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Abstract

A novel Carnot-type cycle is proposed to allow ideal theoretical performances for sub-critical and trans-critical applications. Three natural fluids, ammonia, carbon dioxide and nitrogen corresponding to one of the application were chosen. Compared to conventional cycles, the Carnot-type cycle leads to COP increase by 4% to 70% depending on the application. For the most suitable case (CO_2), a simpler design resulting from the trade-off between complexity and performances still exhibits high performances (92.5% of COP_{Carnot} vs. 95.4% for Carnot-type cycle). Compared to conventional one, this simpler design appears to be more sensitive to the expander efficiency but still achieves higher performances: 80% vs. 50% of COP_{Carnot}. The impact of the cooling efficiency was explored too. Considering a compressor temperature pinch equal to 20K, despite a decrease by 24% of the COP the performances are still better than those of the conventional one (+21% of COP).

Keywords

Natural refrigerants; Refrigerating system; Compression system; Isothermal compression; Thermodynamic cycle

Nomenclature

$\dot{W}$	mechanical power (W)
$\dot{Q}$	heat capacity (W)
$\dot{Ex}_d$	exergy destruction (W)
T	temperature (K)
$\tilde{T}$	entropic temperature (K)
ΔT	temperature difference (K)
P	pressure
h	enthalpy (kJ.kg^{-1})
s	entropy (kJ.kg^{-1}.K^{-1})
$\dot{m}$	mass flow rate (kg.s^{-1})
x	mass flow rate fraction (%)
η	efficiency (-)

Subscripts

Dis	discharge
LT	low-temperature
HT	high-temperature
LP	low-pressure
HP	high-pressure
In	inlet
Out	outlet
CP	compressor
CD	condenser
E	expander

EV	evaporator
Crit	critical
Ref	reference
Iso_T	isothermal
Abbreviation	
COP	Coefficient Of Performance
IHX	Internal Heat eXchanger

1 Introduction

In cooling applications, the reverse Rankine cycle, consisting in an isentropic compression, a throttling valve expansion and two isobaric transformations, is mostly used because of practical considerations. Depending on the fluid and heat source temperatures, two main configurations can be distinguished: sub-critical and trans-critical.

Due to evaporation and condensation at constant temperature, reverse Rankine cycle is appropriate for reaching high efficiency with sub-critical cycles. Nevertheless, throttling device and discharged compressor vapour desuperheating, limit its efficiency compared to Carnot cycle. Therefore, designs have been developed to reduce those impacts. Cascades [1] are usually used for very important heat source temperature differences. The disadvantage is to add a temperature pinch in the intermediate heat exchanger. The multi-stage compression is another possibility to achieve better performances. A flash gas bypass is generally used to get a temperature drop between the compressors, meaning that this mass flow fraction is not employed at the evaporator.
The use of an internal heat exchanger (IHX) can be also considered but cannot be generalized with all sub-critical cycles as its interest depends on the fluid [2]. Currently, ejector refrigeration systems are under development [3]. They consist in creating an intermediate pressure to reduce the mechanical power input. Nevertheless, for a given ejector sizing, performances are optimised for nominal operating conditions [4].

The reverse Rankine cycle, as trans-critical cycle, produces important irreversibilities during super-critical gas cooling. However, it is possible to enhance it thanks to an IHX. Indeed, it allows to decrease the optimal high-pressure and the temperature before expansion, which increases cooling capacity, but the mechanical power input too. However, in most cases, cycle performances are enhanced and more particularly with carbon dioxide [5]. Other improvement possibilities are under development. Among them, the ejector system and above all the work-recovering expander could be considered as crucial for trans-critical cycle prospects [6].

In this paper, a new cycle [7] based on isothermal compression is described and compared with sub-critical and trans-critical conventional cycles. From a thermodynamical point of view, it is well-known that the isothermal compression corresponds to the minimal net mechanical work requirement. For attractive cases, a simpler design still including an isothermal compression and an expander, having both efficiencies, is explored.

2 Novel Carnot-type cycle description

The aim of the proposed thermodynamic cycle [7] is to reach Carnot cycle performances between two temperatures called T_{LT} and T_{HT} (Figure 1). All transformations are taken as reversible when possible.

As the compressor is cooled at the warm heat source temperature, a gas-cooler is not required with trans-critical cycle. However, with a sub-critical cycle and according to technical considerations, the condensation cannot occur inside the compressor, so the heat is rejected successively through the compressor and the condenser.

