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Parametric Study of NH_3/CO_2 Cascade Refrigeration Cycles for Hot Climates

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ABSTRACT

This article makes a parametric study of cascade refrigeration systems with a low carbon footprint in tropical climates. An NH3/ CO_2 cascade refrigeration system is modelled, simulated under Engineering Equation Solver (EES) and the results are discussed. It is noted that the use of an internal heat exchanger in the R717/R744 cascade system reduces performance by 4.7%, whereas subcooling in both circuits increases the performance of the cascade systems by 16.50%. Exercise analysis of the cascade refrigeration system identified the temperature of -10°C as an optimal coupling temperature that allows for a low total irreversibility (i.e. a total irreversibility of 78.072 kW for a system with a cooling capacity of270 kW).The condenser and the compressor of the high temperature cell have been identified as the most critical components with a total irreversibility of 48%. Based on Total Equivalent Warning Impact (TEWI) calculation, 8 cascade refrigeration configurations are compared in the weather conditions of Benin (Cotonou). The natural refrigerants R717 and R600 seem to be suitable candidates in combination with $_{CO2}$ for cascade systems. Furthermore, the refrigerant R717 (ammonia) appears to be a good choice to achieve high performance with a relatively low TEWI.

Keywords: Refrigeration, Cascade cycle, Carbon dioxide, Ammonia, tropical countries.

INTRODUCTION

Refrigerants, which enabled and supported the development of refrigeration and conditioning from the 1930s onwards, were synthetic products. [1] The substitution of polluting refrigerants by hydrofluorocarbons (HFCs) is an effective approach to avoid aggravating the destruction of ozone, but this generation of HFCs contributes to global warming. [1] Hydrocarbons are organic fluids with good thermodynamic properties, but are dangerous because of their flammability. Hydrofluoroolefins (HFOs) are new, environmentally friendly synthetic refrigerants that are becoming more widely available, but their low flammability should be taken into account for applications with high refrigerant requirements. Driven by the desire to use environmentally friendly refrigerants, the use of $_{CO2}$ in refrigeration cycles is very recent and significant advantages are noted. [2] CO_2 is the most promising natural fluid in refrigeration cycles, due to its low global warming potential (GWP) and zero ozone depletion potential (ODP). [3] Non-flammable, non-explosive and non-toxic in low concentrations, CO_2 has become an increasingly relevant argument for reducing the carbon footprint. It does not need to be recovered or recycled compared to HFCs, which means that CO_2 is very attractive where infrastructure is either non-existent or too expensive, such as in developing countries. Despite these advantages as a refrigerant in refrigeration cycles, the major disadvantage of CO_2 is its low coefficient of performance and reduced cooling capacity, especially in hot environments. [4] The low critical temperature of 31.1°C makes it difficult to achieve the usual refrigeration cycles. Its thermo-physical properties allow it to adapt favourably to cascade refrigeration systems, which are remarkable systems in the refrigeration industry. [5] Single-stage systems are inefficient and impractical to reach low temperatures (-30°C to -100°C), and cascade systems overcome this obstacle. Ammonia (NH3) is a natural refrigerant that does not contribute to the greenhouse effect but its use has been greatly reduced due to its toxicity. The use of NH_3 can be perpetuated because of a certain interest in CO_2 at low temperatures. The simultaneous use of two natural refrigerants (R717 and R744) with aquasi-zero GWP can be an advantageous solution for a food chain to preserve foodstuffs. From a safety point of view, the toxicity of R717 can be avoided by channelling the high-temperature cell into a technical room far from the sales area and establishments receiving the public. In addition to the safety aspect, the cascade system guarantees the subcritical operation of the $_{CO2}$ system and could potentially lead to a reduction in the system's electricity consumption and a considerable reduction in indirect emissions. Therefore, the fundamental aim of this study is to carry out a parametric study of an NH_3/CO_2 cascade refrigeration system in tropical climatic conditions.

