# BIOMETHANE GRID INJECTION OR BIOMETHANE LIQUEFACTION: A TECHNICAL-ECONOMIC ANALYSIS

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**ABSTRACT**

The gradual reduction of subsidies for electricity production from biogas and the raising interest of bio-methane as an integration to natural gas market forces the biogas plant owners to choose alternative solutions for biogas exploitation.

In this study two systems for biomethane distribution have been compared: liquefied biomethane and biomethane grid injection were investigated as a function of gas connection cost, electric tariff, selling price and type of expander adopted in the liquefaction cycle (radial turbines or screw expanders). A nitrogen Joule-Brayton reverse cycle was considered for liquefaction and a detailed model of the cycle was developed in Aspen Hysys, by minimizing the energy specific consumption. Results showed that expander efficiency has a key role in the liquefaction scenario and screw expanders led to specific consumption 1.45 times higher than radial turbines but reduced capital costs by a factor of 1.39.

Biomethane grid injection was the preferable solution in terms of investment risk if connection cost was below 500 k$, independently from the electricity price, whilst liquefaction through a cycle equipped with radial turbines maximized the profit up to electricity price of 0.23 $/kWh. A sensitivity analysis on products selling price showed that biomethane grid injection was always the most convenient solution when connection costs are low while for higher connections cost, liquefaction with radial turbines has major probability of minimizing the investment risk and maximizing the profit for a given combination of selling prices.

Keywords: Biomethane, Liquefaction, Biomethane Grid Injection, Aspen Hysys, Economic Analysis

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## 1 INTRODUCTION

Biogas is a renewable energy source which can be produced from waste organic materials such as manure, organic waste from agricultural activities and sewage from water treatment plant. Sewage treatment may represent a significant contribution to the increase of biogas production [1] especially if co-digested with municipal bio-waste to increase methane concentration of the resulting biogas [2–5]. In this sense, anaerobic digestion plants satisfy two vital functions of the society: clean power generation, by producing biogas, and provide sanitation services, by using organic wastes as an input [6]. Biogas is normally burned in a cogenerator to provide electric energy and heat for the process [7,8]. Recently, many European countries promoted, and supported through governmental subsides, the production of biomethane to be injected into natural gas grid or to be liquefied. In this perspective, anaerobic digestion plants may be seen as poly-generation units, where electricity, heat and fuel are produced [9]. For this reason, biomethane produced from biogas has been attracting more and more interest, since it is a renewable fuel which can integrate natural gas [10] and liquefied natural gas (LNG) [11]. LNG is an attractive energy vector since has a volume about 600 times lower than natural gas at standard conditions, thus making it easy to store and ship. The high energy density allows LNG to be adopted as heavy-duty vehicle fuel, such as ship and trucks [12–14], with much lower emissions than traditional diesel and heavy-fuel oil [15–18].

Liquefaction of natural gas requires the removal of heat over a wide range of temperatures by using one or more refrigerants in a complex refrigeration system [19,20]. Commercial processes currently used for natural gas liquefaction are designed for large scale LNG productions (6 000 tonnes of LNG per day) and can be classified into three general categories:

- cascade processes using pure refrigerants;
- mixed refrigerant processes using refrigerant mixtures;
- expander processes using expanders instead of Joule−Thomson (J−T) valves.

These plants are known to be energy consuming, since they require a large amount of power for compression and refrigeration. In addition, they require special equipment as cryogenic heat exchangers, compressors, drivers and turbines. An increased attention has been recently put to the study and development of solutions

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for small and micro-scale plants (from 5 to 500 tonnes of LNG per day) [21–23]. This trend is confirmed by the recent construction of plants of this capacity, such as the Cryo Pur plant, which continuously produces about 3 tonnes per day of liquefied biomethane in Northern Ireland [24], or the world biggest plant of LNG from landfill gas at Altamont Landfill and Resource Recovery Facility in Livermore, California [25], generating roughly 24 000 litres of liquefied biomethane per day. The most suitable process for small-scale liquefaction plants may differ considerably from a large-scale application since these solutions are neither practical nor economical when applied to small plants.

The key factor for the development of small-scale plants is the trade-off between efficiency and costs. Khan et al. [26] claim a specific electric energy requirement of 0.745 kWh/kg for a single nitrogen expander processes and 0.501 kWh/kg for dual nitrogen expander processes.

Ancona et al. [27] investigated two different LNG production layouts and showed the benefit of using an expander which considerably reduces the system consumption.

Qyyum et al. in [28] proposed a refrigeration cycle based on a vortex tube integrated with a turbo-expander to reduce the overall energy request for LNG production. The hybrid vortex-tube turbo-expander LNG process resulted in a specific energy requirement of 0.59 kWh/kg.

Nitrogen expansion technology (with several variations, as single or dual expansion process, with or without a precooling cycle, etc.) seems to be the most suitable process for small-scale LNG production plants, as it combines several positive aspects in terms of energy consumption, economic performance, safety, and operability.

The production of compressed biomethane to be injected in the grid has been analyzed in various studies, most of which focuses on the economic feasibility of this solution. Hoo et al. in [29] determined, the value of the carbon tax which makes biomethane grid injection feasible. Cucchiella et al. in [30] estimated the profitability of injecting biomethane in the gas grid as a function of the anaerobic digester feed. They found that the most profitable case was achieved when the digester was fed with municipal organic waste.

This study compares the technical and economic feasibility of two systems for biomethane distribution:

- biomethane liquefaction;

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- biomethane injection into the natural gas grid.

The analysis is carried out by considering an anaerobic digestion plant producing about 23 300 $m^3/d$[1] of biogas (60% $CH_4$), corresponding to a capacity of about 10 tonnes/day of biomethane. The analysis was carried out by considering:

- liquefaction and transportation to final users for the first system (liquefaction)
- compression system and piping for the second (biomethane grid injection).

The pressure of the biomethane at the liquefaction unit or gas grid injection system was assumed of 0.4 MPa, a typical value of a PSA (Pressure Swing Adsorption) upgrader. The liquefaction system is based on a reverse dual-pressure Joule-Brayton regenerated cycle with nitrogen as a working fluid. It was designed by using the commercial code ASPEN Hysys and optimized from the electric specific consumption point of view ($kWh/m^3$ of raw biogas processed). Two different expanders were considered for the cycle: a radial turbine and a screw expander. These components have a key role in the liquefaction process because they largely decrease the nitrogen temperature in addition to recover part of the compression work. The higher is the isentropic efficiency of these machines the lower is the temperature reached. Radial turbines can achieve very high efficiency but are characterized by high costs and may not represent the best solution. For this reason, screw expanders, which have lower costs and efficiency, were also considered in this work. After modeling the liquefaction system and obtaining cost information, an economic analysis of the two solutions has been conducted for the two types of expander in terms of Net Present Value (NPV) and Profitability Index (PI). Several variables were considered to find out the best option for biomethane distribution as biomethane and liquefied biomethane selling price, electricity cost, connection cost (for the biomethane injection).

## 2 CASE STUDIES

The two systems described in this paper are schematized in fig. 1. In the first system (liquefied biomethane production), after an upgrading unit, the gas is processed in a liquefaction unit based on a reverse Brayton cycle. After liquefaction, biomethane is stored in a tank and delivered to final customers (heavy duty-vehicles) by trucks. Due to the limited production, a local user was assumed.

[1]All volume flow rates are evaluated at 0 °C and 101 kPa.

In the second system, after an upgrading unit, the gas is processed in a compression station and then injected in the nearest natural gas pipeline.

