Available online at www.sciencedirect.com

SciVerse ScienceDirect

Energy Procedia 14 (2012) 56 – 65

Energy Procedia

ICAEE 2011

R744-R717 Cascade Refrigeration System: Performance Evaluation compared with a HFC Two-Stage System

Antonio Messineo*

Engineering and Architecture Faculty, University of Enna Kore, Cittadella Universitaria, Enna, 94100, Italy

Abstract

In this study is presented a thermodynamic analysis of a cascade refrigeration system using as refrigerant carbon dioxide in low-temperature circuit and ammonia in high-temperature circuit. The operating parameters considered in this paper include condensing, evaporating, superheating and subcooling temperatures in the ammonia (R717) high-temperature circuit and in the carbon dioxide (R744) low-temperature circuit. Diagrams of COP versus operating parameters have been obtained. In addition, values for R744-R717 cascade refrigeration system are compared with the values obtained for a partial injection two-stage refrigeration system using the synthetic refrigerant R404A, a nearly azeotropic blend, specially used for commercial refrigeration. Results show that a carbon dioxide-ammonia cascade refrigeration system is an interesting alternative to R404A two-stage refrigeration system for low evaporating temperatures (–30°C ÷ –50°C) in commercial refrigeration for energy, security and environmental reasons.

Keywords: Refrigeration systems; cascade systems; two-stage compression systems; ammonia; carbon dioxide; R404A.

1. Introduction

As a result of environmental problems related to global warming and ozone-depleting effects caused by the use of synthetic refrigerants experienced over the last decades, the return to use of natural substance for refrigeration purposes seems to be the best long-term alternative [1-3]. Therefore, the use of natural fluids as refrigerants has attracted renewed interest during the last years. They must not be less energy efficient than the fluids that they replace. They must be proven to be safe, both for proximate neighbourhood and global environment.

*Corresponding author. Tel.: +39 0935 536448; fax: +39 0935 5369513.
E-mail address: antonio.messineo@unikore.it.

doi:10.1016/j.egypro.2011.12.896

Nomenclature

HTC	high-temperature circuit
LTC	low-temperature circuit
COP	coefficient of performance
sub	subcooling
sup	superheating
η	efficiency
Subscripts	
E	evaporation
C	condensation
ME	evaporating temperature of HTC
MC	condensing temperature of LTC
is	isentropic
max	maximum
H	high-temperature circuit
L	low-temperature circuit
OPT	optimum

They must be simple to use and cost-effective, immediately available and ideally they should not require any significantly new or unfamiliar technology [4]. In low-temperature applications, including rapid freezing and the storage of frozen food, the required evaporating temperature of the refrigeration system ranges from –40°C to –55°C, so a single-stage vapour-compression refrigeration system is insufficient, while two-stage or cascade refrigeration systems are used for low-temperature applications. The high- and low-pressure sides of a two-stage refrigeration system are charged with the same refrigerant, whereas the high and low-temperature circuits in a cascade system are filled separately with appropriate refrigerants. Therefore, using natural refrigerants in both two-stage and cascade refrigeration system helps to meet the requirements of environmental regulations [5]. Cascade refrigeration systems are appropriate for industrial applications, especially in the supermarket refrigeration industry, where the evaporating temperature of frozen-food cabinets ranges from –30°C to –50°C [6]. In these units, two single-stage systems are thermally coupled through a cascade condenser. The high-temperature stage of a cascade refrigeration system usually uses ammonia (R717), propane (R290), propylene (R1270), ethanol or R404A, whereas the low-temperature circuit of the refrigeration system can be charged with carbon dioxide (R744) [7].

