# $CO_2$ CARNOT-TYPE CYCLE BASED ON ISOTHERMAL VAPOUR COMPRESSION


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## ABSTRACT

The purpose of this work is to explore the possibility of enhancing the $CO_2$ trans-critical cycle by performing a novel Carnot-type cycle between two heat sources at $T_{High}$ and $T_{Low}$. From a thermodynamical point of view, the proposed design, with ideal evolutions, achieves to get close to Carnot COP. The influence of expansion irreversibilities was explored too. Finally, a design resulting from a compromise between complexity and performance is retained.


## 1. INTRODUCTION

Current earth environmental situation leads to avoid or limit direct greenhouse effect due to HFC emissions. Natural fluids like ammonia, hydrocarbons are already used in industry applications and carbon dioxide should be a long-term substitute if $CO_2$ systems become as efficient as current HFC ones. During the last two decades, $CO_2$ research programs have brought this fluid to a competitive level with HFC in mobile air conditioning applications for a large area of operating temperatures (Pettersen and Neskå, 2003). Indeed, its excellent transport properties are adapted with specific constraints like space and weight, but are balanced with poor thermodynamic aspect. Different solutions to improve $CO_2$ cycle performances are currently under development: two-stage compressors (Hwang et al., 2004), new designed heat exchangers (Hafner, 2004), expansion work recovering (Robinson and Groll, 1998) and implementation of an ejector system (Denso, 2005).

In this paper, the possibility to enhance carbon dioxide cycle performances by using isothermal compression associated with expansion energy recovery will be explored (Meunier, 2006).

## 2. ISOTHERMAL COMPRESSION CARNOT-TYPE CYCLE

The aim of the proposed thermodynamic cycle is to reach Carnot cycle performances between two heat sources at $T_{Low}$ and $T_{High}$. The usual isentropic vapour compression shows its own limits due to carbon dioxide thermodynamic properties. The only solution to get a great improvement of $CO_2$ refrigerant performances is to change significantly the cycle design as shown on Figure 1.

Herein, a Carnot-type cycle based on isothermal compression is studied (Meunier, 2006). During this nearly reversible cycle, the refrigerant exchanges heat with two external heat sources at constant temperature ($T_{Low}$ and $T_{High}$) and exchanges work with two expanders and with a compressor. An ideal adiabatic heat exchanger (IHX) is also used.

An originality of this cycle is that only a fraction (x), of the high-pressure discharged refrigerant is transferred to the IHX to allow it to reach $T_{Low}$. The expansion of the high-pressure liquid at the outlet of the IHX must be isothermal to allow the cycle to be a two-temperature reversible cycle. The other high-pressure discharged fraction (1-x), is expanded through an adiabatic and reversible expansion device.

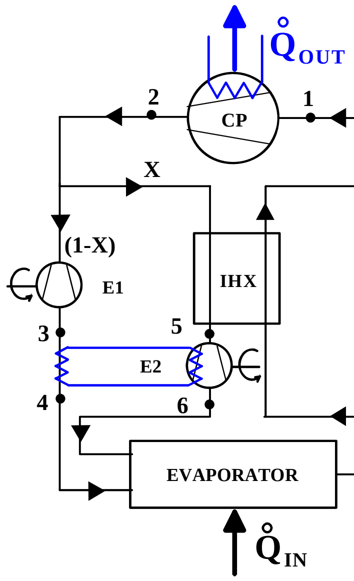


Figure 1: Proposed configuration

We then get a Carnot-type cycle, including a reversible isothermal compression, two reversible expansions (one isentropic and the other isothermal) and an internal heat exchanger. Assuming the IHX to be ideal, this cycle yields a Carnot coefficient of performance (COP) since it is a reversible cycle with two isotherm and one isentropic transformations. However, the IHX is not ideal, herein, we study the influence of the IHX on the COP. Figure 2 represents the proposed cycle for two different high-pressures: 60 bar (1,2',3',4',5',6,7) and 100 bar (1,2,3,4,5,6,7).

