COMPARATIVE ANALISIS OF DIFFERENT CONFIGURATION OF HEAT PUMPS OPERATING WITH NATURAL REFRIGERANTS

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The article compares, based on the exergetic analysis, the performances of a cogeneration system for heating and air conditioning that works with CO_2 or NH_3.

The analyzed refrigeration schemes are in two stages.

The most efficient scheme for NH_3 was the one with flash intermediary cooling (COP_{NH3}=5.79, η_{exNH3}=0.42) and for CO_2, the one with intermediary cooling through the injection of saturated gas at intermediate pressure in the gas stream discharged by the first compression stage (COP_{CO2} =3.56 and η_{exCO2}=0.29).

Keywords: Heat Pump, Exergetic Analysis, Ammonia, Carbon dioxide, Optimization.

1. Introduction

Sustainable development in the field of cold technology requires the use of refrigerants with a harmless effect on the external environment.

Such refrigerants must not have a destructive effect on the ozone layer and their contribution to global warming must be minimal. Natural substances such as water, air, CO_2, NH_3 or hydrocarbons fully fulfill these requirements [1].

Each of these natural refrigerants, due to its thermo-physical properties, is used in specific applications.

Ammonia is an excellent refrigerant recommended for the refrigeration and air conditioning industry. No other refrigerant can compete with the efficiency of NH_3 refrigeration systems. Its toxicity and flammability require the use of an intermediary cold carrying fluid between NH_3 and the cold space. Special ventilation measures are required that require the separation of ammonia circuits from closed spaces where people are.

Carbon dioxide is non-toxic and can be used in heat exchangers in direct contact with the product being cooled in the cold room.

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CO_2 is not flammable which justifies its use on board ships in maritime transport and in general in the case of large refrigeration transport units.

Significant research has been carried out to find ways to improve the efficiency of CO2 refrigeration plants.

Jing Luo et al. [2] studied the optimized operation of some transcritical cycles operating with CO_2. The objective of the optimization is the value of the exergetic efficiency of the refrigeration cycle. The study reports a maximum of the exergetic performance coefficient in the case of operation as a refrigeration installation between the condensing temperature of 35 ^{o}C and the vaporization temperature of 5 ^{o}C. A five-fold increase in the exergetic efficiency can be achieved in the cogeneration regime of cold and heat between the temperatures of 45 ^{o}C and 5 ^{o}C. The process of simulating the system was carried out in Aspen HYSYS, and a genetic algorithm was used as a tool to search for the optimal performance coefficient.

Zhenying Zhang et al. [3] analyzed the effect of intermediate cooling in different two stage transcritical CO_2 refrigeration cycles, on the optimum value of the intermediate and final pressure in the gas cooler. To reduce the dissipative effect of throttling, in some versions the throttling valve is replaced with an expander. In the case of intermediate cooling with a flash intercooler, compared to an external intermediate cooler, the values of the optimal intermediate and final pressures are lower, which leads to the reduction of mechanical energy consumption for driving the compressors. The effect of replacing the throttling valves with expanders increases the value of the performance coefficient of the considered cycle by reducing the dissipation of the adiabatic expansion process as well as by recovering part of the mechanical work of compression.

To find solutions for replacing refrigerants with a destructive effect on the environment, work [4] carries out a review of the schemes and functional characteristics of CO_2 systems, which proves to be extremely attractive in many applications where this natural agent is used, due to its properties of not being toxic or flammable. Schemes with multiple constructive and functional modifications are presented and analyzed from the perspective of energy efficiency. Special attention is paid to the application in the vehicle field.

The paper [5] investigates the possibilities of reducing the mechanical energy consumption in CO_2 cogeneration systems where cooling and heating at high temperature are performed. To double the positive effect of the temperature shift in the gas cooler, the temperature shift is also introduced in the vaporizer by using CO_2-based refrigerant mixtures. The optimization study is based on the exergetic performance coefficient. The minimum temperature in the refrigeration area reaches -32.4 ^{o}C in the case of the CO_2/R601 mixture, where a maximum weight of 30% of the cold exergy is obtained in the total amount of exergetic product, cooling and heating of the cogeneration system.

