A gas turbine/closed cycle solution for effective recovery in LNG regassifiers

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Abstract:
The problem of energy recovery from LNG regasification terminals is attracting scientific as well as industrial attention. Still few LNG regasification terminals are operating (with special reference to Europe), and a substantial margin of performance improvement appears to be possible with respect to current industrial solutions. Advanced research schemes for power recovery have been proposed, which however often entail complicated plant layouts. The idea here developed is to couple a modern gas turbine with an intermediate closed gas cycle, taking care of keeping low values of heat transfer exergy destruction. The closed gas cycle is coupled to the cold-end (LNG) side in two supercritical heat exchangers, that is, the heat rejection at closed cycle turbine exhaust, and the compressor intercooler. The intermediate pressure of the two-stage compressor is matched to guarantee uniform conditions between the two critical heat exchangers, so that the GNL flow is split in two approximately equal separate streams. Two different working fluids suitable for cryogenic operation were considered for the closed cycle: Helium and Nitrogen. The natural gas properties were modeled as a real mixture using Refprop. The performance estimation of the plant includes the calculation of the exergy balance and main environmental indicators. Attractive performance figures appear to be possible, both in terms of efficiency and of carbon footprint.

Keywords:
LNG regasification, Gas Turbines, Exergy.

1. LNG Regasification

Liquified Natural gas (LNG) has become a major energy resource, covering nowadays about 10% of the overall gas consumption; it is worth to recall that even if utilization of natural gas is always increasing (1,6% worldwide[1], with decreasing market in the EU), the rate of increase of LNG is much larger (7%) [2], reflecting a lower price and a high interest in developing this trade,with price of LNG largely depending on regional conditions. At 2015, the total capacity of liquefaction has reached 836 MTons per year –and the same capacity is approximately available at regasification terminals.

The present study is focused on the situation of Italian regasification terminals. At present, three terminals are operational as stated in Table 1. On the whole, the three terminals in Italy have an overall capacity of 15Bsm^3/yr[1]; however, the national gas consumption has decreased (-11,6 % from 2013 to 2014), so that only 4,5 Bsm^3 were imported as LNG in 2014: that is, something less than 30% of the Regasification capacity installed. It appears there is thus an overcapacity; even if the market price of LNG has scored values as low as 3,4 $/MBTU (about 3€/GJ), the reduced industrial production has hindered the planned further development of Regasification terminals. Of the three existing in Italy, two are relatively new, while the oldest one, located in La Spezia, uses a

[1]Correspponding to about 10% of the gas imported by pipeline [1].

reliable but relatively obsolete technology, that is, submerged gas burners, which are responsible of a consumption of about 1,5% of the carrier natural gas during the Regasification process. Under this scenario, it makes sense to consider the economic and environmental potential advantages of introducing a power production island in an existing regasification terminal.

Table 1- Regasification terminals in Italy [2]

Terminal	Capacity, MTPY	Location	Type	Regasification Technology	Year
Adriatic LNG	5,8	Monfalcone	Offshore	Seawater + indirect WHR from GT	2009
OLT LNG	2,7	Livorno	Ship	Seawater	2013
GNL Italy	2,5	La Spezia	On-Shore	Submerged Gas Burners	1971

2. Plant layout and model setup

2.1. Reference case

Considering the situation in Table 1, it was decided to consider the possible refurbishment of the Regasification plant of GNL Italy as reference case; the idea was to propose a simple but effective way of coupling power production by a modern gas turbine unit, of a size compatible with the existing terminal, with the regasification heat demand.

With respect to documented alternatives for large terminals, using effective but complex pressure-staged Organic Rankine Cycles [3], it was decided to adopt a gas turbine combined with an intermediate closed-cycle, capable of taking advantage of cryogenic heat rejection; the idea is inspired by other researchers [4, 5, 6, 7], but relevant changes were introduced as shown in Fig. 1. The Closed Gas Turbine (CGT) cycle uses a non-flammable working fluid (WF) suitable for use at cryogenic conditions (Nitrogen or Helium in alternative, with adjustable base pressure).The LNG is gasified splitting the design flow rate in two streams: one is directed to the outlet of the intercooled compressor (streams 12 – 13for LNG; 2 – 3 for WF); another recovers heat at CGT discharge (streams 10 – 11 for LNG; 7 – 1 for WF). The flow rates and the interstage intercooler pressure are adjusted so that equal and acceptable delivery temperature conditions (-5 to 10°C) are achieved at the exit of the two heat exchangers. At final compressor discharge, the CGT working fluid exchanges heat first with seawater SW (indirect shell and tube heat exchanger, streams 4 – 5 for WF; 14 – 15 for SW), then recovering heat at the discharge of a gas turbine (HRSG, streams 5 – 6for WF; 16 – 17for exhaust gases).