For both the two systems, a mass flow rate of 10 tonnes/day of biomethane from the upgrading system was assumed, this corresponding to a biogas production of about 23 300 $m^3$/d (60% $CH_4$). The pressure of the biomethane from upgrading was 0.4 MPa for both the systems. This is a common pressure if a PSA upgrading system is adopted as an upgrader. The control volume adopted in this study considers only the liquefaction unit and truck transportation (distribution) for the first system and the compression unit and pipeline up to natural gas grid connection for the second system, as reported in fig. 1. Anaerobic digestion and upgrading facilities are not included in the analysis since they are common to the two systems, and an equivalent cost of biogas production and upgrading was considered.

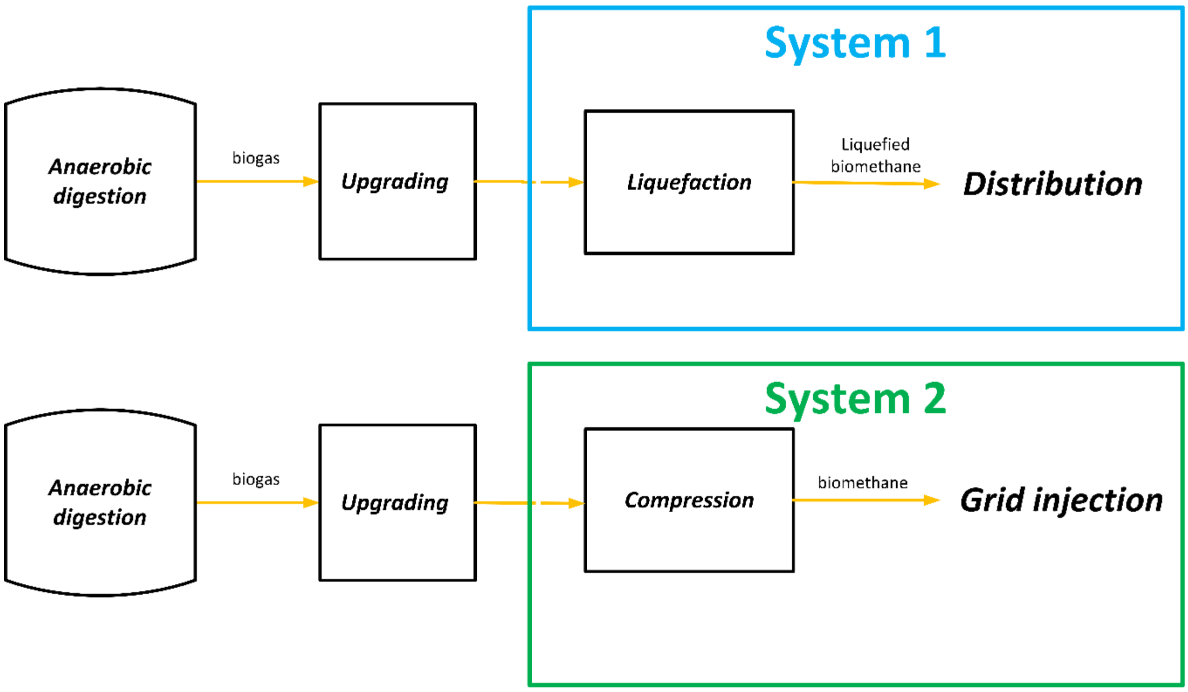


***Figure 1**: Scheme of the two systems analyzed*

## 3. METHODOLOGY

*3.1. First system: production of liquefied biomethane.*

In this system, biogas from upgrading is sent to the liquefaction unit, where it is compressed in a three-stage intercooled compressor and sent to the cold-box where the temperature is lowered by cold nitrogen (fig. 2). After the cold-box, most of the methane is in the liquid form and is therefore sent to a storage tank after a

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lamination. The boil-off due to the incomplete liquefaction of bio-methane and to thermal losses is recompressed and recirculated back to the first stage of the bio-methane compressor.

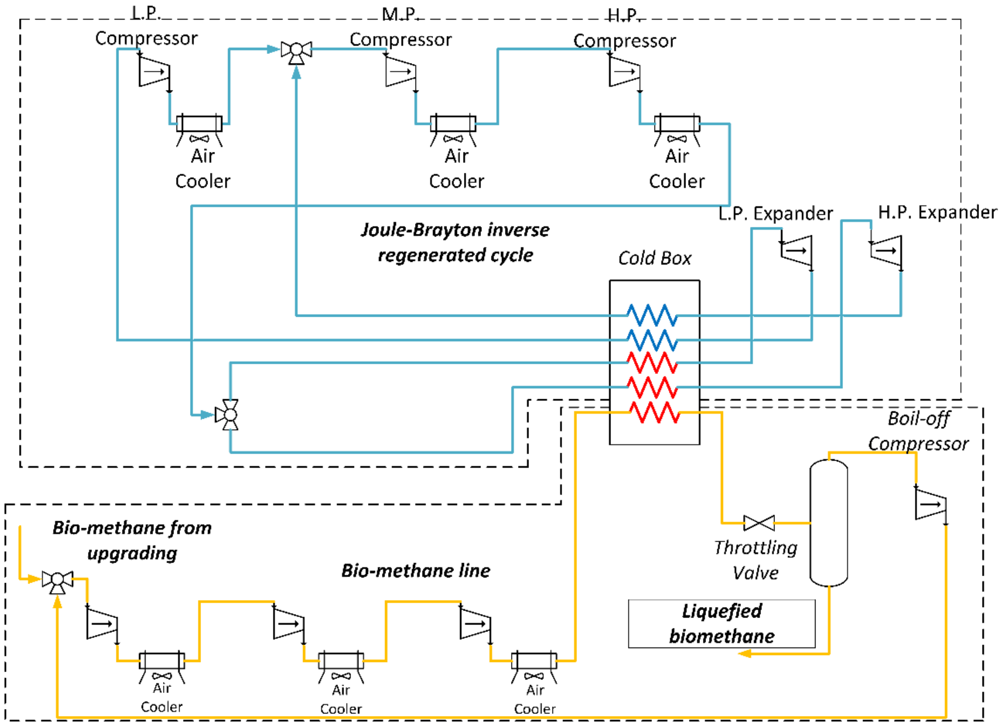


***Figure 2***: *Liquefaction system*

Regarding the double-pressure Joule Brayton cycle, the whole mass flow rate of nitrogen is compressed from medium pressure up to the maximum pressure of the cycle in a two-stages intercooled compressor. After the high-pressure stage compressor, the heat is removed in an air cooler. Nitrogen flow rate is divided in two streams: a first stream flows in the cold box for cycle regeneration and then in the high-pressure expander. This expander operates under low expansion ratio and the temperature of the stream at expander outlet is not enough for methane liquefaction. However, this stream is useful to regenerate the cycle, by cooling down the flow temperature of the second stream in the cold box and to partly pre-cool methane. After the expansion in a HP expander, cold nitrogen is sent back to the cold box where it regenerates the Brayton cycle and then flows back to the medium pressure compressor. The second stream of nitrogen flows in the cold box where it transfers the heat to the first stream coming from the expander and then is directed to the low-pressure expander. This expander works with a high expansion ratio and, after the expansion, nitrogen temperature is very low (below -160°C). This stream flows to the cold box to cool down and liquefy the biomethane. After the cold box the

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stream is sent to the low-pressure compressor, cooled in an air cooler and then mixed with the first stream to restart the cycle.

The compressors of both methane and nitrogen were assumed to be screw type compressors whilst for the expanders two different cases were considered:

- Screw expander;
- Radial turbine.

The first type of expander is a volumetric machine, characterized by low speeds and therefore by low costs. A reverse Joule-Brayton cycle working with this type of device was patented by [31].