Alireza Zendehboudi [6] investigates the operation of an air-water heat pump with CO_2 for heating living spaces. In search of optimal functional and constructive solutions, the research calls for energetic, exergetic and exergoeconomic analysis. The study highlights the advantage of the temperature glide in the gas cooler between the water that heats up and the CO_2 gas that cools down. The high temperature zone of the gas cooler is used for domestic water heating, and the lower temperature zone, for space heating.

In the paper [7], the authors study the constructive and functional solutions for a heat pump where it is desired to replace the R134a freon with CO_2. The aim is to increase the efficiency of CO_2 transcritical cycles by adopting new schemes, in two stages of compression, with various intermediate cooling solutions between stages. From an energetic point of view, the two-stage CO_2 schemes prove to be superior to the initial version of the heat pump with R134a, which operates in one compression stage.

Hongwei Zhang et al. [8] highlight the superior heat transfer properties of CO_2 in the vaporization process and study the performance characteristics of some cascade heat pumps for low temperature environments. CO_2 is used in the lower cascade and R134a in the upper one. The simulation of the functioning of the cascade system is experimentally validated and presents a viable solution for heating in cold regions.

The bibliographic analysis highlighted the researchers' interest in improving the performance of CO_2 refrigeration systems. Exergetic analysis proves to be a powerful tool in the optimization procedure. The conclusions of the exergetic analysis will specify the direction to follow to find the optimal parameters of a certain scheme and finally to structurally improve the analyzed scheme.

2. Comparative analysis of different schemes of refrigeration and heat pump systems

2.1 Two-stage refrigeration system with intermediate cooling achieved by injection of a cold current

The scheme and representation of the cycle in a T-s diagram, both for NH_3 and CO_2, are shown in figure 1.

The operating characteristics are: heater inlet temperature of the heat carrier $t_{h,wi} = t_{10} = 45^{o}C$, heater outlet temperature of the heat carrier $t_{12} = 55^{o}C$, evaporator inlet temperature of the cold carrier $t_{v,wi} = 9^{o}C$, evaporator outlet temperature of the cold carrier $t_{v,we} = 4^{o}C$, temperature of vaporization $t_v = 0^{o}C$, the minimum temperature difference in the heater $\Delta T_P = T_5 - T_{12} = 5K$ (figure 1, d).