MATERIAL AND METHOD

1. NH_3/CO_2 cascade refrigeration cycles

A cascade refrigeration system consists of two refrigeration units (cells), one operating at a lower temperature and the other operating at a higher temperature that is thermally coupled to an internal cascade heat exchanger. Carbon dioxide (CO_2) is used for the low-temperature cell because of these thermodynamic properties. [6] The internal heat exchanger (evapo-condenser) acts as a condenser for the low temperature unit and as an evaporator for the high temperature unit and increases the performance of the refrigeration machine. As the efficiency of the exchanger increases, the optimum pressure decreases and the coefficient of performance increases. [9,10] The model of such a device is shown in Figure 1. At point (1), the outlet of the evaporator-condenser, ammonia (NH_3) in the form of saturated vapour enters the compressor and exits under the conditions of point (2) with a pressure corresponding to the saturating pressure of the condenser. During this step, the NH_3 transfers the heat that has been absorbed at the evaporator to the ambient environment as a liquid. After condensation (3), the liquid passes through the expansion valve where it undergoes an enthalpic transformation. At the outlet, the refrigerant is at low pressure (4), enters the evaporator-condenser and a heat exchange with carbon dioxide ($_{CO2)}$ causes it to pass as vapour (1), which constitutes the NH3 cell cycle. At point (5), at the outlet of the evaporator the CO_2 in the form of saturated vapour enters the compressor and exits under the conditions of point (6) with a pressure corresponding to the saturating pressure of the condenser. It passes through the evaporator-condenser and heat exchange with NH_3 causes it to pass as a liquid. During this step, the CO_2 releases the heat that has been absorbed at the evaporator plus the work of the compressor degenerated into heat. That heat is absorbed by the NH3, allowing the CO_2 to evaporate. After condensation (7), the liquid passes through the expansion valve where it undergoes an enthalpic transformation. At the outlet, the refrigerant is at low pressure (8), enters the evaporator, the CO_2absorbs the heat from the cooled medium and is transformed into vapour (5), which constitutes the $_{CO2}$ cell cycle (figure 1). Several research and experimental studies have been carried out on cascade systems. Recently, some authors [5] have studied two NH_3/CO_2 cascade refrigeration systems. In 2018 Gholamian et al. conducted an advanced exercise analysis of an NH_3/CO_2 cascade refrigeration system. Compared to conventional exergetic analysis, the advanced exergetic analysis makes several improvements and allows designers to find where these improvements are needed. It is a method of dividing the exergetic destruction to find the true source of irreversibility in the system in order to improve cycle performance. [9] In the advanced exergetic analysis which is explained in detail in reference, [10] the rate of exergetic destruction in each component is composed of two distinct parts, namely exogenous and endogenous destruction. Endogenous destruction is due to irreversibility within the component while exogenous destruction is imposed on the component by other components.

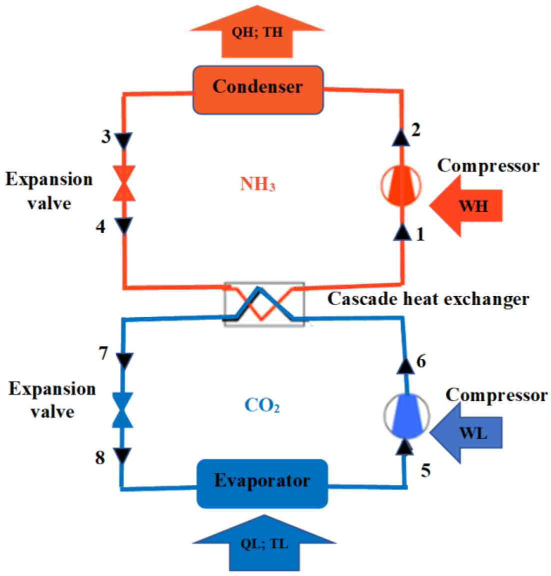

Figure 1: NH_3/CO_2 cascade cooling system

2. Modelling of the NH_3/CO_2 cascade refrigeration cycle

2.1. Assumptions for modelling the NH_3/CO_2 refrigeration cycle

In order to analyse the performance of the operating cycle of the cascade refrigeration system, the following assumptions have been made. These refrigeration needs and assumptions correspond to those of a shopping centre in Benin.

