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## Research Paper

# CO<sub>2</sub> Refrigeration at Very Low Temperatures Using Natural Gas Expansion

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## ABSTRACT

The pressure reduction of natural gas in distribution networks generally leads to a significant loss of available energy due to the use of throttling valves. Recovering this pressure energy represents an important opportunity to improve the global efficiency of gas infrastructures. In this study, a refrigeration system operating with carbon dioxide (CO<sub>2</sub>) and driven by the expansion of natural gas is investigated. The proposed system replaces the conventional expansion valve by a turbo-expander able to convert the pressure energy of natural gas into mechanical energy. This energy is then used to drive the compressor of a vapor compression refrigeration cycle. A thermodynamic model of the system is developed based on mass and energy balances applied to each component. The performance of the system is evaluated for different operating conditions corresponding to pressure reduction stations used in Algeria. The results show that the integration of a turbo-expander allows simultaneous production of refrigeration at very low temperatures while recovering part of the pressure energy of natural gas. This approach contributes to improving the overall energy efficiency of gas distribution networks while reducing energy losses.

## 1 Introduction

Global energy demand is experiencing sustained growth and is projected to increase by more than 50% over the next three decades, primarily driven by the rapid expansion of developing countries [1]. Concurrently, worldwide natural gas demand is expected to rise significantly, potentially reaching 30% by 2040 [1]. In this context, producing countries such as Algeria are witnessing a substantial increase in domestic consumption, facilitated by the expansion of distribution networks through large-scale pipeline infrastructure. However, prior to delivery to end consumers, natural gas must be reduced to an appropriate operating pressure—a process traditionally performed using thermostatic expansion valves. Although simple and cost-effective, this method represents a major source of exergy destruction, wasting a considerable amount of potentially recoverable energy.

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To minimize these losses, numerous studies have been conducted in recent years to integrate innovative technologies such as turbo-expanders [2]. Installed in pressure reduction stations (PRS) or city gate stations (CGS), these devices convert the exergy of high-pressure natural gas into mechanical work, which can be valorized in multiple ways: electricity generation, driving compressors in vapor-compression refrigeration machines, or coupling with energy storage systems. Scientific literature has confirmed for over a decade the technological maturity of these recovery systems. Early studies demonstrated that replacing valves with turbo-expanders enables electricity generation ranging from several hundred kilowatts to multiple megawatts depending on flow rate and pressure ratio, with typical annual outputs between 1 and 6 GWh per station [3–5]. Economically, Rahmatpour and Shaibani showed that an Iranian station could produce up to 4.17 MW with an internal rate of return close to 29% and a payback period under 4 years [3], while European and Turkish analyses highlighted highly competitive specific costs for recovered electricity (0.01–0.03 \$/kWh) [6, 7].

A major challenge, however, lies in the increased temperature drop at the turbo-expander outlet, which necessitates additional thermal energy to prevent hydrate formation. This constraint has spurred innovation in preheating strategies: integrating renewable sources (solar or geothermal energy) reduces gas consumption by 10–15% [8, 9], while advanced architectures coupling Organic Rankine Cycles (ORC) or CO<sub>2</sub> cycles with energy recovery achieve energy efficiencies exceeding 40% and exergy recovery rates above 85% [10, 11]. Furthermore, flow rate variability—particularly pronounced in urban networks—has driven the development of adaptive-geometry expanders (variable inlet guide vanes, twin-screw expanders) that maintain high efficiencies even under off-design conditions [12, 13].