The whole plant was designed to process a nominal LNG flow rate of 36,77 kg/s. The composition of LNG and its thermodynamic conditions at carrier discharge and plant outlet are listed in Table 2.For the Gas Turbine (GT), several options were considered: the best performance were achieved with a Siemens SGT-800 unit[2]. The GT was modelled as an equivalent air unit, adjusting the cooling by-pass (5% of compressor flow rate) so that the actual unit main design parameters were met (W = 41,6 MWe; m_g = 131,5 kg/s, T_{20} = 1233°C; T_{21} = 539,4 °C). With this GT unit, the value

[2]http://www.energy.siemens.com/nl/en/fossil-power-generation/gas-turbines/sgt-800.htm

of T_{21}= 539,4 °C results high enough, so that $T_{16} = T_{21}$ and it is not necessary to activate the PostCombustion unit (PC), which is required when considering GTs with lower exhaust temperature.

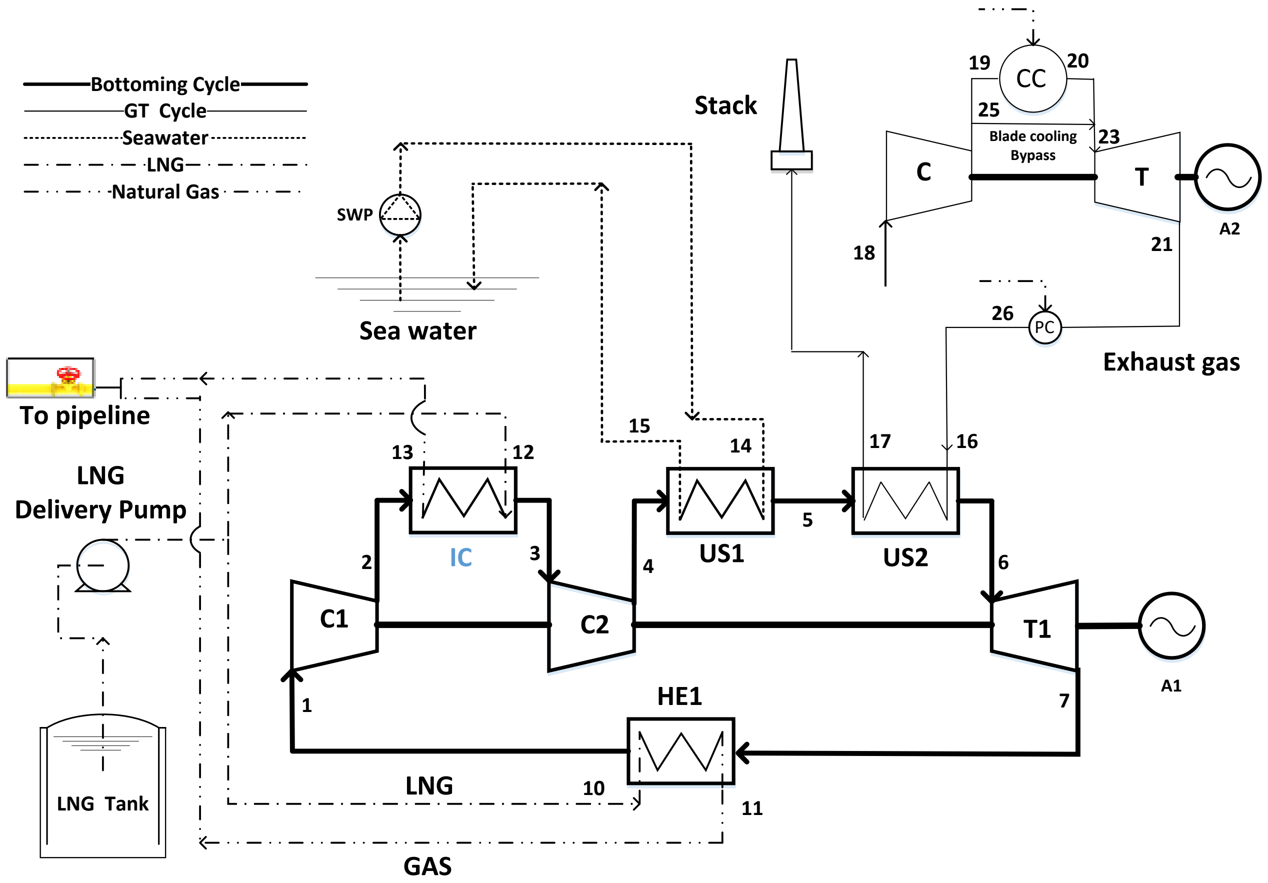


Fig. 1 – Proposed Regasification plant layout.

Table 2- Natural Gas composition and inlet/outlet conditions.

LNG and NG main process data	
CH_4	0,888
C_2H_6	0,076
C_3H_8	0,026
C_4H_{10}	0,010
$T_{LNG, in}$	-160°C
$p_{LNG,in}$[3]	80 bar
$T_{NG, out}$	>3°C
$p_{NG, out}$	75 bar

[3] After compression in LNG pump, supercritical conditions

2.2. Model equations

The CGT with intercooled compression is modelled with well-known equations for gas turbine cycles. The fundamental input data are resumed in Table 3.