- Each component of the cascade refrigeration system can be treated as a control volume.
- All components are assumed to be in a steady state, constant flow process. Variations in the potential energy and kinetic energy of the components are negligible.
- The compressors of both cells are not isentropic, and the isentropic efficiency can be expressed as a function of the pressure ratio. Combined with the electrical and mechanical efficiency of each compressor, the total efficiency is assumed to be 0.80.
- Heat losses and pressure drops in pipe connections and components are negligible.
- All relaxation devices are isenthalpic.
- The output states of the condenser and cascade condenser are in the saturated liquid state and the evaporator is in the saturated vapour state.
- The cooling capacity is fixed (270 kW).
- The condensation temperatures used in the parametric study are in the range [25 °C; 55 °C].
- Evaporation temperatures are in the range [-50°C; -10°C].
- The temperature difference in the cascade exchanger is ΔT=5° C.

2.2 Simulation models

The thermodynamic models applied to the refrigeration cycle shown in Figure 1 are summarised in Table 1. For each component of the system, the mass, energy and exergy balance equations are applied. Each process is represented mathematically and integrated in Engineering Equation Solver (EES). In Table 1, Q_L is the cooling capacity, Q_M is the capacity of the cascade exchanger, W_H is the work done by the compressor of the high-temperature cell, W_L is the work done by the compressor of the low-temperature cell, $\dot{m}_L$ is the carbon dioxide mass flow rate, $\dot{m}_H$: ammonia mass flow rate, h_i: enthalpy of the refrigerant at different points in the system, s_i:entropy of the refrigerant at different points in the system, s_{gen}: entropy generated, η_m: mechanical efficiency, η_e: electrical efficiency, η_{is}: isentropic efficiency.