The refrigeration sector represents a particularly relevant application domain for this energy valorization. Facing growing demand for refrigeration—encompassing freezing, blast freezing, food storage, and air conditioning—the industry is actively seeking energy-efficient solutions. Among existing technologies (compression [14–18], absorption [19], ejector systems [20]), the vapor-compression cycle remains the most widespread due to its technological maturity and relatively high coefficient of performance. Nevertheless, according to the International Institute of Refrigeration (IIR), refrigeration, air conditioning, and heat pump systems already accounted for 20% of global electricity consumption in 2019 [21], making their energy optimization an absolute priority [22] for both economic reasons and to limit environmental impacts associated with refrigerants. In this perspective, integrated solutions are emerging that combine pressure energy recovery with eco-friendly refrigerants. Autonomous refrigeration systems using carbon dioxide (CO<sub>2</sub>) as a working fluid (GWP = 1, ODP = 0) have been proposed, directly activated by converting natural gas pressure energy into rotational mechanical energy transmitted to the compressor. Particularly suited for applications requiring very low evaporation temperatures, these concepts have undergone comprehensive modeling, with evaluation of their overall coefficient of performance and exergy efficiency through parametric studies under various operating conditions.

More recently, attention has shifted toward integrating storage systems to address the imbalance between electricity production and demand. Mechanical storage solutions—hydraulic pumping into water towers or turbo-pump systems directly driven by the expander—enable storage capacities ranging from 1 to 15 GWh/year depending on station capacity, with round-trip efficiencies of 70–75% and payback periods under 3 years [23–25]. Other studies have integrated these systems into broader hybrid configurations, such as photovoltaic plants coupled with compressed air energy storage, significantly improving production stability and overall profitability [26]. An emerging trend also explores using recovered electricity for high-value applications, particularly hydrogen production via electrolysis, with potentials exceeding 1.5 million Nm<sup>3</sup>/year for large-capacity stations [27, 28].

Overall, the literature converges on the conclusion that natural gas pressure energy recovery has matured into an economically viable technology fully aligned with decarbonization objectives. Recent research is steering the field toward multifunctional integrated systems combining electricity generation, storage, thermal valorization, and emission reduction, progressively transforming pressure reduction stations into genuine local energy hubs at the heart of the energy transition [2, 5, 7, 11]. This energy recovery approach fully supports energy transition and circular economy goals, contributing to CO<sub>2</sub> emission reductions while enhancing the efficiency of gas infrastructure.

## 2 Problem description and proposed solution

Pipeline transportation represents the preferred method for conveying natural gas over long distances, ensuring continuity between production fields and consumption centers. This tubular network, typically buried underground, transports gas at high pressure (up to 100 bar) using regularly spaced compressor stations. Constructed from high-strength steel, pipelines offer massive transport capacity, high operational reliability, and a lower carbon footprint compared to alternatives such as

LNG carriers or trucks. They form the backbone of continental energy infrastructure, with several million kilometers deployed worldwide. Despite high initial investment costs, their energy efficiency (losses  $< 0.1\%$ /100 km) and safety—enhanced by real-time monitoring systems—make them an indispensable component of modern gas supply chains.

Pressure reduction stations (Figure 1a), also known as delivery stations or "city gates," constitute essential links in natural gas distribution networks. Their primary function is to reduce the pressure of gas transported through transmission pipelines (typically 40 to 70 bar) to levels suitable for local distribution (4 to 16 bar) and subsequently for low-pressure networks serving end consumers ( $< 0.5$  bar). This isenthalpic expansion, performed through control valves or pressure regulators, causes significant gas cooling (Joule-Thomson effect), potentially leading to ice or hydrate formation; thermal exchangers or heaters are therefore integrated to maintain a minimum safety temperature (generally  $> 0^\circ\text{C}$ ). These installations also incorporate metering equipment (flowmeters), filtration systems, odorization units (tetrahydrothiophene injection), and precise pressure regulation devices. To address the energy losses inherent in conventional expansion, this study proposes replacing the thermostatic expansion valve with a modern station integrating turbo-expanders (Figure 1b) capable of recovering pressure energy in mechanical form, thereby enhancing the overall energy efficiency of the gas system. These stations thus ensure not only network safety and stability but also a controlled transition between transmission infrastructure and end-use gas applications.

