A FEASIBILITY ANALYSIS OF REPLACING THE STANDARD AMMONIA REFRIGERATION DEVICE WITH THE CASCADE NH_3/CO_2 REFRIGERATION DEVICE IN THE FOOD INDUSTRY

by

Dennis JANKOVICH [a] * ***and Kresimir OSMAN*** [b]

[a] Gavrilović Ltd., Petrinja, Croatia
[b] Faculty of Mechanical Engineering and Naval Architecture, University of Zagreb, Zagreb, Croatia

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The thermodynamic analysis demonstrates the feasibility of replacing the standard ammonia refrigeration device with the cascade NH_3/CO_2 refrigeration device in the food industry. The main reason for replacement is to reduce the total amount of ammonia in spaces like deep-freezing chambers, daily chambers, working rooms, and technical passageways. An ammonia-contaminated area is hazardous to human health and the safety of food products. Therefore, the preferred reduced amount of ammonia is accumulated in the central refrigeration engine room, where the cascade NH_3/CO_2 device is installed as well. Furthermore, the analysis discusses and compares two Carnot's refrigeration cycles, one for the standard ammonia device and the other for the cascade NH_3/CO_2 device. Both cycles are processes with two-stage compression and two-stage throttling. The thermodynamic analysis demonstrates that the selected refrigeration cycle is the most cost-effective process because it provides the best numerical values for the total coefficient of performance with respect to the observed refrigeration cycle. The chief analyzed influential parameters of the cascade device are: total refrigeration load, total compressor power, mean temperature of the heat exchanger, evaporating and condensing temperature of the low-temperature part.

Key words: *natural refrigerants, ammonia, carbon dioxide, refrigeration device, refrigeration factor and load, reactive power, heat exchanger mean temperature, evaporating and condensing temperature*

Introduction

In Earth's atmosphere, the impact of adverse emissions from most used refrigerants and refrigeration devices has been greatly increased, which also significantly increases the risk of the *ozone depletion effect* [1] and the *greenhouse effect* [2]. In developed industrial countries, these phenomena have accelerated the retrofitting to natural alternative refrigerants such as ammonia (NH_3) and carbon dioxide (CO_2). In modern food industry, it is essential to use such substances that are harmless or insignificantly harmful both to human health, and for the refrigerated food product. Because of its low production price, a high and efficient heat transmission coefficient, specificity and common use in the management of various NH_3 refrigerated processes for many years, NH_3 [3] is being used again as a very interesting refrigerant. It clearly arises from the foregoing that one of the possible effective solutions to the presented problems

* Corresponding author; e-mail: dennis.jankovich@gmail.com

is to introduce a new cascade NH_3/CO_2 refrigeration device [4] in the existing NH_3 refrigeration system, thus substantially decreasing the total amount of hazardous refrigerants such as NH_3 across the entire NH_3 pipeline spreading through the entire plant. By centralizing the NH_3 portion of the cascade device, NH_3 will be restricted to the central refrigeration engine room, which is physically separated from the rest of the manufacturing plant. This significantly increases the safety of employees, property, and food products being refrigerated. Using NH_3/CO_2 refrigeration device the achievable low temperature will be between –40 °C and –50 °C [5]. To effectively compare and assess such newly introduced refrigeration process, it is necessary to test the presumed main influential parameters for NH_3 and CO_2 as refrigerants. The main influential parameters are: condensation pressure and evaporation pressure ratio p_c/p_e, superheated vapor temperature T_4, condensing temperature T_c, evaporating temperature T_e, total condensation load Q_c, total compressor power P, total refrigeration load Q_0, and total coefficient of performance for the observed refrigeration cycle ε. This research aims to provide quick and accurate feedback regarding the replacement of the standard NH_3 refrigeration device by a cascade NH_3/CO_2 refrigeration device.

Motivation

The authors' primary motive was to define effective criteria for selecting NH_3 and CO_2 [6] as natural refrigerants that do not deplete the ozone layer around the Earth, do not produce (or produce little of) the greenhouse effect [7], and are already being used or may be used in large existing NH_3 refrigeration systems in almost all branches of the food industry. The paper focuses on defining certain criteria for their use. The general criteria to be met at all times by NH_3 and CO_2 as the selected natural refrigerants are: the refrigerant evaporating temperature must be as low as possible at a saturation pressure of 1 bar (security against potential air penetration into the refrigeration device), the condensation pressure must range between 15 and 20 bar at a condensing temperature of +30 ºC, the refrigerant density must be as high as possible at the evaporating temperature, the critical temperature must be high to give us more freedom in working parameters selection [8], the refrigerant should not be corrosive and should not dissolve materials, the refrigerant should be chemically stable because its decomposition into other chemical compounds momentarily results in a change in the values of all parameters, the refrigerant should be nonflammable and non-explosive, the refrigerant should not form a homogeneous blend with a lubricant, and the refrigerant should be easily detected in case of leakage.

Definition of the problem

The present-day use of refrigerants in refrigeration and air conditioning engineering, especially chlorofluorocarbons (CFC) and hydrochlorofluorocarbons (HCFC) for their substantial impact on ozone depletion, and hydrofluorocarbons for their substantial impact on the greenhouse effect [2, 7, 9], is the subject of extensive discussions initiated by a number of adverse environmental problems. These discussions should significantly affect human awareness and ultimately result in more efficient minimization of emissions of these harmful refrigerants to the atmosphere. In many cases, inadequate maintenance of refrigeration and air conditioning devices results in the emission of excessive amounts of such refrigerants directly into the environment. Their purchase price was low and these refrigerants were not considered harmful to the environment, so their emission was common. However, this study gives us now reliable information indicating that this type of refrigerant causes considerable damage to Earth's ozone layer [1] and that their emission into the air in our environment is prohibited.