Ammonia (R717), as a natural refrigerant, is widely used in industrial and commercial refrigeration. In the last few years, the advantageous thermo-physical properties and good environmental compatibility have increased interest in using ammonia, but its flammability and toxicity limits have also to be taken into account. Carbon dioxide has been used as a refrigerant in a number of vapour compression systems for over 130 years, but it has only been fully exploited during the last decade [8]. The main advantages of carbon dioxide are that it is nontoxic and incombustible. Moreover, compared with ammonia two-stage refrigeration system, the R744-R717 cascade refrigeration system has a significantly lower charge amount of ammonia, and the COP of the cascade system is comparable to a two-stage system at low temperatures [9-11]. Therefore, many investigations of the R744-R717 cascade refrigeration system are gaining attention [9, 12-14]. Researches like Eggen and Aflekt [15], Pearson and Cable [16] and Van Riessen [17] show practical examples of the use of a cascade system of R744-R717 for cooling in supermarkets. Eggen and Aflekt [15] developed research based on a prototype of a cooling system built in Norway. Pearson and Cable [16] showed data from a cooling system used in a Scottish supermarket line (ASDA) and Van Riessen [17] carried out a technical energy and economical analysis of a cooling system used in a Dutch supermarket. Lee et al. [5] analyzed a R744-R717 cascade system from the thermodynamics point of view in order to determine the optimum condensing temperature of R744 in the low-temperature circuit. Bansal and Jain [18] evaluated the optimum cascade condensing temperatures of R744 for different refrigerants such as R717, R290, R1270 and R404A, which are in the high temperature circuits of a cascade system. Cascade systems using ammonia or R404A in the high-temperature part and carbon dioxide in the low-temperature one with direct distribution of cold have been built in recent years, mainly in Scandinavian countries.

The scope of the present paper is focused on the thermodynamic analysis of a cascade refrigeration system using as refrigerant carbon dioxide, because of its thermo-physical properties, in low-temperature circuit and ammonia in high-temperature circuit. The operating parameters considered in this study include (1) condensing, evaporating, superheating and subcooling temperatures in the ammonia (R717) high-temperature circuit (HTC); (2) condensing, evaporating, superheating and subcooling temperatures in the carbon dioxide (R744) low-temperature circuit (LTC). The results for cascade refrigeration system are compared with the values obtained for a partial injection two-stage refrigeration system using the synthetic refrigerant R404A.

2. Thermodynamic analysis of R744-R717 cascade system

A cascade system comprises two separate one-stage refrigeration cycles, each working with a different refrigerant, best suited for the working conditions. It is necessary to use a cascade system when the difference between the temperature at which heat is rejected and the temperature at which refrigeration is required is so large that a single refrigerant with suitable properties cannot be found.

A schematic diagram of a two-stage cascade refrigeration system is shown in Fig. 1a. This refrigeration system comprises two separate refrigeration circuits: the high-temperature circuit (HTC) and the low-temperature circuit (LTC). Each refrigeration system consists of a compressor, a condenser, an expansion valve and an evaporator. In this study ammonia is the refrigerant in HTC, whereas carbon dioxide, because of its thermo-physical properties, is the refrigerant in LTC. The circuits are thermally connected to each other through a cascade condenser, which acts as an evaporator for the HTC and a condenser for the LTC. Fig. 1a shows that the condenser in this cascade refrigeration system rejects a heat flow $\dot{Q}_H$ from the condenser at condensing temperature of T_C, to its condensing medium or environment. The evaporator of this cascade system absorbs the cooling load $\dot{Q}_L$ from the cooling space to the evaporating temperature T_E. The heat absorbed by the evaporator of the LTC plus the work input to the LTC compressor equals the heat absorbed by the evaporator of the HTC. T_{MC} and T_{ME} represent the condensing and evaporating temperatures of the cascade condenser, respectively. $\Delta T = T_{MC} - T_{ME}$ represents the difference between the condensing temperature of LTC and the evaporating temperature of HTC.