The second type of expander considered in this study, i.e. a radial turbine, can achieve higher efficiency than volumetric expanders, but is characterized by high cost and high rotational speed due to the small size of the plant.

The cold box is generally made up by a multi-stream plate and fin heat exchanger. This type of exchangers is common in natural gas liquefaction processes and allows the simultaneous heat transfer from various flows (up to 10).

The storage tank is a cryogenic tank with an operative pressure of 0.15 MPa.

The model of the system was developed in Aspen Hysys. For the compressor, a constant isentropic efficiency of 0.75 was considered, while for the expander the constant isentropic efficiencies of 0.65 and 0.85 were considered for screw expanders and radial turbines respectively.

The work requested by the compressor was estimated as:

$$\dot{W}_c = \frac{\dot{m} \cdot \Delta h_{is}}{\eta_{is,c}} \tag{1}$$

The power from the expander was estimated as:

$$\dot{W}_e = \dot{m} \cdot \Delta h_{is} \cdot \eta_{is,e} \tag{2}$$

Main heat transfer processes were optimized by using a multi-stream exchanger, where pinch analysis was adopted to determine the solution that optimize the heat transfer process.

For the estimation of the electric load of the air coolers, a specific consumption of 0.017 kW for each thermal kW removed was assumed [32].

For the economic analysis, the cost functions reported in [33,34] were adopted to estimate the cost of air coolers, radial turbines, storage tank and liquid-gas separator. For these devices, the purchased equipment cost was estimated as:

$$log_{10} C_{p0} = K_1 + K_2 \, log_{10} A + K_3 (log_{10} A)^2 \quad (3)$$

where A is the size parameter of the air cooler and turbines (exchange area for the air coolers and mechanical power for turbine, volume for the storage and separator) and the constants $K_i$ are a function of the equipment typology.

The heat transfer coefficient for the air coolers was evaluated by interpolating the values reported in [35].

The pressure factor was estimated according to the procedure reported in [33] to consider the dependence of operating pressure on the cost of the devicesb

$$log_{10} F_p = C_1 + C_2 \, log_{10}(P_g \cdot 10) + C_3 \left(log_{10} P_g\right)^2 \left[log_{10}(P_g \cdot 10)\right]^2 \quad (4)$$

where $F_p$ is the pressure factor, $P_g$ is the operating gauge pressure and $C_i$ are constants which depend on the type of the device. In the case of the storage tank and of the separator, the pressure factor was estimated according to the following equation [33]:

$$F_p = \frac{\frac{(P_g \cdot 10+1)\cdot D}{2\left(850-0.6\cdot(P_g \cdot 10+1)\right)}+0.00315}{0.0063} \quad (5)$$

The final direct and indirect capital costs of the system, were therefore estimated by considering operating pressure, material of construction, piping, control, and labour cost as:

$$C_{TBM} = C_{p0} \cdot \left(B_1 + B_2 \cdot F_m \cdot F_p\right) \quad (6)$$

where $F_m$ is the material factor, which takes into account the material of construction of the device, and $B_i$ are constants depending on the device. For the turbines, due to the low temperature, stainless steel was considered as a construction material.

The current direct capital cost of the turbines and of the air cooler were obtained from the ratio between the CEPCI indexes of 2001 (394.3) and of 2015 (556.8) and was equal to 1.4.

Regarding the cold-box, a multi-stream plate-fin heat exchanger was considered. The cost of this type of heat exchanger depends on the volumetric heat transfer coefficient as reported in [36]. The calculation method

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is based on the analysis of the hot and cold composite heat transfer curves, representing the global heat transfer in the heat exchanger. The hot and cold composite curve are directly evaluated by Aspen Hysys, through the pinch analysis. The procedure for the cost determination is reported in the following steps:

- Partition of the curves in different regions (determined by straight lines), in order to evaluate each section independently;
- Estimation of the volumetric coefficient [36], according to the following equation:

$$\frac{\dot{Q}_z}{B_z} = \sum_{i=1}^{n} \frac{\dot{Q}_i}{B_i} \tag{7}$$

where $\dot{Q}_z$ is the thermal power transferred in the z zone, $B_z$ is the average volumetric heat exchange coefficient in the z zone, $\dot{Q}_i$ is the heat transferred by the i$^{th}$ stream in the zone z, $B_i$ is the heat exchange coefficient of the i$^{th}$ stream in the zone z;

- Estimation of the logarithmic mean temperature difference in the zone z ($\Delta T_{m,z}$), obtained through the hot and cold composite curve;
- Estimation of the volume of each zone [36]

$$\mathrm{V}_z = \frac{\frac{Q_z}{\Delta T_{m,z}}}{B_z} \tag{8}$$

- Estimation of the total volume of the heat exchanger [36]

$$V = 1.15 \cdot \sum_{i=1}^{n} V_z \tag{9}$$

Once the volume is known the specific cost of the heat exchanger is evaluated from function reported in [37].

The current direct capital cost was obtained by multiplying the direct capital cost for the ratio between the CEPCI indexes of 2015 and of 2007.

For the cost of the screw compressors, an analytic expression was found, by interpolating the costs from a catalogue [38], and finding the following linear relation, as from fig. 2:

$$C_{p0} = 138\,682 \cdot (\dot{V}) + 3\,253$$

(10)

where $\dot{V}$ is the volume flow rate of the compressor.

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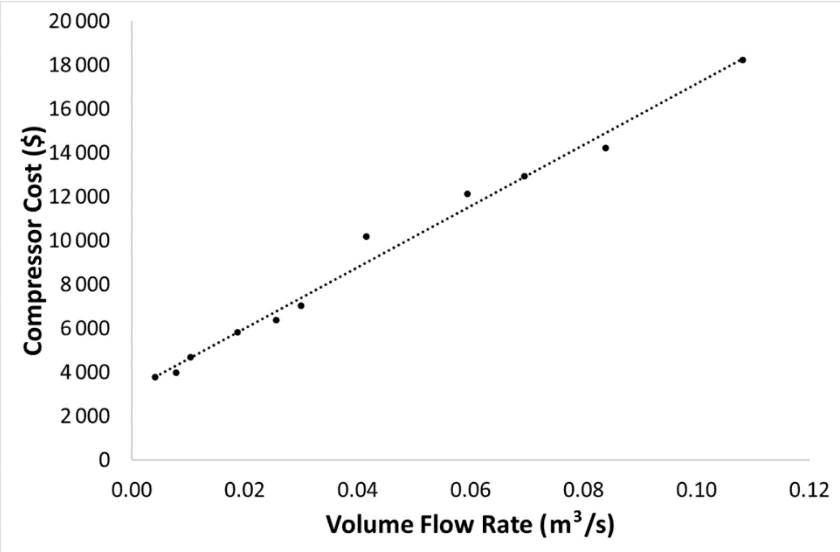


***Figure 3**: Cost of screw compressors as a function of the volume flow rate*

In the case of the screw expander the cost was estimated with the same equation used for the compressors, but multiplying the volume flow rate by the volumetric expansion ratio, as in [39]:

$$C_{p0} = 138\,682 \cdot (\dot{V} \cdot r) + 3\,253 \tag{11}$$

where $r$ is the volumetric expansion ratio of the expanders and $\dot{V}$ is the volume flow rate in $m^3/s$.

The costs of cold box, screw expanders and compressors, were multiplied by 1.45 and by 1.1 as from [34] to consider material, installation costs and indirect costs. The total direct and indirect cost were multiplied by the factor 1.18 in order to obtain the total capital expenditure (CapEx) of the system [33] by considering contingencies and fees. All the cost function has been implemented in the numerical model, in order to estimate the CapEx as a function of various variables. The O&M costs were assumed to be the 2.5% of the global direct and indirect costs.