Table 1: Mass, energy, entropy and exergy balance of the system components

Components	Mass balance [$kg.s^{-1}$]	Energy balance [kW]	Entropy balance [kW/K]	Operating exergy balance [kW]
High temperature compressor	$\dot{m}_1=\dot{m}_2=\dot{m}_H$	$W_H=\dot{m}_H\frac{(h_{2is}-h_1)}{\eta_{is}.\eta_m.\eta_e}$ $=\dot{m}_H\frac{(h_2-h_1)}{\eta_m.\eta_e}$	$Sgen=\dot{m}_H(s_2-s_1)$	$Exd=\dot{m}_H(Ex_1-Ex_2)-W_H$ $=T0\,[\dot{m}_H(s_2-s_1)]$
Condenser	$\dot{m}_2=\dot{m}_3=\dot{m}_H$	$Q_H=\dot{m}_H(h_2-h_3)$	$Sgen=\dot{m}_H(s_3-s_2)-\frac{Q_H}{T_c}$	$Exd=(1-\frac{T0}{T_c})Q_H+\dot{m}_H(Ex_2-Ex_3)$ $=T_0\,[\dot{m}_H(s_3-s_2)-\frac{Q_H}{Tc}]$
High-temperature circuit expansion valve	$\dot{m}_3=\dot{m}_4=\dot{m}_H$	$h_3=h_4$	$Sgen=\dot{m}L(s_4-s_3)$	$Exd=\dot{m}_H(Ex_3-Ex_4)$ $=T_0[\dot{m}_H(s_4-s_3)$
Evapo-condenser	$\dot{m}_6=\dot{m}_7=\dot{m}_L$ $\dot{m}_4=\dot{m}_1=\dot{m}_H$	$Q_M=\dot{m}_H(h_1-h_4)$ $=\dot{m}L(h_6-h_7)$	$Sgen=\dot{m}_L(s7-s6)+\dot{m}_H.(s_1-s_4)$	$Exd=\dot{m}_L(Ex_6-Ex_7)+\dot{m}H(Ex4-Ex1)$ $=T0\,[\dot{m}_L(s_7-s_6)+\dot{m}_H(s_1-s_4)]$
Low temperature circuit compressor	$\dot{m}_5=\dot{m}_6=\dot{m}_L$	$WL=\dot{m}_L\frac{(h_{6is}-h_5)}{\eta_{is}.\eta_m.\eta_e}$ $=\dot{m}_H\frac{(h_6-h_5)}{\eta_m.\eta_e}$	$Sgen=\dot{m}_L(s_6-s_5)$	$Exd=\dot{m}_L.(Ex_5-Ex_6)-WL$ $=T_0\dot{m}_L(s_6-s_5)]$
Expansion valve low temperature circuit	$\dot{m}_7=\dot{m}_8=\dot{m}_L$	$h_7=h_8$	$Sgen=\dot{m}_L(s_8-s_7)$	$Exd=\dot{m}_L(Ex_7-Ex_8)$ $=T_0(\dot{m}_L(s_8-s_7)$
Evaporator	$\dot{m}_8=\dot{m}_5=\dot{m}_L$	$Q_L=\dot{m}_L(h_8-h_5)$ $\dot{m}_L=\frac{Q_L}{(h_8-h_5)}$	$Sgen=\dot{m}_L(s_8-s_5)-\frac{Q_L}{T_e}$	$Exd=(1-\frac{T0}{T_e})Q_L+\dot{m}_H(Ex_8-Ex_5)$ $=T_0[\dot{m}L(s_8-s_5)-\frac{Q_L}{T_e}]$
Coefficient of performance of the high-temperature cycle (COP_H)				$COP_H=\frac{Q_M}{W_H}$
Coefficient of performance of the low temperature cycle (COP_L)				$COP_L=\frac{Q_L}{W_L}$
Coefficient of performance of the cascade refrigeration system (COP)				$COP=\frac{COP_H.COP_L}{1+COP_H+COP_L}$

2.3 Environmental assessment of the NH_3/CO_2 cascade refrigeration system

In addition to energy and exergy analysis, environmental assessment is crucial for NH_3/CO_2 cascade systems to examine their environmental footprint. TEWI (Total Equivalent Warming Impact) is a concept for assessing global warming during the operational life of a refrigeration system. Details on the calculation of this index can be found in many studies. [15] As an indication, it is given by the following formula:

$$TEWI_{direct} = GWP100_{CO2}\,[L_{CO2}.N+M_{CO2}(1-a_1)] + GWP100_{NH3}\,[L_{NH3}.N+M_{NH3}(1-a_2)] \quad (1)$$

GWP_{100}: global warming potential for 100 years.

The mass of the refrigerant is assumed to be 1 kg/kW of cooling capacity for carbon dioxide and 2 kg/kW of cooling capacity for ammonia. The leakage (L) is assumed to be 15% of the total refrigerant mass (M), [6,13]

$$L_{CO2}=0.15M_{CO2} \quad (2)$$

$$L_{NH3}=0.15M_{NH3} \quad (3)$$

The years of analysis (N) are assumed to be 15 and the recycling factors (a_1 and a_2) should be 95%. [6,13]

$TEWI_{Indirect}$ is a function of the duration N of use, the annual electrical energy consumption (E) by the equipment in kWh and the amount (A) in kilogram equivalent of $_{CO2}$ emitted to produce 1 kWh of electricity consumed by the cooling equipment . A depends mainly on the primary energy used to produce electricity.

$$TEWI_{indirect}=E.N.A. \quad (4)$$

According to Vivendi Group's methodological note for calculating greenhouse gas emissions the emission factor for electricity production in Benin is A= 0.720 $kgCO_2/kWh$.