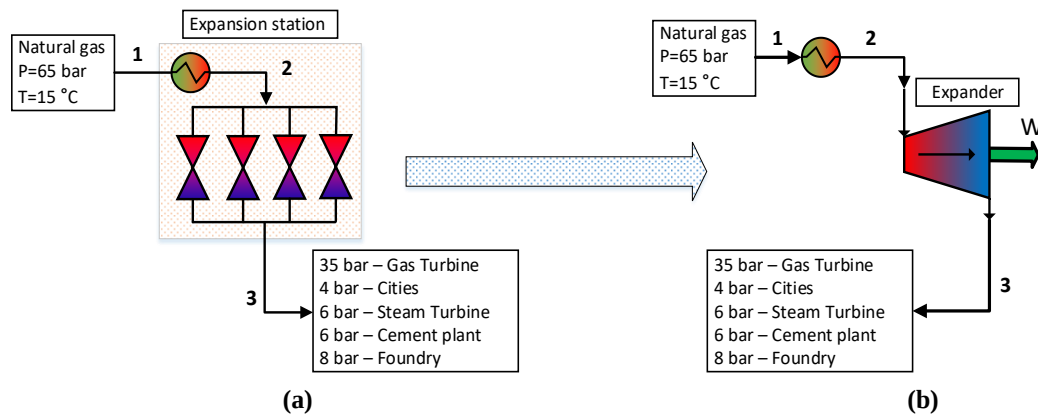


**Fig. 1 – Natural gas expansion system**

The diagram shown in Figure 2 is a basic layout of a combination of an expander with a compressor of a refrigeration machine. The expander allows the recovery and conversion of the pressure energy contained in natural gas into mechanical energy (2g-3g), which is then transmitted and used to activate the compressor of the refrigeration machine. In the compressor (1-2), the refrigerant is delivered at the condensing pressure. At the condenser level (2-3), the quantity of heat released to ensure desuperheating and then condensation of the refrigerant is absorbed by the natural gas through a heat exchanger (1g-2g) where the natural gas enters into the exchanger with a temperature of  $15^\circ\text{C}$  and a pressure of 65 bar. The condensed refrigerant flows through an isenthalpic expansion valve (3-4) and then heads to the evaporator to produce the required cold (4-1). After that, the saturated vapor is compressed through the compressor (1-2), thus completing and closing the cycle.

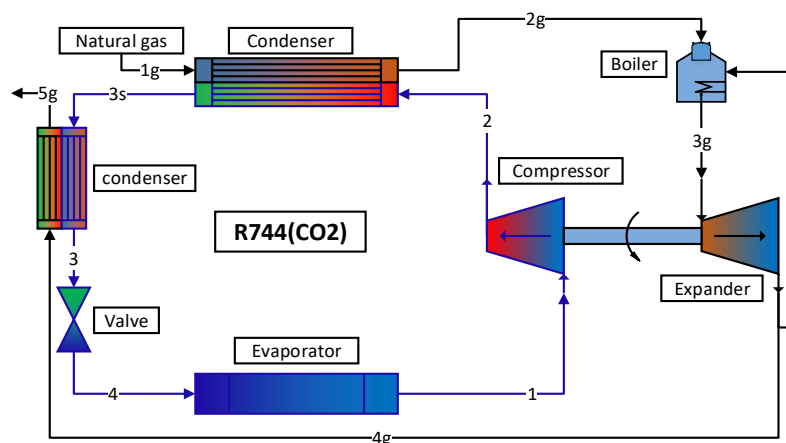


**Fig. 2 – Refrigeration systems operated by turbo-expander**

Table 1 gives the operating pressures of some expansion stations as well as the expansion ratio for a supply pressure of 65 bars (cases from Algeria).

**Table 1 – Operating pressures**

|               | Transport pressure<br>[bar] | Operating pressure<br>[bar] | Expansion ratio |
|---------------|-----------------------------|-----------------------------|-----------------|
| City          | 65                          | 4                           | 16.25           |
| Cement plant  | 65                          | 6                           | 10.83           |
| Gas turbine   | 65                          | 18 to 35                    | 3.61 to 1.86    |
| Steam turbine | 65                          | 6                           | 10.83           |
| Foundry       | 65                          | 8                           | 8.13            |

### 3 Natural gas composition

Natural gas is primarily composed of methane (70 to 90 %), accompanied by heavier hydrocarbons (ethane, propane), nitrogen, CO<sub>2</sub>, and trace impurities (Hydrogen, Helium, sulfur compounds, water). This variable composition depends on the geological origin of the reservoir and determines the required treatments prior to distribution or liquefaction. Table 2 summarizes the composition of natural gas considered in the simulations.