A minimum pinch point of 5°C was considered as a constraint in the optimization process for the cold box. This is a value which can be easily achieved in multi-stream plate and fin heat exchanger. The optimization variables were:

- compression ratio of the methane compressors and of the three stages of nitrogen compressors (the expansion ratio of the expander resulted from the compression ratio of the compressors and from the heat exchangers pressure drops);
- nitrogen mass flow rate;

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- ratio between the mass flow rate of nitrogen flowing in the two expanders;
- temperature of the bio-methane at the outlet of the cold box (from this variable depends the vapor quality of liquefied biomethane after the lamination valve and therefore the amount of natural boil-off from the separator).

The system was optimized from the thermodynamic point of view and all the economic assessments were based on the thermodynamically optimized system. The minimum specific consumption was defined as the objective function:

$$F = \min\left(\frac{\dot{W}_{el}}{\dot{V}_{bio}}\right) \tag{12}$$

where $\dot{W}_{el}$ is the total power requested by the liquefaction (sum of all the compressors minus sum of all the expanders) and $\dot{V}_{bio}$ is the normal volume flow rate of biomethane liquefied.

In the simulation, the RefProp Equation of State (EoS) was adopted for the simulation of both nitrogen and bio-methane, which was assumed as pure methane. This simplified assumption can be considered as reasonable since upgrading systems are able to provide high purity methane (>97%) [40].

*3.2. Second system: production of biomethane and injection in the natural gas grid*

In the second system (fig. 4), the cost of biomethane grid injection was estimated as a function of the connection and compression costs. Firstly, the power requested by the compressor to increase the pressure of the biomethane from the PSA outlet to the grid injection was calculated as:

$$\dot{W}_c = \frac{\dot{m} \cdot \Delta h}{\eta_{is,c}} \tag{13}$$

where $\dot{m}$ is the biomethane mass flow rate, $\Delta h$ is the enthalpy difference between compressor outlet and inlet and $\eta_{is}$ is the isentropic efficiency of the compressor, assumed as 75%. A single-stage volumetric compressor was considered (fig. 4), due to the small amount of produced biomethane. An air-cooled heat exchanger reducing the gas temperature to 45°C was considered.

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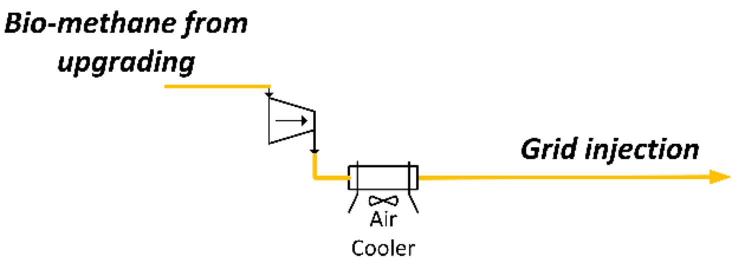


***Figure 4***: *Scheme of the second system.*

Regarding the cost of this system, compressor and air cooler cost as well as cost of pipeline for grid connection was considered. This last is a crucial investment of the plant and various cases are possible: the pipeline cost is a function of the distance between the plant and the gas network, the amount of biomethane produced and the complexity of the civil work requested (i.e. burial, crossing of rivers, motorways, railroads etc.).

Due to the large number of variables involved, which are responsible of an extremely high variability of the pipeline cost (from 55 up to 1000 $/m [41] ), three different costs of connection to the gas network were considered:

- 50 k$: order of magnitude of the cost corresponding to an easily accessible and close gas network (a few hundred meters), without the need of any burial or other civil works;
- 1 M$: order of magnitude for the cost of the connection to a gas network far from the biogas plant between one and two kilometres with the need of burial and rehabilitation of the land;
- 2 M$: order of magnitude of the cost of the connection to a far (more than two kilometres) or hardly accessible gas network, providing the need of burial and rehabilitation of the land and crossing of river or civil infrastructures.

The cost of compression system including air cooler was calculated by using the cost functions reported in paragraph 3.1 and was about 39 k$. The pressure drop in the pipe was considered but was negligible.

*3.3. Economic analysis*

Once the system cost was estimated, the net present value (NPV) was calculated for each system as:

$$NPV = -C_{TBM} + \sum_{i=1}^{n} \frac{(-C_{OpEx,i} - C_p + R_i)}{(1+j)^i} \quad (14)$$

Where $C_{OpEx,i}$ are operative costs (O&M and electricity), $C_p$ is biomethane production and upgrading cost

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and $R_i$ are the revenues. $C_{TBM}$ is the total cost of the installation (CapEx). $j$ is the discount rate and was considered equal to 0.07. The cost of electricity and maintenance was considered for the evaluation of OpEx. In the case of liquefied biomethane production an overall transportation cost of about 100 $/day was considered by assuming a ten kilometres route to the closest gas station and a fleet composed of only one truck [42], which correspond to an annual cost of 25 000 $, and by considering 6 000 equivalent hours of operation of the plant.

By assuming to deal with anaerobic digesters processing sludge and organic waste, the biomethane production and upgrading cost was assumed equal to 0.3 $/m$^3$, according to [41]. The profitability index, representing the ratio between the cash flows and the plant costs was estimated for both the two systems as:

$$PI = \frac{C_{TBM}+NPV}{C_{TBM}} \tag{15}$$

The higher is the PI, the lower is the risk of the investment, because for the same investment return, the investment costs are lower.

Since the values of NPV and PI for the two systems is a function of the selling prices of liquefied biomethane and grid injected one, the NPV difference was considered to estimate the economic convenience of each system:

$$\Delta NPV = NPV_{system1} - NPV_{system2} \tag{16}$$

This means that if $\Delta NPV$ is greater than zero, liquefaction is the most convenient technology.

For the same reason a ratio between the profitability indexes was defined:

$$PI_r = \frac{PI_{system1}}{PI_{system2}} \tag{17}$$

Meaning that if $PI_r$ was higher than 1, liquefaction would result the technology which minimize the investment risk.

A list reporting the assumption of the models for the two studied systems is reported in the Appendix in table A1.

## 4. RESULTS

### *4.1 Thermodynamic analysis*

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As stated above, the systems were optimized from the thermodynamic point of view by minimizing the specific electric consumption referred to the standard cubic meter of raw biogas. The results of the optimization are reported in fig. 3 for both radial turbo-expanders and screw expanders in the Joule-Brayton reverse cycle, together with the specific consumption of the second system (grid injection).