Natural alternative refrigerants are being used again within large industrial-process and food production refrigeration systems, primarily NH_3 and CO_2.

Background – natural refrigerants and ecology

Thanks to the ability to compare different refrigerants against ozone depletion [1, 10], the dimensionless *ozone depletion potential* (ODP) number has been introduced as a function of the chlorine (Cl) and bromine (Br) releasing ability and weather stability in the stratospheric layers of the atmosphere. The ODP tells us how much ozone will be depleted by the release of a particular refrigerant as compared with the same amount of trichlorofluoromethane ($CFCl_3$ – R11, ODP = 1). To compare the impacts of different refrigerants on producing the greenhouse effect [7, 11], the dimensionless *global warming potential* (GWP) number has been introduced to indicate the extent of the impact of a particular refrigerant released in the stratospheric layers of the atmosphere on the production of the greenhouse effect as compared to the same amount of CO_2 (GWP = 1).

As a natural alternative refrigerant, anhydrous ammonia (NH_3 – R717) is suitable to replace CFC and HCFC in modern refrigeration equipment. From the environmental viewpoint, NH_3 is the most acceptable refrigerant and long-term alternative [12, 13] because its release into the atmosphere has no impact on ozone depletion (ODP = 0) or the greenhouse effect (GWP = 0). Thermodynamically, NH_3 is the best refrigerant and almost irreplaceable in large industrial facilities. The NH_3 [14, 15] has a very high level of latent evaporation heat r and is used up to an evaporating temperature T_e = –50 °C. Its critical temperature and pressure are very high, T_{Cr} = +132.4 °C, and p_{Cr} = 112.97 bar. Relative to other refrigerants, NH_3 has a high heat transfer coefficient α as a result of its low kinematic viscosity and almost the highest specific refrigeration load q_0 (only that of CO_2 is higher). It is very cheap to produce (synthesis of nitrogen and hydrogen) and it is used for refrigeration loads in excess of 30 kW.

The CO_2 – R744 [15, 16] is another very important natural alternative refrigerant, affordable and easy to produce. The CO_2 has no impact on ozone depletion (ODP = 0), and an insignificant impact on the greenhouse effect (GWP = 1). The costs of producing CO_2 are low and no additional costs are incurred for its disposal. Although it provides considerable benefits as a refrigerant, the main reason why it is not widely used in practice are its unfavorable thermodynamic characteristics for standard refrigeration applications, which result in technical problems in device's performance. The CO_2 has a very high level of latent evaporation heat r, but also a very low critical temperature T_{Cr} = +31.1 ºC, combined with a very high critical pressure p_{Cr} = 74 bar. In case of one-stage systems, this requires trans critical working parameters with a condensation pressure in excess of 100 bar. Its volumetric refrigeration load is 5-8 times higher than in NH_3, which substantially reduces the dimensions of the device. The present-day use of CO_2 is acceptable both in industry and in large commercial refrigeration systems [15] where it is used as a refrigerant in cascade NH_3/CO_2 refrigeration devices, in the lower cascade (at temperatures between –10 ºC and –50 ºC). The CO_2 provides great electricity savings, its energy efficiency is excellent, and it has a good heat transfer coefficient α.

Description of the testing methodology

In most cases, using the cascade NH_3/CO_2 refrigeration device, the achievable low temperature and the coefficient of performance (*COP*) of the cascade system are the best among all the systems when the evaporating temperatures are below –40 °C [5]. Such low refrigeration temperatures are used for laboratory purposes, for tissue transplantations in medicine, for the purposes of various biochemical processes, for cry freezing researches [17], to obtain liquefied gases (nitrogen, helium) and in the process and food industries. The proposed and applied testing methodology consists of a comparison between two cycles: two-stage-compression cycle and two-stage-throttling Carnot's refrigeration cycle. The first refrigeration cycle is a process typical for the standard NH_3 refrigeration device, while the other one is typical for the cascade NH_3/CO_2 refrigeration device.

The testing methodology applied must meet three basic requirements [18, 19]:

- it is necessary to use such a refrigerant that has high density because it provides a higher mass flow rate m, which ultimately increases the refrigeration load q_0,
- there is no need to refrigerate deeper than necessary because of the increase in total compressor power P in the form of technical work expended w (a drop in the value of the process coefficient of performance ε and specific refrigeration load q_0), and
- there is no need to heat more than necessary because of the increase in the compression ratio and a decrease in the value of the total refrigeration device *COP* ε:

$$\varepsilon = \frac{1}{\left(\dfrac{p}{p_0}\right)^{\frac{\kappa-1}{\kappa}}} - 1 \tag{1}$$

