and high flexibility, together with a lower initial investment, the liquefaction technology that was chosen and studied for recovering BOG on the Trafaria Terminal was the expander cycle using nitrogen as refrigerant.

The expander (or turbo-expander) liquefaction cycle, is based on the reversed-Brayton cycle, as illustrated in Figure 1. While less efficient than the other explored options, it is considered the most adequate solution for BOG re-liquefaction when performing an analysis of cost versus benefit. Anyhow, and as referred by Mokhatab et al. [8], these installations are limited to 1–2 MTPA (million ton per annum) of single train capacity.

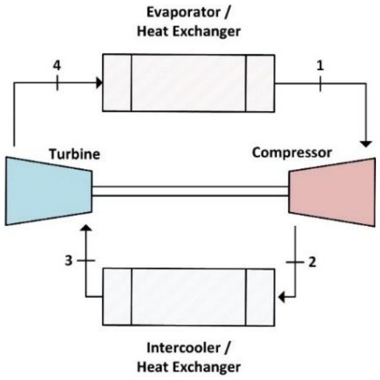

Figure 1 – Reversed-Brayton Cycle

Analysing the processes of Figure 1, the cycle occurs as follow: in process 1-2 the compressor compresses nitrogen and the fluid achieves high temperature and pressure; in process 2-3, the fluid exchanges heat with the colder environment in an intercooler, resulting in a decrease of its temperature; in process 3-4 the fluid is expanded in the turbine, causing a decrease of pressure and temperature; in process 4-1, the nitrogen passes through the evaporator (heat exchanger), absorbing heat from the cold refrigerated fluid (LNG, in this case), which then changes back to the conditions of state 1.

He & Ju [11] accomplished an analysis of different configurations of the turbo-expander LNG liquefaction cycle in order to attain the maximum efficiency possible. In their work, the thermodynamic models obtained are computed according to the following equations:

i) Compressors

Compressors have the function of pressurizing the refrigerant (nitrogen) in the expansion cycle, generating exergy losses caused by friction and heat losses in the process. The energy balance (equation (1)) is calculated as follows:

$$\dot{W}_c = \dot{m}_{N_2} \frac{\left(h_{o,isentropic} - h_{i,c}\right)}{\eta_c} = \dot{m}_{N_2}\left(h_{o,c} - h_{i,c}\right) \quad (1)$$

$\dot{W}_c$ is the work performed by the compressor, η_c is the isentropic efficiency of the compressor, $\dot{m}_{N_2}$ is the mass of nitrogen, and $h_{o,c}$ and $h_{i,c}$ are respectively the outlet and inlet enthalpies in the compressor.

ii) Expanders (Turbines)

Turbines execute the depressurization of the refrigerant with the purpose of lowering its temperature. Simultaneously, these components transmit mechanical power to the compressor by means of a connecting shaft, driving it. The energy balance for expanders (equation (2)) is calculated as follows:

$$\dot{W}_e = \dot{m}_{N_2}\left(h_{o,isentropic} - h_{i,e}\right)\eta_e = \dot{m}_{N_2}\left(h_{o,e} - h_{i,e}\right) \quad (2)$$

$\dot{W}_e$ is the work performed by the expander, η_e is the isentropic efficiency of the expander, $\dot{m}_{N_2}$ is the mass of nitrogen, and $h_{o,e}$ and $h_{i,e}$ are respectively the outlet and inlet enthalpies in the expander. Note that, according to equation (2), $\dot{W}_e$ is negative.

iii) Intercooler (Heat Exchanger)

This heat exchanger is used to lower the discharge temperature of the fluid at the outlet of the compressors, so that the work to the booster compressors becomes smaller. The energy balance for heat exchangers (equation (3)) is calculated as follows:

$$\dot{Q}_i = \dot{m}_{N_2}\left(h_{o,i} - h_{i,i}\right) \quad (3)$$

where $\dot{Q}_i$ is the heat transferred on the intercooler, $\dot{m}_{N_2}$ is the mass of nitrogen, and $h_{o,i}$ and $h_{i,i}$ are respectively the outlet and inlet enthalpies in the condenser.

