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CASCADE REFRIGERATION SYSTEM WITH INVERSE BRAYTON CYCLE ON THE COLD SIDE

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Keywords: Cold-store refrigeration; Cascade system; Inverse Brayton cycle; Performance analysis

ABSTRACT

Low temperature refrigeration of cold stores poses some specific issues: single stage, vapour compression cycles have modest COP at low evaporation temperature; cold evaporator surfaces require de-frosting and a fan for air circulation; a part of the refrigeration load may be delivered at intermediate temperature levels, e.g. for the cold store loading dock.

Cascade system may improve the COP and add flexibility on the temperature levels and working fluids, but the problems related to the cold evaporator surface remain unsolved.

The refrigeration system presented herein features a cascade configuration combining a vapour compression cycle and an inverse Brayton cycle. Both cycles use “natural” fluids, complying with strictest regulations. The top cycle uses Ammonia in order to increase efficiency, while the bottom cycle uses air, which directly circulates in the cold space and hence eliminates the cold heat exchanger. A detailed thermodynamic analysis allows a complete screening of the relevant design parameters for an overall system optimization.

The results show that, notwithstanding the intrinsic gap of efficiency suffered by the Brayton cycle, the proposed system features an acceptable global performance and widens the implementation field of this technology. This system configuration shows a COP 50% higher than the corresponding simple Brayton cycle at temperatures of the refrigerated storage of -50°C.

28 **Nomenclature**

29	A	Cross-section area [m^2]
30	c_p	Isobaric specific heat [$\text{kJ}\cdot\text{kg}^{-1}\text{K}^{-1}$]
31	c_v	Isochoric specific heat [$\text{kJ}\cdot\text{kg}^{-1}\text{K}^{-1}$]
32	COP	Coefficient of performance
33	G	Mass velocity [$\text{kg}\cdot\text{m}^{-2}\text{s}^{-1}$]
34	h	Specific enthalpy [$\text{kJ}\cdot\text{kg}^{-1}$]
35	J	Humid air specific enthalpy [$\text{kJ}\cdot\text{kg}^{-1}$]
36	k	Heat capacity ratio
37	L	Axial length [m]
38	\dot{m}	Mass flow rate [$\text{kg}\cdot\text{s}^{-1}$]
39	P	Absolute pressure [kPa]
40	Q_f	Cooling load [kW]
41	r	Latent heat [$\text{kJ}\cdot\text{kg}^{-1}$]
42	s	Specific entropy [$\text{kJ}\cdot\text{kg}^{-1}\text{K}^{-1}$]
43	S	Transfer surface [m^2]
44	T	Temperature [K]
45	W	Power [kW]
46	x	absolute humidity
47	z	Compressibility factor
48		

49 **Greek symbols**

50	α	Transfer area per unit volume [m^{-1}]
51	β	Heat transfer coefficient [$\text{W}\cdot\text{m}^{-2}\text{K}^{-1}$]
52	ε	Thermal effectiveness
53	η	Efficiency

54 π Compression ratio

55

56 **Subscripts**

57	air	Air
58	amb	Ambient
59	aux	Auxiliary fan
60	B	Inverse Brayton
61	C	Compressor
62	$cond$	Condenser
63	$corr$	Accounting for fan and defrost loads
64	el	Electrical
65	eva	Evaporator/Cascade heat exchanger
66	ice	Ice
67	is	Iso-entropic
68	LD	Loading dock
69	max	Maximum
70	$mech$	Mechanical
71	min	Minimum
72	R	Regenerative heat exchanger
73	RS	Cold storage
74	sat	Saturation
75	sh	Super-heat
76	T	Expander
77	v	Water vapour
78	VC	Vapour compression
79		
80		

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1. INTRODUCTION

1.1 Cascade systems

The low storing temperature required by many perishable goods may be efficiently guaranteed by a staged refrigeration system, featuring one or more intermediate temperature levels. This reduces the irreversibilities of the thermodynamic cycle, both on the compression and on the expansion side. Cascade refrigeration systems, when compared to other staged configurations (e.g. double stage with economizer), offer an additional degree of freedom for their optimization, i.e. the choice of a suitable combination of refrigerant fluids, and may offer enhanced performance [1]. Ideally, the refrigerants should be environmentally friendly, non-flammable, non-toxic, and yield the lowest possible initial investment and operative cost.

Several thermodynamic simulations for low temperature refrigeration systems have been reported in the literature. A summary is presented in Table 1. Apparently, NH_3 is the best choice as a fluid for the high temperature cycle (apart from ethanol). CO_2 is often proposed for the low temperature cycle, as it is environmentally safe, non-flammable and non-toxic. However, it has a worsening effect on COP, as shown e.g. by comparing the result of Kilicarslan and Hosoz [2], who used R23, with Lee et al. [3], who used CO_2 .

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Table 1 – Literature results of simulations for cascade refrigeration systems

Reference	HT Fluid	LT Fluid	T_{cond}	T_{eva}	COP
Aminyavari et al. 2014 [5]	NH_3	CO_2	40	-49	1.5
Nasruddin et al. 2016 [6]	C_3H_8	$\text{C}_2\text{H}_6 + \text{CO}_2$	56	-49	0.79
Di Nicola et al. 2011 [4]	NH_3	C_2H_6	40	-70	0.97
		C_3H_8			1.01
		$\text{C}_2\text{H}_6 + \text{CO}_2$			0.93
		$\text{C}_3\text{H}_8 + \text{CO}_2$			0.87
Gettu, Bansal 2008 [7]	NH_3	CO_2	40	-50	1.43
	Ethanol				1.5
	R404				1.35
Lee et al. 2006 [3]	NH_3	CO_2	30	-55 ÷ -45	1.1 ÷ 1.44
			35		1.01 ÷ 1.31
			40		0.92 ÷ 1.2
Kilicarslan, Hosoz 2010 [2]	NH_3	R23	40	-65 ÷ -45	0.89 ÷ 1.25
	R134				0.77 ÷ 1.08
	R404				0.64 ÷ 0.8
Mosaffa et al. 2016 [8]	NH_3	CO_2	35	-45 ÷ -35	1.17 ÷ 1.37

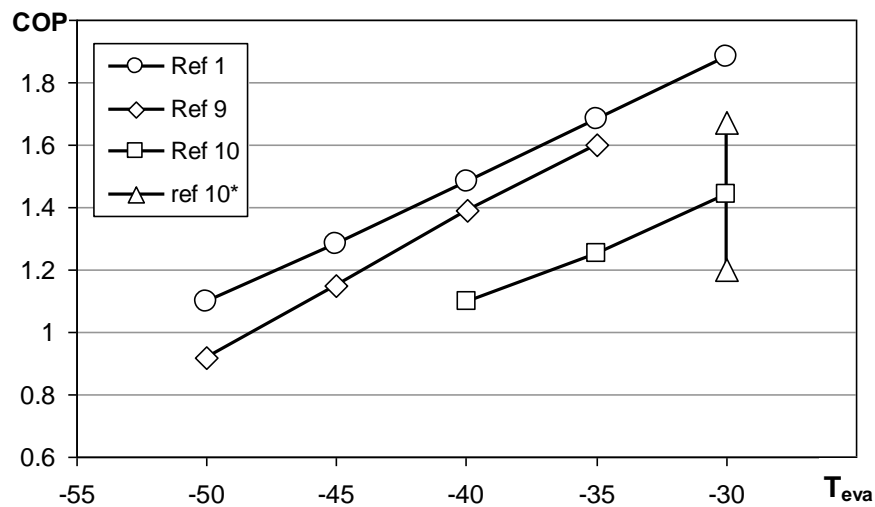
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99

100 The same result was shown by Di Nicola et al. [4] who compared pure hydrocarbons with mixtures of
 101 hydrocarbons and CO₂. Furthermore, CO₂ is unusable at very low temperatures, triple point being at -56.6°C,
 102 and requires high operating pressures, increasing the cost of the refrigeration system.