Equation (1) is used for the case of a Carnot refrigeration cycle with an ideal gas and is therefore applicable only approximately. Where the refrigeration temperature is extremely low, it is unable to use standard refrigerants [20]. Because of that, must be used refrigerants which density levels are very high at such temperature and their specific volumes are low. Therefore, it is used CO_2 as the refrigerant in the lower cascade and NH_3 in the upper cascade. To attain an extremely low refrigeration temperature, it is used the cascade NH_3/CO_2 refrigeration device where two independent refrigeration (circuits) devices [18, 19] are interconnected using a special heat exchanger. Heat is exchanged in this exchanger because the total refrigeration load in upper cascade $Q_{0,\mathrm{UC}}$ equals the total condensation load (heat) in lower cascade $Q_{\mathrm{c,LC}}$, *i. e.* $T_{\mathrm{e,UC}} < T_{\mathrm{c,LC}}$ and $\Delta T = T_{\mathrm{c,LC}} - T_{\mathrm{e,UC}} = 7 - 10$ °C should apply. The temperature difference results in the transfer of heat within the heat exchanger, which thermally connects the upper and lower cascades of the cascade NH_3/CO_2 refrigeration device. The total condensation load of the lower cascade equals the total refrigeration load of the upper cascade $Q_{\mathrm{c,LC}} = Q_{0,\mathrm{UC}}$. As mentioned, the lower cascade uses CO_2 as refrigerant evaporating at a temperature of $T_{\mathrm{e,LC}} = -30/-20$ °C and attaining the required refrigeration load $Q_{0,\mathrm{LC}} = Q_{0,\mathrm{c}}$, while the upper cascade uses NH_3 as refrigerant with a wide saturated area, so $T_{\mathrm{Cr}} \geq T_{\mathrm{en}}$, [21, 22] applies to it. Following the two-stage compression process within the lower cascade of the device [23], CO_2 is condensed to the selected condensing temperature of the lower cascade $T_{\mathrm{c,LC}} = -2$ °C. The CO_2 is then subjected to two-stage throttling at $h = const.$ and brought to an evaporated state at the selected evaporating temperature $T_{\mathrm{e,LC}} = -30/-20$ °C. The NH_3 is used as refrigerant in the upper cascade, where the selected evaporating temperature is $T_{\mathrm{e,UC}} = -10$ °C. It is required to have $p_{\mathrm{e,LC}} \approx 1$ bar to have compressors with real working volumes. The useful consequences of the requirements are reflected in the fact that the compression ratio in both cascades will be lower than the borderline ratio *i. e.* it will be $p_c/p_e \leq 12$, while the maximum temperature will meet the $T_{4,\mathrm{UC}} \leq (T_{\mathrm{lim}} = +140$ °C) requirement. In addition to basing the proposed testing methodology on a comparison between two Carnot's refrigeration cycles with two-stage compression and two-stage throttling, it is also necessary to develop within the scope of such methodology a dedicated process to optimize the conditions for efficient control of the key influential parameters within the analyzed refrigeration cycle of the newly introduced cascade NH_3/CO_2 refrigeration device.

The requirements to be met for proper functioning of such device are:

– it is necessary to have a sufficiently low refrigeration temperature because the primary objective is to reduce the total amount of NH_3 across the refrigeration system,
– the total condensation load (heat) of the lower cascade $Q_{c,LC}$ must equal the total refrigeration load of the upper cascade $Q_{0,UC}$:

$$Q_{c,LC} = Q_{0,UC} \tag{2}$$

– there must be a difference in temperature on the heat exchanger connecting the upper and lower cascades of the refrigeration device. It ensures heat exchange on a natural refrigeration basis *i. e.* based on heat transfer from the higher-temperature tank to the lower-temperature tank, ranging between 7 and 10 °C:

$$\Delta T = T_{c,LC} - T_{e,UC} = 7 - 10\ °C \tag{3}$$

– the $Q_{c,UC}$ eq. (2) is transferred to the environment:

$$Q_{c,UC} = P_{1,LC} + P_{2,LC} + P_{1,UC} + P_{2,UC} + Q_{0,LC} \tag{4}$$

– the $q_{0,UC} = q_{0,LC}$ does not need to apply because the mass flow rates may differ,
– maximum savings in technical work expended w need to be ensured, which is why the isobaric intercooling pressure in the separator equals:

$$p_s = \sqrt{p_c p_e} \tag{5}$$

– the selected additional irreversibility for the cascade refrigeration device contemplated herein is:

$$\Delta T = T_{c,LC} - T_{e,UC} = 8\ °C \tag{6}$$

The prescribed temperature difference of 8 ºC between the two cycles in the cascade system is a matter of the heat exchanger capacity.

Thermodynamic analysis of the results

The thermodynamic analysis considered a Carnot's refrigeration cycle with two-stage compression and two-stage throttling. This study provide a comparison between the obtained numerical values of the key influential parameters for the standard NH_3 refrigeration cycle and the cascade NH_3/CO_2 refrigeration cycle. The authors carried out a thermodynamic analysis for two cooling processes (meat and fruit berries). The both processes are shown in tables, one for the standard NH_3 refrigeration device and the other for the cascade NH_3/CO_2 refrigeration device (for the both cascades).

Analysis of the standard NH_3 refrigeration cycle

Figure 1(a) provides a schematic presentation of the standard NH_3 refrigeration device, while fig. 1(b) presents the standard NH_3 refrigeration cycle with two-stage compression and two-stage throttling in a *T-s* diagram. The two-stage-compression and two-stage--throttling process was selected to analyze the standard NH_3 refrigeration cycle.