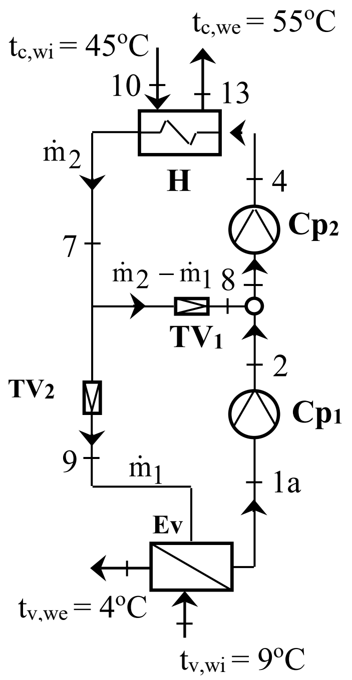

Fig. 1,a

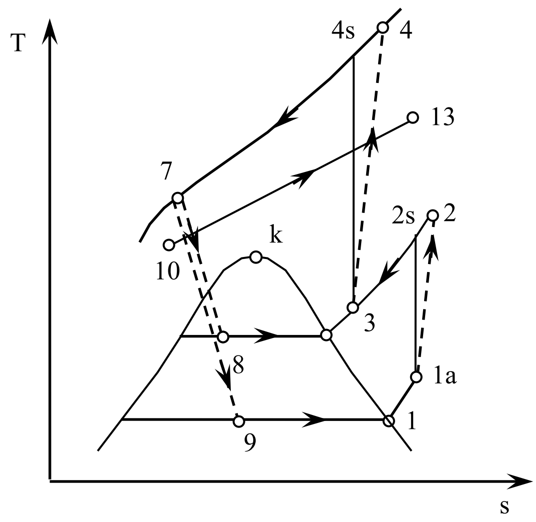

Fig. 1,b

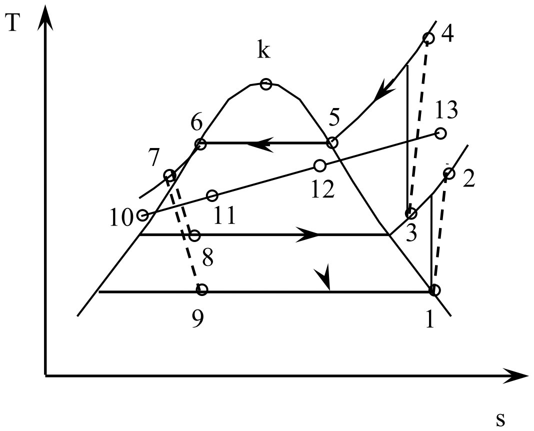

Fig. 1,c

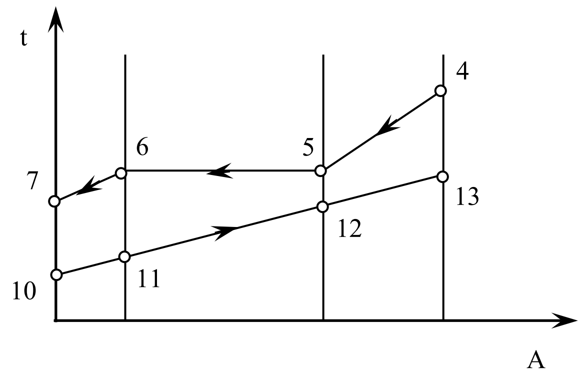

Fig. 1,d

Fig. 1. A two-stage refrigerating system with intermediate cooling achieved by injecting a cold stream in a biphasic state into the gas stream discharged from the first compression stage.
1,a: system diagram; 1,b: T-s diagram for CO_2; 1,c: T-s diagram for NH_3; 1,d: t-A diagram for the NH_3 scheme heater

2.1.1 The mathematical model

Energetic analysis

To achieve heat transfer in the heater (condenser for NH_3) a minimum temperature difference is required at Pinch (Fig. 1,d).

For NH_3, the mathematical model constructed to satisfy all constraints, starts with the choice of vaporization and condensation temperatures (Figures 1,c,d).

The designer's decisions are:

a. Vaporization temperature: $t_v = 0^o C$

b. Condensation temperature: $t_c = t_{12} + \Delta T_P$ (figure 1,d) (1)

c. Intermediate pressure: $p_i = \sqrt{p_v \cdot p_c}$ (2)

Knowing the values of the thermodynamic parameters in the main states of the refrigeration cycle, the refrigerant mass flow rates are calculated.

For 1 kW of heat transferred to the consumer in the condenser (heater), the flow rate $\dot{m}_2$ in the upper stage becomes:

$$\dot{m}_2 = 1/(h_4 - h_7) \tag{3}$$

The energy balance of the mixing process corresponding to state 3 gives the mass flow rate in the evaporator (first stage):

$$(\dot{m}_2 - \dot{m}_1) \cdot h_8 + \dot{m}_1 \cdot h_2 = \dot{m}_2 \cdot h_3 \tag{4}$$

The energy balance of the condenser-subcooler group imposes the mass flow rate of the heat carrier (water):

$$\dot{m}_w (h_{12} - h_{10}) = \dot{m}_2 (h_5 - h_7) \tag{5}$$

The desuperheat imposes the exit temperature of the heat carrier from the heater:

$$\dot{m}_w (h_{13} - h_{12}) = \dot{m}_2 (h_4 - h_5) \tag{6}$$

$$t_{13} = t(Water, p_w, h_{13}) \tag{7}$$

Exergetic analysis

The exergetic balance equation is [9]:

$$\left|\dot{W}_{cp1}\right| + \left|\dot{W}_{cp2}\right| = \dot{E}x_{Q_h}^{T_{h,w}} + \dot{E}x_{Q_v}^{T_{v,w}} + \dot{I}_{cp1} + \dot{I}_{cp2} + \dot{I}_m + \dot{I}_{h,\Delta T} + \dot{I}_{t1} + \dot{I}_{t2} + \dot{I}_{v,\Delta T} \tag{8}$$

where $\dot{I}_{cp1,2}$, $\dot{I}_m$, $\dot{I}_{h,\Delta T}$, $\dot{I}_{t1,2}$, $\dot{I}_{v,\Delta T}$, are in order, the exergy destructions caused by the irreversibility of the work processes in the two compression stages, intercooling by mixing, heat transfer at finite temperature difference in the heater, throttling in the throttling valves and transfer of heat at a finite temperature difference in the evaporator.