iv) LNG Heat Exchangers / Cycle evaporator

This equipment constitutes the heart of the liquefaction process. Here, the heat is removed from the refrigerated fluid to the refrigerant, which absorbs it. In the present case, natural gas delivers heat to the nitrogen, this way achieving liquid state. The energy balance for LNG heat exchangers (equation (4)) is calculated as follows:

$$\dot{Q}_{hex} = \sum_{n=1}^{z} \dot{m}_{N_2,n}\left(h_{o,hx\,n} - h_{i,hxn}\right) \quad (\dot{m} = 1, 2, \ldots, z) \quad (4)$$

where $\dot{Q}_{hex}$ is the heat transferred within the heat exchanger, $\dot{m}_{N_2}$ is the mass of nitrogen in each heat exchanger, z is the number of heat exchangers on the system, and $h_{o,hxn}$, and $h_{i,hxn}$ are respectively the outlet and inlet enthalpies in each heat exchanger.

The (COP) for a refrigeration (or liquefaction) cycle, relating the amount of heat transferred by the hot source (Q_{evap}) with the work performed (W_{comp}), is as shown in equation (5):

$$COP = \frac{Q_{evap}}{W_{comp}} \quad (5)$$

These equations were applied to both expander cycle configurations simulated, in order to conclude what is the most suitable technology for Trafaria Terminal, and guarantee the best relation of cost and efficiency possible.

4. Methodology

The models used in this work address specifically the heat ingress, and resulting BOG production and management as the themes of greatest interest and importance, especially in terms of energy efficiency.

Heat Ingress in the Tank and LNG System

The daily BOG production rate was defined as 0.067%, and later compared with the results for a common value found in the literature, 0.050%, and a lower value, 0.039%, for comparison. This value allowed to define the insulation of the tank (computed according to existing industrial projects). With the tank dimensions defined and with relevant and local data of solar radiation and ambient temperature that characterize each season of the year, the heat ingress in the tank was calculated as follows:

Total Heat input to the tank

The total heat input to the tank is given by equation (6):

(6) $$q_{tank} = q_{bottom} + q_{walls} + q_{dome}$$

Total Heat input through the Bottom

The heat input through the bottom of the tank was analysed as a regular infinite plane wall, which allows the application of the Fourier Law of heat conduction for one dimension. Equation (7) allows to compute the heat input through the bottom, while equation (8) allows to calculate the surface temperature of each layer of the bottom:

(7) $$q_{bottom} = \frac{T_{SOIL} - T_{LNG}}{\sum Thermal\ Resistances} \quad [\,W\,]$$

(8) $$T_1 = T_2 + q\frac{L}{k.A} \quad [\,K\,]$$

Total Heat input through the Walls

Following a similar approach to the heat input through the bottom, to calculate the heat input through the walls of the tank, this section is analysed as one-dimensional heat flow through multiple cylinder sections, allowing to apply the Fourier Law of heat conduction for the several layers that compose the walls. Equation (9) allows to compute the heat input through the walls, while equation (10) allows to calculate the surface temperature of each layer of the walls:

(9) $$q_{walls} = \frac{T_{SOIL} - T_{LNG}}{\sum Thermal\ Resistances} \quad [\,W\,]$$

(10) $$T_1 = T_2 + \frac{\ln(R_2/R_1)}{2\pi.k.L} \quad [\,K\,]$$

Total Heat input through the Roof

Besides accounting for the ambient temperature it was also necessary to account for the solar radiation incident on the outer surface of the roof. The determination of the heat input through the roof is then influenced by climacteric conditions as temperature, radiation and also wind conditions can affect the results.