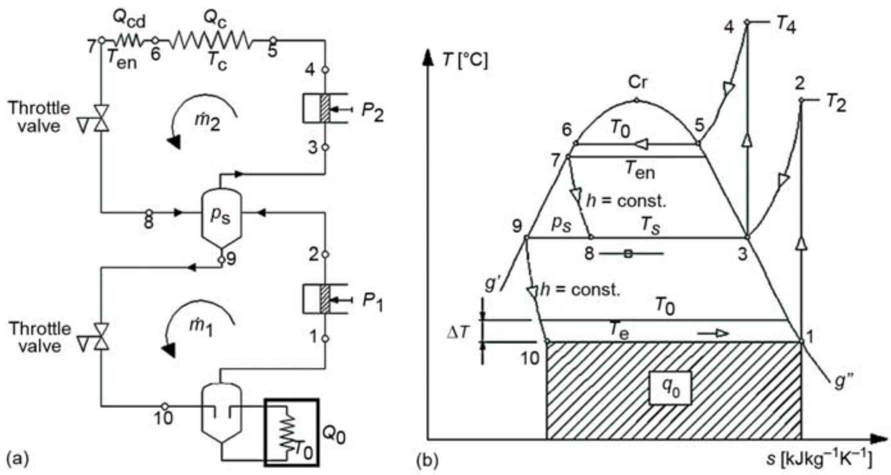

Figure 1. (a) Scheme of the standard NH_3 device; (b) The standard NH_3 cycle in a *T-s* diagram

Table 1. The process points for the standard NH_3 cycle

Point	T [°C]	p [bar]	h [kJkg^{-1}]	s [kJkg^{-1}K^{-1}]
1	– 30/–20	1.2/1.9	1343.1/1356.9	3.6/3.5
2	+57/+34.6	4.3/4.3	1515.8/1463.4	3.6/3.5
3	0/0	4.3/4.3	1379.1/1379.1	3.2/3.2
4	+73/+73	11.7/11.7	1518.6/1518.6	3.2/3.2
5	+30/+30	11.7/11.7	1396.6/1396.6	2.8/2.8
6	+30/+30	11.7/11.7	264.8/264.8	–0.9/–0.9
7	+25/+25	11.7/11.7	241.1/241.1	–1.1/–1.1
8	0/0	4.3/4.3	241.1/241.1	–
9	0/0	4.3/4.3	121.8/121.8	–1.4/–1.4
10	–30/–20	1.2/1.9	121.8/121.8	–

Table 1 presents two cases: the process points for the standard NH_3 cycle [13] for evaporating temperatures –30/–20 °C, while tab. 2 presents preset and calculated parameters for the standard NH_3 cycle for the both evaporating temperatures. In this cycle the influential parameters are: total refrigeration load of the process Q_0, eq. (9), NH_3 condensing temperature T_c, eq. (10), environmental temperature T_{en}, eq. (11), and NH_3 evaporating temperature T_e, eq. (12). Table 2 uses a thermodynamic analysis process to present the obtained numerical values for the two main and most influential parameters of this cycle – total compressor power of the device P_a eq. (25) and total coefficient of performance ε_1 eq. (26). These values are used below to compare the standard NH_3 refrigeration cycle with the cascade NH_3/CO_2 refrigeration cycle.

Analysis of the refrigeration cycle for the cascade device

The entire cascade refrigeration cycle with two-stage compression and two-stage throttling was thoroughly thermodynamically analyzed for the upper and the lower cascades of the refrigeration device. The high-temperature part uses NH_3 in an evaporation range of –10 °C, while the low-temperature part uses CO_2 for evaporating temperatures –30/–20 °C. A cascade refrigeration device so designed uses a minimum amount of NH_3 because it is only used in the high-temperature part. The crucial role of CO_2 is to replace hazardous NH_3 for safety reasons in areas where people spend time and where refrigerated final products are stored [25, 26]. In the thermodynamic analysis, the total coefficient of performance ε_2, eq. (64), of the cascade NH_3/CO_2 refrigeration device was analyzed by changing the evaporating temperature value in the lower cascade $T_{e,LC}$, eq. (35). This provides a good theoretical basis for convenient and optimal operation of the cascade NH_3/CO_2 refrigeration device. Figure 2(a) provides a schematic presentation of the cascade NH_3/CO_2 refrigeration device, while fig. 2(b) presents the cascade NH_3/CO_2 refrigeration cycle with two-stage compression and two-stage throttling in a *T-s* diagram.

Table 2. Preset and calculated parameters for the standard NH_3 cycle [24]

Equation number	Preset and calculated parameters		Equation number	Preset and calculated parameters	
	Equation	Value		Equation	Value
(7)	$p_c/p_e \leq 12$	9.8 ≤ 12/6.2 ≤ 12 bar	(17)	$x = (h_9 - h_2)/(h_8 - h_3)$	1.2/1.2
(8)	$T_4 \leq T_{lim}$	73 ≤ 140/73 ≤ 140 °C	(18)	$m_1 = Q_0/q_0$	1.9/1.9 kg/s
(9)	Q_0	2.3/2.3 MW	(19)	$m_2 = m_1 x$	2.3/2.2 kg/s
(10)	T_c	+30/+30 °C	(20)	$w_1 = h_2 - h_1$	172.8/106.5 kJ/kg
(11)	T_{en}	+25/+25 °C	(21)	$w_2 = h_4 - h_3$	139.5/139.5 kJ/kg
(12)	T_e	–30/–20 °C	(22)	$Q_c = m_2 q_c$	2947.1/2804.3 kW
(13)	$p_s = (p_c p_i)^{1/2}$	3.7/4.7 bar	(23)	$P_1 = m_1 w_1$	325.4/198.3 kW
(14)	p_{sat}	4.3/4.3 bar	(24)	$P_2 = m_2 w_2$	321.7/306.1 kW
(15)	$q_0 = h_1 - h_{10}$	1221.3/1235.1 kJ/kg	(25)	$P_a = P_1 + P_2$	647.1/504.5 kW
(16)	$q_c = h_4 - h_7$	1277.6/1277.6 kJ/kg	(26)	$\varepsilon_1 = Q_0/P_a$	3.6/4.6