Table 3- Main model input data.

T_1, °C	-150
η_{c1}, η_{c2}	0,85
η_t	0,9
$\Delta T_{c,IC,}$ °C	25
$\Delta T_{h,HE1,}$ °C	25
T_5, °C	15
T_6, °C	500

Starting from the closed cycle low-pressure compressor inlet,

$$h_2 = h_1 + (h_{2s} - h_1)/\eta_{c1} \quad (1)$$

Where h_{2s} is the isentropic enthalpy at end of compression, calculated using real fluid properties and with $s_{2s} = s_1$. The power absorbed by compressor 1 is then given by:

$$\dot{W}_{c1} = \dot{m}_{wf}(h_2 - h_1) \quad (2)$$

Assuming to cool down at temperature $T_3 = T_1$, it is possible to calculate the IC heat duty:

$$\dot{Q}_{IC} = \dot{m}_{wf}(h_2 - h_3) = \dot{m}_{LNG,IC}(h_{13} - h_{12}) \quad (3)$$

The conditions at point 4 (discharge of second compression stage) are calculated as for the first compressor:

$$h_4 = h_3 + (h_{4s} - h_3)/\eta_{c2} \quad (4)$$

$$\dot{W}_{c2} = \dot{m}_{wf}(h_4 - h_3) \quad (5)$$

Taking T_5 from Table 3, the seawater heat exchanger heat duty can be calculated as:

$$\dot{Q}_{US1} = \dot{m}_{wf}(h_5 - h_4) = \dot{m}_{SW}(h_{14} - h_{15}) \quad (6)$$

And, for the HRSG at GT discharge:

$$\dot{Q}_{US2} = \dot{m}_{wf}(h_6 - h_5) = \dot{m}_g(h_{16} - h_{17}) \quad (7)$$

The other closed cycle points are calculated in a similar way; the turbine is modelled by:

$$h_7 = h_6 - (h_6 - h_{7s}) * \eta_t \tag{8}$$

$$\dot{W}_t = \dot{m}_{wf}(h_6 - h_7) \tag{9}$$

The energy balance of the heat recovery heat exchanger can be written as:

$$\dot{Q}_{HE1} = \dot{m}_{wf}(h_7 - h_1) = \dot{m}_{LNG,HE1}(h_{11} - h_{10}) \tag{10}$$

with $h_{10} = h_{12}$ and $h_{11} = h_{13}$ (same conditions for LNG inlet and NG delivery).

The gas turbine cycle was modelled following consolidated practice for performance prediction, including the choice of the cooling by-pass flow, which was adjusted to match the power and outlet gas temperature.

On the whole, the set of equations 1 - 10 allows to calculate the points and the power production of the closed cycle, as well as the productivity $\dot{m}_{LNG,HE1}$ and $\dot{m}_{LNG,IC}$. The system of equations was programmed in EES and a standard dispersed matrix system solution approach was applied [8].

2.3. Thermodynamic properties; variable specific heat.

For the thermodynamic properties of all fluids, a real fluid model was used. The working fluid properties were takenfrom the internal EES libraries, which extend accurately for both Helium and Nitrogen to cryogenic conditions. At the beginning of the study, natural gas was modelled as pure methaneusing the same libraries. At a further stage of development, it was decided to provide a more accurate description of the properties of natural gas using the EES-REFPROP interface, which allows themodellingof a mixture of up to five components, accurately selected among the most representative of average natural gas composition (Table 2). The same library was also used to model the properties of the gas turbine working fluid downstream the combustion chamber.

The applied models include variable specific heat for all fluids; this is particularly relevant for the mixture of natural gas under supercritical conditions, as encountered in the Regasification process. Consequently, for both heat exchangers HE1 and IC, a piecewise description was adopted, obtained segmenting the enthalpy interval in small sub-intervals (typically, 20 to 50). Examples of the resulting calculation of the Q-T curves are shown in Figs. 2 (Nitrogen) and 3 (Helium). It is clear that the working fluid (upper line) is well described as having a nearly-constant specific heat[4], while Natural Gas (lower line), as a complex mixture, shows a remarkable variation of heat capacity during the supercritical regasification process. This means that local pinch conditions could exist: the graphical representation shown in Fig. 2 was used to check that a positive and controlled temperature difference between the hot and cold stream (here, larger than 5K) is maintained across the heat exchangers.

[4]This is completely true for Helium, which behaves as a perfect gas even at the very low temperatures encountered in this process; it is approximately true for Nitrogen, which shows numerically a marginal deviation from perfect gas behavior for the same temperature conditions.