To perform this calculation and compute the heat input in this complex interface, one may make recourse to the resistances of insulation and structure materials, and the absorptivity of the white paint with which the dome is painted; heat input sources, incident radiation, ambient temperature and associated convection are analysed. Knowing the physical characteristics of the dome (roof), an equivalent heat transfer diagram was made in order to define the calculations to determine the heat input through the roof of the tank, as displayed in Figure 2:

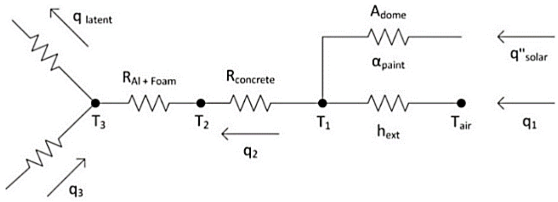

Figure 2 – Equivalent Heat Transfer Diagram – Dome

where q''_{solar} is the solar irradiation, q_1 is the heat transferred by convection, h_{ext} is the convection coefficient considered, αpaint is the absorptivity of the white paint, q_2 is the heat rate that enters through the layer of concrete, q_3 is the heat that enters through the walls and bottom of the tank, and q_{latent} is the latent heat (phase change of LNG) inside the tank; T_{air} is the ambient temperature, T_1 is the surface temperature, T_2 the temperature between the dome and the insulation deck and T_3 the temperature inside the tank. $R_{concrete}$ and $R_{Al+Foam}$ are the thermal resistances (conductivity) of the concrete layer and the deck (foam glass and aluminium).

To simplify the calculations, the convection resistance between the concrete layer (inner interface) and the insulation deck (upper interface) is discarded, as natural convection with high mixture occurs; it is then considered that the natural gas temperature is uniform in both interfaces. Then, according to Figure 2:

(11) $$q_{Dome} = q_2 = q_1 + q''_{solar}.A_{Dome}.\alpha = \\ = \frac{T_1 - T_3}{R_{concrete} + R_{Al+Foam}},$$

(12) $$q_1 = \frac{T_{air} - T_1}{\frac{1}{h_{ext}.A_{dome}}}$$

(13) $$\frac{T_1 - T_3}{R_{concrete+}R_{Al+Foam}} = \\ q''_{solar}.A_{Dome}.\alpha + \frac{T_{air} - T_1}{\frac{1}{A_{Dome}.h_{ext}}}$$

(14) $$T_1 = \frac{A_{Dome} \cdot (R_{concrete} + R_{Al+Foam})(h_{ext} \cdot T_{air} + q''_{solar} \cdot \alpha) + T_3}{A_{Dome}(R_{concrete} + R_{Al+Foam}) \cdot h_{ext} + 1}$$

Knowing T_1, it is possible to find q_2, the total heat input through the dome referring to equation 11, and thus finding the heat input to the tank through the dome. In order to study different effects of temperature and wind, the simulations were performed for different seasons of the year, and assuming convection coefficients that simulate a day with no wind and a very windy day.

Heat Input through Piping and Pumping System

Although only applicable while loading or unloading the tank, there is a substantial heat input through circulation and loading pipelines, between the tank and the vessels at the jetty. The calculations were performed with a similar methodology to the one used by Wordu and Peterside [12].

Simulation Scenarios

Single Expander Re-Liquefaction System

Due to the size of the LNG terminal and storage facilities, only the hypothesis of a single expander re-liquefaction system was studied, as the expected BOG rate is very small and the dual turbo-expander cycle implies a superior initial investment.

In order to determine the capacity of the re-liquefaction system and the quantity of refrigerant fluid (N_2), the Mollier diagrams of methane and nitrogen were used with the aim of determining the enthalpy, pressure and temperature changes of both fluids during the liquefaction process. The cycles studied are depicted in Figure 3, which represents the nitrogen cycle (refrigerant), and the BOG cycle, collected from the tank in gaseous form, and returned to the tank, liquefied, after transferring heat to the refrigerant.