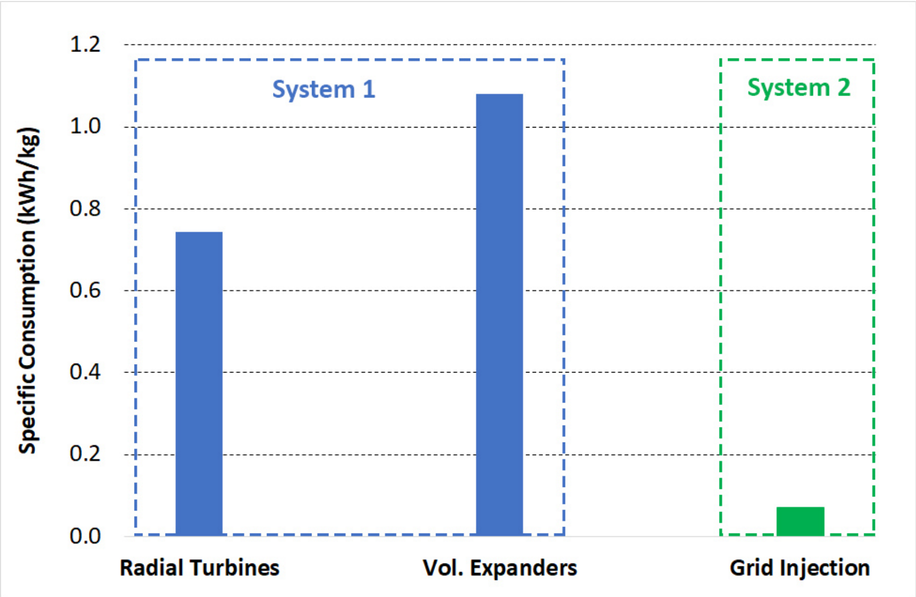


***Figure 5****: Specific consumption of the two systems: in the system 1 the specific consumption achieved with radial turbines and screw expanders is reported*

By analyzing the results reported in fig. 5, it can be observed that grid injection had a specific consumption of 0.07 kWh/kg, about 10.7 times lower than that obtained with the most efficient liquefaction system.

In the case of a liquefaction system, the specific consumption obtained with the Joule-Brayton cycle equipped with screw expander is 1.45 times higher than that obtained with radial turbines. This means that the efficiency of the expander has a key role in the liquefaction process. Under the same expander inlet pressure, the reduction of isentropic efficiency, caused a higher temperature at the expander outlet which negatively affected the capacity of reaching the liquefaction temperature of the methane. To overcome the problem, in the case of a low efficient expander, the system should be designed at higher values of pressure ratio and with a high nitrogen mass flow rate (tab. 1) with a significant increase in the specific consumption. The comparison between the shape of the two liquefaction cycles, with radial turbines and screw expanders, is reported in fig. 6.

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From this comparison, it is clear that the cycle with screw expanders had a larger area, since it required a maximum pressure of 7.25 MPa (1.25 MPa more than the configuration with radial turbines) and a minimum pressure of 0.25 MPa. The global compression ratio of the cycle with screw expanders at optimal thermodynamic conditions resulted to be 29 while the one with radial expander resulted to be 20. Due to the higher compression ratio, the highest temperature of the cycle resulted to be 248°C, 32 K higher than the one reached with the turbines.

Data of mass flow rate, pressure and temperature at the inlet and at the outlet of the various devices for both the two configurations at optimal thermodynamic conditions are reported in Appendix in tables A2 and A3.

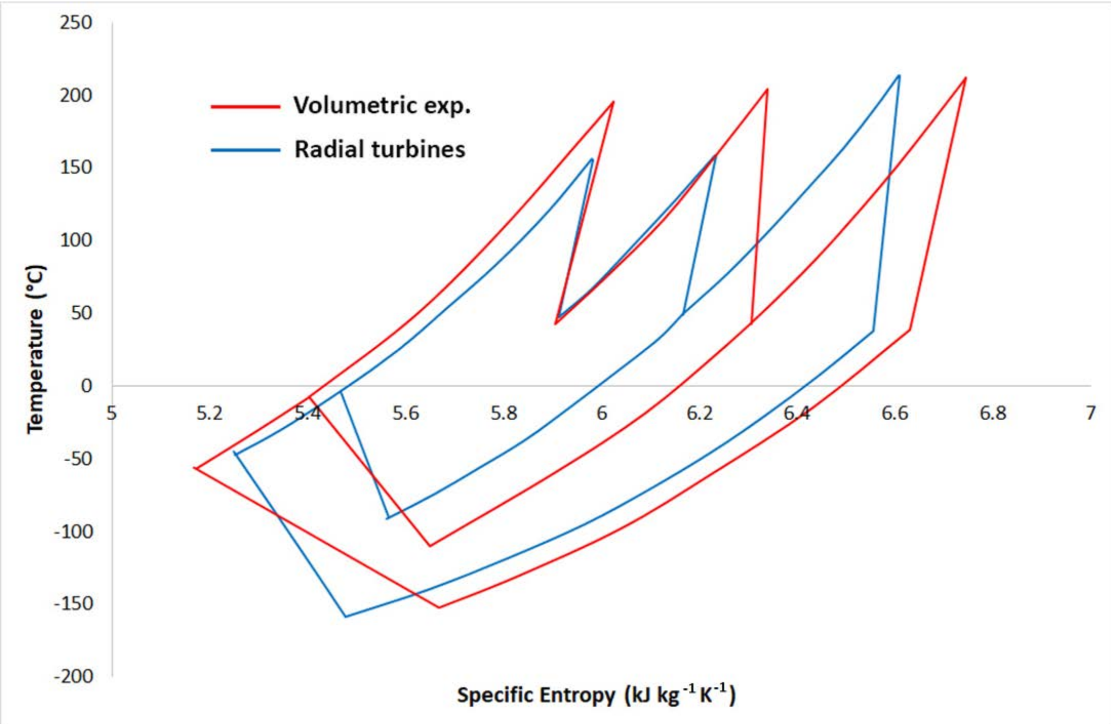


***Figure 6**: Comparison between the Joule-Brayton cycle with radial turbines (up) and Joule-Brayton with screw expanders.*

Total power requested for the liquefaction of 10 tonnes/d of methane was 310 kW with radial turbines, and 450 kW with screw expanders. It is possible to state that a decrease of about 24% in the isentropic efficiency of expanders lead to an increase of almost 45% in plant electric energy consumption.

From the energy point of view, biomethane injection resulted the less consuming strategy since, total power requested to compress the methane to 1.2 MPa (natural gas grid pressure) was only 29 kW.

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*4.2Economic analysis*

The economic evaluation was carried out by considering cost functions reported in the previous paragraph with system operating at optimal thermodynamic conditions. The estimated cost of the two systems, by considering the two types of expanders for liquefaction and the three costs for natural gas injection are reported in table 1.

*Table 1: Cost and specific cost of the system for the two systems.*

| | System 1 | | System 2 | | |
|---|---|---|---|---|---|
| | Radial turbines | Screw expanders | | Radial turbines | Screw expanders |
| Cost ($) | 940 500 | 692 500 | Cost ($) | 940 500 | 692 500 |
| Specific annual cost ($ $tonne^{-1}$ $year^{-1}$) | 258 | 190 | Specific annual cost ($ $tonne^{-1}$ $year^{-1}$) | 258 | 190 |

For the liquefaction system, the cost of the plant with radial turbines resulted to be 1.35 times higher than the cost with screw expanders. Radial turbines cost is extremely high, especially due to the material of construction (stainless steel). As an example, the cost of the low-pressure radial turbine approached 200 000 $. The order of magnitude of the cost for the turbines was confirmed by data provided by a commercial partner.

System 2 connection cost are strictly related to the distance from the natural gas grid, as mentioned before, from 50k$ up to 2M$.

A sensitivity analysis on the electricity price was carried out by fixing liquefied biomethane and biomethane prices to the actual market levels in Europe [43,44] (0.51 $/kg of LNG and 0.29 $/kg of natural gas) and by considering an average incentive of 0.50 $/kg [45]. Results in terms of PI and NPV are reported in fig. 7 and 8, by assuming 6 000 equivalent hours per year of operation.

Figure 7 shows that grid injection with a connection cost below 500 k$ has a higher PI than system 1 and therefore below this value, system 2 minimized the investment risk, for every electricity price. By increasing the natural gas grid connection cost (1-2 M$) liquefaction with screw expanders maximized the profitability index and minimized the investment risk up to an electricity price of 0.16 $/kWh, due to the low cost of the screw expanders. By increasing the price of electricity over this value, liquefaction with screw expanders resulted the technology which minimized the investment risk.