103 Experimental data on NH₃/CO₂ cascade refrigeration systems have been collected e.g. by Bingmin et
 104 al. [1] and by Dopazo and Fernandez-Seara [9]. A comparison between these data and other experimental
 105 results on a R134a/CO₂ cascade system (Sanz-Kock et al. [10]) is shown in Fig.1. Once again, NH₃ seems to
 106 be a preferable option for the high temperature cycle. Ammonia does pose safety problems, but the
 107 refrigeration industry has been using it from the very beginning and has acquired the due experience for
 108 managing any possible risk. Ammonia is commonly included among the “natural” refrigerants and is
 109 receiving an increasing attention from many global players in the refrigeration area [11]. Furthermore, in a
 110 cascade system ammonia may be confined to a restricted part of the plant.

111



112

113 Fig. 1 – Experimental data from the literature on cascade refrigeration systems

114 Ref. [1]: Bingmin et al.; NH₃/CO₂; T_{cond} = 40°C; Ref. [9]: Dopazo, Fernandez-Seara; NH₃/CO₂; T_{cond} = 30°C;

115 Ref. [10]: Sanz-Kock et al.; R134a/CO₂; T_{cond} = 40°C; Ref. [10]*: same with T_{cond} = 30-50°C

116

117 1.2 Air cycle

118 A further option for the low-temperature working fluid is air, which may be used in an inverse Brayton
 119 cycle. This cycle has been extensively analysed in various configurations and under different evaluation

approaches. Zhang et al. [12] developed an irreversible model and compared the optimal performance of a regenerative cycle with that of a simple Brayton cycle. Chen et al. [13] derived a general expression for the exergetic efficiency of a regenerated air refrigerator, whereas Ust [14] based his cycle optimization on an ecological coefficient of performance (ECOP). Obvious advantages of air as a refrigerant are absolute environmental and operational safety. Air is available everywhere at no cost, which has prompted its use as working fluid in many devices from the early stages of civilization, immediately after water [15]. Not surprisingly, among the first refrigeration systems we find the air machine patented by John Gorrie that dates back to the first half of the 19th century and had a good success until the introduction of synthetic refrigerants [16]. Perishable goods are normally stored in air, so that this same air can be used as working fluid by opening the cycle on the cold side (**high pressure cycle** – Fig. 2a), whence the low temperature heat exchanger is eliminated. Alternatively, we may use the ambient air as working fluid, eliminating the high temperature heat exchanger (**low pressure cycle** – Fig. 2b).

Grazzini and Milazzo [17] have shown that, in both cases, an open cycle yields a significant increase in efficiency, but the low pressure configuration of Fig.2b is somewhat better. On the other hand, the high pressure scheme of Fig. 2a has the major advantage of eliminating the cold heat exchanger which, when operated below 0°C, is prone to frost accumulation.

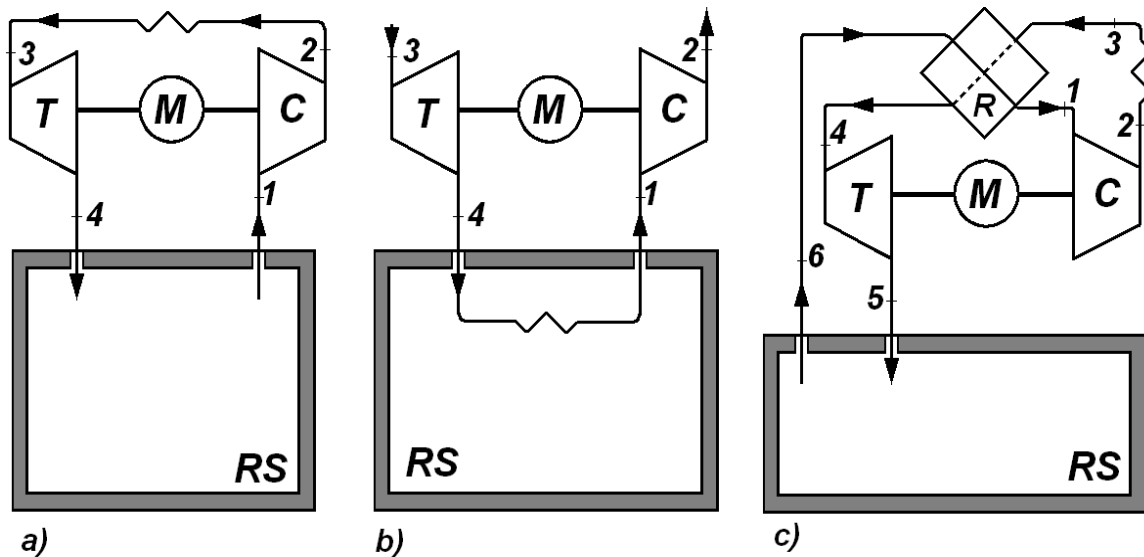


Figure 2. Inverse Brayton cycles; a) high pressure; b) low pressure; c) regenerated

M: electric motor; C: compressor; T: turbine; R: regenerator, RS: refrigerated space.

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141 The inevitable ingestion of the air humidity causes water condensation and eventually icing in the cold
142 sections of the system, but the ice particles are formed within the air stream and, as far as they are small, they
143 move with the air flow, instead of sticking on a cold surface which must be periodically defrosted. The ice
144 particles can be captured, allowing some humidity control.

145 As far as the residual air velocity at turbine exit is sufficient, even the electric fan that normally
146 circulates the cold air within the cell (and adds a further electric and thermal load) may be eliminated.

147 The inclusion of a regenerative heat exchanger (Fig. 2c), increases efficiency and reduces the
148 performance gap between the high and low pressure configuration as shown by Giannetti and Milazzo [18].
149 This component plays a crucial role in the thermodynamic optimization of the whole system, as will be
150 shown later.

151 A cascade configuration employing an inverse Brayton as the bottom cycle was suggested by Nobrega
152 and Sphaier [19] or Elsayed et al. [20], who used a desiccant top cycle. However, to the authors' knowledge,
153 a cascade of a standard vapour compression cycle and an open inverse Brayton was not discussed in previous
154 literature, and the integration of these two well-established technologies could be useful to overcome some
155 of their downsides for a cold store application. The present proposal may be near to market application and
156 worth of a detailed analysis in terms of design parameters and expected performance. Additionally, the effect
157 of the phase change of the humidity carried from the air stream circulating within the Brayton cycle is
158 accounted for in the present modelling effort. The air expansion is reconstructed as a series of equilibrium
159 states to accurately predict the conditions of the air at the outlet of the expander, which can considerably
160 deviate from the results obtained for an ideal-gas behaviour. An accurate prediction of the outlet stream
161 properties in the cold store is critical for an accurate design and optimization of the specific plant. Moreover,
162 ice separation from the expanding air stream represents a promising alternative to the traditional defrosting
163 process of aero-evaporators used in cold stores and an evaluation of its potential may be useful.

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2. SYSTEM ANALYSIS

166 From a practical point of view, an air cycle refrigeration system may be fairly simple, with compressor and
167 turbine on a single shaft, driven by a high speed electric motor. Magnetic bearings reduce friction and allow

168 a completely oil-free operation. These concepts are well proven by state-of-the-art centrifugal compressors
169 which are gaining an increasing market share in many refrigeration applications. The avoidance of a closed
170 cycle (which could present fluid leakage) increases reliability. Highly efficient compressor, turbine and heat
171 exchangers are required in order to achieve acceptable COP. The residual efficiency gap with respect to
172 vapour compression plants must be evaluated on a system basis and over a seasonal operating period that
173 includes defrosting cycles.