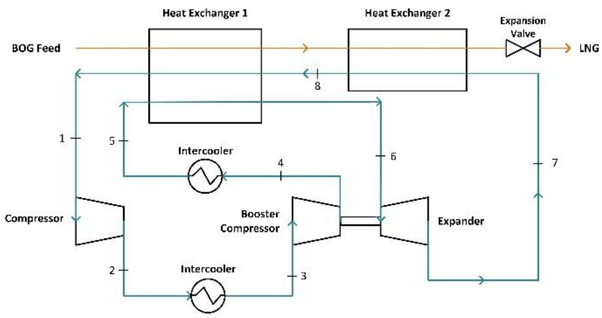

Figure 3 - Single Expander Nitrogen Cycle

Knowing the enthalpies at the beginning (h_{BOG}) and end (h_{LNG}) of the liquefaction cycle, it was possible to determine the heat necessary to remove from the BOG in order to condense to LNG (h_{ref}). This is defined by using equation 15:

(15) $$h_{ref} = h_{BOG} - h_{LNG}$$

Knowing the BOG mass flow rate it is then possible to calculate the power of the refrigeration/liquefaction cycle (P_{ref}) by using equation 16:

(16) $$P_{ref} = \dot{m}_{BOG} \times h_{ref}$$

According to the examples from the works of Gerdsmeyer and Isalski [13], A. Vorkapić et al. [14], and Gómez et al. [15], the nitrogen pressure at the inlet of the first compressor (state 1) shall be between 0.8 and 1 MPa and the pressure at the expander inlet shall be between 3.5 and 6.0 MPa. The efficiency of the turbo-compressors shall be considered 0.8. Knowing the pressure ratio of the cycle (p_R) given by equation 17, and through equation 18, the intermediate pressure (p_i), pressure of states 2 and 3, can be calculated as follows:

(17) $$p_R = \frac{p_2}{p_1},$$

where p_1 is the inlet pressure, and p_2 is the discharge pressure. To guarantee that the work done by each compressor is minimal and divided evenly, the following equation is applied:

(18) $$p_R = \frac{p_i}{p_1} \times \frac{p_2}{p_i} \Rightarrow p_i = \sqrt{p_1 p_2}$$

The outlet temperature of the expander shall be lowered to a temperature between -168ºC to -180ºC, in order to achieve sub-cooled LNG. The subcooled LNG shall be delivered to tank at -165ºC. With the required refrigeration power of the liquefaction system (P_{ref}), the mass flow rate of refrigerant shall be computed using equation 19:

(19) $$\dot{m}_{N_2} = \frac{P_{ref}}{(h_1 - h_7) + (h_6 - h_5)},$$

where $\dot{m}_{N_2}$ is the mass flow rate of nitrogen, h_1 and h_7 are the enthalpies at states 1 and 7, respectively, and h_5 and h_6 are the enthalpies at states 5 and 6, respectively.

The total work of the compressors ($\dot{W}_c$) is expressed by equation 20, while the total heat dissipated in the intercoolers (Q_i) is expressed by equation 21:

(20) $$\dot{W}_c = \dot{m}_{N_2}[(h_2 - h_1) + (h_4 - h_3)]$$

(21) $$\dot{Q}_i = \dot{m}_{N_2}[(h_3 - h_2) + (h_5 - h_4) + (h_6 - h_5)]$$

The total heat dissipated from the nitrogen to the surroundings shall be calculated using equation 22:

(22) $$\dot{Q}_{diss} = \dot{Q}_i + \dot{Q}_{he}$$

where $\dot{Q}_{he}$ is the heat absorbed by the refrigerant stream in the heat exchangers.