An ideal IHX is assumed to heat the low-pressure refrigerant vapour from T_{LT} to T_{HT} when it cools down a fraction of the high-pressure refrigerant from the T_{HT} to the T_{LT}. Only a fraction (x), of the whole mass flow rate, is transferred to the IHX to allow it to reach T_{LT}. The IHX energy balance settles it.

$$\dot{m}_{CP} \cdot \left(h_7 - h_1 \right) + \dot{m}_{CP} \cdot x \cdot \left(h_3 - h_4 \right) = 0 \tag{1}$$

This fluid mass fraction (x) being at the cold heat source temperature, the best thermodynamical way to expand it is the isothermal one. The other mass flow fraction (1-x) is directly expanded from the compressor or the condenser outlet, depending on the cycle configuration, to the evaporator through a reversible adiabatic expander. Then, mechanical work and heat are produced and/or absorbed during expansions and compression.

We then get a Carnot-type cycle, including a reversible isothermal compression, two reversible expansions (isentropic and isothermal), an internal heat exchanger and an optional condenser (isothermal). Assuming the IHX to be ideal, this cycle yields the Carnot coefficient of performance (COP_{Carnot}).

3 Mathematical model

In this section, the needed equations for mass and energy balances are presented. All indicated symbols in the following equations must be referred to Figure 1.

- Internal Heat eXchanger

As seen in section 2, Eq. (1) gives the internal heat exchanger energy balance.

Without any IHX temperature pinch, the exchanger is thermally perfect. Nevertheless, according to the second law of thermodynamics (Eq. 2), all configurations are not possible as some of them result in negative entropy production.

$$P(s) = \dot{m}_{CP} \left(s_7 - s_1 \right) + \dot{m}_{CP} \cdot x \cdot \left(s_3 - s_4 \right) \geq 0 \tag{2}$$

Consequently, to keep the isothermal expansion, we propose to reduce the IHX low-pressure outlet temperature (T_1) until to have no IHX entropy production. As a result, we could not get an isothermal reversible compression.

The input mechanical work and heat extraction during a cooled compression depend on the thermodynamical transformation. To calculate the energy balance on the compressor, either heat transfer phenomena are taken into account [8], or the compression is divided between distinguished unitary compressions including possibly heat transfer assumptions [9]. Here, in order to keep a pure thermodynamical approach, the second solution, without heat transfer phenomena, is retained.

Then, we consider an endoreversible compression, divided in two serial compressions: an isentropic one allows leading the IHX low-pressure outlet fluid from T_1 to T_{HT}, then the fluid reaches the HP at constant temperature manner.

Herein, this reversible cycle yields the COP_{Carnot}. However, in most cases, the IHX entropy generation is positive.

Nevertheless, positive or null global entropy generation is not sufficient to affirm that the IHX can run practically. Indeed, we have to verify that it is still accurate locally. The IHX is discretized in finite elements, each exchanging the same heat capacity so that the temperature profiles on the LP and HP sides can be calculated. If the hot stream (HP side here) is always at higher or equal temperature than the cold stream (LP), then the transformation is possible.
Otherwise, T_1 must be adjusted to satisfy this condition. Consequently, the cycle is built with an endoreversible compression as previously.

- Compressor (CP) and optional condenser (CD)

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

$$\dot{Q}_{CP} = T_{HT} \cdot \dot{m}_{CP} \cdot \left(s_2 - s_1 \right) \tag{3}$$

$$\dot{W}_{CP} = \dot{m}_{CP} \cdot \left[\left(h_2 - T_{HT} \cdot s_2 \right) - \left(h_1 - T_{HT} \cdot s_1 \right) \right] \tag{4}$$

For sub-critical cycle, the heat rejected at the condenser is given by:

$$\dot{Q}_{CD} = \dot{m}_{CP} \cdot \left(h_3 - h_2 \right) \tag{5}$$

- Expanders (E1 and E2)

The fraction (1-x), of the whole high-pressure mass flow rate at T_{HT}, enters into an isentropic expansion device, which produces mechanical work.

$$\dot{W}_{E1} = \dot{m}_{CP} \cdot (1-x) \cdot \left(h_4 - h_3 \right) \tag{6}$$

The fraction (x) leaving the IHX is expanded through an isothermal reversible expander. Heat capacity and mechanical expansion work are given by Eq. (7) and Eq. (8):

$$\dot{Q}_{E2} = T_{LT} \cdot \dot{m}_{CP} \cdot x \cdot \left(s_6 - s_5 \right) \tag{7}$$

$$\dot{W}_{E2} = \dot{m}_{CP} \cdot x \cdot \left[\left(h_6 - T_{LT} \cdot s_6 \right) - \left(h_5 - T_{LT} \cdot s_5 \right) \right] \tag{8}$$

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

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

- Evaporator (EV)

The two-phase flow rate coming from E1 and saturated liquid coming from E2 enter the evaporator at the same temperature T_{LT}. The evaporator outlet is taken as saturated vapour at T_{LT}. Then, the evaporator cooling capacity is given by:

$$\dot{Q}_{EV} = \dot{m}_{CP} \cdot \left(h_7 - x \cdot h_6 - (1-x) \cdot h_4 \right) + \dot{Q}_{E2} \tag{10}$$

Here, the heat capacity absorbed by the expander 2 ($\dot{Q}_{E2}$) is used to increase liquid quality at the evaporator inlet.