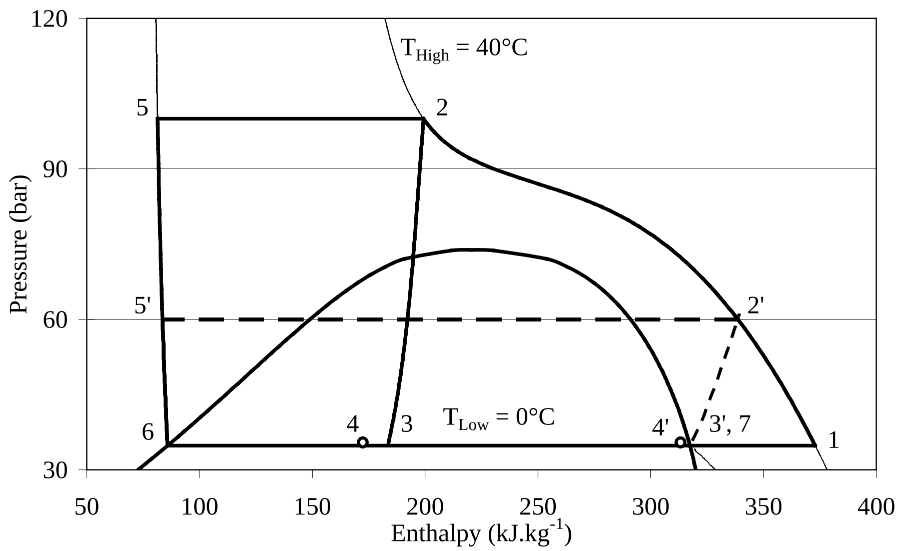


Figure 2: Cycle with two different high-pressures.

### 2.1. Components description

As the compressor is cooled at the ambient temperature, a gas-cooler is not needed (Figures 1 & 2), which could result in a more compact and lighter system.

- Compressor (CP)

Assuming the isothermal compression to be reversible without heat losses yields:

$$\dot{Q}_{OUT} = T_{High} \cdot \dot{m}_{CP} \cdot (s_2 - s_1) \tag{1}$$

$$\dot{W}_{CP} = \dot{m}_{CP} \cdot \left(h_2 - h_1\right) - \dot{Q}_{OUT} \tag{2}$$

- Internal Heat eXchanger (IHX)

A fraction (x) of the total discharged mass flow is transferred to the IHX high-pressure side (Figure 1), so that the refrigerant is cooled from $T_{High}$ to $T_{Low}$ into the IHX. The energy balance at the IHX gives:

$$\left(h_7 - h_1\right) + x \cdot \left(h_2 - h_5\right) = 0 \tag{3}$$

For such configuration, the entropy generation is due to the non-ideal heat transfer in the IHX:

$$P_{IHX}(S) = \dot{m}_{CP} \cdot \left(x \cdot \left(s_5 - s_2\right) + \left(s_1 - s_7\right)\right) \text{ with } P_{IHX}(S) \geq 0 \tag{4}$$

- Expanders (E)

The other fraction (1-x) of the discharged mass flow at $T_{High}$ enters into an isentropic expansion device. This expansion presents two advantages: it recovers mechanical power (eq. (5)) and increases outlet liquid quality. This aspect is all the more true as the inlet temperature is high (Figure 2).

$$\dot{W}_{E1} = \dot{m}_{CP} \cdot (1 - x) \cdot \left(h_3 - h_2\right) \tag{5}$$

The fraction (x) leaving the IHX is expanded through an isothermal reversible expander. Working expansion power and cooling capacity are recovered as given by eq. (6) and eq. (7):

$$\dot{Q}_{E2} = T_{Low} \cdot \dot{m}_{CP} \cdot x \cdot \left(s_6 - s_5\right) \tag{6}$$

$$\dot{W}_{E2} = \dot{m}_{CP} \cdot x \cdot \left(h_6 - h_5\right) - \dot{Q}_{E2} \tag{7}$$