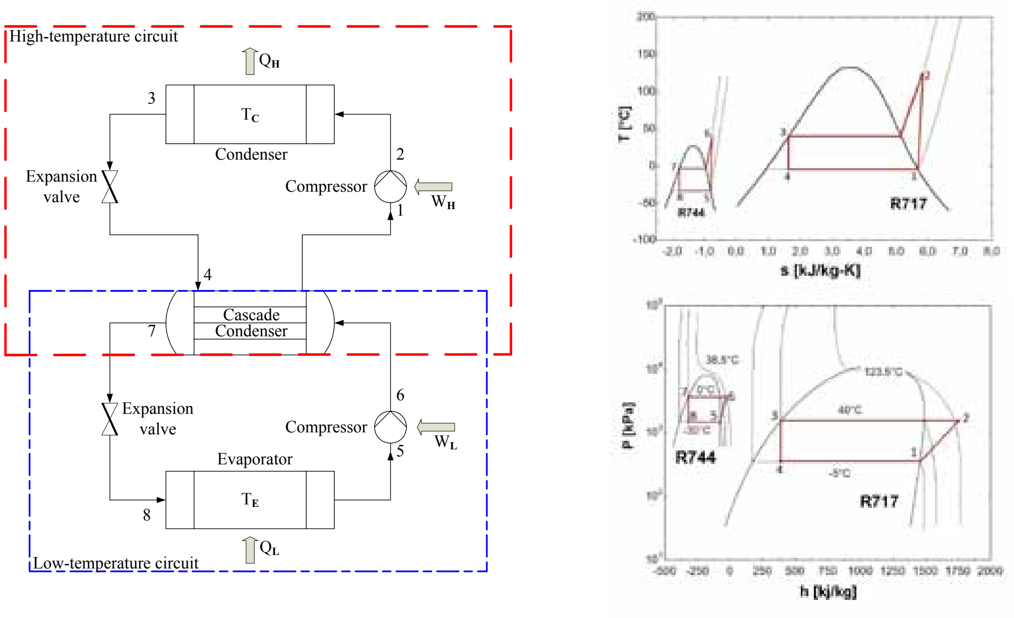


Fig. 1. a) Schematic for R744-R717 two-stage cascade system. b) T–s and log P-h diagrams.

The evaporating temperature T_E, the condensing temperature T_C, and the temperature difference in the cascade condenser are three important design parameters of a R744-R717 cascade refrigeration system.

Fig. 1b shows the process evolution for both the R744 and R717 cycles in a T–s and log (P)–h diagrams. The thermodynamic analysis of the two-stage cascade refrigeration system was performed based on the following general assumptions:

- all components are assumed to be a steady-state and steady-flow process. The changes in the potential and the kinetic energy of the components are negligible;
- adiabatic compression with an isentropic efficiency (η_{is}) equal to 0.70 for both high- and low-temperature compressors;
- the expansion process is isenthalpic;
- negligible pressure and heat losses/gains in the pipe networks or system components;
- $\Delta T = 5°C$ in the cascade condenser.

The thermo-physical properties of the refrigerants specified in this paper were calculated using a software package called Engineering Equation Solver [19], which contains built-in property functions of many refrigerants. Taking into account the assumptions previously made, mass and energy balances are given by Eqs. (1)-(2), respectively.

$$\sum_{in} \dot{m} = \sum_{out} \dot{m} \tag{1}$$

$$\dot{Q} - \dot{W} + \sum_{in} \dot{m}\,h - \sum_{out} \dot{m}\,h = 0 \tag{2}$$

In Table 1 specific equations for each system's component are summarized. The system's COP has been calculated by the following equation:

$$COP = \frac{(COP_{LTC})(COP_{HTC})}{1 + COP_{LTC} + COP_{HTC}} \tag{3}$$

with

$$COP_{LTC} = \frac{\dot{Q}_L}{\dot{W}_L} \tag{4}$$

$$COP_{HTC} = \frac{\dot{Q}_M}{\dot{W}_H} \tag{5}$$

Table 1. Energy and mass balance for R744-R717 cascade system.