The corresponding COP$_{Carnot}$ and the cycle COP are given by:

$$COP_{CARNOT} = \frac{T_{LT}}{T_{HT} - T_{LT}} \text{ and } COP = \frac{\dot{Q}_{EV}}{\dot{W}_{NET}} \tag{11}$$

All thermodynamical properties were calculated with Refprop7.

Three different cycle configurations are considered: a sub-critical, a trans-critical and a cryocooler cycles. Each of them uses a natural refrigerant: ammonia, carbon dioxide and nitrogen respectively. The revival of ammonia and particularly carbon dioxide is due to environmental preoccupations: ozone layer depletion and above all greenhouse gases emissions.

4   Sub-critical cycle

Because of its high temperature overheating during compression, ammonia was chosen. Hence, the expected benefit of isothermal compression should be more noticeable than with hydrocarbons or fluorine gases.

A Carnot-type ammonia sub-critical cycle operating between 40°C and -20°C, corresponding to refrigeration applications, is considered (Figure 2). With given operating conditions, an ideal IHX cannot be achieved without temperature pinch and the calculated temperature profiles are thermally appropriate.
Our novel Carnot-type performances are compared with that of usual ammonia refrigeration cycle (A) made of an adiabatic reversible compressor, a throttling device, a condenser without sub-cooling and an evaporator without super-heating. Furthermore, we consider another similar design equipped with an expander (B) instead of a throttling device.

The comparison results are reported in Table 1.

With these operating conditions, the performances of the Carnot-type cycle are very close to Carnot ones (99.8%). It proves that IHX irreversibilities are negligible. Compared with the conventional design (A), the novel cycle performances improvement is above 20%. However, considering a conventional cycle with an expander (B), our proposed cycle enhancement reaches only 12%.

Table 1 highlights the fact that volumetric capacity is reduced by more than 10% with the Carnot-type cycle compared with the conventional design. This reduction is the consequence of compressor sucked vapours density and specific cooling energy deviations. Indeed, the Carnot-type cycle mass flow split allows increasing the inlet evaporator liquid quality and consequently the specific cooling capacity by 15% and 11% compared to conventional cycles (A) and (B) respectively. In the other hand, the Carnot-type cycle compressor sucked vapour is superheated and its density decreases by

27%. As a result, the density drop being more important than the specific cooling capacity increase, the volumetric capacity decreases.

It has to be noticed that the Carnot-type cycle needs to reject less heat than conventional cycles (-6% and -3%). However, the difference is minimal as the rejected heat contribution during the compression (Carnot-type cycle) or the desuperheating (conventional cycles) is quite similar and small as they are equal to 35% and 24% respectively of the latent heat rejected during condensation.
Lastly, the expansion work recovery gain represents less than 10% for both cases; this poor contribution is due to liquid properties at the expander inlet. That is particularly true with Carnot-type cycle as half of the mass flow rate has to be isothermally expanded to get close to COP_{Carnot}.

Despite important ammonia vapour superheating during compression, the Carnot-type cycle, or more generally the isothermal compression, interest seems limited (COP increased by 10%). Calculations were made with propane and R-134a at previous operating conditions: the Carnot-type cycle gain is less than 1%. Hence, the Carnot-type cycle no interest might be generalized for all sub-critical cycles, conventional cycles having already high performances.

5   Trans-critical cycle

The carbon dioxide used as refrigerant in air-conditioning applications is currently under development. Research programs have brought CO_2 mobile air-conditioning systems at a competitive level with hydrofluorocarbons ones in spite of its poor thermodynamic characteristics. Indeed, because of CO_2 low critical point (31°C, 73 bar) the CO_2 cycle is often trans-critical. Here, the T_{HT} is still equal to 40°C and T_{LT} is set at 0°C.

In this section, the Carnot-type cycle is compared to the CO_2 conventional one with recovering of expansion work (Figure 3). As trans-critical cycles are sensitive to the high-pressure, the influence of this parameter is addressed on Figure 4.