This cooling capacity can be used as cooling load or can be either used to increase the liquid quality (3-4 in Figure 2) of the other outlet expansion fluid as shown by equation 8:

$$\dot{Q}_{E2} = \dot{m}_{CP} \cdot (1-x) \cdot \left(h_3 - h_4\right) \tag{8}$$

Mechanical power from each expander is recovered, so that the net compressor input power is:

$$\dot{W}_{NET} = \left|\dot{W}_{CP}\right| - \left|\dot{W}_{E1}\right| - \left|\dot{W}_{E2}\right| \tag{9}$$

- Evaporator

Outlet two-phase flow from E1 and saturated liquid from E2 are at the same temperature $T_{Low.}$ In consequence, the cooling rate of the constant and uniform temperature evaporator is given by:

$$\dot{Q}_{IN} = \dot{m}_{CP} \cdot \left(h_7 - x \cdot h_6 - (1 - x) \cdot h_4\right) \tag{10}$$

### 2.2. Numerical comparison with classical $CO_2$ cycle

Results are presented for an air conditioning application where the two heat sources are $T_{Low}$ = 0°C and $T_{High}$ = 40°C and assuming a unitary total mass flow rate. All thermodynamical properties are determined with Refprop7. The corresponding Carnot Coefficient Of Performance (COP), calculated by eq. (11), is equal to 6.829:

$$COP_{CARNOT} = \frac{T_{Low}}{T_{High} - T_{Low}} \quad \text{and} \quad COP = \frac{\dot{Q}_{IN}}{\dot{W}_{NET}} \tag{11}$$

For the cycle to operate correctly, the HP has to be higher than 60 bar. In the following, we study the COP versus HP for the two following cycles: conventional design with an isentropic expansion device (A - IsoS(S) design in Figure 5), and the new proposed cycle (B). In each case, expansion work is recovered.

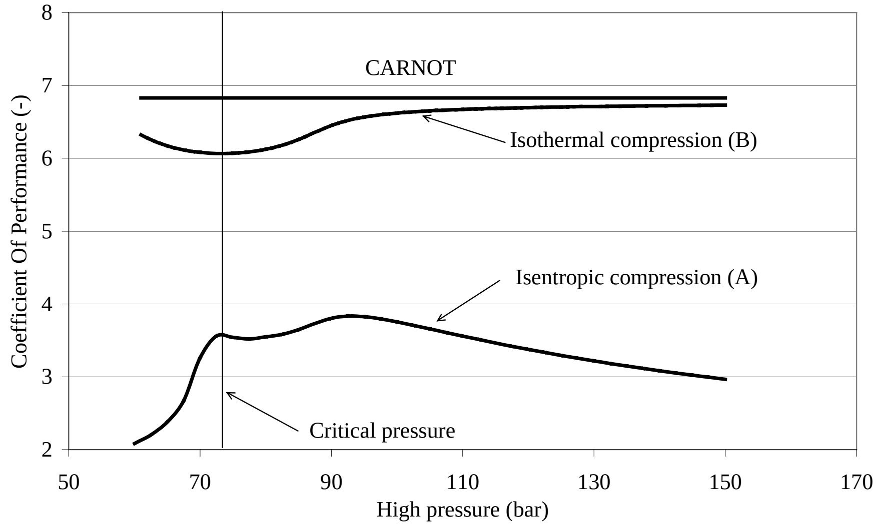


Figure 3: Evolution of COP vs. High Pressure for the two cycles.

For the proposed system (B), the performances are always greatly higher than for previous design (A) and close to Carnot COP. For conventional trans-critical cycle, an optimal HP exists at approximately 95 bar corresponding to a COP equal to 3.825. For the new design cycle, a minimum is observed and the deviation with the Carnot COP is due to the irreversibilities generated inside the IHX.
To explain this deviation, we consider the entropy production. This can be performed using the entropy number ($N_S$), which is introduced in equation 12 resulting from the combination of the 1$^{st}$ and 2$^{nd}$ laws for the global system:

$$COP = COP_{CARNOT} \cdot \left(1 - N_S\right) \text{ with } N_S = \frac{T_{High} \cdot P_{IHX}(S)}{\dot{W}_{NET}} \qquad (12)$$

Figure 4 represents the evolution of entropy production number history of the global system as function of the high-pressure. The entropy production number (Ns) is maximum and the COP is minimum when the high-pressure is equal to the critical pressure. At higher pressure, the COP is higher too and tends towards an asymptote. For example at 150 bar, COP is about 99% of Carnot COP.