$$E=\int_{t=0}^{t=8760} Pdt. \quad (5)$$

The annual mean performance coefficient is given by the relation: [7]

$$COP_m=\frac{\int_{t=0}^{t=8760} Q_L dt}{\int_{t=0}^{t=8760} Pdt} \quad (6)$$

RESULTS AND DISCUSSIONS

1. The effect of the evaporation temperature on the NH_3/CO_2 cascade system

Figure 2 shows the effect of the evaporating temperature on the coefficient of performance of the cascade system, the effect of the refrigeration capacity (Q_L) and the work required (W_C) in the compressors (sum of the work of the two compressors). The results are obtained by setting the condensing temperature at 35°C and the condensing temperature of the R744 in the cascade heat exchanger at 15°C. As the evaporating temperature increases, the coefficient of performance increases slightly (the COP is equal to 1.211 at -50°C and equal to 3.22 at -10°C). The work required to operate the compressors decreases as the performance of the cascade system increases. Figure 3 shows that the compression work required in the low-temperature cell decreases as the evaporating temperature increases because the pressure ratio decreases. The work has dropped from 169.1 kW (at the evaporating temperature of -50°C) to 45.02 kW (at the evaporating temperature of -10°C). The compression work required in the high-temperature cell has decreased slightly. It has decreased from 53.83 kW to 38.62 kW at the same evaporating temperatures. As a result, the combined work (W_C) required also decreases.

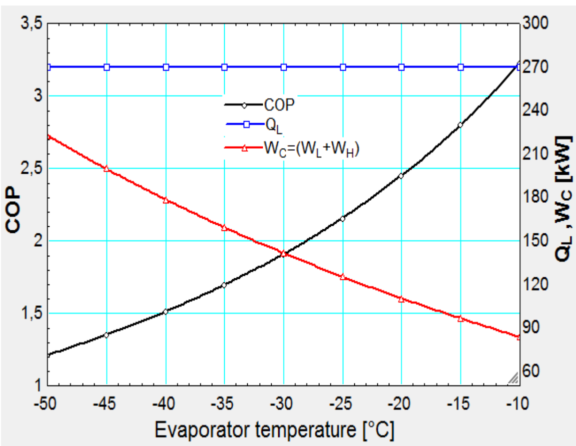

Figure 2: Variation in COP, compressor work and the effect of the evaporating temperature of the cascade system

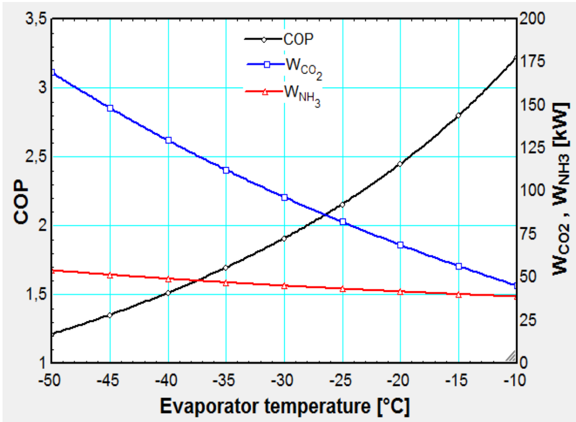

Figure 3: Effect of the evaporating temperature of the cascade system on compressor operation

2. Effect of the condensation temperature of the cascade refrigeration system

When the evaporating temperature is kept constant at -35°C and the condensing temperature from R744 to -15°C in the evaporator-condenser, the coefficient of performance of the cascade refrigeration system will decrease as the condensing temperature increases; the cooling capacity (Q_L) will be constant over the entire condensing temperature range (Figure 4). However, the work required at the high-temperature cell compressor increases due to the increase in the compression ratio of the cell. The combined work ($W_C=W_L+W_H$) increases, therefore the COP of the cascade system decreases.