**Table 2 – Natural gas characteristic**

| Components     | Chemical formulas              | Mole fraction (%) | $P_c$ (bar) | $T_c$ [K] | HHV (MJ/kg) | LHV (MJ/kg) |
|----------------|--------------------------------|-------------------|-------------|-----------|-------------|-------------|
| Methane        | CH <sub>4</sub>                | 83.50%            | 45.992      | 190.56    | 55.536      | 50.048      |
| Ethane         | C <sub>2</sub> H <sub>6</sub>  | 6.90%             | 48.718      | 305.33    | 51.902      | 47.611      |
| Propane        | C <sub>3</sub> H <sub>8</sub>  | 2.10%             | 42.771      | 369.83    | 50.322      | 46.33       |
| i-Butane       | iC <sub>4</sub>                | 0.35%             | 36.40       | 407.82    | 49.363      | 45.577      |
| n-Butane       | C <sub>4</sub> H <sub>10</sub> | 0.53%             | 37.96       | 425.13    | 49.511      | 45.725      |
| i-Pentane      | iC <sub>5</sub>                | 0.11%             | 33.957      | 460.35    | 48.909      | 45.249      |
| n-Pentane      | C <sub>5</sub> H <sub>12</sub> | 0.12%             | 31.60       | 433.75    | 49.003      | 45.343      |
| Hexane         | C <sub>6</sub> H <sub>14</sub> | 0.14%             | 30.340      | 507.82    | 48.454      | 44.879      |
| Nitrogen       | N <sub>2</sub>                 | 5.85%             | 33.958      | 126.19    | 0           | 0           |
| Carbon Dioxide | CO <sub>2</sub>                | 0.21%             | 73.773      | 304.13    | 0           | 0           |
| Helium         | He                             | 0.19%             | 0.2274      | 5.1953    | 0           | 0           |

### 4 Modeling

The modeling of the system was formulated by establishing the mass and thermal balances through each element of the installation, based on the following hypotheses: steady state, pressure and heat losses through the pipes are negligible and variations in kinetic and potential energies in compressors and heat exchangers are ignored.

Drawing upon an in-depth thermodynamic analysis, this study evaluates the system through energy and exergy balances. Tables 3 and 4 present a comprehensive summary of the detailed balance equations for each system component

**Table 3 – Energy balance for the different components of the installation**

| Component       | Formula                                                                                                             | number |
|-----------------|---------------------------------------------------------------------------------------------------------------------|--------|
| Compressor      | $\dot{m}_r h_1 + \dot{W}_c = \dot{m}_r h_2$                                                                         | (1)    |
| Condenser       | $\dot{m}_r h_2 = \dot{Q}_c + \dot{m}_r h_3$                                                                         | (2)    |
|                 | $\dot{m}_g h_{1g} + (\dot{m}_g - \dot{m}_b) h_{4g} + \dot{Q}_c = \dot{m}_g h_{2g} + (\dot{m}_g - \dot{m}_b) h_{5g}$ | (3)    |
| Expansion valve | $\dot{m}_r h_3 = \dot{m}_r h_4$                                                                                     | (4)    |
| Evaporator      | $\dot{m}_r h_4 = \dot{Q}_e + \dot{m}_r h_1$                                                                         | (5)    |
| Expander        | $\dot{m}_g h_{3g} = \dot{W}_t + \dot{m}_g h_{4g}$                                                                   | (6)    |
| Boiler          | $\dot{m}_g h_{2g} + \eta_{ch} \cdot \dot{Q}_H = \dot{m}_g h_{3g}$                                                   | (7)    |
|                 | $\dot{Q}_H = \dot{m}_b LHV$                                                                                         | (8)    |