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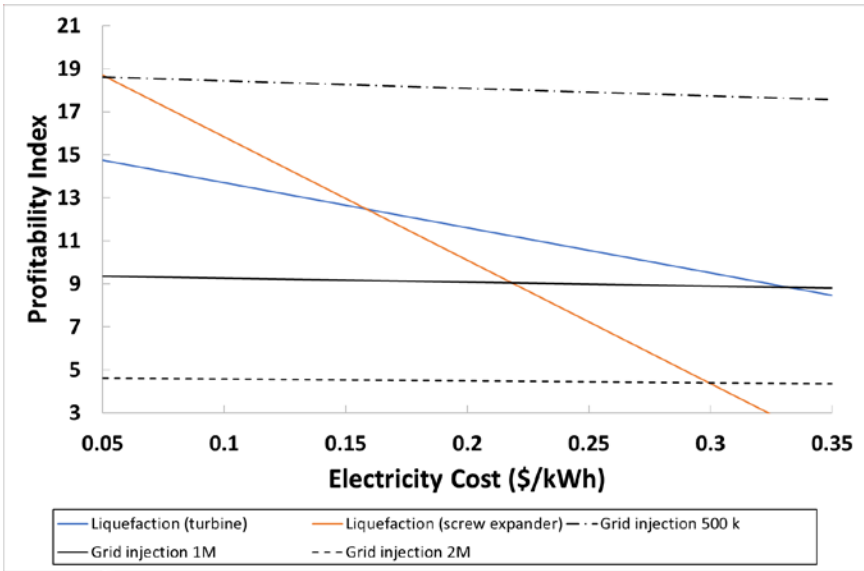


***Figure 7****: Profitability index (PI) after 20 years for system 1 (radial turbines, screw expanders) and system 2 (different connection costs for grid injection), 6 000 equivalent hours.*

Regarding NPV after 20 years (figure 8), liquefaction plants showed better results for the lowest electricity cost. This means that up to an electricity cost of 0.23 $/kWh biomethane liquefaction with a reverse Brayton Cycle and a radial expander maximized the investment return for every cost of connection in the system 2. The use of a screw expander in the liquefaction cycle was not competitive with radial turbines in terms of investment profits. Therefore, despite the lower cost of the liquefaction plant which minimized the investment risk (fig. 7) up to an electricity price of 0.16 $/kWh, the use of a screw expander decreased the system profit for every price of electric energy. Biomethane grid injection resulted profitable only if electricity price was higher than 0.23 $/kg.

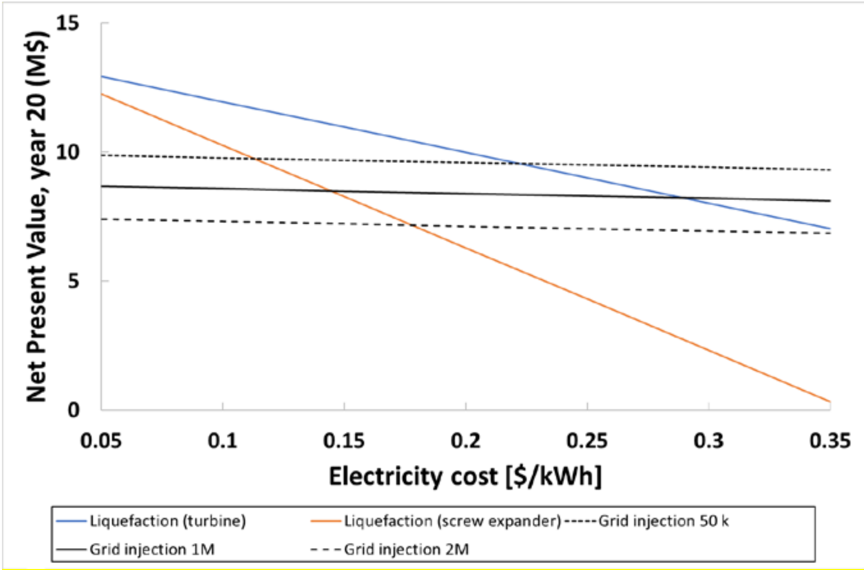


***Figure 8****: Net present value after 20 years for system 1 (radial turbines, screw expanders) and system 2 (different connection costs for grid injection), 6 000 equivalent hours.*

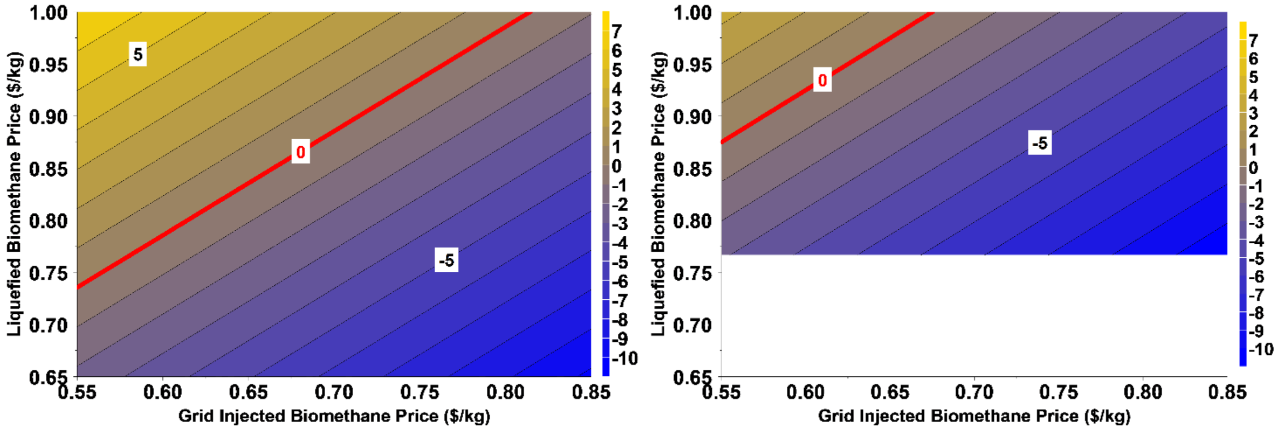


***Figure 9****: ΔNPV after 20 years (in M$) between system 1 (radial turbines left, screw expanders right) and system 2 (grid injection connection costs of 50k$), 6 000 equivalent hours/year and electricity price of 0.20 $/kWh.*

Since in many European countries different type of governmental subsides are in place both for biomethane liquefaction and for biomethane injection, a sensitivity analysis on the final selling price was carried out. Results in terms of the difference between the NPV ($\Delta NPV$) of system 1 and system 2 (by considering radial turbines and screw expanders in the case of liquefaction) and for a connection cost of 50 k$ (for system 2) are reported in fig. 9, for an electricity cost of 0.20 $/kWh. The red line represents the limit of convenience between the two systems: positive values mean that the liquefaction system is more profitable than gas grid injection and vice versa for the negative region.

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It is worth to notice that in these conditions, liquefaction with a Joule-Brayton reverse cycle equipped with screw expanders (fig. 9 right) resulted economically unfeasible when the selling price of liquefied biomethane was lower than 0.77 $/kg (blank area on the diagram).

By comparing fig 9. left and right, the area of convenience for liquefaction is larger in the case of radial turbines than in the case of screw expanders. This means that in these conditions, radial turbines should be adopted for those combinations of prices that made liquefaction (system 1) profitable.

Regarding PI, in these conditions (connection cost of 50 k$), biomethane grid injection (system 2) was the technology which maximized this indicator, for every price combination considered. This means that despite in some cases liquefaction may provide higher values of NPV, gas grid injection is the technology which always maximized the investment multiplier and minimized the risk.