Component	Mass	Energy
HTC compressor	$\dot{m}_H = \dot{m}_1 = \dot{m}_2$	$\dot{W}_H = \dot{m}_H (h_2 - h_1)/\eta_{is}$
Condenser	$\dot{m}_H = \dot{m}_2 = \dot{m}_3$	$\dot{Q}_H = \dot{m}_H (h_2 - h_3)$
HTC exp. device	$\dot{m}_H = \dot{m}_3 = \dot{m}_4$	$h_3 = h_4$
Cascade condenser	$\dot{m}_L = \dot{m}_6 = \dot{m}_7$ $\dot{m}_H = \dot{m}_1 = \dot{m}_4$	$\dot{Q}_M = \dot{m}_L (h_6 - h_7) = \dot{m}_H (h_1 - h_4)$
LTC compressor	$\dot{m}_L = \dot{m}_5 = \dot{m}_6$	$\dot{W}_L = \dot{m}_L (h_6 - h_5)/\eta_{is}$
LTC exp. device	$\dot{m}_L = \dot{m}_7 = \dot{m}_8$	$h_7 = h_8$
Evaporator	$\dot{m}_L = \dot{m}_5 = \dot{m}_8$	$\dot{Q}_L = \dot{m}_L (h_5 - h_8)$

The hypothesis of a constant η_{is}=0.70, for every working conditions, does not allow changes of efficiency with pressure ratio, but this value is an average value for the most modern compressors. Comparison of system performance was investigated as a function of the variation of downstream R744 temperature from cascade condenser, keeping constant the evaporating temperature (T_E), condensing temperature (T_C) and temperature difference in the cascade condenser (ΔT). System's performance curve for R744-R717 are reported in Fig. 2 for different degrees of superheating and subcooling, considering T_E = –35°C, T_C = 35°C. Table 2 presents the obtained optimum condensing temperatures of the cascade condenser and the corresponding maximum COPs for different degrees of subcooling and superheating. As it is possible to observe in Fig. 2, a maximum COP and its corresponding optimal T_{MC} exists. The effects of subcooling on the COP are significant for these refrigeration systems. In particular, concerning zero degree of subcooling, the general trend is that COP_{MAX} increases as the degree of subcooling increases, i.e. the COP_{MAX} rises by 3% for a degree of subcooling equal to 5°C in both cycles, and 7% for a degree of subcooling equal to 10°C.

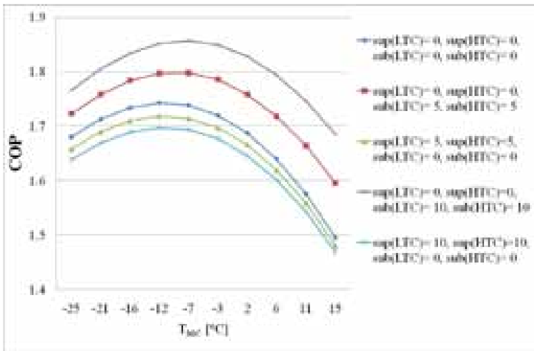


Fig. 2. Effect of T_{MC} on the overall COP of a R744-R717 cascade refrigeration system.

Table 2. T_{MC} and COP_{MAX} for different degrees of subcooling and superheat.

Degrees of subcooling and superheat [°C]	COP_{MAX}	$T_{MC, OPT}$ [°C]
sup (LTC)= 0, sup (HTC)= 0, sub (LTC)= 0, sub (HTC)= 0	1.71	-11.65
sup (LTC)= 0, sup (HTC)= 0, sub (LTC)= 5, sub (HTC)= 5	1.76	-9.88
sup (LTC)= 5, sup (HTC)= 5, sub (LTC)= 0, sub (HTC)= 0	1.68	-11.63
sup (LTC)= 0, sup (HTC)= 0, sub (LTC)= 10, sub (HTC)= 10	1.82	-8.24
sup (LTC)= 10, sup (HTC)= 10, sub (LTC)= 0, sub (HTC)= 0	1.66	-11.62

In the opposite direction, the effects of superheating reduce the overall performance of cascade systems. Concerning zero degree of superheating, the general trend is that COP_{MAX} decreases as the degree of superheating increases, i.e. the COP_{MAX} decreases by 1.5% for a degree of superheating equal to 5°C in both cycles, and 3% for a degree of superheating equal to 10°C. These results closely agree with other researches [1, 5, 20]. Fig. 3 depicts the variation of COP as a function of the condensing temperature (left) and evaporating (right) for different degrees of subcooling and superheating.