Cogeneration

The study of the cogeneration scenarios starts by having the data of the different BOG rate scenarios and considering the BOG mass flow as the feed gas for the cogeneration plant (at reference conditions – $T_{ref} = 273.15\,K, p_{ref} = 1.01325\,bar(a)$). This way, and assuming the electric efficiency as 40% and the thermal efficiency as 35%, it is possible to calculate several parameters associated with the implementation of a cogeneration.

Through equation 23 the consumption of natural gas, assumed as an ideal gas, is calculated for reference conditions:

$$Q_{ref} = Q \times \frac{T_{ref}}{T} \times \frac{p}{p_{ref}} \quad (23)$$

Where Q is the volumetric rate consumption of natural gas at a given pressure, p, and temperature, T.

Equation 24 gives the power of the cogeneration,

$$P_{cogeneration} = Q_{ref} \times \frac{LHV}{3600} \quad (24)$$

where $P_{cogeneration}$ is the power of the cogeneration (in kW), Q_{ref} is the consumption of natural gas (in Nm^3/h), and LHV is the lower heating value in kJ/Nm^3.

Equations 25 and 26 are, respectively, the electric and thermal power of the cogeneration:

$$P_{electric} = P_{cogeneration} \times E\eta \quad (25)$$

$$P_{thermal} = P_{cogeneration} \times H\eta \quad (26)$$

where $P_{electric}$ and $E\eta$ are the electrical power (in kW) and efficiency, and $P_{thermal}$ and $H\eta$ are the thermal power (in kW) and efficiency.

Following the guidelines found in the Portuguese and European legislation dedicated to this subject (Decree-Law n.23/2010 [16] and EU directive 2004/8/CE [17]), the Primary Energy Saving (PES), during a certain period of time, that results from the cogeneration operation can be calculated as shown in equation 27:

$$PES = \frac{H_{CHP}}{Ref\ H\eta} + \frac{E_{CHP}}{Ref\ E\eta} - \mathrm{F}_{\mathrm{Total}} \quad (27)$$

where H_{CHP} and E_{CHP} are the thermal and electrical energy produced in cogeneration, $Ref\ H\eta$ and $Ref\ E\eta$ are, respectively, the considered reference values of thermal and electrical efficiencies for the combustion of natural gas as fuel, and $\mathrm{F}_{\mathrm{Total}}$ is the total amount of fuel consumed during the analysed period.

Also as a reference, it is possible to calculate the total CO2 emissions avoided during the operation in cogeneration, applying equation 27:

$$(A.E.CO_2)_{CHP} = \frac{PES}{E_{CHP}} \times (E.CO_2) \quad (27)$$

where $E.CO_2$ is the CO_2 emission factor calculated for natural gas [18], correspondent to 201,96 gCO_2/kWh, and $(A.E.CO_2)_{CHP}$ are the avoided emissions of CO_2 by producing electricity in cogeneration.

Emergency System

The emergency system, a flare, shall be only used in case no other option to handle the BOG is available, and as last resource. Flaring represents a great loss for the operator, and can endanger the profitability of the terminal. It is estimated that this system shall only be used in case the BOG re-liquefaction system, cogeneration, or other recovery system is damaged or under maintenance, and in case a phenomenon like the rollover effect occurs.

5. Results

Designed Heat Ingress to the Tank

The BOG rates simulated were 0.039%, 0.05%, and 0.067%. The winter scenarios were considered virtually windless and, the summer scenarios, very windy, in order to depict, respectively, the best and worst scenarios.