To operate properly the Carnot-type cycle, the outlet isentropic expander refrigerant has to be saturated vapour or liquid-vapour mixture. In consequence, the high-pressure has to be higher than 60 bar to match this requirement. The temperature profiles along the IHX were calculated as explained in section 3. They are reported as T_{HP} vs. T_{LP} on Figure 5.

From the calculations, it appears that the high-pressure must be higher than 78.5 bar to operate normally the cycle. At lower high-pressure, an endoreversible compression is used achieving COP_{Carnot}.
The following results are done above 78.5 bar and the deviation with COP_{Carnot} is only due to the IHX irreversibilities. For example, Figure 2 depicts a Carnot-type cycle operating at 100 bar.
For the proposed system (B - Figure 4), the performances are always higher than the conventional design (A - Figure 4) and close to COP_{Carnot}. For conventional trans-critical cycle, an optimal high-pressure exists at approximately 92.5 bar corresponding to a COP equal to 3.83. For the new design cycle, no optimal high-pressure is noticed in the high-pressure studied scale. However, the coefficient of

performance tends asymptotically towards COP_{Carnot} when increasing the high-pressure (98% of COP_{Carnot} at 150 bar).

Table 2 shows some important figures for a high-pressure set at 92.5 bar corresponding to the optimal performance of conventional cycle.

Performances are greatly improved, (70% at 92.5 bar), without changing significantly the refrigerating effect per unit of swept volume. Unlike ammonia, the CO_2 Carnot-type cycle improves the volumetric capacity. Indeed, as trans-critical cycle has an internal heat exchanger, the compressor inlet is fixed; the volumetric capacity depends only on specific cooling capacity and not on suction density. So, CO_2 isothermal compressors, as well as conventional ones, keep the advantage to be smaller than corresponding hydrofluorocarbon compressors.

Unlike sub-critical cycle, expansion energy recovering is relevant to trans-critical cycle. It is particularly true with proposed design, as it represents approximately half net-mechanical work against 25% with the conventional case. Indeed, the mechanical power input is minimal with an isothermal compression and despite the lower mass flow fraction expanded through E1 (Figure 1) work recovered during expansion is similar to the conventional cycle one.

Lastly, the Carnot-type cycle rejects approximately 7% less heat than the conventional cycle.

For a high-pressure set at 92.5 bar, Carnot-type gain is substantial compared with the state of the art conventional cycle. Furthermore, as the high-pressure increases, benefit is growing too.

6   Cryocooler cycle

In this section, heat source temperatures correspond to nitrogen liquefaction applications. T_{HP} is still equal to 40°C while T_{LP} is equal to -195°C corresponding to the atmospheric saturation temperature.

With the Carnot-type cycle, whatever the high-pressure in the studied scale (40 bar to 200 bar), the two-temperatures Carnot-type cycle yields to negative entropy production at the IHX. Then, we have to consider the endoreversible compression by reducing T_1 (-67°C at 40 bar and +13°C at 200 bar). Consequently COP is always equal to COP_{Carnot}.

Claude-type cycle is the reference liquefaction cycle. With ideal expansion processes, only an expander and two IHX are required. One, assumed perfect because of the different mass flows in each side, allows limiting considerably irreversibilities of the second IHX. At 40 bar, which is approximately the practical operating high-pressure, the Claude-type cycle reaches around 96% of COP_{Carnot}.

In conclusion, possible profit does not seem significant (4%). Indeed, cycles are quite similar as based on compression cooling.

7 Impact of design and components efficiency on trans-critical cycle performances

As seen, the profit due to Carnot-type cycle depends on the application. With the sub-critical or the cryocooler cycle, a profit exists, but is small compared with the CO_2 trans-critical cycle.

Hence, we will only focus on CO_2 trans-critical cycles degradations. Three types of degradations are studied:
- cycle design,
- expander efficiency,
- compression cooling efficiency.

7.1 Cycle design

A simpler design (Figure 6), being a trade off between complexity and performances but still based on an isothermal compression, is proposed [10]. It offers high performances (Table 3): COP already reaches 87% of COP_{Carnot} at 92.5 bar (94% at 150 bar) against 95.4% for Carnot-type cycle at the same pressure. Then, the COP is still highly improved (55%) when compared to conventional CO_2 cycle.

7.2 Expander efficiency

In this section, simpler and conventional (Figure 3) designs isentropic expander efficiencies equal to 60% are assumed [6].

We notice, in Table 3, that an optimal high-pressure exists for both cycles: 122 bar with simpler design against 95 bar with conventional one. Therefore, the proposed design COP is more sensitive to the expansion efficiency.