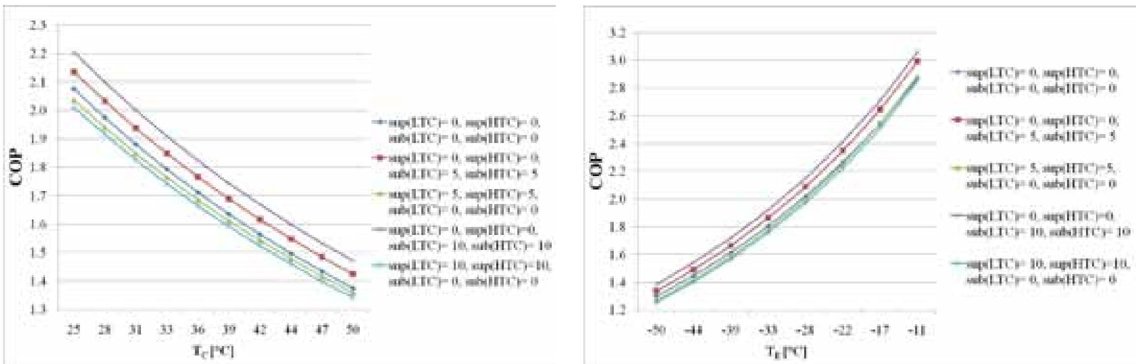


Fig. 3. a) Effect of condensing (left) and b) evaporating temperature (right) on system performance.

The ratio of mass flow of the refrigerants in the high-temperature circuit and in the low-temperature circuit was also analyzed. Fig. 4 shows the variation of R744-R717 mass ratio with respect to condensing temperature of the cascade condenser, considering T_E= –35°C, T_C = 35°C.

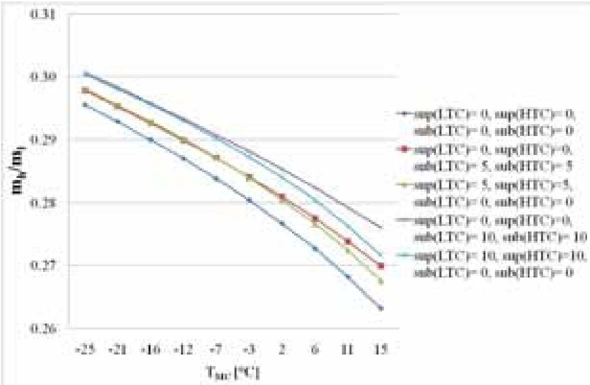


Fig. 4. Variation of R744-R717 mass ratio as a function of T_{MC}.

These curves, obtained for different degrees of superheating and subcooling, show a mass flow ratio value ranging from 0.26 to 0.30. This value is imputable to ammonia thermo-physical properties and in particular due to its high latent heat of vaporization. Moreover this aspect has positive consequences on the system safety, since ammonia is toxic and inflammable.

3. Thermodynamic analysis of R404A two-stage system

The objective of the tests was to determine how the vapour-compression refrigeration plant actually behaved when the different types of refrigerant fluids were used. This way it was possible to obtain adequate information on the plant with replacement fluids other than R22 while working under different conditions. The most used plant engineering solution in low-temperature applications (–30°C ÷ –50°C), is based on multi-stage systems, which are able to guarantee reliability and safety of the plants. With the aim of a making a direct comparison with the cascade system previously analysed, a two- stage cycle working with R404A was taken into account. The blend of HFC R404A is one of the working fluids mostly used for commercial and industrial refrigeration. The considered two-stage system is of partial injection type. In this typology of plant the downstream superheated steam of the first compression stage is brought back to the saturated dry steam conditions exploiting the expansion of part of the fluid flowing out from the condenser in a intermediate receiver. The fluid, in saturated dry steam conditions, is compressed in the second stage up to the condensing pressure. In good practice, each compressor will operate with almost the same pressure ratio. So the intermediate pressure was obtained using the equation:

$$p_m = \sqrt{p_{cond} \cdot p_{evap}} \tag{6}$$

The temperature of the upstream fluid at the expansion valve of the low pressure stage was assumed equal to the evaporating temperature of the fluid in the intermediate receiver, increased by a ΔT of 5°C.