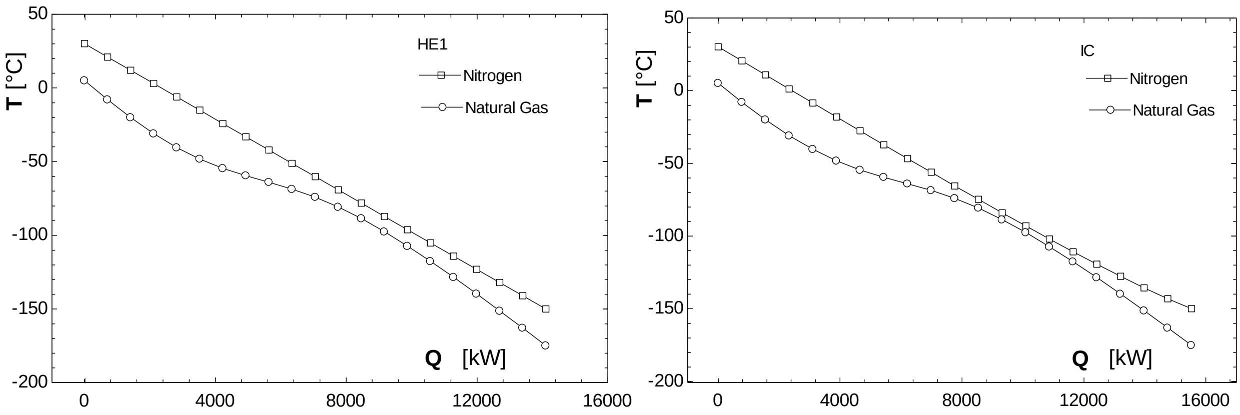


Fig. 2 – Example of calculated Q-T Profiles, Nitrogen (a) HE1 (b) IC.

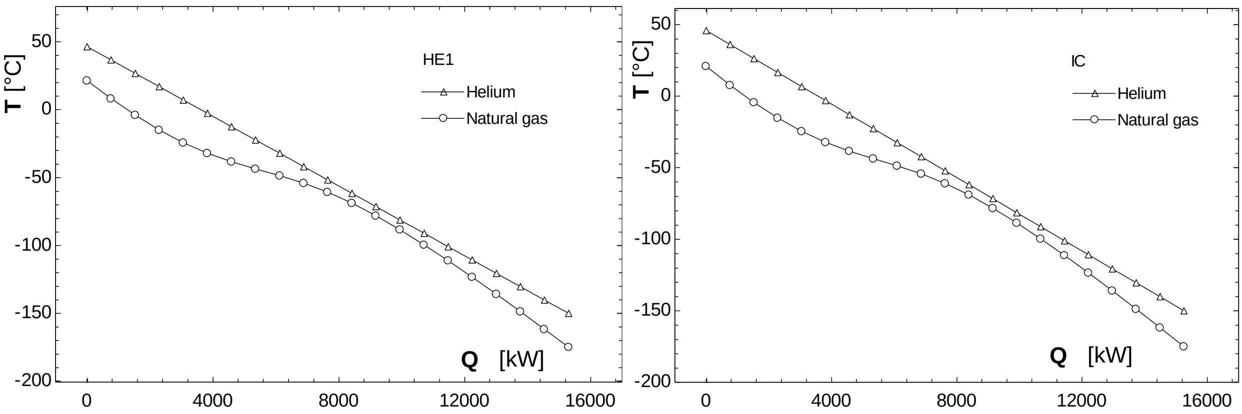


Fig. 3 – Example of calculated Q-T Profiles, Helium (a) HE1 (b) IC.

2.4. Energy and exergy balance

For all calculated thermodynamic points – including the gas turbine cycle, exergy was calculated as:

$$\varepsilon_i = (h_i - h_o) - T_o(s_i - s_o) \tag{11}$$

$$E_i = \dot{m}_i \varepsilon_i \tag{12}$$

Once known the exergy ofeach stream, it is possible to calculate the exergy destructions and losses in all the plant components [9]. The exergy inputwas calculated approximating the LNG chemical exergy to the lower heating value in standard conditions, thus:

$$\dot{E}_{in} = \dot{m}_f HCI + \dot{m}_w(\varepsilon_{14} - \varepsilon_{15}) + \dot{m}_{LNG\_IC}\varepsilon_{12} + \dot{m}_{LNG\_HE1}\varepsilon_{10} \tag{13}$$

The useful exergy output of the system is given by:

$$\dot{E}_{out} = \dot{W}_{GT} + \dot{W}_{CC} + \dot{m}_{LNG\_IC}\varepsilon_{13} + \dot{m}_{LNG\_HE1}\varepsilon_{11} \tag{14}$$

The system exergy efficiency can be calculated either directly or indirectly, thereby checking the correctness of the exergy balance:

$$\eta_x = \frac{\dot{E}_{out}}{\dot{E}_{in}} = 1 - \sum \frac{\dot{E}_{D-L}}{\dot{E}_{in}} \tag{15}$$

3. Results

3.1. Operating conditions for the closed cycle

A key issue is the selection of the most appealing operating conditions for the closed cycle. The main performance parameters are the cycle efficiency $\eta$and specific workW_s. The main design variables are:

a) The closed cycle pressure ratio β
b) The closed cycle base pressure p_1
c) The fractional partition of β between compressors C1 and C2

In order to examine parametrically point (c), the following power law partition rule was applied:

$$\beta_1 = \beta^z \tag{13}$$

With 0<Z<1 covering the entire range of options[5]. The response to the Z parameter can be best interpreted in terms of performance map η versus W_{sp}; these are shown respectively in Figs. 4 and 5 for Nitrogen and Helium.