Indeed, the simpler cycle HP is higher (122 bar) than that of the conventional design (95 bar). Then the expansion leads to higher work recovered and liquid quality at evaporator inlet, +45% and +17% respectively. Furthermore with expander efficiency diminution, the optimal HP decreases a lot with simpler design and increases in little proportion with conventional design, which impacts work-recovering expansion and cooling capacity.

Both COP decrease, by 14% and 10% respectively, when compared with corresponding ideal processes. However, the simpler cycle still keeps a large profit compared with usual cycle one (+59% of COP).

7.3 Compression cooling efficiency

Lastly, we have to evaluate the non-isothermal compression impact on performances. An endoreversible compression is considered (Figure 7) as explained in section 3 but two aspects differ. Despite cooling compression, the compressor discharge temperature (T_{Dis}) is higher than T_{HT}. Then a small gas-cooler is used to cool outlet compressor vapours from T_{Dis} to T_{HT}. The cooled compression is still assumed reversible, but it exchanges heat at T_{Dis} with a heat source at T_{HT}.

Results for two discharge temperatures are given in Table 3.

Whatever the outlet compressor temperatures pinch, it exists an optimal high-pressure on the studied scale, which decreases as the outlet temperature increase: with T_{Dis} set at 50°C, the optimal high-pressure is about 112 bar against 100 bar at 60°C. It corresponds

to COP equals to 4.59 and 4.15 respectively. Therefore, compared to the isothermal compression (COP=6.32), COP decreases quickly with the temperature pinch increase. Although COP is sensitive to T_{Dis}, it is still greatly higher than that of conventional cycle (+21% at 60°C and 34% at 50°C), provided that T_{Dis} remains at acceptable values (<60°C in this case).

It has be noticed that for given expander efficiency, the volumetric capacity, the ratio $\overset{\&}{W}_E / \overset{\&}{W}_{NET}$ and $|\overset{\&}{Q}_{Out} / \overset{\&}{Q}_{EV}|$ are of the same order of magnitude for the three cycles (Table 3). The differences are due to the difference in high-pressure. A higher HP implies better cooling capacity because of increased liquid quality at evaporator inlet. Similarly, an increase in HP results in an increase of the recovered work at the expanders. However, $\overset{\&}{W}_E / \overset{\&}{W}_{NET}$ depends on $\overset{\&}{W}_{CP}$ too. Hence, comparing the endoreversible and the simpler design, despites decrease by 6% of $\overset{\&}{W}_E$ as the HP drops from 122 bar to 112 bar, $\overset{\&}{W}_E / \overset{\&}{W}_{NET}$ decreases by 19% as in the same time $\overset{\&}{W}_{CP}$ is increased by 12%.

8   Conclusions

From a thermodynamical point of view, the proposed Carnot-type cycle allows to reach close COP_{Carnot}. Thus, it is possible to improve each conventional sub-critical and trans-critical cycles. However, the profit is not always so relevant according to the configuration.

With conventional sub-critical cycles, the condensation permits already to have high performances. Consequently, the Carnot-type cycle benefit gets only to 12% compared with the best conventional one. With cryocooling applications, as both cycle are based on cooling compression, the profit is not more substantial.
Concerning the CO_2 trans-critical Carnot-type cycle, the expected gain is about 70% and still 65% with a simpler cycle. So that isothermal compression is relevant with CO_2 cycles and it confirms that they need to have a cryogenics approach [11].
If expansion efficiency is taken into account (60%), the profit is still about 60% despite a more important sensitivity to this parameter, compared with the conventional cycle. Finally, the compressor heat extraction has to be efficient. For a 20K at compressor discharge, it is suitable to put a gas-cooler, then COP decreases by 24% compared to the isothermal case but is still improved by 21% compared to the conventional cycle COP or 34% if pinch is reduced to 10K.

These encouraging results prove that the isothermal compression is relevant for CO_2 trans-critical cycle, according to an efficient cooling during compression. Furthermore, compression enhancement is the main way to improve significantly the conventional COP cycle, as an increase in expander efficiency will have only a small impact on the performances.

Further work will consist in investigating possible ways to get an efficient compression cooling. Consequently, we have to explore different compression designs taking into account heat transfer phenomena.