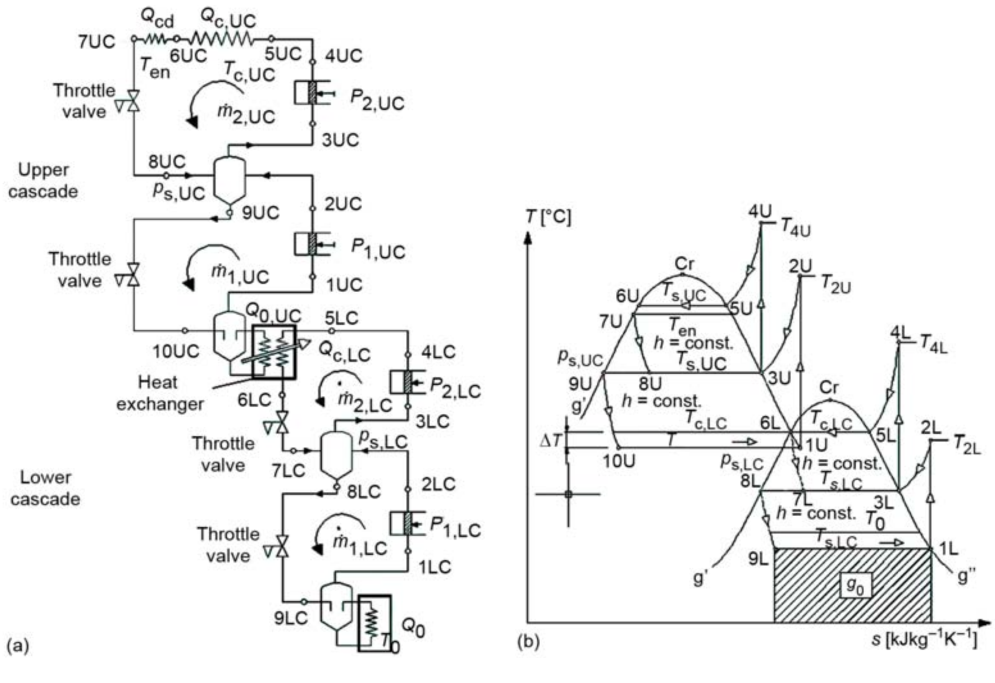

Figure 2. (a) Scheme of the cascade NH_3/CO_2 device; (b) The cascade NH_3/CO_2 cycle in a *T-s* diagram

Thermodynamic estimation for the lower cascade

The heat exchanger is the most important part of the cascade NH_3/CO_2 refrigeration device whose working parameters directly affect the *COP* characteristic of the entire device. When used as a refrigerant, CO_2 circulates the low-temperature part and is used as a refrigerant for normal cooling of space like other standard refrigerants [6, 27, 28]. The issue of safety with regard to a potential disaster resulting from NH_3 leakage was resolved by installing and placing the NH_3 circuit of the upper cascade only within the central refrigeration engine room area, which must be physically separated from the rest of the plant within the food manufacturing facility. The total compressor power of the cascade device P_c, eq. (62), consists of two parts: the total compressor power in the low-temperature part P_{LC}, eq. (46), with CO_2 as refrigerant [29] and total compressor power in the high-temperature NH_3 part P_{UC}, eq. (61). Table 3 presents all process points of the NH_3 refrigeration cycle in the upper cascade and all process points of the CO_2 refrigeration cycle in the lower cascade of the cascade NH_3/CO_2 refrigeration device. Table 4 provides preset and calculated parameters of the cascade cycle for the lower cascade of the cascade NH_3/CO_2 refrigeration device.

Table 3. The process points of the cascade cycle for lower and upper cascades

Lower cascade – CO_2					Upper cascade – NH_3				
Point	T [°C]	p [bar]	h [kJkg^{-1}]	s [kJkg^{-1}K^{-1}]	Point	T [°C]	p [bar]	h [kJkg^{-1}]	s [kJkg^{-1}K^{-1}]
1LC	–30/–20	14.3/ 19.7	358.3/436.9	1.5/1.9	1UC	– 10/–10	2.9/2.9	1369/*	3.3/3.3
2LC	–5/–10	21.6/ 21.6	375.0/440.0	1.5/1.9	2UC	+42/+42	6.2/6.2	1468.7/*	3.3/3.3
3LC	–17/–17	21.6/ 21.6	358.2/436.6	1.4/1.9	3UC	+10/+10	6.2/6.2	1387.2/*	3.1/3.1
4LC	+23/+10	33.1/ 33.1	393.0/455.0	1.4/1.9	4UC	+57/+57	11.7/11.7	1474.2/*	3.1/3.1
5LC	–2/–2	33.1/ 33.1	353.7/432.0	1.3/1.9	5UC	+30/+30	11.7/11.7	1396.6/*	2.8/2.8
6LC	–2/–2	33.1/ 33.1	116.9/195.2	0.5/1.0	6UC	+30/+30	11.7/11.7	264.8/*	-0.9/-0.9
7LC	–17/–17	21.6/ 21.6	116.9/195.2	–	7UC	+25/+25	11.7/11.7	241.1/*	-1.1/-1.1
8LC	–17/–17	21.6/ 21.6	83.1/160.9	0.3/0.9	8UC	+10/+10	6.2/6.2	241.1/*	–
9LC	–30/–20	14.3/ 19.7	83.1/160.9	–	9UC	+10/+10	6.2/6.2	169.4/*	–1.3/–1.3
–	–	–	–	–	10UC	–10/–10	2.9/2.9	169.4/*	–