174 Regeneration, i.e. internal heat exchange between the warm air on the high-pressure side of the cycle
175 and the cold air on the low-pressure side (Fig 2c), allows to overcome the constraint that links the
176 compression ratio of the simple cycle to the temperature range to be covered.

177 The approach featuring a high pressure, regenerated cycle with single-stage compression is confirmed
178 by a refrigeration system named “Pascal Air”, manufactured in Japan by Mayekawa [21] and a similar
179 system developed by Mitsubishi [22]. Another system, the “AIRS50”, featuring a staged compression [23]
180 was developed by Kajima Inc. and later by Earthship Ltd. and reached a pre-commercial stage of
181 development, but is no longer available.

182 In the case of a food storage facility, a loading/shipping dock is common everywhere and absolutely
183 necessary in humid and warm climates. The benefits of a refrigerated loading dock are [24]:

- 184 - The refrigeration load in the low-temperature storage (where the energy demand per unit capacity is
185 higher) is reduced;
- 186 - A lower amount of humid air is infiltrated in the low-temperature area, reducing ice formation;
- 187 - Refrigerated products held on the dock for loading/shipping maintain their quality;
- 188 - Products packaging, equipment and floor areas stay drier, increasing goods quality, system reliability
189 and operators’ safety.

190 The loading dock (*LD*) is normally kept around 5°C. The combined need for a low temperature
191 refrigerated space and intermediate temperature loading dock may be favourably satisfied by a cascade
192 refrigeration concept that has an intermediate heat exchanger between the two cycles.

193 A hybrid vapour compression / inverse Brayton cascade system may be envisaged as follows (Fig. 3):
194 the top cycle 7-8-9-10 could be any high temperature vapour compression system, featuring a compressor *C*,
195 a condenser *K*, an expansion valve *EV* and two evaporators *E*₁ and *E*₂. The first evaporator refrigerates the

196 loading dock *LD*, while the second couples the vapour compression and the inverse Brayton cycle, cooling
 197 down the hot compressed air between points 2 and 3. The regenerator further cools the air flow between
 198 points 3 and 4. The expansion brings the air back to ambient pressure and produces the minimum system
 199 temperature at point 5. The exhaust air 6 from the refrigerated space *RS* is warmed up in the regenerator
 200 before entering the compressor in 1. Auxiliary components (e.g. liquid receiver at condenser exit) and other
 201 practical issues (load control, etc.) are not included in this analysis, which is mainly devoted to
 202 thermodynamics for a preliminary system design and screening of its potential.

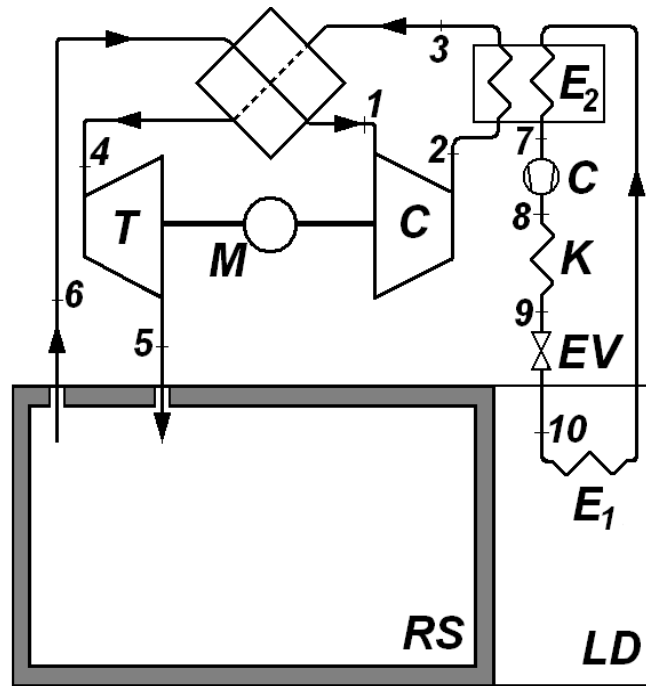


Figure 3. Scheme of the proposed plant

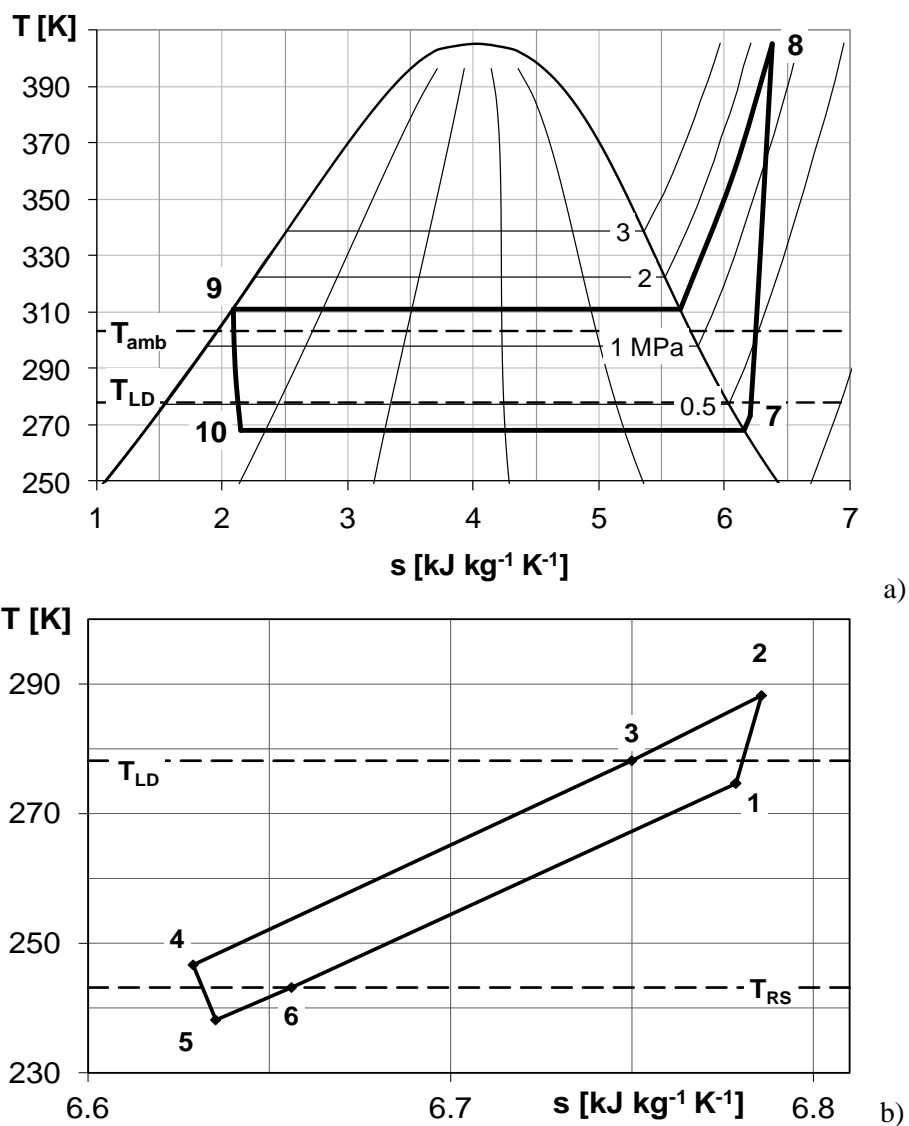
2.1 Thermodynamic modelling

The system efficiency may be expressed by a coefficient of performance defined as:

$$COP = \frac{Q_{f-RS} + Q_{f-LD}}{W_B + W_{VC} + W_{aux}} \quad (1)$$

where Q_f = cooling load [kW], W_B = input power for Brayton cycle [kW], W_{VC} = input power for vapour compression cycle, W_{aux} = auxiliary power input (condenser fan, etc.).