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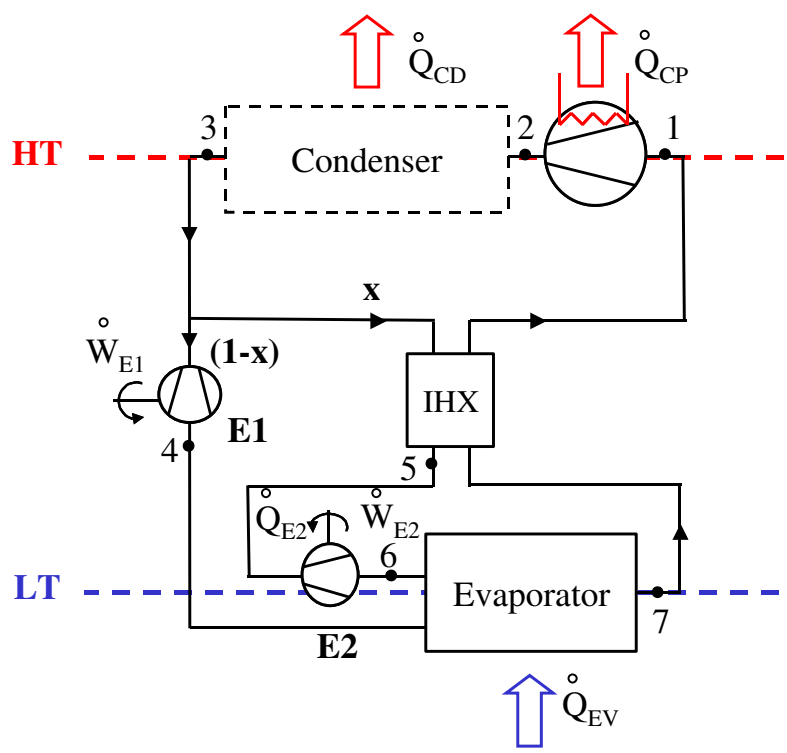[11] Vaisman IB.COP Analysis of carbon dioxide cycles. ASHRAE Transactions 2002, 108; 252-262; USA.
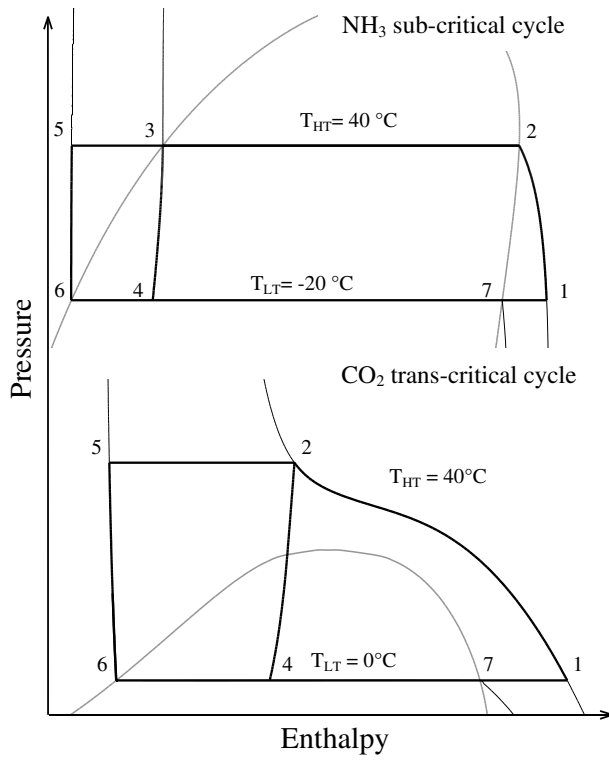
Figure 1: Ideal Carnot-type cycle.

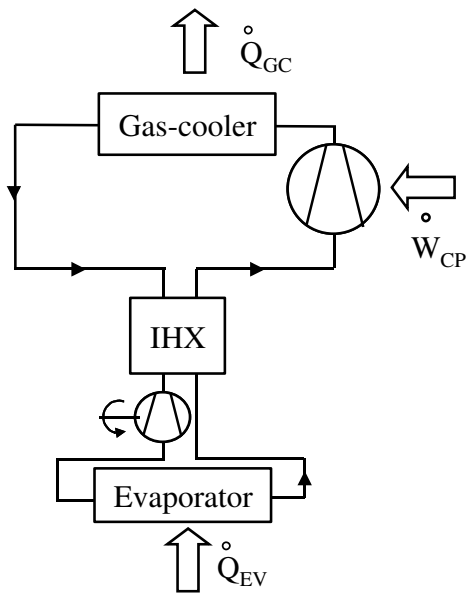Figure 2: Carnot-type cycle in (h, P) NH_3 and CO_2 diagrams.