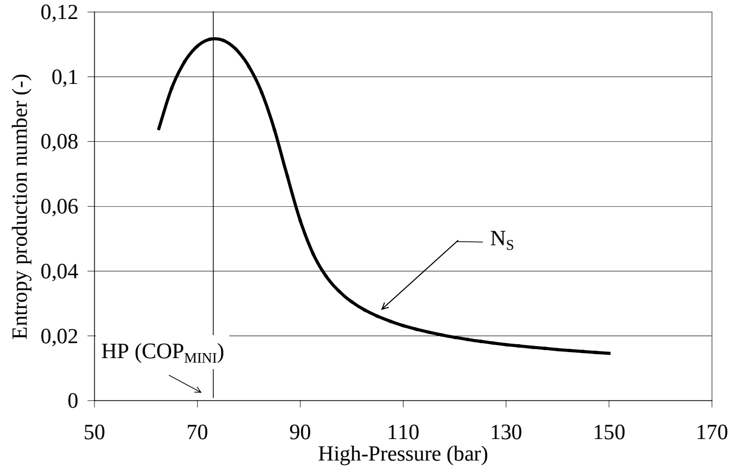


Figure 4: Evolution of $N_S$ versus High-Pressure for new cycle.

To compare the proposed design with the conventional one, Table 1 shows some important figures for two different high pressures: 92.5 bar corresponding to the optimal performance of isentropic cycle and 150 bar corresponding to the highest value of the COP observed for isothermal cycle.

| Cycle | Iso. S compression | Iso. T compression | |
|---|---|---|---|
| $HP$ $(bar)$ | 92.5 | 92.5 | 150 |
| $COP$ $(-)$ | 3.833 | 6.518 | 6.729 |
| $Volumetric\ \ capacity$ $\left(kJ\cdot m^{-3}_{Suc.}\right)$ | 11633 | 11769 | 15166 |
| $\left(\lvert\dot{W}_{E1}\rvert+\lvert\dot{W}_{E2}\rvert\right)/\dot{W}_{NET}$ $(\%)$ | 25% | 49% | 45% |
| $\dot{Q}_{IN}/\dot{Q}_{OUT}$ $(\%)$ | 79% | 87% | 87% |

Table 1: Comparison of conventional and isothermal compression.

In contrast with the HFC, expansion energy recovering is important for carbon dioxide (Pettersen et al., 1998), in particular for proposed design: it represents more than 45% of the effective input power against 25% for classical isentropic case.
For a same high pressure, performances are greatly improved (about 70-75%) without changing significantly the refrigerating effect per unit of swept volume and it is true for all the high-pressure scale. So, it should be highlighted that isothermal compressors, as conventional $CO_2$ ones, keep the advantage of being smaller than HFC compressors.
In fact, with the same cooling capacity, the isothermal cycle needs to reject approximately 10% less of heat energy than conventional cycle. Moreover, thanks to the cooling during compression, the needed mechanical power is less important for an isothermal compression than for the equivalent isentropic one.

## 3. OTHER POSSIBILITIES OF CYCLE DESIGN

In this paragraph, we will evaluate the influence of design and expander efficiency.

### 3.1. Influence of cycle design

Let us highlight the impact of cycle changes on the performances. A variety of cycles based on the two cycles presented previously, will be discussed:

One expansion cvcle
- A conventional design (reversible adiabatic compressor, gas-cooler, IHX, evaporator and expander with work recovering): **IsoS(S)**.
- Same design as IsoS(S) with an isothermal compression: **IsoT(S)**.
- Same design as IsoT(S) with an adiabatic expansion: **IsoT(H)**.

Two expansions cvcle
- Design described on Figure 1: **IsoT(E1:S-E2:T)**.
- Adiabatic expansion replaces the isothermal one: **IsoT(E1:S-E2:H)**.
- Two adiabatic expansions are used: **IsoT(E1:H-E2:H)**.