Figs. 4 and 5 show that a very good performance can be achieved, with a cycle efficiency in the range of 40 to 50%. Nitrogen and Helium behave quite differently in terms of specific work: this is in the range of Ws = 250 kJ/kg for Nitrogen, while the choice of Helium is attractive as it allows to achieve a high value of Ws≅ 1000 kJ/kg. As anticipated, Nitrogen displays some sensitivity to the base pressure, while with Helium, as an ideal gas,the performance are insensitive to pressure.

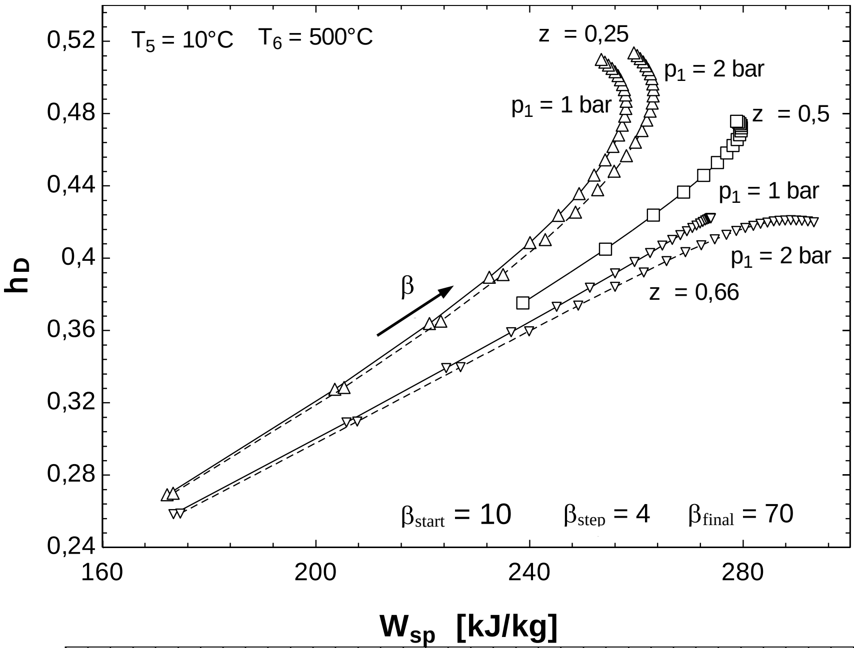
The two working fluids display a significantly different W_{sp}- η map behaviour:

- Nitrogen has a typical initially linear increasing map; at low z (z = 0,25) a maximum condition is reached for specific work; at high Z (Z = 0,66), with special reference to a base pressure p_2 = 2 bar, a maximum in efficiency is achieved. High values of β (β> 40) can be recommended.

[5] z = 0,5 corresponds to the traditional even partition of the cycle pressure ratio between the two compressors, commonly applied in intercooled gas turbine cycle practice as the rule approximately maximizing power output; here, the IC is also used to produce a part of the regasification heat duty, so that the criterion for selection of z is not only represented by power output.

- Helium displays a typical perfect – gas simple cycle behaviour, with optimization possible with increasing β: first for W_{sp} and then for η; values of β between 16 and 20 can be recommended

Fig.4 – Closed cycle performance maps; Nitrogen, p_1 = 1 or 2 bar.



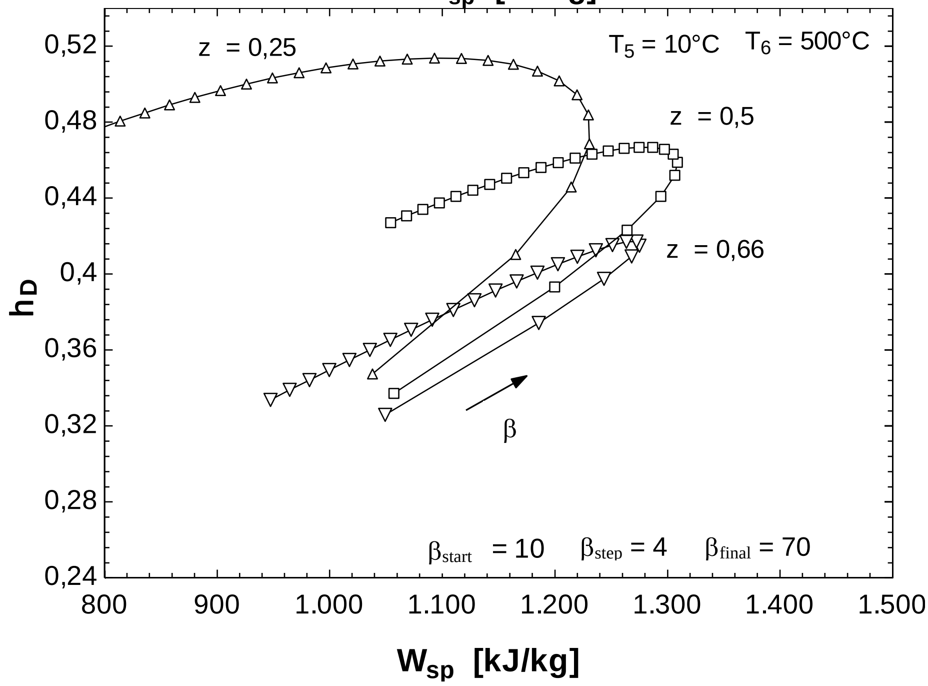


Fig.5 – Closed cycle performance maps; Helium.