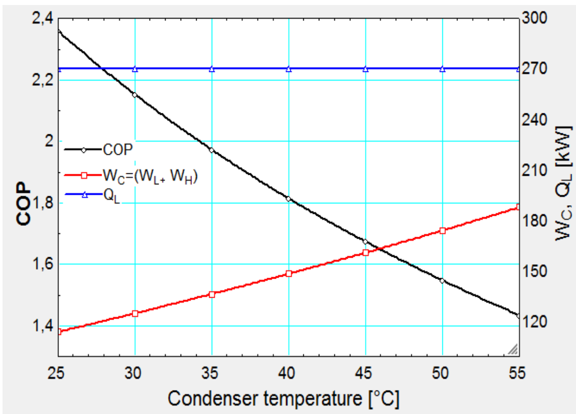

Figure 4: Variation in COP and compressor work of the cascade system as a function of condensing temperature

3. Effect of the condensation temperature of CO_2 in the evapo-condenser

When the condensing temperature is kept constant at 45°C and the evaporating temperature at -45°C, the Coefficient of

Performance (COP) of the cascade refrigeration system and the Coefficient of Performance (COP_L and COP_H) of each cell vary according to the condensing temperature of the low-temperature cell. According to Figure 5, it is observed that the COP_L of the low-temperature cell decreases as the carbon dioxide condensing temperature increases although the COP_H of the high-temperature cell increases. By varying the carbon dioxide condensing temperature (-25°C to 10°C) the COP of the cascade system first increases; reaches the maximum value (1.407 to -15°C) and then decreases. The condensing temperature of the low temperature cell at which the COP of the cascade system is maximum is known as the optimum coupling temperature of the cascade system.

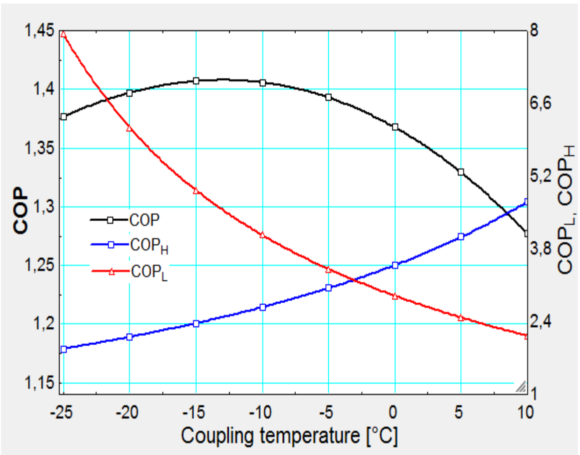

Figure5: Variation of COP, COP_H and COP_L as a function of coupling temperature

4. Effect of cascade system compressor efficiency

The variation of the coefficient of performance of the cascade refrigeration system with the coupling temperature is shown in Figure 6 for different isentropic compressor efficiencies (ε). It is observed in Figure 6 that the COP of the cascade system increases as the compressor efficiency increases, due to the reduction in compressor work.

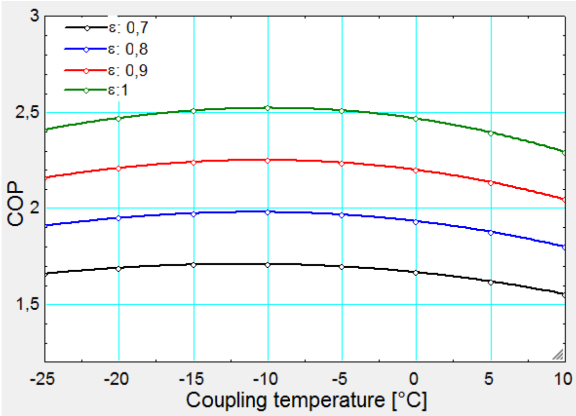

Figure 6: COP variation for different isentropic compressor efficiencies.