**Table 4 – Exergy destruction for the different components of the installation**

| Component       | Formula                                                                                                                                                                            | number |
|-----------------|------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|--------|
| Compressor      | $\dot{X}_W = \dot{m}_r T_0 (s_2 - s_1)$                                                                                                                                            | (9)    |
| Condenser       | $\dot{X}_{CD} = \dot{m}_r [h_2 - h_3 - T_0 (s_2 - s_3)] + \dot{m}_g [h_{1g} - h_{2g} - T_0 (s_{1g} - s_{2g})] + (\dot{m}_g - \dot{m}_s) [h_{4g} - h_{5g} - T_0 (s_{4g} - s_{5g})]$ | (10)   |
| Expansion valve | $\dot{X}_D = \dot{m}_r T_0 (s_4 - s_3)$                                                                                                                                            | (11)   |
| Evaporator      | $\dot{X}_{EV} = \dot{m}_r [h_1 - h_4 - T_0 (s_1 - s_4)] - \left[ \dot{Q}_e \left( 1 - \frac{T_0}{T_E + \Delta T} \right) \right]$                                                  | (12)   |
| Expander        | $\dot{X}_T = \dot{m}_{gt} T_0 (s_{3g} - s_{4g})$                                                                                                                                   | (13)   |
| Boiler          | $\dot{X}_H = \dot{m}_g T_0 (s_{3g} - s_{2g})$                                                                                                                                      | (14)   |

with,

$$\eta_{is} = \frac{h_{2is} - h_1}{h_2 - h_1} \quad (15)$$

In the literature, there are several empirical correlations relating the isentropic efficiency to the compression ratio. Here, the chosen correlation is given as [29]:

$$\eta_{isc} = 0.8981 - 0.092384 * \tau_c + 0.00476 * \tau_c^2 \quad (16)$$

$$\text{With } \tau_c = \frac{P_2}{P_1} \quad (17)$$

The isentropic efficiency of the expander as  $\eta_{isT} = 0.8$  and the boiler efficiency as  $\eta_{ch} = 0.9$

## 5 Performance

Global coefficient of performance  $COP_g$  and that of the compression system coefficient of performance  $COP_c$  are given by:

$$COP_g = \frac{\dot{Q}_e}{\dot{Q}_H} \quad \text{and} \quad COP_c = \frac{\dot{Q}_e}{\dot{W}_c} \quad (18)$$

With  $W_C = \eta_m W_T$  and  $\eta_m = 0.9$ , as well as the exergy efficiency is defined by:

$$\eta_{ex} = 1 - \frac{\dot{X}_{SYS}}{\dot{E}x_{in}} \quad (19)$$

With

$$\dot{X}_{SYS} = \dot{X}_W + \dot{X}_{CD} + \dot{X}_D + \dot{X}_{EV} + \dot{X}_T + \dot{X}_H \quad (20)$$

And

$$\dot{E}x_{in} = \dot{E}x_{3g} - \dot{E}x_{2g} \quad (21)$$

## 6 Results and discussions

At this stage of the study, it should be noted that the developed computational model is implemented in MATLAB and coupled with the REFPROP database, in accordance with the mathematical formulations presented in the previous sections. The use of REFPROP enables an accurate evaluation of the real thermophysical properties for both the pure refrigerant and natural gas. Several operating parameters are considered as input variables during the simulations; their values and ranges are summarized in Table 5.

**Table 5 – Operating conditions**

|                 | Temperature [°C] | Transport Pressure [bar] | Flow rate [kg/s] | Operating Pressure [bar] |
|-----------------|------------------|--------------------------|------------------|--------------------------|
| Studied Case    | 15               | 65                       | 1.5              | 35                       |
| Variance margin | 15               | 45 – 65                  | 0.5 - 50         | 5 - 35                   |

### 6.1 Effect of expander inlet temperature

All expansion stations on the national natural gas network are equipped with conventional boilers; this is to prevent the cooling of natural gas below the temperature threshold for the formation of hydrates. In fact, this motivated the idea to modernize these expansion stations and make them more energy efficient by installing turbo-expanders in order to recover the energy contained in the natural gas dissipated in the thermostatic expansion valves. However, a major technical problem could arise at the end of the expansion, namely, the formation of methane hydrates. To avoid this, studying the effect of temperature at the expander inlet is crucial to determine the operating parameters.