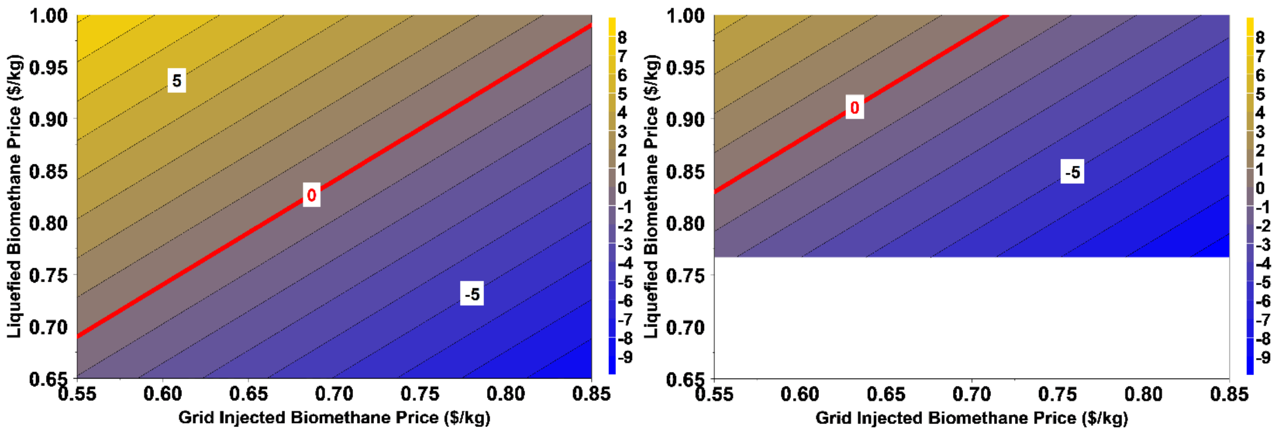


***Figure 10**: ΔNPV after 20 years (in M$) between system 1 (radial turbines left, screw expanders right) and system 2 (grid injection connection costs of 1 M$), 6 000 equivalent hours/year and electricity price of 0.20 $/kWh.*

By increasing grid connection costs for system 2 up to 1 M$, the limit of convenience of gas grid injection decreased and the difference of NPV between system 1 and 2 increased (fig.10). Even in this case, turbines are the best choice for the liquefaction system from the profit point of view. With this connection cost, the capital investment of the two systems were of the same order of magnitude, but the high cost of electricity in the case of liquefaction, made biomethane grid injection the technology with the highest probability for various combinations of prices.

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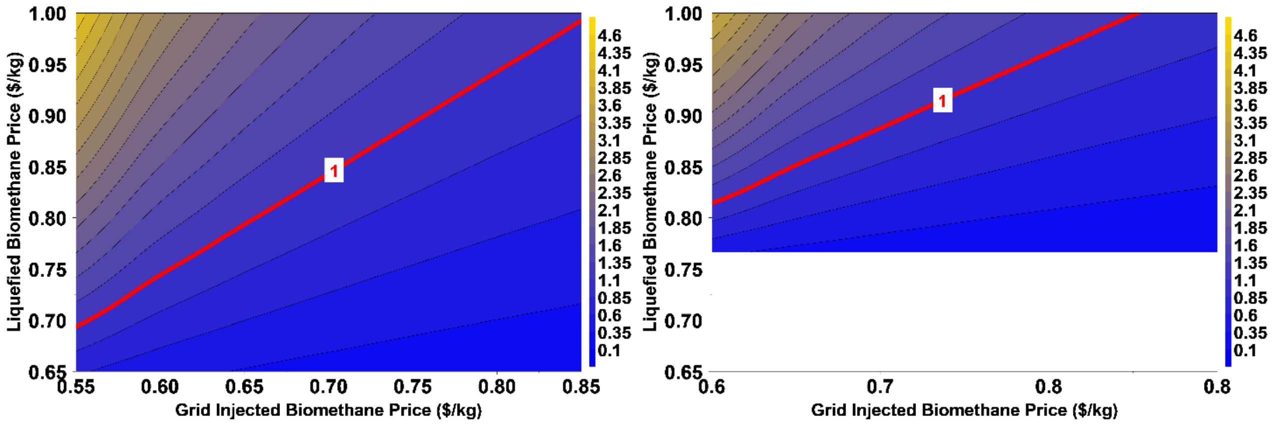


***Figure 11****: Profitability index ratio ($PI_r$) after 20 years between system 1 (radial turbines left, screw expanders right) and system 2 (grid injection connection costs of 1M$), 6 000 equivalent hours/year and electricity price of 0.20 $/kWh.*

Capital costs of system 1 and 2 are similar and a limit of convenience for the PI may be found in the case of liquefaction for both the cycle equipped with radial turbines and that equipped with screw expanders. In fig. 11 the ratio between the PI of system 1 and that of system 2 is reported: the red line corresponding to the ratio of 1 indicates the limit of convenience with respect to the PI between the two systems. Values higher than 1 indicates the convenience of liquefaction. In this case, with a connection cost of 1 M$ for system 2, biomethane grid injection resulted the system which provided a larger area of profitability. Therefore for several price combinations is the technology that minimized the investment risk. In those regions where liquefaction should be preferred, the use of a high-efficient liquefaction cycle with radial turbines should be always preferred from the point of view of both profit maximization (fig. 10) and investment risk (fig. 11).

Finally, in the case of high cost of connection for system 2 (connection cost of 2M$), the limit of convenience decreased (difference between NPV, fig 12) and the red line translated towards the lower end of the map, thus increasing the suitability area of liquefaction. This is valid only in the case of liquefaction by a Joule-Brayton reverse cycle equipped with a radial turbine (fig. 12 left). The low efficiency of the cycle with screw expanders led to high electricity purchasing costs which made the system unprofitable (fig. 12 right) in comparison to that equipped with turbines. As in the previous case (grid injection cost of 1M$), radial turbines should be always preferred to screw expanders since they maximized the investment return. With a connection cost of 2 M$, liquefaction resulted the system which is more profitable and less risky for most of the combinations of selling prices, even in those cases where the price difference between liquefied and gaseous biomethane is very low (fig. 13). In a similar way to previous cases, even in this case of high connection costs, liquefaction with screw expanders increased the investment risk (fig. 13 right).

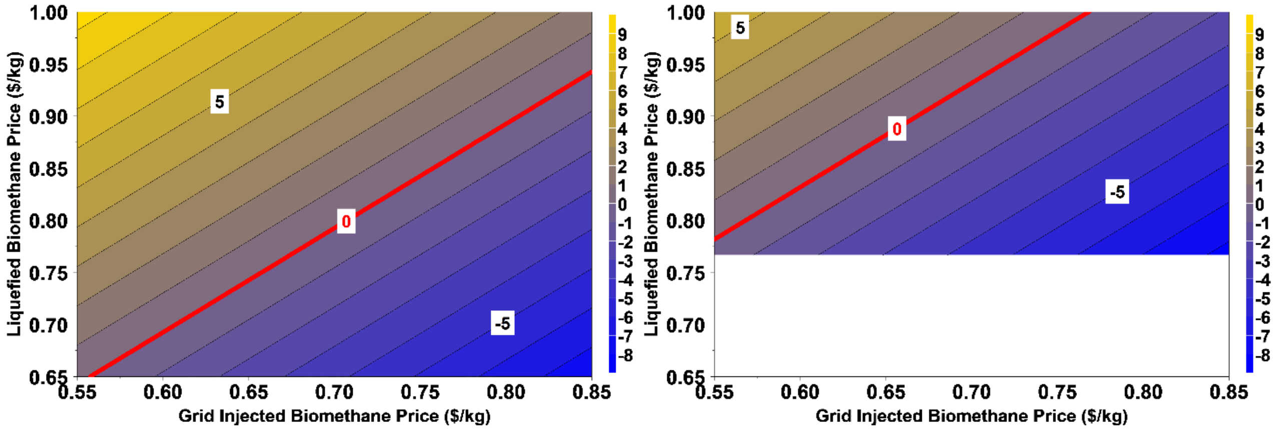


***Figure 12***: *Difference of Net Present Value at 20th year (in M$) between system 1 (radial turbines left, screw expanders right) and system 2 (grid injection connection costs of 2M$), 6 000 equivalent hours/year and electricity price of 0.20 $/kWh.*