212 The corresponding thermodynamic cycles are shown on a Ts diagram in Fig. 4 for the preliminary
 213 design values of the operative parameters listed in Table 2.
 214



216
 217 Figure 4. Temperature-entropy diagrams of the top (a) and bottom (b) cycles
 218

Table 2 – Reference system parameters.

Refrigerated store temperature (°C)	T_{RS}	-30
Loading dock temperature (°C)	T_{LD}	5
Ambient temperature	T_{amb}	30
Minimum temperature difference at heat exchangers (°C)	$\Delta T_{eva,min}$	10
Top cycle		
Fluid		NH ₃
Evaporation temperature (°C)	T_{10}	-5
Condensation temperature (°C)	T_9	40
Superheating at evaporator (°C)	ΔT_{sh}	5
Subcooling at condenser outlet (°C)	ΔT_{sc}	0
Cooling capacity (kW)	Q_{f-VC}	103
Input power (kW)	W_{VC}	26.5
Fluid temperature at compressor outlet (°C)	T_8	132
Air cycle		
Temperature difference between inlet cold air and store (°C)	ΔT_{5-6}	10
Specific heat ratio (air)	k	1.4
Regenerative heat exchanger effectiveness	ε_R	0.9
Compression efficiency	η_C	0.85
Expansion efficiency	η_T	0.85
Combined mechanical and electrical efficiencies	$\eta_{mech}\eta_{el}$	0.9

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The high temperature cycle is calculated using the selection software available at the compressor manufacturer website (<https://www.bitzer.de/websoftware/Calculate.aspx> [25]). NH₃ is assumed as the working fluid. The compressor performance and the working cycle are specified according to EN 12900. The selected compressor is an open, reciprocating unit with 6 cylinders, operating at 50 Hz (Bitzer W6FA). The performance of the top cycle is summarized in Fig. 5 for two evaporation temperatures. Note that this temperature is related to the loading dock temperature and hence is not used as a design parameter in this analysis. The conservative value $T_{eva} = -5^\circ\text{C}$ (i.e. 10°C temperature difference between evaporation and loading dock temperature) is assumed hereafter. The Ts diagram in Fig. 4a is calculated by NIST REFPROP [26] functions.

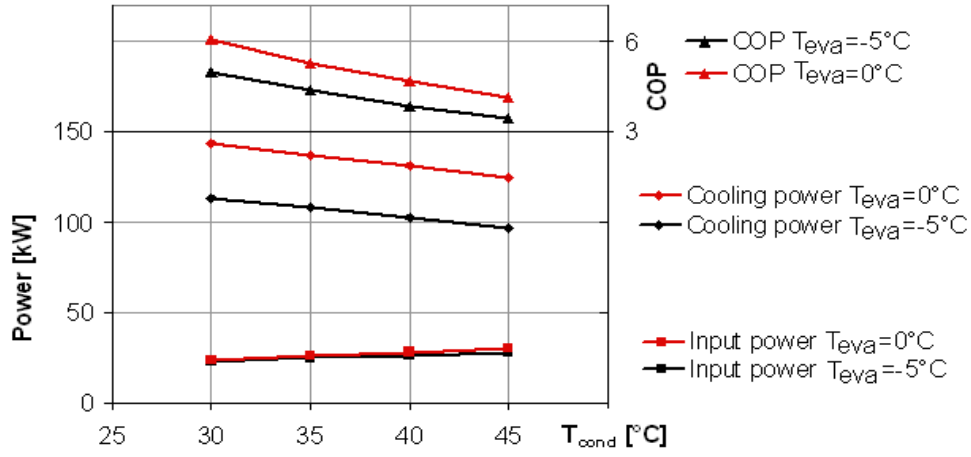


Fig. 5. High temperature cycle performance

For the bottom cycle (Fig. 4b), air is modelled as an ideal gas, compressibility factor $z = Pv / RT$ being unitary within 2‰ in the relevant pressure and temperature range ($230 < T < 310$ K; $0.1 < P < 0.2$ Mpa). Air specific heat ratio $k = c_p / c_v$ is assumed constant in the same range, its variations being of the same order. Air enthalpy and entropy are calculated by NIST REFPROP [26] functions.

Air humidity is taken into account, assuming saturation within the refrigerated space. Unfortunately, NIST [26] functions for water are unavailable below 0°C. Therefore, water vapour properties are taken from ASHRAE Fundamentals Handbook [27]. Unless an extremely efficient regenerator is available, absolute humidity remains constant through the cycle, until phase change occurs in the turbine. Ice particles may be captured by suitable filters (NEDO documents [28]) placed at inlet and outlet air ports within the refrigerated space, in order to perform active de-humidification (air introduced in the refrigerated space at point 5 has lower absolute humidity than that drawn at point 6).

Losses within compressors and turbine are accounted for by isentropic efficiencies:

$$\eta_C = \frac{W_{C-is}}{W_C}; \quad \eta_T = \frac{W_T}{W_{T-is}} \quad (2)$$

When the compressor and expander are assembled on a single shaft, the electric motor of the Brayton cycle consumes a power

$$W_B = \frac{W_C - W_T}{\eta_{mech}\eta_{el}} \quad (3)$$

251 The regenerator effectiveness ε_R , neglecting the specific heat variation, is given by:

$$252 \quad \varepsilon_R = \frac{T_3 - T_4}{T_3 - T_6} = \frac{T_1 - T_6}{T_3 - T_6} \quad (4)$$

253 Cycle point numbering is specified in Fig. 4.

254 The regenerator inlet temperatures are $T_6 = T_{RS}$ and T_3 , which is fixed by setting a minimum
 255 temperature difference between air flow and evaporating fluid temperature T_{10} . The inverse Brayton cycle is
 256 hence calculated as follows (cycle points are numbered as in Fig. 2):

$$257 \quad T_5 = T_{RS} - \Delta T_{5-6}; \quad T_3 = T_{10} + \Delta T_{\min}; \quad T_4 = T_3 - \varepsilon_R(T_3 - T_6); \quad T_1 = T_6 + \varepsilon_R(T_3 - T_6) \quad (5)$$

258 Temperature T_2 is a function of the compression ratio $\pi = P_2 / P_1$:

$$259 \quad T_2 = T_1 \left[1 + \frac{1}{\eta_c} \left(\pi^{\frac{k-1}{k}} - 1 \right) \right] \quad (6)$$

260 The pressure ratio is obtained from a simulation of the expansion 4-5 accounting for ice formation.
 261 The transformation is reconstructed as a series of equilibrium states, iteratively calculating the inlet condition
 262 that gives the required outlet temperature after a “humid expansion” process, which accounts for the possible
 263 phase-change of the vapour. Pressure losses in the heat exchangers being neglected, compression ratio and
 264 expansion ratio are equal. The adopted method for the calculation of π accounts for the deviation of the real
 265 expansion process from the ideal adiabatic-isentropic transformation of the humid air by means of the
 266 expansion efficiency defined in eq. (7).

$$267 \quad \eta_T = \frac{J_4 - J_5}{J_4 - J_{5is}} \quad (7)$$

268 where point 5_{is} is defined with reference to the thermodynamic state at turbine inlet ($s_{5is}=s_4$). Enthalpy
 269 and entropy of the humid air are calculated as follows:

$$270 \quad J = h_{air} + x_4 h_v \quad (8)$$

$$271 \quad s = s_{air} + x_4 s_v \quad (9)$$

272 where x is the absolute humidity of the working air stream. Water enthalpy and entropy are evaluated
 273 assuming ideal gas behaviour until saturation, whereas after saturation they are given, respectively, by eq.s
 274 10 and 11.

$$275 \quad h_v = h_{v,sat} - \left(1 - \frac{x_5}{x_4}\right) r \quad (10)$$

$$276 \quad s_v = s_{v,sat} - \left(1 - \frac{x_5}{x_4}\right) \frac{r}{T} \quad (11)$$

277 r being the latent heat of ice formation. When phase change occurs, the total flowrate includes the
 278 three contributions of ice, vapour and dry air flowrates. If the absolute humidity decreases with respect to the
 279 preceding step, the difference is taken as the quantity of ice produced. The ice formation is considered as
 280 instantaneously defined by eq. 12; where the vapour and the ice flowrates change with the local temperature,
 281 whereas the dry air's flowrate stays constant.

$$282 \quad \dot{m}_{ice} = (x_4 - x_{sat}) \dot{m}_{air} \quad (12)$$

283 The corresponding latent heat release is added to the air enthalpy (eq. 13);

$$284 \quad J = \frac{\dot{m}_{air} h_{air} + \dot{m}_v h_{v,sat} + \dot{m}_{ice} r}{\dot{m}_{air}} \quad (13)$$

285 and a corrected temperature is calculated via NIST functions. Pressure P_4 is calculated by iteratively
 286 increasing its value, from an initial one that gives the due state (state 5) at the end of an isentropic expansion,
 287 until the eq. 7 is verified with a preset tolerance.