Figs. 6 and 7 show typical T-s diagram for the two working fluids. For the choice of β, it is necessary to consider the constraints imposed by the process: specifically, the acceptable delivery temperature should be kept within -5°C < T_{11}< 10°C; and the closed circuit maximum pressure should not be raised beyond 100 bar for safety and cost reasons. On the other hand, in the low-pressure section, it is necessary to maintain a reasonable volumetric flow rate, and thereby a certain degree of pressurization is needed to avoid excessive splitting of the piping lines.

The value of Z, which was kept as a free parametric variable in Section 3.1, is now automatically determined imposing the additional condition of equal delivery temperature for the two regasified natural gas streams, that is, $T_{11} = T_{13}$.

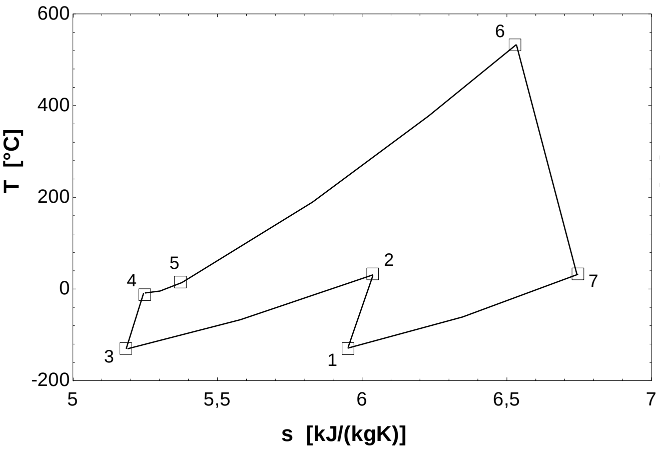


Fig. 6 – T-s diagram, Nitrogen.

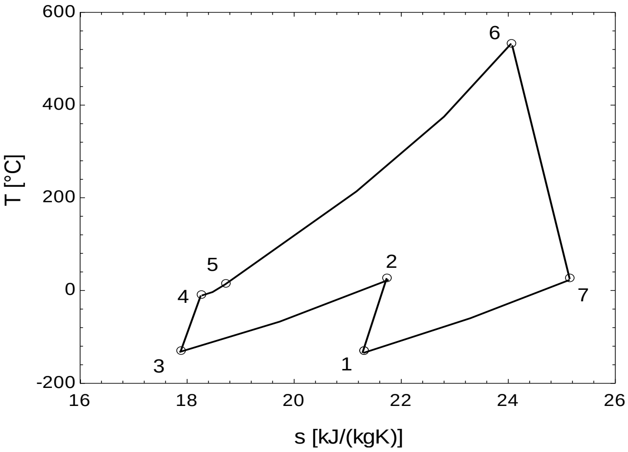


Fig.7 – T-s diagram, Helium.

3.2. Performance

Table 4 reports the calculated points and the main performance indicators of the GT cycle. The same data for the Nitrogen and Helium cycles are reported in Tables 5 and 6 respectively.The comparison of Tables 5 and 6 indicates that a similar overall performance can be obtained with both fluids; however, Nitrogen requires a high pressurization of the closed cycle, and achieves a somewhat larger value of T_{11} = 4,6°C (compared to Helium with T_{11} = -1,3°C). The Closed Cycle power and efficiency are similar for both fluids, with a marginal advantage for Nitrogen. It is important to notice that, for Helium, the closed cycle works at lower pressure ratio (β = 20 instead of β = 66 for Nitrogen), which allows the adoptionof a higher base pressure (p_1 = 4 bar for Helium instead of p_1 = 1,5 bar for Nitrogen) and thus a smaller volume flow rate, that is, a more compact power plant. Table 7 compares the overall performance of the whole system for the two working fluids.

Table 4 - Calculated points and performance of the GT cycle.

Point	p, bar	T, °C	h, kJ/kg	s, kJ/(kgK)	Calculated Performance			
18	1	15	288,4	6,83	$\dot{m}_a$, kg/s	131,5	$\dot{Q}_{GT}$, kW	133111
19	19,9	457,5	747,1	6,93	$\dot{m}_f$, kg/s	2,737	$\dot{W}_{GT}$, kW	41621
20	19,9	1233	1790	7,66	β	20	$\dot{Q}_{HRSG}$, kW	65003
21-16	1	539,4	942,7	7,77	η_t	0,9	η_{GT}	0,313
17	1	95	458,4	6,91	η_c	0,85	η_{HRSG}	0,74

Table 5 - Calculated points and performance of the Nitrogen cycle.