(*) the same value

The heat exchanger requires the amount of NH_3 $m_{1,UC}$ eq. (54) to refrigerate and condense the amount of CO_2 $m_{2,LC}$ eq. (40), provided that there is no heat loss and that heat transfer is only possible from a point with a higher CO_2 condensing temperature $T_{c,LC}$ eq. (33) to a point of a lower NH_3 evaporating temperature $T_{e,UC}$ eq. (50).

For evaporating temperatures –30/–20 °C, the total mass flow rate of CO_2 in the lower cascade is 3.8/4.0 times higher than the total mass flow rate of NH_3 in the upper cascade, which results in substantial reduction of the liquid phase of NH_3 within the cascade refrigeration device compared to the standard NH_3 refrigeration device [30]. Due to high latent evaporation heat r required to change the liquid phase of CO_2 to the vapor phase, heat transfer was significantly improved, while the effective area of the cascade refrigeration device's heat exchanger was optimized.

Table 4. Preset and calculated parameters of the cascade cycle for the lower cascade [24]

Equation number	Preset and calculated parameters		Equation number	Preset and calculated parameters	
	Equation	Value		Equation	Value
(27)	$p_{c,LC}/p_{e.LC} \leq 12$	2.1 ≤ 12/2.3 ≤ 12 bar	(37)	$q_{c.LC} = h_{4LC} - h_{6LC}$	276.1/259.8 kJ/kg
(28)	$p_{c.UC}/p_{e.UC} \leq 12$	4.0 ≤ 12/4.1 ≤ 12 bar	(38)	$x_{LC} = \frac{h_{8LC} - h_{2LC}}{h_{7LC} - h_{3LC}}$	1.2/1.2
(29)	$T_{4.UC} \leq T_{lim}$	57 ≤ 140/57 ≤ 140 ºC	(39)	$m_{1.LC} = Q_{0.LC}/q_{0.LC}$	8.4/8.3 kg/s
(30)	$p_{s.LC} = (p_{c.LC}\, p_{e.LC})^{1/2}$	21.7/21.7 bar	(40)	$m_{2.LC} = m_{1.LC}\, x_{LC}$	10.1/9.6 kg/s
(31)	$p_{sat.LC}$	21.6 bar	(41)	$w_{1.LC} = h_{2LC} - h_{1.LC}$	16.7/3.1 kJ/kg
(32)	$Q_0 = Q_{0.c}$	2.3/2.3 MW	(42)	$w_{2.LC} = h_{4LC} - h_{3LC}$	34.8/18.4 kJ/kg
(33)	$T_{c.LC}$	–2/–2 °C	(43)	$Q_{c.LC} = m_{2.LC}\, q_{c.LC}$	2791.4/2503.4 kW
(34)	$T_{0.LC}$	–25/–25 °C	(44)	$P_{1.LC} = m_{1.LC}\, w_{1.LC}$	139.2/25.9 kW
(35)	$T_{e.LC}$	–30/–20 °C	(45)	$P_{2.LC} = m_{2.LC}\, w_{2.LC}$	352.2/177.5 kW
(36)	$q_{0.LC} = h_{1.LC} - h_{9LC}$	275.3/275.9 kJ/kg	(46)	$P_{LC} = P_{1.LC} + P_{2.LC}$	491.4/203.4 kW

Thermodynamic estimation for the upper cascade

Table 5 presents preset and calculated values of all relevant influential parameters for the upper cascade and the main influential parameters for the entire cascade NH_3/CO_2 refrigeration device. When combined, the selected NH_3 evaporating temperature $T_{e,UC}$ eq. (50) in the high-temperature part and the selected CO_2 condensing temperature $T_{c,LC}$ eq. (33) in the low-temperature part must provide an optimal *COP* value for the cascade device. This is exactly why the optimal mean temperature of the common heat exchanger $T_{m,opt}$ eq. (63) should be carefully selected.