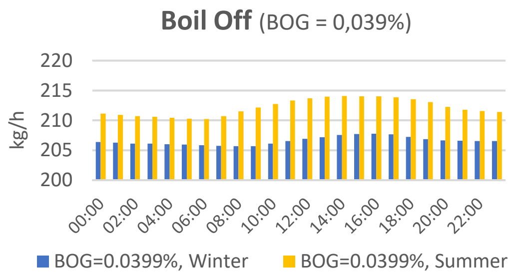

Figure 4 - Daily Mass Flow Rate - BOG=0.039%

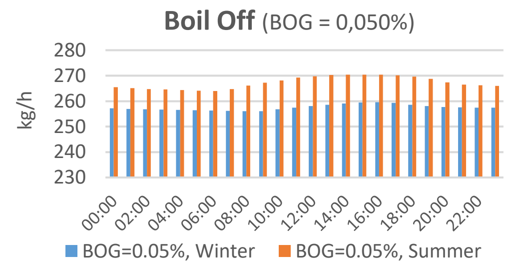

Figure 5 - Daily Mass Flow Rate - BOG=0.050%

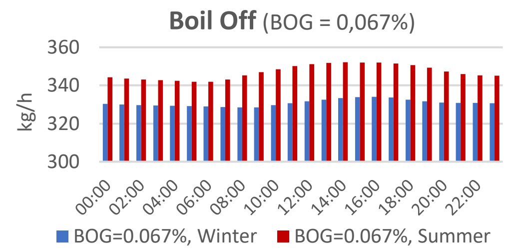

Figure 6 - Daily Mass Flow Rate - BOG=0.067%

For a BOG rate of 0.039% the maximum expected BOG is 214.1 kg of BOG per hour, resulting in approximately 5092.2 kg per day; for a BOG rate of 0.067% the maximum expected BOG is 352.1 kg of BOG per hour, resulting in approximately 8328.3 kg per day. The summer BOG production for BOG rates of 0.039 and 0.067 % is, respectively, approximately 3 and 5% higher than the winter BOG production in each case, resulting in the need for processing, respectively 134 and 388 more kg of BOG per day. For a BOG rate of 0.067%, it is expected that the BOG rate results in a maximum production of BOG 35.9% higher than for a BOG rate of 0.039%, highlighting the effect of a poorer insulation.

Total Heat Ingress

With the calculation of heat ingress through the different tank areas, piping, and pumping system, the total heat ingress for each operation mode was calculated and is presented in Table 2.

Table 2 – Heat Ingress in the LNG System during each Operation Mode

Operation Mode	Best Scenario	Worst Scenario
	Tank [kW]	
Holding Mode (BOG = 0.039%)	29,31	30,11
Loading / Unloading	29,31	30,11
Holding Mode (BOG = 0.05%)	36,54	37,92
Loading / Unloading	36,54	37,92
Holding Mode (BOG = 0.067%)	46,95	49,24
Loading / Unloading	46,95	49,24
	Piping [kW]	
Loading / Unloading	112,32	112,32
	Pumping System [kW]	
Loading / Unloading	23,59	23,59
	TOTAL [kW]	
Holding Mode (BOG = 0.039%)	29,31	29,31
Loading / Unloading	165,23	165,23
Holding Mode (BOG = 0.05%)	36,54	36,54
Loading / Unloading	172,45	172,45
Holding Mode (BOG = 0.067%)	46,95	49,24
Loading / Unloading	182,86	196,45

Insulation Costs and Holding Mode Operation

The initial investment difference between the best and worse insulation is approximately of 301 448 €, being this the estimated cost of having a smaller BOG rate. According to a basis of estimate for LNG storage tank cost analysis [19], it is also necessary to account for other relevant components: the labour component corresponds to 2.5% of the cost of the tank, the LNG tank foundations and the civil works, piping, electrical and structural systems, and infrastructures construction. To these systems, it is necessary to add the cost of the cryogenic pumps and fittings on the tank, as well as the BOG management system.

For a BOG rate of 0.039%, the tank is expected to withstand 6.86 years in holding mode, and 4.11 years for a BOG rate of 0.067%.