The quantity of heat necessary to reach the temperature  $T_{3g}$  is produced, in the case of conventional installations, by the combustion of small quantities of NG. Contrary to our case, a very large part of this heat is recovered from that released during the condensation of CO<sub>2</sub>. Figure 3 shows the evolution cycle of NG in both T-s and h-s diagrams, meanwhile, Figure 4 depicts the refrigerant cycle (R744) on both T-s and p-h diagrams.

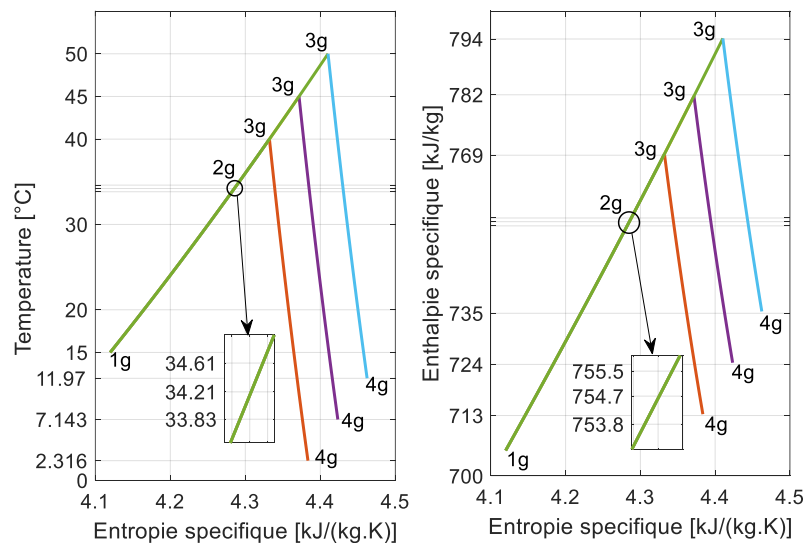


**Fig. 3 – T-s and h-s cycle of natural gas expansion**

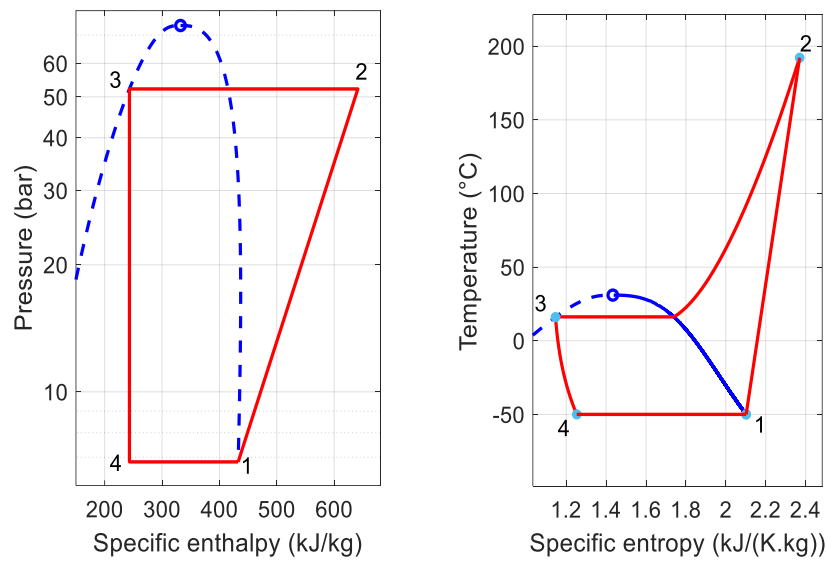


**Fig. 4 – T-s and p-h diagram of refrigeration system**

The 1g-2g evolution represents the increase in the NG temperature by the recovery of the heat evacuated by the condenser. As a result, the increase in temperature (also in enthalpy) from 2g to 3g is due to the heat released by the combustion of NG in the boiler. Furthermore, the mechanical power produced is generated following the expansion of NG (3g-4g). In other words, the temperature  $T_{3g}$  influences the operating parameters of the installation as well as the powers produced. Namely, the cooling power  $\dot{Q}_E$  and the power of the expander  $\dot{W}_T$  (see Figure 5). The power  $\dot{Q}_E$  decreases with the increase in temperature  $T_{3g}$  because of the higher temperature of NG at the outlet of the expander which leads to an increase in the condensation temperature, thus reducing the refrigeration effect ( $h_1 - h_4$ ), while the power  $\dot{W}_T$  increases with the increase in temperature  $T_{3g}$  (hence the increase in enthalpy at the entrance to the expander). Furthermore, one of the most important design parameters having a substantial effect on system performance is the natural gas preheating temperature  $T_{3g}$ . A low value of this temperature has negative effects on the operation of the expander. On the other hand, for higher values of  $T_{3g}$  affects the performance coefficients of the installation (see Figure 5). The  $COP_g$  decreases from 20.45 to 0.762 and the  $COP_c$  decreases from 1.073 to 0.755, following an increase in  $T_{3g}$  from 36 °C to 55 °C. This observation can be justified; firstly, by the reduction in the cooling power and secondly, by the increase in the thermal power and the power consumed by the compressor following the increase in the condensing temperature

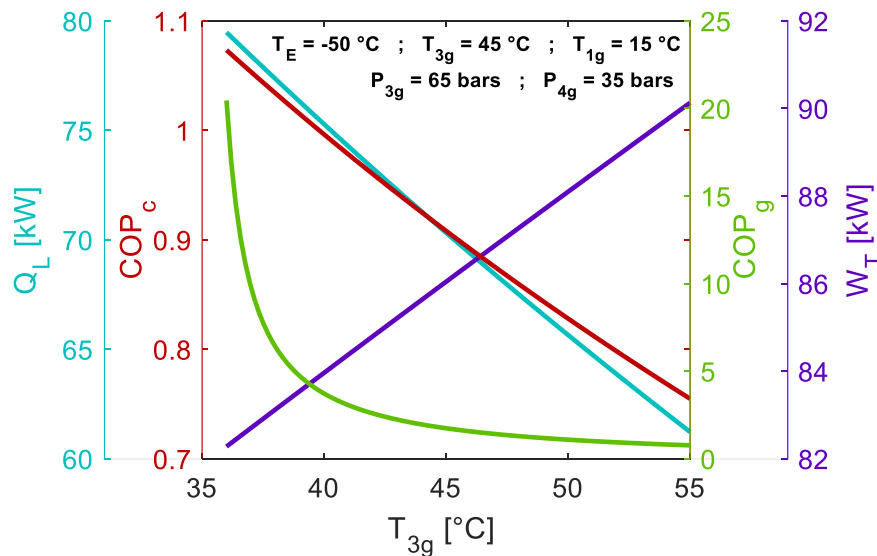


**Fig. 5 – Powers and performances**

## 6.2 Case study

For the case dealt with here, by keeping the gas temperature at the expander inlet at the equivalent value of a conventional expansion station ( $T_{3g} = 45\text{ °C}$ ). The main purposes are the global thermodynamic assessment of the system at this operating point. The values of the main operating parameters, at each point of the cycle considered, are listed in table 6. The results obtained show that the overall performance and compression coefficients are respectively 1.7371 and 0.9081 as well as the exergy efficiency is 88.39 %. As a result, the power developed by the expander is 86.0419 kW and that of the evaporator is 70.3207 kW (20.03439 kTR).