Pt.	p bar	T °C	h kJ/kg	s kJ/(kgK)	e kJ/kg	Calculated Performance			
1	1,5	-130	147,1	5,951	102,7	z	0,56	$\dot{Q}_{US1}$, kW	3645
2	15,7	31,6	313	6,036	243,3	β	66	$\dot{Q}_{NG}$, kW	39300
3	15,7	-130	133,4	5,188	316,4	ΔT_{IC}, °C	27	W_s, kJ/kg	260,3
4	99	-11,	245,5	5,254	408,9	ΔT_{HE1}, °C	27	$\dot{W}_{CC}$, kW	29349
5	99	15	277,9	5,372	406,1	$\dot{m}_{WF}$, kg/s	113	η_{CC}	0,428
6	99	529,	854,3	6,524	639,1	$\dot{m}_{LNG1}$, kg/s	26,8	T_{11}, °C	4,6
7	1,5	31,6	316	6,741	35,95	$\dot{m}_{LNGIC}$, kg/s	28,5	$\dot{V}_1$, m3/s	31,62

Table 6- Calculated points and performance of the Helium cycle.

Pt.	p bar	T °C	h kJ/kg	s kJ/(kgK)	e kJ/kg	Calculated Performance			
1	4	-130	749,7	21,31	1191	z	0,55	$\dot{Q}_{US1}$, kW	2937
2	20,6	32,2	1564	21,74	1878	β	20	$\dot{Q}_{NG}$, kW	39393
3	20,6	-130	754,5	17,91	2209	ΔT_{IC}, °C	27	W_s, kJ/kg	1172
4	80	-8,2	1407	18,3	2748	ΔT_{HE1}, °C	27	$\dot{W}_{CC}$, kW	28547
5	80	15	1528	18,73	2738	$\dot{m}_{WF}$, kg/s	24,4	η_{CC}	0,420
6	80	529	4197	24,05	3823	$\dot{m}_{LNG1}$, kg/s	28,5	T_{11}, °C	-1,3
7	4	25,7	1558	25,14	859,5	$\dot{m}_{LNGIC}$, kg/s	28,5	$\dot{V}_1$, m3/s	18,17

Table 7 – Comparison of performance, combined cycle; Nitrogen and Helium working fluid.

Parameter	Nitrogen	Helium
Combined Cycle Efficiency	0,5301	0,5264
Power Output, kW	70970	70168
Relative consumption of regasified LNG, %	4,95	4,81
Annual electricity Generation, GWh	621,7	614,7
Annual Regasification Capacity, Bm^3/yr	2,366	2,408
CO_2 emission Factor, g/kWh	516,2	522,1

3.3. Exergy analysis

Further insight in the interpretation of performance - as well as in the margins for process improvement - can be retrieved performing an exergy analysis of the combined cycle. This last is resumed in Figure 8, reporting the main exergy destructions and losses ($\dot{E}_{D-L}$, non–dimensional with respect to the system inlet exergy). The exergy efficiency results to be η_x = 0,547 for Helium and η_x = 0,551 for Nitrogen –close to the combined-cycle efficiency, as the main product of the process is electricity (the exergy of the natural gas at pipeline inlet represents a marginal contribution to the inlet exergy).