288 Once the compression ratio is known, all properties can be calculated throughout the cycle. The
 289 amount of ice collected is define by eq. 12 and the cooling capacity is

$$290 \quad Q_{f-RS} = \dot{m}_{air} (J_6 - J_5) \quad (14)$$

291

292 2.2 Heat Transfer surfaces

293 The cascade system presented eliminates the low temperature heat transfer surface, but features an air-
 294 refrigerant heat exchanger between the top and bottom cycles, and introduces an air-air regenerator, which
 295 might suffer from high pressure drops. A good design requires a compromise between volume of the heat

exchanger and related pressure drops. Considering the definition from Kays and London [29], the evaporator has effectiveness,

$$\varepsilon_{eva} = \frac{Q_{eva}}{Q_{max}} = \frac{Q_{eva}}{C_{min}(T_2 - T_{10})} = \left(1 + \frac{\Delta T_{eva,min}}{T_2 - T_{LD}}\right)^{-1} \quad (15)$$

where, due to the refrigerant phase-change occurring on the vapour compression cycle side, the minimum thermal capacity C_{min} pertains to the air flux circulating in the inverse Brayton cycle. Referring to eq.s 4-6,

$$\varepsilon_{eva} = \left(1 + \frac{\Delta T_{eva}}{\left[T_{RS} + \varepsilon_R(T_{LD} - T_{RS})\right] \left\{1 + \frac{1}{\eta_C} \left[\frac{(1 - \varepsilon_R)(T_{LD} - T_{RS}) + \Delta T_{5-6}}{(T_{RS} - \Delta T_{5-6})\eta_T}\right]\right\} - T_{LD}}}\right)^{-1} \quad (16)$$

Equation (16) highlights the dependence of the effectiveness of the cascade heat exchanger (evaporator of the top cycle) on the effectiveness of the regenerative heat exchanger of the Brayton cycle. Introducing the general expression of the effectiveness of a counter-current flow heat exchanger as a function of the number of transfer unit $NTU = \beta S / C_{min}$,

$$\varepsilon = \frac{1 - e^{-NTU(1 - C_{min}/C_{max})}}{1 - \frac{C_{min}}{C_{max}} e^{-NTU(1 - C_{min}/C_{max})}} \quad (17)$$

the total transfer surface required by the low temperature cycle may be written as a function of the regenerator effectiveness (Fig. 7). Specifically, C_{min} / C_{max} being 0 for the cascade Heat exchanger and 1 for the regenerative heat exchanger, equation (17) yields:

$$\varepsilon_R = \frac{NTU}{1 + NTU} \quad , \quad \varepsilon_{eva} = 1 - e^{-NTU} \quad (18)$$

A higher regenerator effectiveness ε_R reduces the load at the cascade heat exchangers $\dot{m}_{air} c_p (T_2 - T_3)$ for the same air flowrate and, consequently, the size of the heat exchanger (Fig. 7) required to achieve a target minimum temperature difference $\Delta T_{eva,min}$.

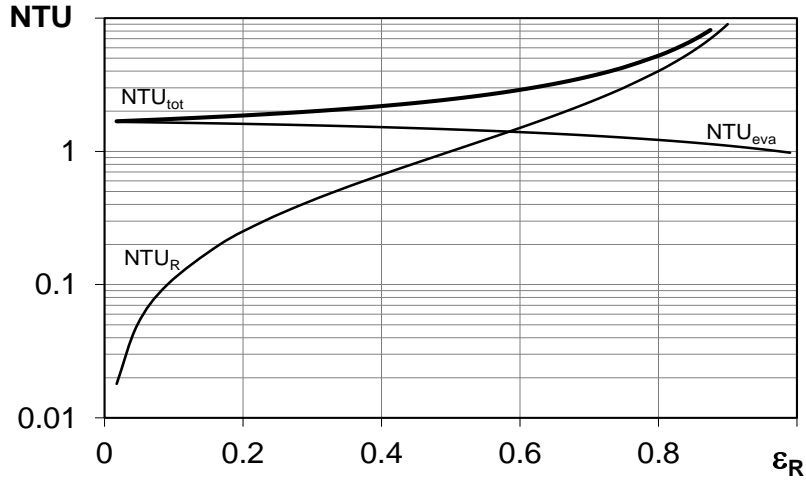


Figure 7. $NTU-\varepsilon_R$ diagram of the bottom cycle

A higher regenerator effectiveness reduces also the compression ratio required for a given operative temperature difference $T_{LD}-T_{RS}$, therefore increasing cycle efficiency.

In the regenerator, the hot and cold flows have practically equal thermal capacities and the driving temperature difference is approximately constant along the axial length of the heat exchanger L (eq. 19).

$$Q_R = \dot{m}_{air} c_p (T_{LD} - T_4) = \dot{m}_{air} c_p \varepsilon_R (T_{LD} - T_{RS}) = \int_0^L \beta \eta \alpha A \Delta T dl = \beta \eta \alpha A \Delta T L \quad (19)$$

where β is the convective heat transfer coefficient, α the value of transfer area per unit volume, η is the global effectiveness of the exchange surface and A is the frontal flow area. Neglecting the wall thermal resistance, the solid surface temperature T_p corresponds to the mean value between the two streams,

$$|T - T_p| = \frac{\varepsilon_R (T_{LD} - T_{RS})}{2} = \Delta T \quad (20)$$

Accordingly, by introducing the mass velocity of the air flow G_{air} circulating in the low temperature cycle,

$$\varepsilon_R = \frac{\beta \eta \alpha L}{2G_{air} c_p + \beta \eta \alpha L} \quad (21)$$

The heat transfer coefficient β refers to the experimental correlation of Colburn factor reported by Kays and London [29] and η is assumed to be 0.77 for realistic values of convective heat transfer coefficient

and usual fin geometry. The same reference gives the calculation procedure for the global pressure losses (inlet boundary-layer-separation drop, distributed losses through the passages and outlet recovery)

Table 3 - Possible features of a counter-flow regenerator with $\varepsilon = 0.9$.

Transfer surface per unit volume (m^{-1})			α	899
Regeneration effectiveness			ε_R	0.9
Global effectiveness of the exchange surface			η	0.77
Case 1			Case 2	
Mass velocity ($\text{kg}\cdot\text{m}^{-2}\text{s}^{-1}$)	G_{air}	13.7	Mass velocity ($\text{kg}\cdot\text{m}^{-2}\text{s}^{-1}$)	G_{air} 6.37
Pressure drops hot side (kPa)	ΔP	2.59	Pressure drops hot side (kPa)	ΔP 0.64
Pressure drops cold side (kPa)	ΔP	4.15	Pressure drops cold side (kPa)	ΔP 1.02
Axial length (m)	L	2.00	Axial length (m)	L 1.20
Frontal area (m^2)	A	0.95	Frontal area (m^2)	A 2.04

The regenerator features are listed in Table 3, for a target effectiveness of 0.9. This tentative sizing of the regenerator supports the feasibility of the selected design parameters shown in Table 1, where the assumed efficiency ε_R is conservative if compared to the claimed value ($\varepsilon_R \sim 95\%$) of the counter-flow regenerator featured by the 30 kW commercial product “Pascal Air” operating at design condition (NEDO documents [28]).