Exergy destruction in each functional zone of the refrigeration system is calculated based on the Gouy-Stodola theorem or with the help of the exergy balance equation [10].

Exergy destructions associated with compression processes are:

$$\dot{I}_{cp1} = \dot{m}_1 \cdot T_0 (s_2 - s_1) \tag{9}$$

$$\dot{I}_{cp2} = \dot{m}_2 \cdot T_0 (s_4 - s_3) \tag{10}$$

Exergy destruction in the cooling process by mixing is:

$$\dot{I}_m = T_0 (\dot{S}_3 - \dot{S}_2 - \dot{S}_8) = T_0 (\dot{m}_2 \cdot s_3 - \dot{m}_1 \cdot s_1 - (\dot{m}_2 - \dot{m}_1) s_8) \tag{11}$$

Exergy destructions in the heater (desuperheater, condenser and subcooler) is:

$$\dot{I}_{h,\Delta T} = T_0 [\dot{m}_2 (s_7 - s_4) + \dot{m}_w (s_{13} - s_{10})] \tag{12}$$

Exergy destruction in throttling valves:

$$\dot{I}_{t1} = T_0 (\dot{m}_2 - \dot{m}_1)(s_8 - s_7) \tag{13}$$

$$\dot{I}_{t2} = T_0 \cdot \dot{m}_1 (s_9 - s_7) \tag{14}$$

Exergy destruction in the evaporator caused by heat transfer at finite temperature difference:

$$\dot{I}_{v,\Delta T} = \left| \dot{E}x_{Q_v}^{T_v} \right| - \dot{E}x_{Q_v}^{T_{v,w}} \tag{15}$$

The exergetic efficiency of the refrigeration and heat pump system is:

$$\eta_{ex} = \frac{\dot{E}x_{Q_v}^{T_{v,w}} + \dot{E}x_{Q_h}^{T_{h,w}}}{\dot{W}_{cp1} + \dot{W}_{cp2}} \tag{16}$$

Exergy destruction related to mechanical work consumed:

$$\Psi_i = \frac{\dot{I}_i}{\left|\dot{W}_{cp1}\right| + \left|\dot{W}_{cp2}\right|} \tag{17}$$

2.1.2 Comparative analysis of two-stage refrigeration system with injection intercooler when operating with NH_3 or CO_2

The calculus has been performed with the EES simulator [11].

The exergetic analysis shows that compared to NH_3, the better behavior of CO_2 in the heater (lower exergy destruction relative to mechanical work consumption $\psi_{\Delta T,c}$) and evaporator – the cooler of the system – (lower relative exergy destruction $\psi_{\Delta T,v}$), is strongly penalized by the destructive effect of throttling in the throttling valve (higher relative exergy destruction ψ_{TV}) (Table 1).

In the case of CO_2, the important destructive effect of the throttling process (ψ_{TV}) is due to the lower value of the energy performance coefficient (COP), or exergetic (η_{ex}) compared to NH_3 (table 1).

Table 1

Two-stage refrigeration system with intermediate cooling by injecting a two-phase cold stream into the gas stream discharged from the first stage (Figure 1)

$t_0 = 25^oC$; $t_{h,wi} = 45^oC$; $t_{h,we} = 55^oC$; $t_{v,wi} = 9^oC$; $t_{v,we} = 4^oC$; $\Delta T_P = 5K$;