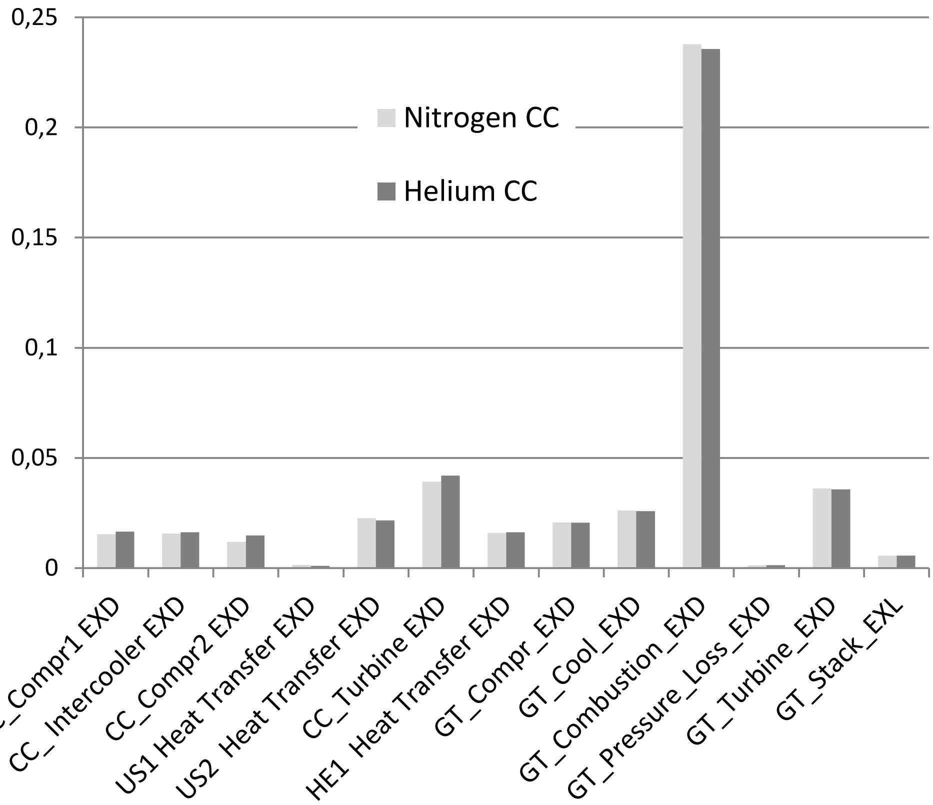


Figure 8 –Exergy destructions and losses, combined cycle; Nitrogen and Helium.

The analysis of the exergy destructions and losses confirms that the largest contribution comes from the Gas Turbine combustion process, with a score of about 24% of the inlet exergy (which is anyway somewhat lower than commonly encountered by isolated gas turbine cycles, typically ranging between 27 and 30% for the combustion irreversibility). All other exergy destructions are lower than 4%, the largest ones in this group being the closed – cycle turbine and the gas turbine. It is remarkable that all terms related to heat transfer exergy destruction (IC, HE1, US1, US2) are below 2%, indicating that the system is performing well in terms of optimization of the heat transfer/recovery network. However, the plant is relatively fragmented – with 13 $\dot{E}_{D-L}$ terms, so that the exergy efficiency is about 0,55. The exergy balance represented in Figure 8 looks similar for the two working fluids of the CC, which show only slight differences in some exergy destruction terms.

4. Conclusions

The proposed combined-cycle layout, built adding an intercooled closed-cycle to a modern gas turbine cycle, provides a performance level in terms of efficiency (about 55%) exceeding the best-performing combined cycles in its power range (70 MW). Another interesting feature is the attractive CO_2 footprint (about 520 g/kWh). This should be compared to the 350-400 g/kWh for the best available gas turbine combined cycles – typically of much larger size; however, if fed by LNG, about 50 – 100 g/kWh should be added to the gas turbine combined cycle figures depending on the technology applied at the regasification terminal.

The exergy analysis shows that the largest exergy destructions are due to gas turbine combustion, and that the heat recovery/transfer network is well optimized. Improvements should be sought in improving the performance of expanders and examining promising schemes for compressor precooling, which appear possible within the context of LNG regasification plants.

As for the working fluid, both Nitrogen and Helium provide good values of efficiency (about 52%). Nitrogen represents an attractive low-cost alternative, partially hindered by the higher values of optimizing pressure ratios (β = 66 in the present case) and by scores of specific work in line with air-breathing gas turbines ($W_s \cong$ 250 kJ/kg). Helium pays a small penalty in efficiency, but is distinguished by a very high specific work ($W_s \cong$ 1200 kJ/kg), with a moderate pressure ratio (β = 20); however, it is a much more expensive working fluid . The base pressure should also be adapted (p_1 = 1,5 for Nitrogen and p_1 = 4 bar for Helium in the present case) and the consequences in terms of size of piping and heat exchangers should be well considered, with the large volumetric flow rate in the low-pressure circuit section penalizing the Nitrogen solution. Both options are considered as possible valid alternatives, and the final choice should mainly consider the different technological challenges connected to the two fluids.

On the whole, the proposed cycle layout appears relatively simple and suitable for application in small/medium size regasification terminals. Considering the necessity of developing dedicated components (specifically, compressors and expanders), it appears that the Nitrogen option can be more easily developed with respect to the Helium closed cycle loop. A complete exergo-economic analysis, and a study of the adaptability to seasonal operation and capability of compensating the off-design of the gas turbine unit will be a subject of future development, as the scheme appears to offer interesting developments within this context. The competitive low carbon footprint ($\cong$ 520 gCO_2/kWh) and the high efficiency should guarantee that this type of plants have priority in power generation with respect to other power plants, producing considerable environmental benefits.

Nomenclature

E	exergy (overall), kW
$\dot{E}_{D-L}$	exergy destruction or loss, kW
h	enthalpy, kJ/kg
$\dot{m}$	mass flow rate, kg/s
p	pressure, bar
$\dot{Q}$	heat rate, kW
s	entropy, kJ/(kg K)
T	temperature, °C
W_{sp}	specific work, kJ/kg
$\dot{W}$	power, W

Greek symbols

ε	exergy, kJ/kg
η	efficiency

Subscripts and superscripts

C1 compressor 1
C2 compressor 2
CC closed cycle
f fuel
GT Gas Turbine
HE1 heat exchanger 1
i point i
IC intercooler
LNG Liquefied natural Gas
o reference conditions
t turbine
w seawater (US1)
wf working fluid

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