Case 1 features a smaller frontal cross-section area (higher mass velocity) and overall volume, but has higher pressure drop. Case 2 has a length $L=1.2$ m and the volume increases, but the pressure drop is reduced significantly ($\Delta P=1.02$ kPa). Therefore, the pressure drops of the regenerative heat exchanger are preliminarily considered to be negligible when compared to the overall pressure ratio π of the bottom cycle and disregarded with respect to the whole system performance.

3. RESULTS AND DISCUSSION

As a first step, we consider the cascade system serving the refrigerated space RS without any additional cooling load due to the loading dock. In this case, as all the cooling power is delivered at the

352 minimum temperature, the system operates in its most severe condition. Assuming the reference parameter
 353 values listed in Table 2, the system performance is summarized in Table 4.

354 In this configuration the compression ratio turns out to be $\pi=1.26$, yielding a very low temperature rise
 355 within the compressor ($T_1=1.5^\circ\text{C}$; $T_2=23.6^\circ\text{C}$). With respect to state-of-the-art centrifugal compressors used
 356 for refrigeration, π is lower by one order of magnitude. This means that the turbine and compressor may
 357 have a very simple design and fairly high efficiency, e.g. they may both have a single axial stage.
 358 Furthermore, this temperature increase copes well with the superheating at the exit of the cascade heat
 359 exchanger (evaporator E_2 in Fig. 2), yielding acceptable heat transfer irreversibility.

360

361 Table 4 - Performance parameters with 100% cooling capacity given to RS .

Brayton cycle mass flow rate ($\text{kg}\cdot\text{s}^{-1}$)	\dot{m}_{air}	5.49
Cooling capacity (kW)	Q_{f-RS}	57.6
Power consumption, vapour compression cycle (kW)	W_{VC}	26.5
Power consumption, Brayton cycle (kW)	W_B	50.1
Coefficient Of Performance	COP	0.75
Water vapour subtracted ($\text{kg}\cdot\text{h}^{-1}$)	\dot{m}_{H_2O}	3.06

362

363

364 Note also the amount of vapour subtracted to the RS , which highlights the significant de-
 365 humidification potential of this system. Obviously the details of the ice-capturing device are outside the
 366 scope of this paper and must be tailored on the quantity and quality (particle dimension, density, etc.) of the
 367 removed ice by a careful experimental analysis.

368 The COP value shown in Table 4 may be compared with the data listed in Table 1 and Fig. 1.
 369 However, these values should be decreased to account for internal loads related to evaporator fans and
 370 defrosting. According to ASHRAE Refrigeration Handbook [24], these contributions may amount to 15% of
 371 the total cooling load. This power penalty enters in the COP calculation both as a reduction of cooling
 372 capacity and as an increase of energy consumption. Another term of comparison can be a stand-alone

373 Brayton cycle covering all the temperature range between ambient and refrigerated space. In this case the
 374 COP would fall to 0.55.

375 As the cooling power delivered to the loading dock increases, the cascade system shows a relentless
 376 performance improvement, up to a COP of 1.4 when half of the cooling capacity from the top cycle is
 377 delivered at the temperature level of the LD. Obviously the aforementioned results are strictly dependent on
 378 the parameters chosen for the simulation.

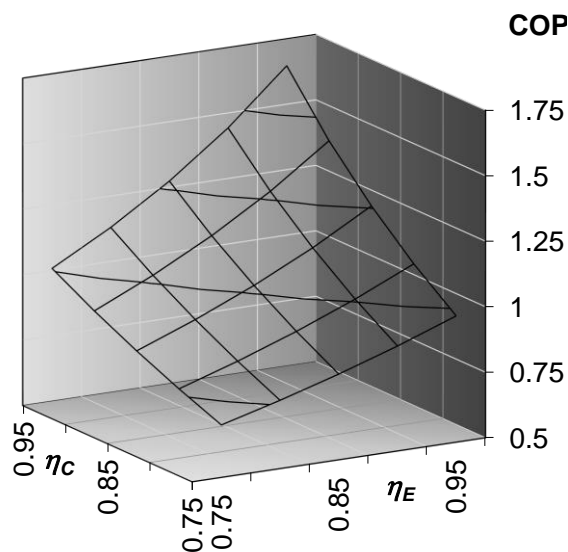
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380 3.1 – Effect of compressor and turbine efficiency

381 If the compressor and/or turbine efficiency are improved or decreased, the system COP is heavily
 382 influenced, as shown in Fig. 8 for $Q_{LD}/Q_{VC} = 0.25$. In general, for given cold store and loading dock
 383 temperatures, higher expander and compressor efficiency are associated to lower required pressure ratios.
 384 The analytic expression of the dependence of the COP of the Brayton cycle on the polytropic efficiency of
 385 the turbine and the compressor has been described by Giannetti and Milazzo in [18].

386 The inclined surface drawn in Fig. 8 is practically symmetric, i.e. the effect of improving the
 387 compressor or the turbine is equally important. This result could change if the effect of pressure losses
 388 through heat exchangers, piping and ice filter were included in the analysis. A further pressure reduction may
 389 be due to the need to introduce the air within the *RS* at a suitable speed in order to promote circulation.

390



391

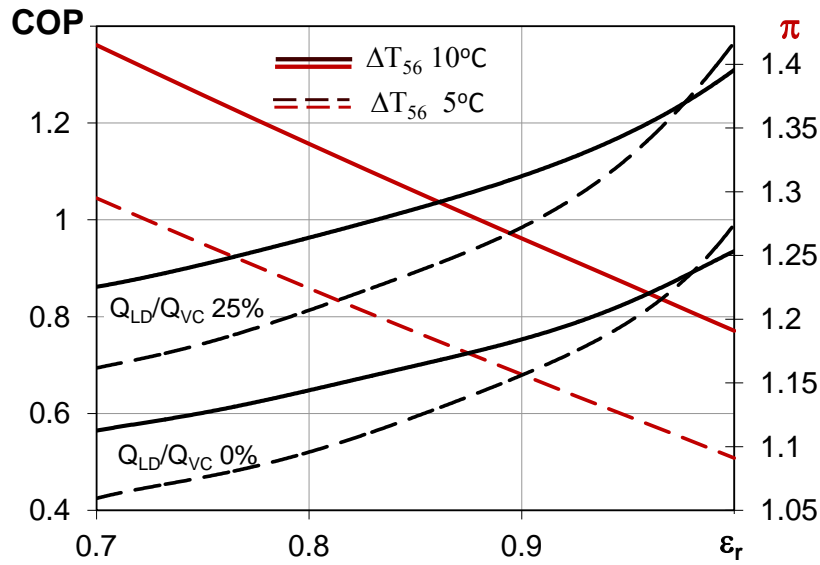
392 Figure 8. Cascade system COP v/s compressor and turbine efficiencies ($Q_{LD}/Q_{VC} = 0.25$)

393

394 3.2 Effect of Regenerator effectiveness

395 The regenerator effectiveness ε_r also has a significantly beneficial influence, especially at high values,
 396 as shown in Fig. 9 for the same value of Q_{LD}/Q_{VC} . *Ceteris paribus*, a higher ε_r reduces the gap between the
 397 source temperature and the inlet temperature to the turbine or compressor, lowering the compression ratio π
 398 required for a specific application case.