$\eta_{scp} = 0.8$; $p_{g,CO_2} = 125bar$; $t_3 = t_i + 5$

	COP	η_{ex}	ψ_{cp1} [%]	ψ_{cp2} [%]	ψ_{cp} [%]	$\psi_{\Delta T,h}$ [%]	ψ_{TV1} [%]	ψ_{TV2} [%]	ψ_{TV} [%]	$\psi_{\Delta T,v}$ [%]
NH_3	5.14	0.38	17.6	20	37.62	10	0.22	6.57	6.7	5.57
CO_2	2.83	0.23	16.13	22.165	38.3	4.25	6.48	22.6	29.1	2.64

	$\dot{Q}_v$ [kW/kWheat]	$\dot{m}_1$ [g/kJheat]	$\dot{m}_2$ [g/kJheat]	$\dot{W}_{cp1}$ [kW/kWheat]	$\dot{W}_{cp2}$ [kW/kWheat]	t_4 [^{o}C]
NH_3	0.74	0.75	0.83	0.156	0.184	112
CO_2	0.58	5	8	0.232	0.327	81

The second throttling has the greatest destructive effect (ψ_{TV2}) (Table 1), a fact due to the large expansion ratio from the highest pressure to the lowest pressure in the system.

To reduce the destructive effect of the second throttling, it is proposed that the intermediate cooling be carried out by heat and mass transfer in an intermediate cylinder by immersing the gas discharged from the first stage in the saturated liquid corresponding to the intermediate pressure. In this way, saturated liquid is throttled from the intermediate pressure to the vaporization pressure in the second throttling valve.

2.2 Intermediate cooling by immersing the discharged gas from the first stage in saturated liquid at intermediate pressure

The diagram of the two-stage refrigeration system with flash intermediary cooling is shown in figure 2

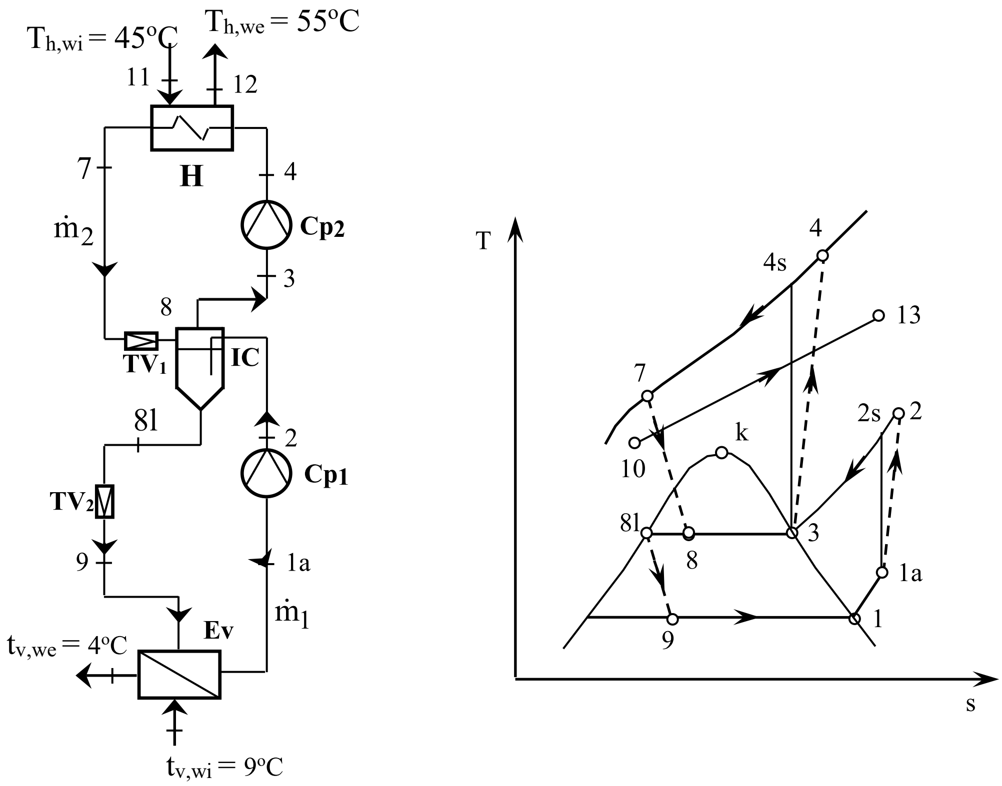

Fig. 2,a Fig. 2,b

Fig. 2. Two-stage refrigeration system with flash intermediary cooling. 2,a: Scheme of the system; 2,b: T-s diagram for CO_2 system