All of the quoted designs are drawn on Figure 5. Concerning the HP, it is set at its optimum value if it exists. If not, the high-pressure will be set at the highest value of the explored scale (60-150 bar), which corresponds to the highest COP value too.

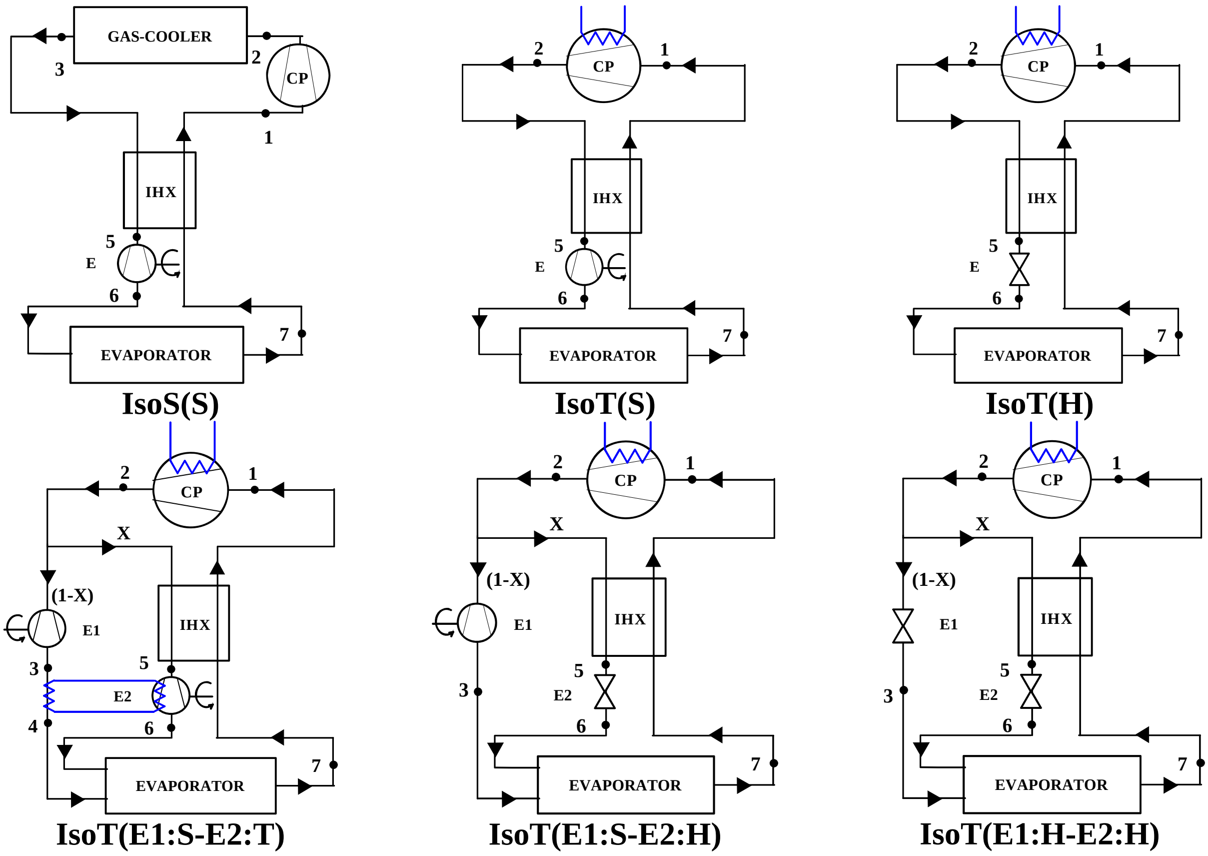

Figure 5: Sketches of the different designs.