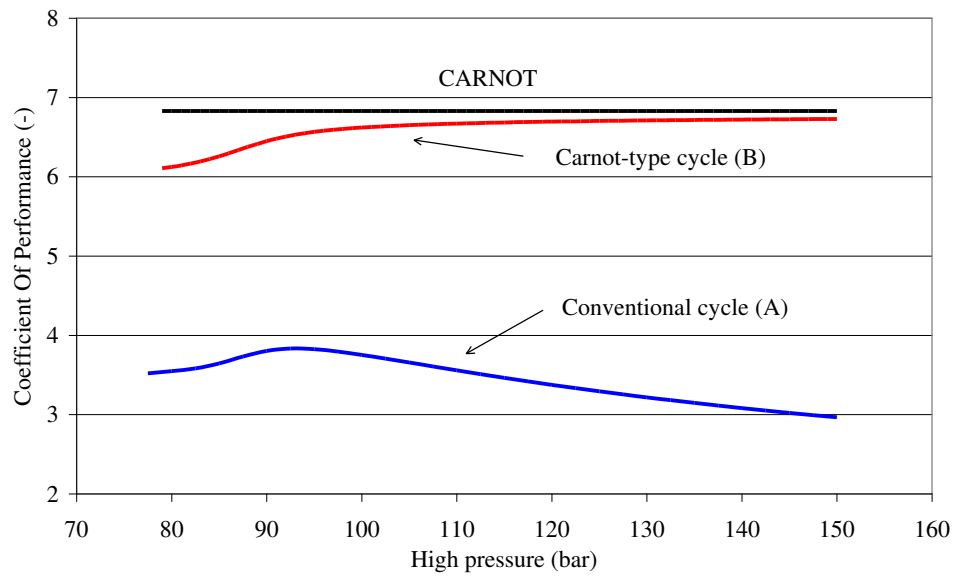Figure 3: State of the art CO_2 trans-critical cycle.

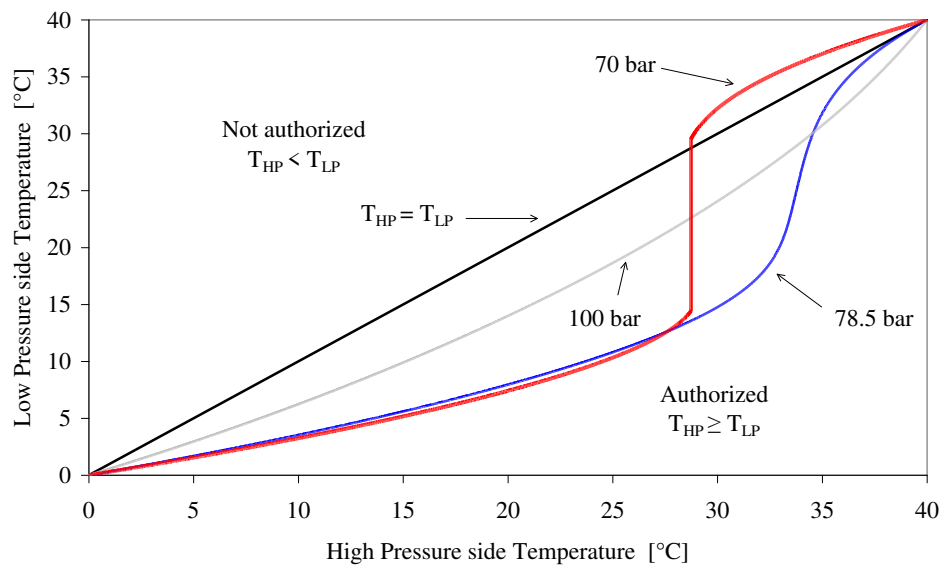Figure 4: Evolution of COP vs. High-Pressure for both CO_2 cycles.

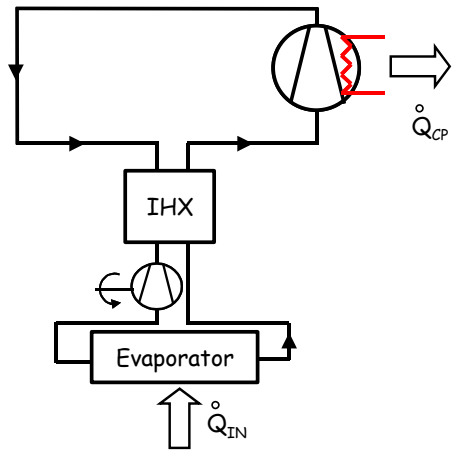Figure 5: Both discretized IHX temperature evolutions.

Figure 6: Simpler trans-critical cycle.

Figure 7: Endoreversible cooled compression design.

Figure captions

Figure 1: Ideal Carnot-type cycle.

Figure 2: Carnot-type cycle in (h, P) NH_3 and CO_2 diagrams.

Figure 3: State of the art CO_2 trans-critical cycle.

Figure 4: Evolution of COP vs. High-Pressure for both CO_2 cycles.