The results of the exergy analysis performed at the level of the refrigeration system in two stages with flash intermediary cooling are shown in table 2

Table 2

Two-stage refrigeration system with flash intermediary cooling (Figure 2)

$t_0 = 25^oC$; $t_{h,wi} = 45^oC$; $t_{h,we} = 55^oC$; $t_{v,wi} = 9^oC$; $t_{v,we} = 4^oC$; $\Delta T_P = 5K$;

$\eta_{scp} = 0.8$; $p_{g,CO_2} = 125bar$

	COP	η_{ex}	ψ_{cp1} [%]	ψ_{cp2} [%]	ψ_{cp} [%]	$\psi_{\Delta T,h}$ [%]	ψ_{TV1} [%]	ψ_{TV2} [%]	ψ_{TV} [%]	$\psi_{\Delta T,v}$ [%]
NH_3	5.79	0.42	15.29	19.35	34.64	10.43	2.45	1.652	4.10	6.4
CO_2	3.16	0.26	10.92	24.26	35.19	2.36	25.36	6.18	31.54	3

	$\dot{Q}_v$ [kW/kWheat]	$\dot{m}_1$ [g/kJheat]	$\dot{m}_2$ [g/kJheat]	$\dot{W}_{cp1}$ [kW/kWheat]	$\dot{W}_{cp2}$ [kW/kWheat]	t_4 [^{o}C]
NH_3	0.771	0.67	0.85	0.133	0.173	102.42
CO_2	0.617	4	12	0.158	0.355	69.46

Each of the two systems, one operating with NH_3 and the other with CO_2, improved their performance.

When operating with immersion intercooling, despite a superior heater and cooler performance, the high exergy destruction in the first throttling valve makes the CO_2 refrigeration system less efficient than the NH_3 one.

It is noted that under the conditions where the temperature difference at Pinch and the pressure in the heater are given, the states at the inlet and outlet of the first throttling valve are specified. Under these conditions, the only way to reduce the exergy destruction in this device is to reduce the mass flow rate expanded to the valve. This can be achieved by increasing the enthalpy at the entrance to the heater, which can be achieved by replacing the flash intermediate cooling by an intermediate cooling, by injecting saturated gas at the intermediate pressure into the gas stream discharged by the first compression stage.

2.3 Two-stage refrigerating system with intermediate cooling by injecting saturated gas at intermediate pressure into the discharged gas stream from the first compression stage

To end the intermediate cooling at a temperature higher than the saturation temperature corresponding to the intermediate pressure, the saturated gas fraction is mixed with the compressed gas discharged by the first compressor (Fig. 3). The temperature of the gas discharged by the second stage compressor increases, and so does its enthalpy.

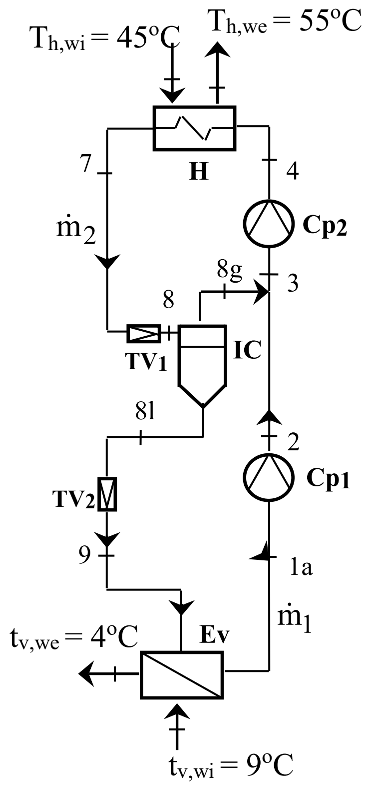

Fig. 3. Two-stage refrigeration plant with intermediate cooling by injection of saturated gas at intermediate pressure

This scheme preserves the advantage of expansion in the second throttling valve of the fraction of saturated liquid corresponding to the intermediate pressure. The specific heating load of the heater on the side of the refrigerant increases leading to the decrease of the mass flow corresponding to a quantity of 1 kW of transferred heat.