5. Effect of the temperature difference in the cascade condenser

The variation of the coefficient of performance as a function of the coupling temperature at different temperature differences (ΔT) in the cascade heat exchanger is shown in Figure 7. In the cascade refrigeration system, the condenser of the low-temperature cell rejects heat to the evaporator of the high-temperature cell. In order for there to be heat transfer between the two cells, there should be a temperature difference (ΔT) in the cascade heat exchanger. When the temperature difference in the cascade heat exchanger increases, the system performance drops although the system cost also decreases (due to the small size of the cascade heat exchanger) vice versa. Therefore, the temperature difference must have an optimal value to balance the cost and performance of the system. Generally, it is recommended that the value of the temperature difference (ΔT) should be equal to 5 °C.

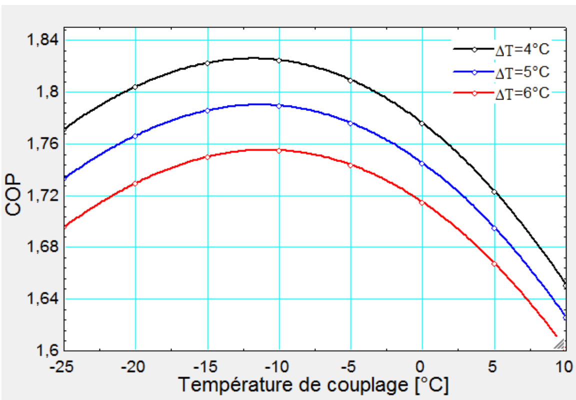

Figure 7: Effect of the temperature difference in the cascade capacitor on the COP

6. Improving the performance of the cascade refrigeration system

It is assumed that the condition of the refrigerant entering the low and high temperature cycle expansion valves is saturated liquid. However, liquid cooling below saturation can increase the refrigeration effect and potentially improve the coefficient of performance. To improve the COP by subcooling, the effects of the internal heat exchanger and condenser subcooling are studied. It is observed in Figure 8 that the internal heat exchanger has a negative effect on the performance of the cascade system. The COP decreased from 1.789 to 1.742. The performance therefore decreased by 4.7%. Therefore, it is concluded that the internal heat exchanger is not advantageous with a pair of ammonia and carbon dioxide refrigerants in a cascade system.

With regard to improving the performance of the cascade system, the effect of undercooling (ΔT_{SC}) and overheating (ΔT_{SH}) has also been studied on the cascade system. Figure 9 shows the evolution of the coefficient of performance as a function of the coupling temperature at various degrees of subcooling and superheating in the two circuits. It can be seen from Figure 9 that the degree of undercooling increases system performance and that overheating always has undesirable consequences on system performance.

At -10°C the COP increased from 1.742 to 1.907 for a subcooling of 10°C, i.e. an increase of 16.50%.

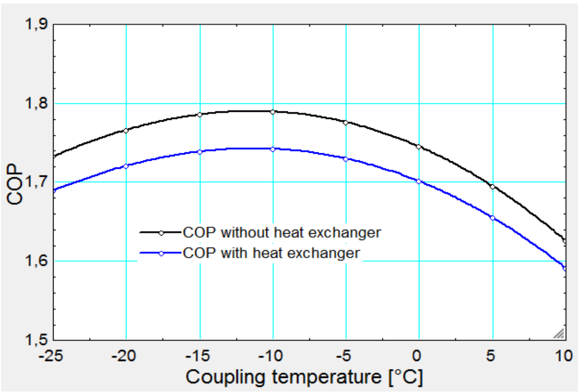

Figure 8: Effect of internal heat exchanger on system performance in cascadee

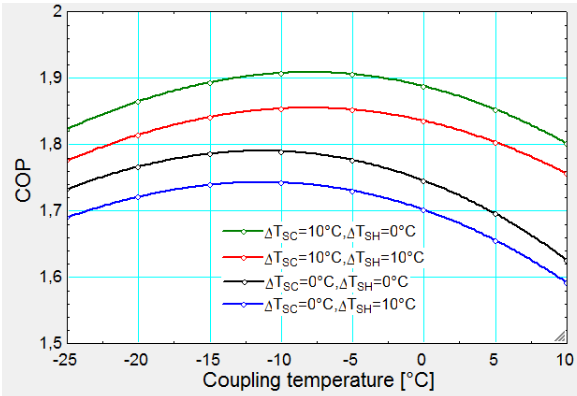

Figure 9: The effect of undercooling and overheating on the COP of the cascade system