399



400

401 Figure 9. Cascade system COP v/s regenerator effectiveness

402

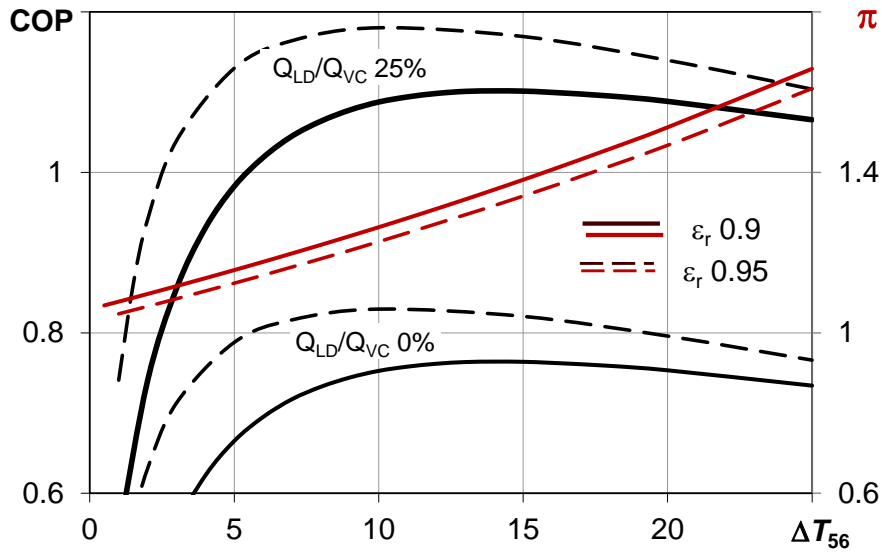
403 Increasing the regenerator surface is probably the simplest way to improve the air cycle performance,
 404 though a great care must be taken to avoid a corresponding increase in pressure loss [18]. The high reference
 405 value assumed for effectiveness has been proven to be feasible in terms of heat exchange surface by the
 406 conceptual design reported in the previous section. The same can be said of evaporators and condenser,
 407 where rather low temperature differences have been imposed.

408

409 3.3 Effect of cold air inlet temperature

410 Another significant parameter is the temperature difference between refrigerated space and air inlet
 411 (ΔT_{5-6}). Figure 10 shows how a reduction of this temperature difference yields a corresponding decrease of π .

412 A higher regenerator effectiveness better matches lower ΔT_{5-6} and, conversely, lower values of the latter
 413 parameter turn out to be beneficial when a highly efficient regenerator is not available. The effect of ΔT_{5-6} on
 414 the system COP is also visible in Fig. 10, which shows a well-defined maximum for each value of Q_{LD}/Q_{VC} .
 415 Specifically, the value of ΔT_{5-6} yielding the maximum performance increases as the regenerator effectiveness
 416 is reduced. The same behaviour was also shown in terms of air flow rate or specific cooling capacity in [18].
 417



418

419 Figure 10. Cascade system COP v/s ΔT_{5-6}

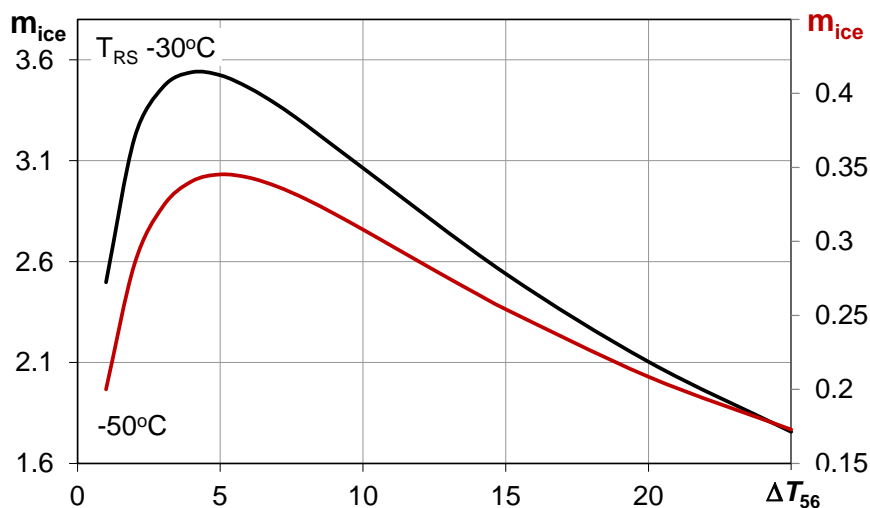
420

421 Every refrigeration system faces finite capacity heat sources that can be approximated to isothermal
 422 sources with different degrees of approximation. However, some applications, counter current products
 423 refrigeration on contact belt freezers for instance, require large temperature variations of the low temperature
 424 heat source. In these cases, if the system is designed to deliver the refrigerant in counter-current to the belt
 425 movement, a higher ΔT_{5-6} could indeed minimize heat transfer irreversibility and lead to higher overall
 426 system efficiency.

427 As the temperature difference increases (i.e. the air flow decreases), so does the difference in absolute
 428 humidity between inlet flow and *RS* air, enhancing the drying capability of the system. On the other hand, a
 429 lower flowrate directly implies a lower rate of humid air processed within the Brayton cycle. Figure 11
 430 highlights a maximum of the humidity removal rate (i.e. amount of ice formed per unit time) at a specific

431 value of ΔT_{5-6} . If the cold-store temperature decreases from -30 to -50, the maximum removal rate moves to
 432 the right.

433



434

435 Figure 11. Rate of humidity extracted for different cold-store temperatures

436

437 3.4 Effect of inter-cooled compression

438 As already mentioned, the efficiency gap of the Brayton cycle decreases at lower cold space
 439 temperatures. Therefore, further calculations have been made for values below -30°C. In this case, an inter-
 440 cooled compression could be advantageous in order to keep the compression ratio and compressor exit
 441 temperature at low values (Fig. 13). A possible scheme is shown in Fig. 12.

442

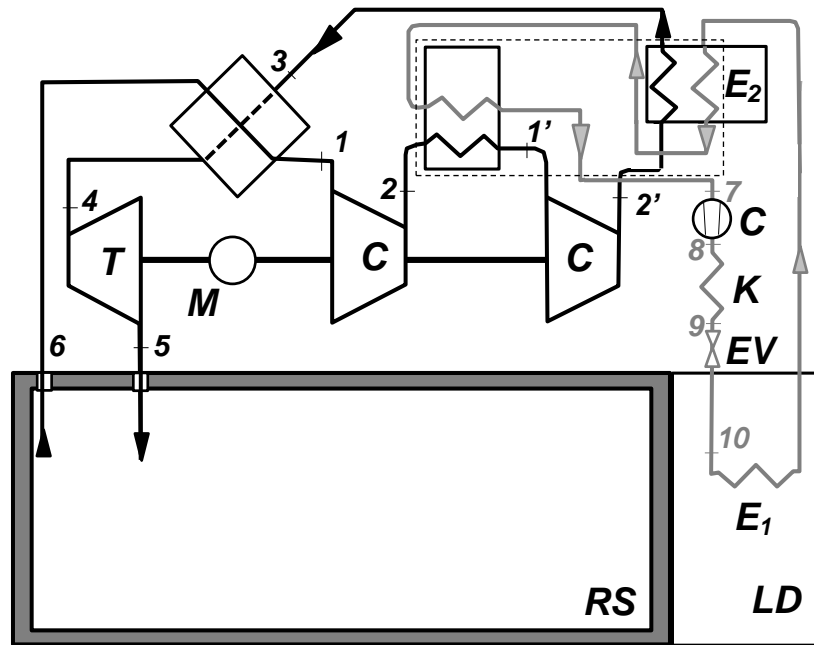


Figure 12. Scheme of the cascade system featuring inter-cooled staged compression on the Brayton cycle