Fig. 5 schematically depicts the considered R404A two-stage refrigeration system.

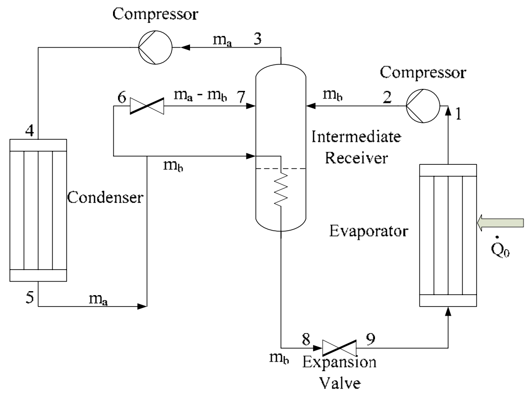


Fig. 5. Schematic of considered R404A two-stage refrigeration system.

The cooling load is defined by:

$$\dot{Q}_0 = \dot{m}_b (h_1 - h_9) \tag{7}$$

The balance equation for the R404A two-stage cycle in the intermediate receiver is:

$$(\dot{m}_a - \dot{m}_b) h_7 + \dot{m}_b h_5 + \dot{m}_b h_2 = \dot{m}_a h_3 + \dot{m}_b h_8 \tag{8}$$

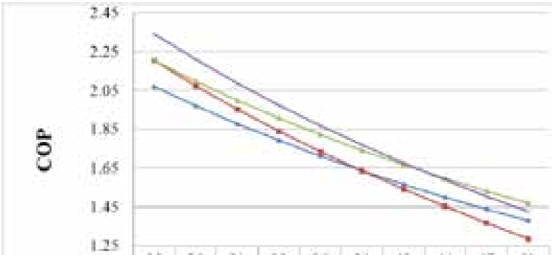
The COP of the system has been calculated by the following equation:

$$COP = \frac{\dot{Q}_0}{\dot{W}_1 + \dot{W}_2} \tag{9}$$

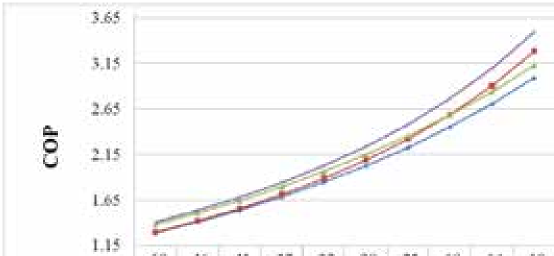
In accordance with thermodynamic analysis of R744-R717 cascade system, the isentropic efficiencies of the compressors (η_{is}) are assumed constant (equal to 70%), for every working condition. The degree of subcooling enhances the system performance, whereas the effect of degree of superheating has little impact on system performance. In particular, the COP rises by 7.5% for a degree of subcooling equal to 10°C, whereas a degree of superheating equal to 10°C does not produce significant effects on the COP value.