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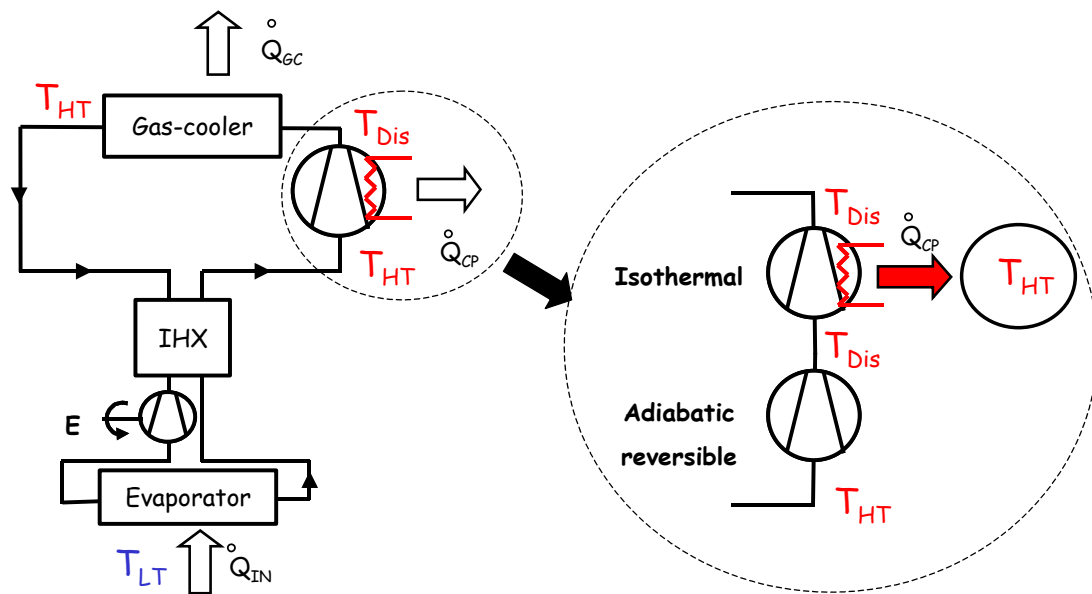
Figure 6: Simpler trans-critical cycle.

Figure 7: Endoreversible cooled compression design.

Tables

Table 1: Comparison of Carnot-type and conventional NH$_3$ cycles.

Cycle		Carnot-type cycle	Conventional NH$_3$ cycles	
			Throttling device (A)	Expander (B)
COP$_{\text{Carnot}}$	$(-)$		4.22	
COP / COP$_{\text{Carnot}}$	$(\%)$	99.8	77.7	88.7
Volumetric capacity	$(kJ{\cdot}m^{-3}_{Suc.})$	1518	1679	1728
$\left\lvert \dot{W}_{E1} + \dot{W}_{E2} \right\rvert \Big/ \dot{W}_{NET}$	$(\%)$	6	-	10
$\left\lvert \left(\dot{Q}_{CP} + \dot{Q}_{CD} \right) \Big/ \dot{Q}_{EV} \right\rvert$	$(\%)$	124	130	127

Table 2: Comparison of Carnot-type and conventional CO_2 cycles.

Cycle		Carnot-type cycle	Conventional CO_2 cycle
COP_{Carnot}	$(-)$	6.83	
HP	(bar)	92.5	
COP / COP_{Carnot}	$(\%)$	95.4	56.1
Volumetric capacity	$(kJ \cdot m_{Suc.}^{-3})$	11769	11633
$\left\| \dot{W}_{E1} + \dot{W}_{E2} \right\| / \dot{W}_{NET}$	$(\%)$	49	25
$\left\| \dot{Q}_{OUT} / \dot{Q}_{EV} \right\|$	$(\%)$	115	126

Table 3: CO_2 cycles degradation.

Cycle	Conventional cycle	Simpler cycle		Endoreversible cycle	
	η_{EXP} 60%	η_{EXP} 100%	η_{EXP} 60%	η_{EXP} 60% ΔT_{Dis} 10K	ΔT_{Dis} 20K
COP_{Carnot} $(-)$			6.83		
HP (bar)	95	122	122	112	100
COP/COP_{Carnot} $(\%)$	50.2	92.5	79.8	67.2	60.7
Volumetric capacity $(kJ \cdot m^{-3}_{Suc.})$	11910	14242	13940	13533	12628

$\left\|\dot{W}_E\right\|\Big/\dot{W}_{NET}$ (%)	12.2	33.4	17.7	14.4	13.5
$\left\|\dot{Q}_{OUT}\Big/\dot{Q}_{EV}\right\|$ (%)	130	116	118	122	124