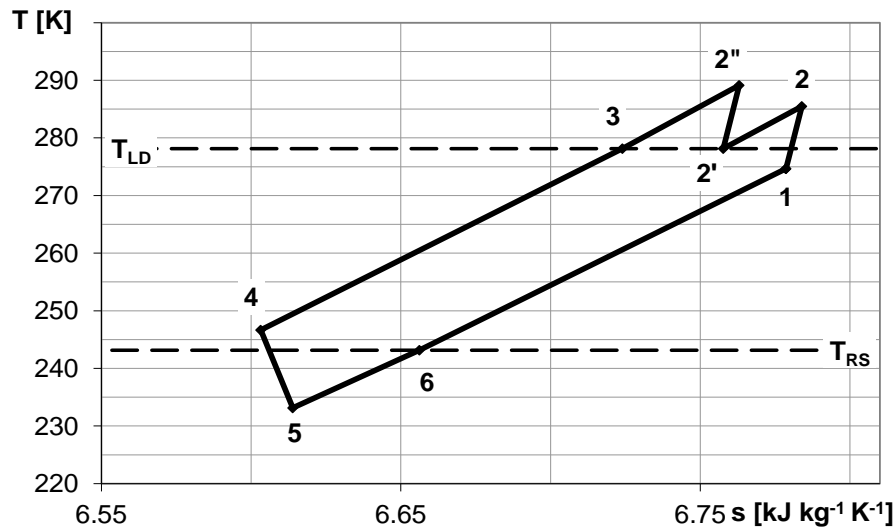


Figure 13. Temperature-entropy diagram for Brayton cycle with inter-cooled compression ($\Delta T_{5-6} = 7^\circ \text{C}$)

When compared with single stage compression, the intercooled configuration copes with higher temperature variation at the cold storage, i.e. lower air flowrates. As shown in Fig.14 the COP gain is not dramatic, but may become worth the effort at high values of ΔT_{5-6} .

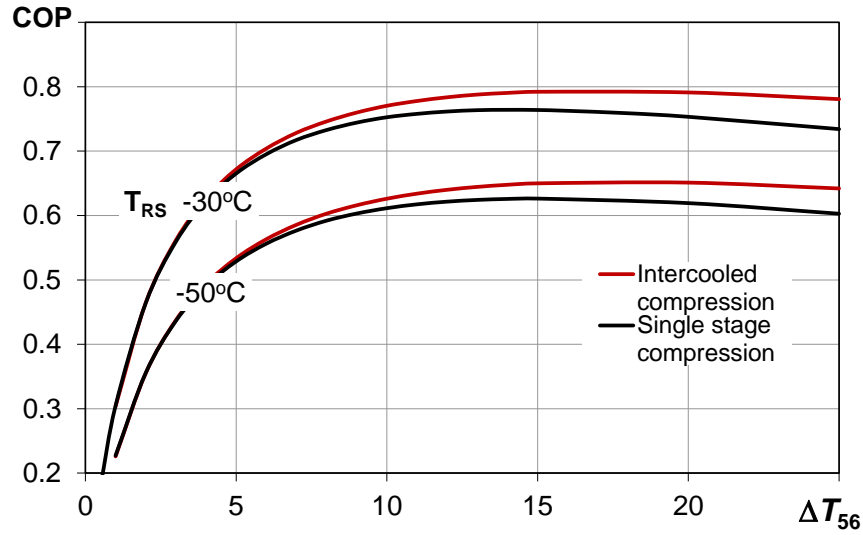


Figure 14. COP as a function of ΔT_{5-6} for various cold space temperatures, with (grey lines) and without (black lines) intercooling ($Q_{LD}/Q_{VC} = 0\%$)

3.5 Effect of cold space temperature

When the cold space temperature is used as a parameter, the comparison between conventional systems (Table 1, Fig. 1) and the one presented here (continuous lines in Figure 15) becomes clearer: accounting for COP losses due to defrosting and cold air circulation of the conventional systems the performance of the cascade Brayton system becomes competitive at low storage temperatures. A pure Brayton cycle is shown for comparison (dashed line in Fig. 15).

Systems requiring very low temperature, e.g. fast freezing of food, may represent a promising application for the proposed scheme. Air, in this case, may be the sole option, as shown e.g. by the experiments performed at -124.5°C by Hongli et al. [30].

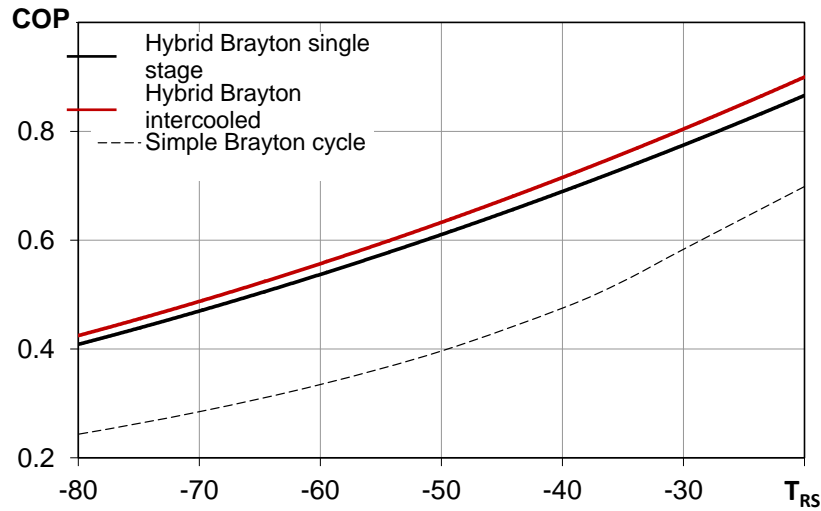


Figure 15. COP as a function of T_{RS} for various system configurations

4. CONCLUSIONS

A zero-GWP cascade configuration featuring an ammonia vapour compression cycle at the top and an inverse Brayton cycle at the bottom is proposed for cold store applications. The use of air as a working fluid in the bottom cycle eliminates the evaporator and the related thermal loads due to defrosting, as well as to the fan for air circulation. In a cascade configuration the regenerated Brayton cycle is characterized by a very low compression ratio, hence enabling the use of simple and highly efficient turbine and compressor. At the same time, this condition yields a lower temperature rise within the compressor and reduces the thermal irreversibility in the cascade heat exchanger.

The top cycle may also serve for loading dock refrigeration. The penalty in terms of COP can be low or even null, as soon as efficient compressor, turbine and regenerator are available. This target is becoming technically feasible as design and manufacturing of turbomachinery and heat exchangers progress. Additionally, as the operating temperature is lowered the efficiency gap with respect to conventional systems decreases or even vanishes, while other fluids (e.g. CO_2) reach their triple point and become unusable.

An optimum value exists for the temperature difference between cold air admission and refrigerated space. Latent load may be handled by an ice capturing device installed at the turbine outlet port.

If the refrigerated space temperature is very low, a staged-intercooled compression may offer a further slight increase in efficiency.

486 The proposed configuration seems promising in terms of safety (low pressure, ammonia used only
487 outside of the cold space), reliability (no need to defrost, no electric fan in the cold space) and environment
488 protection (both cycles use “natural” fluids). The system turns out to be readily feasible, given the well-
489 established background of each of the two integrated refrigeration technologies.

490 As long as the loading dock temperature is moderate (e.g. 5°C as assumed herein), this concept can be
491 extended by coupling the Brayton cycle with a heat driven refrigeration system, i.e. an absorption or ejector
492 chiller, in lieu of the electric-driven ammonia cycle. In this way, the electric energy consumed by the
493 Brayton cycle and the thermal energy for the heat driven chiller could be both produced by a combined heat
494 and power system (CHP), allowing significant energy savings.

495

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