4. Comparison between the R744-R717 cascade cycle and the R404A two-stage cycle

In the following are showed the results obtained from the thermodynamic analysis of the R404A two-stage cycle, compared with those obtained for the cascade cycle as a function of the condensing and evaporating temperatures. Fig. 6a depicts the variation of COPs of two cycles for change in condensing temperature for zero degree of subcooling and degree of subcooling equal to 10°C, for constant evaporating temperature (T_E= –35°C). In R744-R717 cascade cycle the condensing temperature of the cascade condenser (T_{MC}) is equal to –8°C. The general trend of COPs, with zero degree of subcooling and degree of subcooling equal to 10°C, respectively, shows a similar behaviour of the two cycles (in terms of performances) within the range of condensing temperature 35°C ÷ 40°C. This range represents the most common operating conditions. For condensing temperatures higher than 40°C the cascade cycle outperforms the R404A two-stage cycle, whereas for condensing temperatures lower than 35°C the R404A two-stage cycle performs better. Fig. 6b depicts the variation of COPs of two cycles as a function of the evaporating temperature for zero degree of subcooling and degree of subcooling equal to 10°C, for constant condensing temperature (T_C =35°C). In R744-R717 cascade cycle the condensing temperature of the cascade condenser (T_{MC}) is equal to –8°C. The trend of COPs, with zero degree of subcooling and degree of subcooling equal to 10°C, respectively, shows a similar behaviour of the two cycles (in terms of performances) up to an evaporating temperature of -35°C. For higher evaporating temperatures the R404A two-stage cycle is characterized by better performances.



	25	28	31	33	36	39	42	44	47	50
Cascade (0)	2.07	1.97	1.88	1.79	1.71	1.64	1.57	1.50	1.44	1.38
Two Stage (0)	2.20	2.07	1.95	1.84	1.73	1.64	1.54	1.45	1.37	1.29
Cascade (10)	2.20	2.10	2.00	1.91	1.82	1.74	1.67	1.60	1.53	1.47
Two Stage (10)	2.34	2.21	2.09	1.97	1.87	1.77	1.68	1.59	1.50	1.43



	-50	-46	-41	-37	-32	-28	-23	-19	-14	-10
Cascade (0)	1.29	1.41	1.54	1.68	1.85	2.02	2.22	2.45	2.70	2.99
Two Stage (0)	1.30	1.42	1.56	1.71	1.89	2.09	2.32	2.59	2.90	3.28
Cascade (10)	1.38	1.51	1.64	1.80	1.96	2.15	2.35	2.58	2.84	3.13
Two Stage (10)	1.41	1.54	1.68	1.84	2.03	2.24	2.48	2.76	3.09	3.49

Fig. 6. Variation of COPs of two cycles as a function a) of the condensing temperature (left) and b) evaporating temperature (right).

5. Conclusions

In this paper the thermodynamic analysis of a R744–R717 cascade system and of a R404A two-stage system, working in the same operating conditions, is reported. The results obtained from the thermodynamic analysis pointed out:

- a similar behaviour of the two cycles (in terms of performances) within the range of condensing temperature 35°C÷ 40°C, which represents the most common operating conditions;
- a similar behaviour of the two cycles (in terms of performances) within the range of evaporating temperature -50°C÷ 35°C.

In particular, it was noticed a decrease of the COP as the condensing temperature increases from 30°C to 45°C, with zero degree of subcooling and for T_E= –35°C. The decrease is by 27% for the cascade cycle and by 37% for the R404A two-stages cycle, respectively. In the case of subcooling equal to 10°C these decreases amount to 27% and 34%, respectively. On the other hand, it was noticed a increase of the COP as the evaporating temperature rises from -50°C to -30°C, with zero degree of subcooling and for T_C = 35°C. The increase is by 50% for the cascade cycle and by 53% for the two-stages cycle, respectively; in the case of subcooling equal to 10°C these increases change to 48% and 50%, respectively.

Once highlighted the similar perfomances of the two investigated cycles, it is worthy noting a further element which makes particularly interesting the possible utilization of the R744–R717 cascade cycles. The utilization of carbon dioxide (ODP=0 and GWP=1) and ammonia (ODP=0 and GWP=0) is indeed suitable in the light of the always more pressing environmental issues thanks to the fact that they are natural fluids with a null environmental burden, unlike synthetic refrigerants as R404A (ODP=0 and GWP=3700), currently used for commercial and industrial refrigeration. R744–R717 cascade system offer good safety guarantees, because it is possible to confine the high pressure circuit containing R717 within the freezing plant, endowed with all the security devices provided for by the legislative standards. The fluid circulating in low pressure circuit, located within indoor environments with presence of human occupants, is R744, which is not toxic and not flammable.