Single Expander Re-Liquefaction System

Two liquefaction cycles were simulated to achieve the required liquefaction of BOG. The hypotheses were studied for the set resulting of the sum of each worst scenario of the tank BOG rates considered, with the worst scenario of BOG production of piping and pumping system, plus a 5% design margin to guarantee the liquefaction system is capable of processing the maximum expected BOG production. The hypotheses were simulated for four different BOG inlet temperatures, to study the performance of the systems and obtain a well-defined tendency. For the first cycle, hereafter designated as Cycle A, the pressure of the nitrogen at the inlet of the first compressor is 0.8MPa, the calculated intermediate pressure is 1.79 MPa, and the pressure at the inlet of the expander is 4MPa. Cycle B, designation given to the alternative cycle hereafter, intends to assess the effect of expanding to a lower pressure (0.4 MPa) and increasing, this way, the heat the refrigerant can absorb from the BOG. Both cycles start compression at -168ºC, allowing the nitrogen temperature to be always smaller than the BOG temperature, and that the minimum temperature difference between the refrigerant and the BOG is 2ºC; the compression work in both cycles takes place by compressing superheated nitrogen, to guarantee that no liquid nitrogen is present at compressors inlet. As, at this stage, the exact pipes layout and the distance to the re-liquefaction system are not known in detail, the BOG temperature is variable and, as an approximation, its pressure will be considered equal to the tank operating pressure, 0.107 MPa. The LNG shall be delivered slightly subcooled to the tank, at -165ºC. Figure 7 illustrates both simulated cycles in the Mollier diagram.

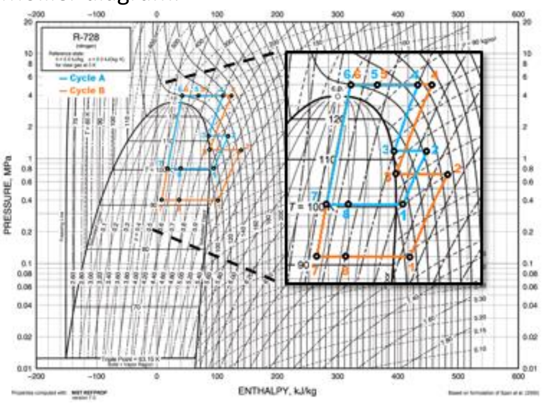

Figure 7 – Liquefaction Cycles A (blue) and B (orange)

As expected, the mass flow rate of nitrogen required to absorb the heat from the BOG increases with its temperature and with its mass flow rate. Cycle A requires a flow rate of nitrogen approximately 31% bigger than Cycle B, which requires more compression work to achieve the same liquefaction power. Results obtained indicate Cycle B as the most efficient one, presenting a significantly smaller specific energy consumption to perform the liquefaction of the BOG, of approximately less 24%.

Figure 8 presents the values of the COP for the cycles, as well as the real COP (COP_{actual}) values. The COP value was calculated using equation 5, and it does not consider the work performed by the turbine as the driver of the second compressor. The KPI COP_{actual} was calculated due to the specific characteristic of this liquefaction cycle, by having a turbo-expander; it was calculated dividing the absorbed heat from the BOG by the work performed by compressor 1, assuming that the work performed by the second compressor is recovered by the work produced by the expander.

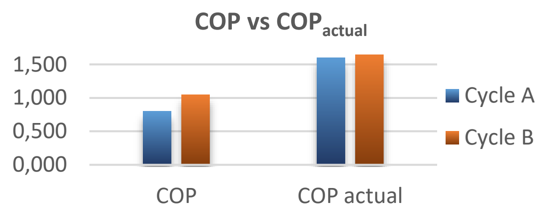

Figure 8 – COP - Cycle Comparison

Cogeneration

The different BOG scenarios enable the operation of a small/medium capacity cogeneration (1.3MW – 2.2MW), allowing the production of electricity on site, for self-consumption or to sell to the electric network, while producing heat that can be sold to the adjacent industries or also be used internally. Cogeneration is a method that allows energy conversion with reduced environmental impacts when compared with other forms of producing energy through fossil fuels. The avoided CO_2 emissions were calculated in order to perceive the positive impact of such an installation, and a monthly estimated reduction of 38 to 64 tons of CO_2 is expected.