Table 5. The preset and calculated values of the cascade cycle for the upper cascade [24]

Equation number	Preset and calculated values		Equation number	Preset and calculated values	
	Equation	Value		Equation	Value
(47)	$p_{s.UC} = (p_{c.UC}\, p_{e.UC})^{1/2}$	5.8/5.8 bar	(56)	$w_{1.UC} = h_{2.UC} - h_{1.UC}$	99.7/99.7 kJ/kg
(48)	$p_{sat.UC}$	6.2/6.2 bar	(57)	$w_{2.UC} = h_{4.UC} - h_{3.UC}$	87.0/87.0 kJ/kg
(49)	$T_{c.UC}$	+30/+30 °C	(58)	$Q_{c.UC} = m_{2.UC}\, q_{c.UC}$	3252.7/2917.3 kW
(50)	$T_{e.UC}$	–10/–10 °C	(59)	$P_{1.UC} = m_{1.UC}\, w_{1.UC}$	232.1/208.1 kW
(51)	$q_{0.UC} = h_{1.UC} - h_{10.UC}$	1199.5/1199.5 kJ/kg	(60)	$P_{2.UC} = m_{2.UC}\, w_{2.UC}$	229.4/205.7 kW
(52)	$q_{c.UC} = h_{4.UC} - h_{7.UC}$	1233.2/1233.2 kJ/kg	(61)	$P_{UC} = P_{1.UC} + P_{2.UC}$	461.4/413.8 kW
(53)	$x_{UC} = \frac{h_{9UC} - h_{2UC}}{h_{8UC} - h_{3UC}}$	1.1/1.1	(62)	$P_c = P_{LC} + P_{UC}$	952.8/617.2 kW
(54)	$m_{1.UC} = Q_{0.UC}/q_{0.UC}$	2.3/2.1 kg/s	(63)	T_m	–6/–6 ºC
(55)	$m_{2.UC} = m_{1.UC}\, x_{UC}$	2.6/2.4 kg/s	(64)	$\varepsilon_2 = Q_0/P_c$	2.4/3.7

In general there are two options for selecting a mean temperature for the heat exchanger T_m. The first one is focused on maximizing the device's *COP*, while the other one is focused on equalizing the pressure ratios for both compressors in the upper and lower cascades of the refrigeration device, to maximize savings in technical work expended w, and compressor power P. This thermodynamic analysis uses the second option. The *COP* characteristic values for the cascade device may be simulated by using a series of equations (27-64) for NH_3 and CO_2.

Discussion of results

By comparing the refrigeration cycles of the standard NH_3 refrigeration device and the cascade NH_3/CO_2 refrigeration device, this study shows us that the total coefficient of performance for the first refrigeration device ε_1, eq. (26), exceeds the second one ε_2, eq. (64), by as much as 32,1%/18,3%. The same goes for the total compressor power used P_a, eq. (25), and P_c, eq. (62). Based on the cascade NH_3/CO_2 refrigeration device functional dependency and change of the evaporating temperature value in the lower cascade $T_{e,LC}$, eq. (35), the study shows that the value of the evaporating temperature $T_{e,LC}$ significantly affects to the change in the *COP* value, so the maximum *COP* value is attained at its optimal mean temperature of the heat exchanger T_m.

After determination of the evaporating temperature $T_{e,LC}$, it is able to calculate the refrigeration load $Q_{0,LC}$ of the cascade device. Table 3 presents the numerical values of all process point state sizes for the low-temperature part of the refrigeration cycle for CO_2 and the high-temperature part of the refrigeration cycle for NH_3 within the cascade device, at the mean temperature of the heat exchanger T_m, eq. (63), refrigeration load $Q_{0,UC}$, eq. (2), and heat load $Q_{c,LC}$, eq. (43), in the heat exchanger, and total compressor power of the upper and lower cascades of the refrigeration device P_c, eq. (62), [31]. As result of a change in the *COP* value in relation to the change in the evaporating temperature in the lower cascade $T_{e,LC,}$ the *COP* value initially rises as the $H_{e,LC}$ value rises. The functional dependency of the change in condensing temperature of the low-temperature part of the refrigeration device $T_{c,LC}$ and the insignificant change in the *COP* value is almost linear [32]. Finally, this comparison isn't addressing to the differences of heat exchangers and compressors in the system. The low temperature stage of the conventional system can be compared with CO_2 cycle of the cascade system because the total refrigeration load Q_0 = 2.3 MW and evaporating temperatures T_e = –30/–20 °C are the same in the both cases of comparison. But, the analysis doesn't show any advantage CO_2 cycle in relation to the size of compressor and heat exchanger.

Conclusions

Selecting the cascade NH_3/CO_2 refrigeration device is the more expensive type of refrigeration compared to the standard NH_3 refrigeration device because it requires more total compressor power P. In addition, the entire low-temperature part of the device where CO_2 circulates must be very well insulated, which ultimately additionally increases the cost of the final cascade refrigeration device version. However, the implementation of the new cascade NH_3/CO_2 refrigeration device in the existing large industrial NH_3 refrigeration systems in all food industry segments underscores the great advantage deriving from the reduced amount of NH_3 in circulation and the resulting safety issue. The cascade NH_3/CO_2 refrigeration device minimizes the amount of NH_3 in the high-temperature part of the device, while the low-temperature part of the device uses CO_2 only as a secondary refrigerant that eliminates hazardous NH_3 for safety reasons from areas where people spend time and where final products are refrigerated. As the evaporating temperature in the lower cascade $T_{e,LC}$ rises, so does the *COP* value. In the thermodynamic analysis [33], this study calculated and considered the *COP* values for the cascade NH_3/CO_2 refrigeration device by changing the evaporating temperature $T_{e,LC}$. This defined a theoretical base [34] to optimize the operating conditions of the new cascade NH_3/CO_2 refrigeration device.