So, in commercial refrigeration, in low-temperature applications including rapid freezing and the storage of frozen foods, the use of cascade systems using ammonia in the high-temperature part and carbon dioxide in the low-temperature one, in place of traditional two-stage systems working with synthetic fluids as R404A, is certainly a valid application for energy, security and environmental reasons.

References

[1] Dopazo JA, Fernàndez-Seara J, Sieres J, Uhìa FJ. Theoretical analysis of a CO_2-NH_3 cascade refrigeration system for cooling applications at low temperatures. *Applied Thermal Engineering* 2009; **29**: 1577-1583.

[2] Messineo A, Panno G. LNG cold energy use in agro-food industry: a case study in Sicily. *Journal of Natural Gas Science and Engineering* 2011; **3**: 356-363.

[3] Messineo A, Panno D. Potential applications using LNG cold energy in Sicily. *International Journal of Energy Research* 2008; **32**: 1058–1064.

[4] Pearson A. Refrigeration with ammonia. *International Journal of Refrigeration* 2008; **31**: 545-551.

[5] Lee TS, Liu CH, Chen TW. Thermodynamic analysis of optimal condensing temperature of cascade-condenser in CO_2/NH_3 cascade refrigeration systems. *International Journal of Refrigeration* 2006; **29**: 1100-1108.

[6] Getu HM, Bansal PK. Modeling and performance analysis of evaporators in frozen food supermarket display cabinets at low temperatures. *International Journal of Refrigeration* 2007; **30**: 1227-1243.

[7] Getu HM, Bansal PK. Thermodynamic analysis of an R744–R717 cascade refrigeration system. *International Journal of Refrigeration* 2008; **31**: 45-54.

[8] Pearson A. Carbon dioxide—new uses for an old refrigerant. *International Journal of Refrigeration* 2005; **28**: 1140–1148.

[9] Pearson A. New developments in industrial refrigeration. *ASHRAE Journal* 2001; **43**: 54-58.

[10] Taylor CR. Carbon dioxide-based refrigerant system. *ASHRAE Journal* 2002; **44**: 22-27.

[11] Rolfsman L. Experiences from CO_2 cascade plants. In: *Proceedings of International Congress of Refrigeration*, Washington, DC, 2003.

[12] Zha S, Ma Y, Wang J, Li M. The thermodynamic analysis and comparison on natural refrigerants cascade refrigeration cycle. In: *Proceedings of Fifth IIR-Gustav Lorentzen Conference on Natural Working Fluids*, Guangzhou, 2002.

[13] Lee TS, Liu CH, Chen TW. Thermodynamic analysis of optimum condensing temperature of cascade condenser for CO_2/NH_3 cascade refrigeration systems. In: *Proceedings of IIR Ammonia Refrigeration Conference*, Ohrid, Republic of Macedonia, 2005.

[14] Yabusita T, Kitaura T. CO_2/NH_3 Cascade Refrigeration System Technical Report. Toyo Engineering Works, LTD., 2005.

[15]Eggen G, K. Aflekt K. Commercial refrigeration with ammonia and CO_2 as working fluids. In: *Proceedings of the Third IIR: Gustav Lorentzen Conference on Natural Working Fluids*, Oslo, Norway, 1998, pp.281-292.

[16] Pearson A, P. Cable P. A distribution warehouse with CO_2 as refrigerant. In: *Proceedings of the International Congress of Refrigeration, Washington*, DC, USA, 2003.

[17] Van Riessen GJ. NH_3/CO_2 Supermarket refrigeration system with CO_2 in the cooling and freezing section. TNO Environment, Energy and Process Innovation, Apeldoorn, Netherlands, 2004.

[18] Bansal PK, S. Jain S. Cascade systems: past, present, and future. *ASHRAE Trans*. 2007; **113**: 245–252.

[19] EES: Engineering Equation Solver (EES), 2006. fChart Software Inc.