The results of the exergetic analysis are presented in table 3.

Table 3

Two-stage refrigeration system with intermediary cooling achieved by injecting saturated gas at intermediate pressure into the discharged gas stream from the first compression stage (Figure 3).

$t_0 = 25^{\circ}C$; $t_{h,wi} = 45^{\circ}C$; $t_{h,we} = 55^{\circ}C$; $t_{v,wi} = 9^{\circ}C$; $t_{v,we} = 4^{\circ}C$; $\Delta T_P = 5K$;

$\eta_{scp} = 0.8$; $p_{g,CO_2} = 125 bar$

	COP	η_{ex}	ψ_{cp1} [%]	ψ_{cp2} [%]	ψ_{cp} [%]	$\psi_{\Delta T,h}$ [%]	ψ_{TV1} [%]	ψ_{TV2} [%]	ψ_{TV} [%]	$\psi_{\Delta T,v}$ [%]
NH_3	5.58	0.41	14.64	19.36	34	14.25	2.12	1.58	3.7	6.13
CO_2	3.56	0.29	12.75	22.11	34.86	6.70	17.23	7.22	24.46	3.57

	$\dot{Q}_V$ [kW/kWheat]	$\dot{m}_1$ [g/kJheat]	$\dot{m}_2$ [kg/kJheat]	$\dot{W}_{cp1}$ [kW/kWheat]	$\dot{W}_{cp2}$ [kW/kWheat]	t_4 [^{o}C]
NH_3	0.764	0.67	0.76	0.132	0.184	152
CO_2	0.653	4.08	7.56	0.167	0.297	89

The decrease in mass flow rate corresponding to the second compression stage diminishes the exergy destruction in the first throttling valve. The cycle efficiency increases and is the highest compared to the other CO_2 schemes studied.

For NH_3, the discharge temperature t_4=152ºC exceeds the admissible value, which eliminates this option. The scheme for CO_2 with intercooling by injection of saturated gas at intermediate pressure will be compared with the best scheme for NH_3 which is with flash intercooling.

Apart from the fact that the exergetic analysis specifies the way forward for improving the structure of the technological scheme, it also highlights other important aspects:

a) Exergy destruction due to finite temperature difference heat transfer in the cooler (evaporator) is less for CO_2 compared to NH_3. This destruction related to vaporization temperatures can be further reduced in the case of CO_2. When using CO_2 as refrigerant there is no need for an intermediary cold carrying fluid as in the case of NH_3.
b) The negative effect of throttling in the case of CO_2 is more important in the high-pressure stage of the two-stage refrigeration system than that corresponding to the first stage. For NH_3, the negative effect of throttling is small, in both stages, compared to the case of CO_2.
c) Exergy destruction in the compression process is comparable in the case of NH_3 and CO_2.

3. Conclusions

A system that provides two products – cold and heat – was studied comparatively, for CO_2 and NH_3.

The exergetic analysis revealed that in the case of CO_2, the throttling process used to lower the temperature of the refrigerant is responsible for the low efficiency of the refrigeration cycle. On the other hand, CO_2 operation is less destructive in the heater and vaporizer. The exergy destruction associated with the cooling process can be reduced even further if it is noted that CO_2 does not need to transfer heat via an intermediary cold-carrying agent.

By quantifying and locating each one of the exergy destructions, exergy analysis specifies the direction to follow for improving the system structure.

In this way the two-stage refrigeration system with CO_2 was improved step by step starting from an intermediary cooling scheme, by injecting into the gas discharged by the first compression stage, a two-phase cold stream at the intermediate pressure. Finally, the best performing scheme was characterized by intermediate cooling by injection of saturated gas at intermediate pressure.

All these schemes of two-stage refrigeration plants with CO_2 have been improved by continuously aiming to reduce the exergy destruction in the throttling valves.

If the cold room is also considered, and if the behavior of the system when operating in both stages with NH_3 or CO_2 is compared, it appears advantageous to connect the two refrigerants in a cascade system with NH_3 in the upper cascade**.**

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