Future research

As the potential direction of future research should be related to an optimization study for the energy input in the cascade NH_3/CO_2 refrigeration device. It is possible to carry out the optimization of the refrigeration process and related influential parameters. The deep-freezing process optimization of various food products must be in detail prepared. For this purpose, the authors propose the use of response surface method [35, 36] by which is possible to predict the unknown values of output parameters based on the known values of the input parameters and the ranking of their mutual influences. As regards everyday engineering issues, highly complex solutions often arise, where system response represents interaction between several influential parameters, which may be variable at the same time. This results in a need to take into account all parameters and conduct experiments for the purpose of testing to determine which of these parameters or their mutual combinations has a maximum or minimum influence on system response, and which empirical laws appear in such experiments. Thus obtained data from these parameters or their mutual combination of maximum or minimum influence the response of the system and empirical regularities in such experiments may occur. The case of two-factor experiment will be defined by two assumed influential input parameters: the mass of the individual food product m_p and the time (duration) of food product refrigeration process t_{RP}, while the assumed output parameter will be the refrigeration temperature of the food product T_{RP}. The case of three-factor experiment will be defined by three assumed influential input parameters, of which the first two ones are identical as in the first case, while the third one will be defined as temperature at the geometric center of the food product T_{GC}, and the assumed output parameter will be identical as in the first case. The both experiments will be performed in all possible combinations of assumed influential input and output parameters. These combinations of parameters should give at least one satisfactory offered model of a valid dependence. Strictly defined operating requirements and limits of the intervals containing the values of the selected influential parameters would maximize the cascade device's *COP* value and would maximize savings on technical work expended (compressor power). The results obtained represent a solid basis for further development of methodology for testing properties of the cascade NH_3/CO_2 refrigeration device in the food industry.

Nomenclature

Cr – critical point
h – specific enthalpy, [$kJkg^{-1}$]
m – mass flow rate, [kgs^{-1}]
m_p – mass of the individual food product, [kg]
m_1 – mass flow rate in the first compression stage, [kgs^{-1}]
m_2 – mass flow rate in the second compression stage, [kgs^{-1}]
P – total compressor power, [kW]
P_a – total compressor power of the NH_3 refrigeration device, [kW]
P_c – total compressor power of the cascade refrigeration device, [kW]
P_1 – total compressor power of the first compression stage, [kW]
P_2 – total compressor power of the second compression stage, [kW]
p_{Cr} – critical pressure, [bar]
p_c – condensing pressure, [bar]
p/p_0 – compression ratio, [–]
p_c/p_e – condensation pressure and evaporation pressure ratio, [–]
p_e – evaporating pressure, [bar]
p_s – calculated pressure in the separator (saved Δw_{max}), [bar]
p_{sat} – rounded p_s value to the standard saturated pressure at 0 °C, [bar]
$p_{sat,LC}$ – rounded $p_{s,LC}$ value to the standard saturated pressure at –17 °C, [bar]
$p_{sat,UC}$ – rounded $p_{s,UC}$ value to the standard saturated pressure at +10 °C, [bar]
Q_c – total condensation load, [kW]
Q_{cd} – condensation down heat, [kW]
Q_0 – total refrigeration load of the NH_3 refrigeration device, [kW]
$Q_{0,c}$ – total refrigeration load of the cascade refrigeration device, [kW]
q_c – specific condensation load, [$kJkg^{-1}$]

q_0 – specific refrigeration load, [kJkg^{-1}]
r – latent evaporation heat, [kJkg^{-1}]
s – specific entropy, [kJkg^{-1}K^{-1}]
$T_{\rm lim}$ – limited temperature in the superheated steam area, [ºC]
$T_{\rm c}$ – condensing temperature, [ºC]
$T_{\rm Cr}$ – critical temperature, [ºC]
$T_{\rm e}$ – evaporating temperature, [ºC]
$T_{\rm en}$ – environmental temperature, [ºC]
$T_{\rm GC}$ – temperature at the geometric center of the food product, [ºC]
$T_{\rm m}$ – mean temperature of the heat exchanger, [ºC]
$T_{\rm RP}$ – refrigeration temperature of the food product, [ºC]
ΔT – temperature difference, [ºC]
T_0 – preset refrigeration temperature, [ºC]
T_4 – superheated vapor temperature , [ºC]
$t_{\rm RP}$ – the time (duration) of the food product refrigeration process, [h, day]
x – refrigerant quantity in the second compression stage, [–]
w – technical work expended, [kJkg^{-1}]
$\Delta w_{\rm max}$ – maximum saved technical work expended, [kJkg^{-1}]
w_1 – technical work expended in the first compression stage, [kJkg^{-1}]
w_2 – technical work expended in the second compression stage, [kJkg^{-1}]

Greek symbols

α – heat transfer coefficient, [Wm^{-2}K^{-1}]
ε – total coefficient of performance, [–]
ε_1 – total coefficient of performance of the NH_3 refrigeration device, [–]
ε_2 – total coefficient of performance of the cascade refrigeration device, [–]
κ – compression isentropic coefficient, [–]

Subscripts

Cr – critical point
LC – lower cascade of the cascade device
UC – upper cascade of the cascade device

Acronimes

CFC – chlorofluorocarbons
$CFCl_3$ – trichlorofluoromethane
COP – coefficient of performance
GWP – global warming potential
HCFC – hydrochlorofluorocarbons
ODP – ozone depletion potential
RSM – response surface method

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