**Table 6 – Cycle point characteristic**

| Points | Pression [bar] | Temperature [°C] | Mass flow rate [kg/s] | Enthalpy [kJ/kg] | Entropy [kJ/kg.K] |
|--------|----------------|------------------|-----------------------|------------------|-------------------|
| 1      | 6.8234         | -50              | 0.37119               | 432.676          | 2.101826          |
| 2      | 58.5151        | 213.98           | 0.37119               | 641.299          | 2.369826          |
| 3      | 58.5151        | 16.07            | 0.37119               | 243.226          | 1.146525          |
| 4      | 6.8234         | -50              | 0.37119               | 243.226          | 1.252849          |
| 1g     | 65             | 15               | 1.5                   | 705.484          | 4.119841          |
| 2g     | 65             | 34.21            | 1.5                   | 754.737          | 4.285351          |
| 3g     | 65             | 45               | 1.5                   | 781.725          | 4.371654          |
| b      | 35             | 7.14             | 9.6386e-07            | 724.363          | 4.371654          |
| 4g     | 35             | 7.14             | 1.499999              | 724.363          | 4.423395          |

The results of the exergy analysis are presented in Figure 6, for each component of the installation. It should be noted that, for the evaporator, the destroyed exergy rate is the highest (40.914%) compared to the other organs, which is due to the temperature difference between the evaporator and the ambient environment, which caused a strong generation of entropy. The destroyed exergy rate in the boiler and in the condenser ranks second (25.091%) and third (19.781%) after the evaporator. The main reason for this behavior is due to the temperature difference between the hot and cold flows in the boiler and the phase change in the condenser, which caused a somewhat high entropy generation rate. Moreover, it is recorded that the rates of the destroyed exergy for the compressor, the expansion valve and the expander are respectively 6.529%, 2.59% and 5.094%, we notice low values compared to the others because the entropies generated are less important compared to the evaporator, the boiler and the condenser.

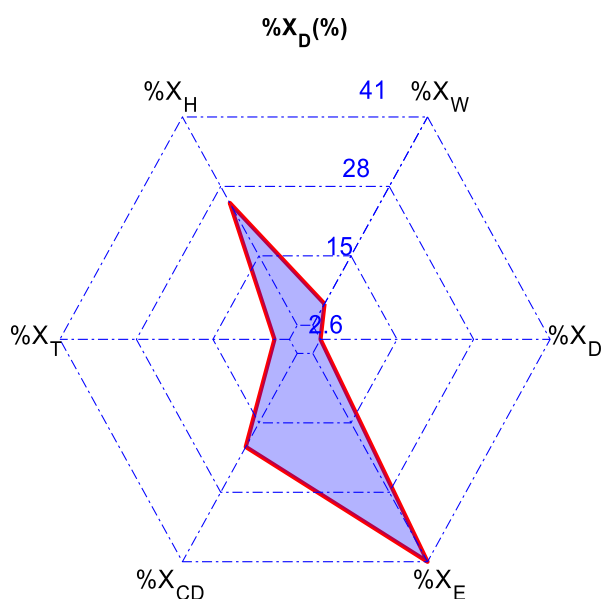


**Fig. 6 – Destroyed Exergy rate in the various components of the system**

## 7 Conclusion

In this article, a turbo-expander is installed in place of a conventional thermostatic expansion valve to enhance pressure energy recovery during natural gas expansion. This innovative configuration serves a dual purpose: it generates mechanical power to drive a refrigeration system while simultaneously exploiting the low temperature of the expanded natural gas, below 20 °C, to condense CO<sub>2</sub>. A detailed simulation coupling MATLAB with NIST-REFPROP 9 was performed to evaluate system performance, yielding several key findings. First, the turbo-expander successfully produces usable mechanical energy even at relatively low natural gas preheating temperatures ( $T_{3g}$ ). Second, the global coefficient of performance consistently exceeds that of conventional compression systems, demonstrating a significant COP improvement. Third, an inverse relationship was observed between natural gas preheating temperature and both COP and cooling capacity, higher preheating temperatures reduce system efficiency. Finally, the mechanical power output from the turbo-expander ( $\dot{W}_T$ ) increases proportionally with  $T_{3g}$  and consistently surpasses the cooling power demand ( $\dot{Q}_E$ ), confirming the technical feasibility and energetic advantage of this integrated approach for simultaneous power recovery and low-temperature CO<sub>2</sub> refrigeration.

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