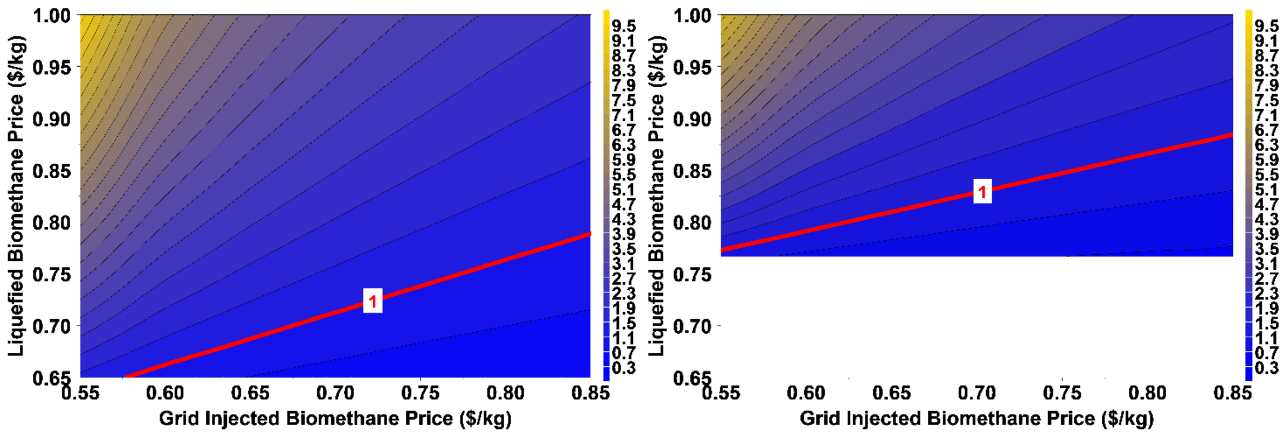


***Figure 13***: *Profitability index ratio ($PI_r$) after 20 years between system 1 (radial turbines left, screw expanders right) and system 2 (grid injection connection costs of 2M$), 6 000 equivalent hours/year and electricity price of 0.20 $/kWh.*

## 5. CONCLUSIONS

The analysis shows that at short distances from the biogas source to a high-pressure distribution network, pressurizing the upgraded methane is the lowest cost option. While injection cost scales, at pipeline costs greater than 2 M$, and typically distances greater than 2 km, better economics are obtained by means of LNG production and trucking that to the point of use. The study also concluded that expander economic efficiency plays a big role in the economics of liquefaction, with radial turbines generally optimal. At low electricity prices, screw expanders with their higher specific energy costs but lower capital investment costs can be

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favored.

**Nomenclature**

| | |
|---|---|
| $\dot{W}$ | Mechanical power (kW) |
| $\dot{m}$ | Mass flow rate (kg $s^{-1}$) |
| $\Delta h_{is}$ | Isentropic enthalpy drop (kJ $kg^{-1}$) |
| $\eta_{is}$ | Isentropic efficiency |
| $\dot{Q}$ | Thermal power (kW) |
| $V$ | Volume ($m^3$) |
| $D$ | Diameter (m) |
| $\Delta T$ | Temperature difference (K) |
| $\dot{V}$ | Volume flow rate ($m^3$ $s^{-1}$) |
| $r$ | Expansion volume ratio |
| $C_{p0}$ | Purchased equipment cost ($) |
| $F_p$ | Pressure factor |
| $P_g$ | Gauge pressure (MPa) |
| $C_{TBM}$ | Total bare module cost ($) |
| $C_{OpEx,i}$ | Operative expenditures ($) |
| $R$ | Revenues ($) |
| $j$ | Discount rate |
| $NPV$ | Net present value ($) |
| $PI$ | Profitability index |

**Subscripts**

| | |
|---|---|
| $e$ | Expander |
| $c$ | Compressor |
| $z$ | Zone |
| $bio$ | Biomethane |

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## APPENDIX

In this section, the main assumption of the models created for the analysis of the two systems are reported (table A1). Thermodynamic states of the Joule Brayton liquefaction cycle are reported for both radial turbines (tab A2) and screw expanders (tab. A3).

*Table A1: Main Assumption of the models for the two analyzed systems*

| **System 1 (Liquefaction) Parameter** | | **System 2 (Biomethane grid injection) Parameter** | |
|---|---|---|---|
| Compressors efficiency | 0.75 | Compressor efficiency | 0.75 |
| Radial turbines efficiency | 0.85 | Air cooler pressure drop [kPa] | 10 |
| Screw expander efficiency | 0.65 | Pipeline pressure drop | Beggs and Brill correlation |
| Heat exchangers pinch point [K] | 5 | | |
| Heat exchangers and air cooler pressure drop [kPa] | 10 | | |
| Final user | Heavy-duty vehicles | Final user | Natural gas users |
| Liquefied biomethane transportation cost | 100 $/day | | |

*Table A2: Thermodynamic points of the Joule-Brayton cycle in the case of radial turbines.*

| **Device** | **Mass flow (kg/h)** | **Pressure (MPa)** | **Temperture (°C)** |
|---|---|---|---|
| L.P. compressor inlet | 1484 | 0.34 | 39.6 |
| L.P. compressor outlet | 1484 | 1.16 | 216.7 |
| M.P. compressor inlet | 3844 | 1.16 | 41.9 |
| M.P. compressor outlet | 3844 | 2.69 | 155.6 |
| H.P. compressor inlet | 3844 | 2.69 | 45 |
| H.P. compressor outlet | 3844 | 6.02 | 155.3 |

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| | | | |
|---|---|---|---|
| M.P. expander inlet | 2360 | 6.01 | -4.4 |
| M.P, expander outlet | 2360 | 1.17 | -94.4 |
| L.P. expander inlet | 1484 | 6.01 | -52 |
| L.P. expander outlet | 1484 | 0.34 | -164.4 |

*Table A3: Thermodynamic points of the Joule-Brayton cycle in the case of screw expander.*

| **Device** | **Mass flow (kg/h)** | **Pressure (MPa)** | **Temperature (°C)** |
|---|---|---|---|
| L.P. compressor inlet | 1898 | 0.25 | 40.0 |
| L.P. compressor outlet | 1898 | 1.03 | 248.4 |
| M.P. compressor inlet | 4789 | 1.03 | 41.3 |
| M.P. compressor outlet | 4789 | 2.68 | 173.0 |
| H.P. compressor inlet | 4789 | 2.68 | 45.1 |
| H.P. compressor outlet | 4789 | 6.29 | 162.5 |
| M.P. expander inlet | 2893 | 6.29 | -3.6 |
| M.P, expander outlet | 2893 | 1.04 | -81.2 |
| L.P. expander inlet | 1896 | 6.29 | -58.1 |
| L.P. expander outlet | 1896 | 0.25 | -153.1 |