Emergency System

Using the price of the LNG of 7$/MMBTU [20], which is equivalent to 0.02€/kWh, and assuming a LHV of 10.85kWh/Nm^3, the price of 1 Nm^3 of BOG is 0.2171€; it is estimated that for a BOG rate of 0.039%, the profit loss is 19 to 19.6k€ per month, while for 0.067% it is estimated 30.5 to 32k€.

6. Conclusions

With the relevant differences found between the two analysed BOG rates, the insulation costs, and the holding mode capacity of each scenario, it is important that the organization assesses further and with more detailed information in project phase, the cost vs benefit of choosing one BOG rate option or another. Either way, the BOG management should be adequate to the chosen option and be as efficient as possible. Simulations were performed to study a re-liquefaction system and a cogeneration plant burning BOG as fuel.

Regarding the re-liquefaction options simulated, Cycle B presents a COP value that is 24% higher than that of Cycle A, while the real COP value is only 3% higher, accounting for the work produced by the turbine to drive the second compressor. Also, the specific energy consumption of Cycle B is 24% smaller than that of Cycle A, which represents a significant amount of energy saved while performing the same liquefaction work; these values indicate Cycle B as the most energy efficient cycle, thus the most adequate to perform the BOG re-liquefaction.

With good results and without a great impact on the LNG storage in holding mode, the cogeneration plant might also be a good option to recover BOG as, despite the inherent inefficiencies of any internal combustion engine, the thermal and electrical energy produced can still be used for the benefit of the terminal, while avoiding CO_2 emissions – approximately 38 to 64 CO_2 tons per month. The cogeneration can be a good method to process the holding mode BOG, while having as backup (and capable of processing that and the BOG produced on loading and unloading events) a re-liquefaction system, based on the most energy efficient cycle simulated - Cycle B. Although the present study indicates some small-scale re-liquefaction plant issues, namely, specific needs in terms of BOG and re-liquefaction control system, the re-liquefaction system can be further improved, namely by adding a pre-cooling cycle to the BOG stream or studying other configurations of the cycle such as the dual expander cycle, although more expensive in terms of investment.

Cogeneration has also been proved an alternative, since the BOG produced is sufficient to power a small cogeneration (between 1.3 MW - scenario of lower BOG production - and 2.2 MW - scenario of higher BOG production). In addition, it enables primary energy savings while producing electricity, as well as avoiding GHG emissions (CO_2). It shall be a decision of the operator of the terminal to decide whether the electric and thermal energy produced can be consumed in the terminal or exported to any facility nearby the terminal.

One possibility of using the heat produced by a cogeneration in the terminal is replacing the electric resistances tracing in the bottom of the tank by a piping system on which heated fluid, capable of preventing the frost heave, circulates; nonetheless, the feasibility of such opportunity requires further study.

7. Future Work

The present thesis was performed as a preliminary work for a possible SSLNG terminal project at Trafaria (Lisbon port). It serves to identify the different available technologies that constitute one such SSLNG terminal. This study identified scenarios for the chosen location and capacities regarding the energy consumption related to the energy-consuming processes that an SSLNG receiving terminal entails, addressing the issue of the BOG management.

Regarding the tank, it might be interesting to simulate the tank design as a completely underground tank, i.e., the dome totally covered by earth, to assess the effect of protecting the tank from the direct effect of convection and irradiation, studying the respective energy efficiency improvement.

As future work regarding the re-liquefaction cycle, the calculations and methodology applied in this thesis may be used to study equipment supplier solutions and respective costs, i.e., a realistic project budget. In order to better understand the operational aspects of this SSLNG terminal, the present simulations should be complemented by performing detailed dynamic simulations regarding the loading and unloading processes of the tank, in order to perceive, conceive, and optimize the adequate solutions for the terminal.

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