The results for the different configurations are reported on Figure 6:

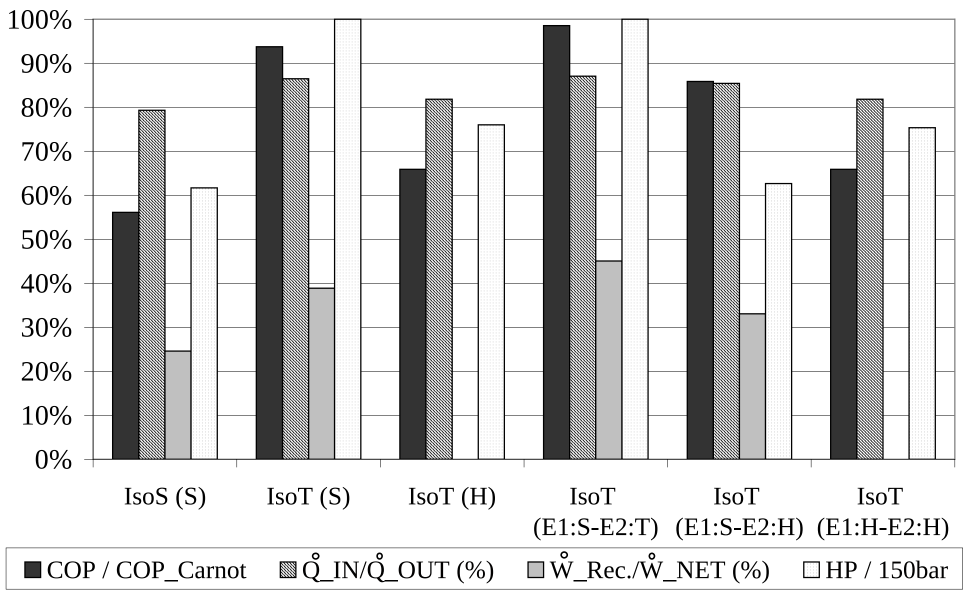

Figure 6: Systems performances with ideal evolutions.

Isothermal compression leads to better performances than the adiabatic one. The higher $COP / COP_{Carnot}$ is obtained with the Carnot-type cycle, IsoT(E1:S-E2:T), and is equal to 98% against only 56% for the conventional one with expansion device, IsoS(S). However, IsoT(S) with a single expansion device achieves a 93% efficiency with a much simpler design. The choice of a throttling device for the expansion leads to a considerable decrease in cycle efficiency: 86% for IsoT(E1:S-E2:H) and only 65% for IsoT(E1:H-E2:H) and IsoT(H).

Regarding the heat flux, we can see that the ratio $\left(\dot{Q}_{IN}/\dot{Q}_{OUT}\right)_{System}$ varies only between 82% and 87% for isothermal cases (to be compared with 79% for IsoS(S)).

### 3.2. Influence of isentropic efficiency

The proposed designs are the same as in section 3.1. but a 60% isentropic efficiency value (Robinson and Groll, 1998) is taken into account for the expansion when available. The new results are reported in Figure 7:

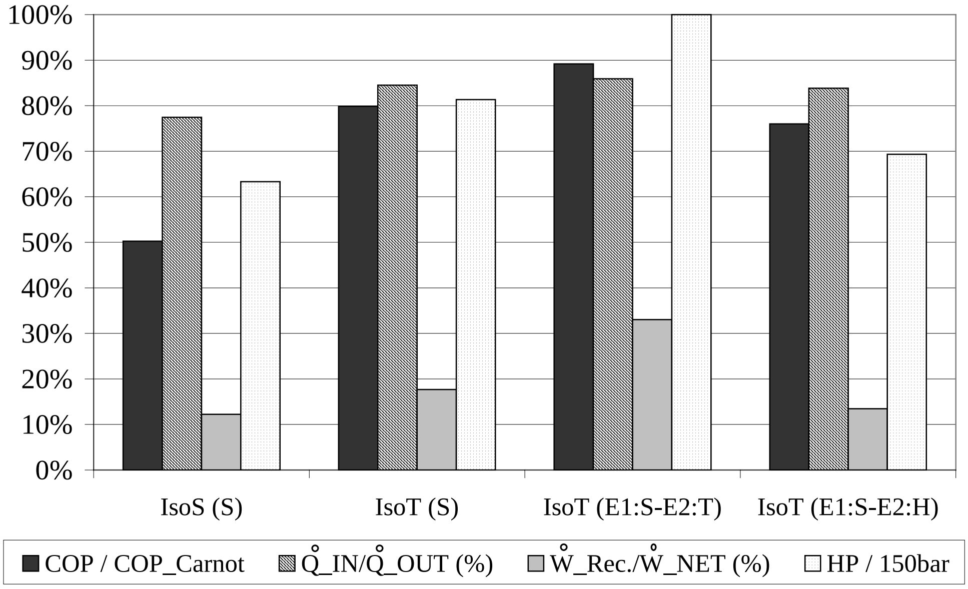


Figure 7: Systems performances with isentropic expansion efficiency (60%).

Of course, expansion efficiency is decreasing both cooling capacity and work recovering then global performances are decreasing: only minus 6% for isentropic compression IsoS(S) and between minus 10 and 13% for isothermal compression. However, isothermal designs are always more efficient. Moreover, IsoT(E1:S-E2:T) and IsoT(S) still have the best performances, respectively 89% and 80%. Except for IsoT(E1:S-E2:T), an optimal high-pressure exists. The optimal HP is more important for isothermal cases (between 105 and 150 bar for IsoT against 95 bar for IsoS(S)).
If expansion efficiencies are reduced from 60% to 40%, performances are decreasing about 6,5% and 4,9% for respectively IsoT(S) and IsoS(S) compression. It means that isothermal case is a little more sensitive to this parameter, but not so radically to invert the choice of the best compromise design.

### 3.3. Influence of isothermal compression efficiency

Finally, an isothermal compression efficiency will be approached for IsoT(S) with a 60% expansion efficiency. The aim is to compare non-isothermal reversible compression with isothermal one. This efficiency is defined as following:

$$\eta_{IsoT\_CP} = \frac{\dot{m}_{CP} \cdot \left[\left(h_{2\_IsoT} - h_1\right) - T_{High} \cdot \left(s_{2\_IsoT} - s_1\right)\right]}{\dot{m}_{CP} \cdot \left(h_2 - h_1\right) - \dot{Q}_{OUT}} \qquad (13)$$

For a discharged compressor temperature fixed at 60°C with $T_{High}$ equal to 40°C, the best thermodynamical efficiency is about 60% (vs. 80% for isothermal case) for maximal high pressure (150 bar) corresponding to an isothermal compression efficiency of 96% (for comparison, this efficiency is 69% for IsoS(S) cf. § 3.2). Performances decrease significantly but are still better than an isentropic compression solution (COP / $COP_{Carnot}$ = 50% in Figure 7).

## 4. CONCLUSION

The proposed Carnot-type design shown in Figure 1, (isothermal compression coupled with one isentropic expansion and one isothermal expansion), leads to performances very close to Carnot ones. The deviation is only due to IHX exchanger irreversibilities, which tend towards zero when the high-pressure increases.
Various design modifications are explored. A simpler cycle, associating only an isentropic expander and an isothermal compression, has a coefficient of performance of 6.35 compared to 6.73 for the Carnot-type design. This simpler design appears to be very attractive. The isentropic expansion efficiency affects the COP but not too drastically.
Next research tasks will be devoted to explore the ways to evacuate the heat during compression.

## NOMENCLATURE

| | | | | |
|---|---|---|---|---|
| $\dot{W}$ | Mechanical power | (W) | Rec | Recoverable |
| $\dot{Q}$ | Heat capacity | (W) | High | Hot heat source |
| HP | High Pressure | (bar) | Low | Cold heat source |
| T | Temperature | (°C) | IN | Input |
| $\dot{m}$ | Mass flow rate | ($kg.s^{-1}$) | OUT | Output |
| x | Mass flow rate fraction | (%) | IHX | Internal Heat eXchanger |
| $N_S$ | Entropy number | (-) | CP | Compressor |
| P(S) | Entropy production | ($W.K^{-1}$) | E | Expander |
| Subscripts | | | NET | Net |
| Rec | Recoverable | | IsoT | Isothermal |

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