7. Exercise analysis of the NH_3/CO_2 cascade refrigeration system

A careful analysis of Figures 10, 11, 12 and 13 illustrates the exercise analysis of the NH_3/CO_2 cascade refrigeration system. The destroyed exergy of some components increased as a function of the condensation temperature of carbon dioxide, while the exergy of some components decreased (Figure 10). The destroyed exertion in the components of the high-temperature cell drops when the condensation temperature of $_{CO2}$ in the evaporator-condenser increases. The destroyed exertion in the cascade exchanger increases as a function of the temperature. In the low-temperature cell the rate of exergy destruction in the compressor, expansion valve and evaporator increases as the CO_2 condensing temperature rises. At -25°C, the condenser has the highest exergy destruction, followed by the high-temperature cell compressor, its expansion valve, the evaporator, the cascade condenser, the low-temperature circuit compressor and its expansion valve. However, when the temperature has risen to 5°C, the condenser is first followed by the $_{CO2}$ cell compressor, its expansion valve, the NH_3 compressor, the cascade exchanger, the evaporator and the NH_3 expansion valve. Other information related to the exertion of the components of the cascade system as a function of the $_{CO2}$ condensing temperature is summarised in Table 2.

Table 2: Exergy of cascade refrigeration system components

Coupling temperature [°C]	Exergy [kW] at condenser	Exergy [kW] at CO_2 compressor	Exergy [kW] at the NH_3 compressor	Exergy [kW] to the CO_2 regulator	Exergy [kW] to the $NH3$ regulator	Exergy [kW] at the intermediate heat exchanger	Exergy [kW] at the evaporator	TotalExergy [kW]
-25	32.25	3.55	17.04	0.9978	12.11	7.534	9.65	83.1318
-20	28.81	5.282	16.06	1.986	10.38	8.028	9.65	80.196
-15	25.86	7.015	15.05	3.363	8.81	8.755	9.65	78.503
-10	23.35	8.774	14	5.191	7.386	9.721	9.65	78.072
-5	21.24	10.59	12.92	7.558	6.103	10.94	9.65	79.001
0	19.5	12.49	11.8	10.58	4.952	12.44	9.65	81.412
5	18.09	14.53	10.63	14.44	3.93	14.28	9.65	85.55
10	17.87	15.54	6.955	21.7	2.24	17.8	19.23	101.335
15	17.55	17.5	6.055	27.75	1.673	20.57	19.23	110.328

Figure 11 illustrates the rate of exergy destruction of the cascade system components for four different CO_2 condensing temperatures (-25°C, -10°C, 0°C, 15°C) in the evaporative condenser when the refrigeration capacity is maintained at 270kW. The condensing (Tc) and evaporating (Te) temperatures of the refrigeration system are kept constant. The values of the exergetic destruction rates of the system components as a function of the coupling temperature and the values of the total exergetic of the refrigeration system as a function of the coupling temperature are determined (Table 2). It can be seen that at the coupling temperature -10°C, the total exertion of the refrigeration system is low (78.072kW) compared to other coupling temperatures.

Figure 12 shows the percentage of exergy destruction of each component of the cascade refrigeration system when the coupling temperature is set at -10°C. The condenser and compressor of the high-temperature cell are the largest contributors to the exergy destruction of the system with an irreversibility of 48%. These results show that heat transfer processes are important in the condenser, so special attention must be paid when designing this exchanger (an irreversibility of 30%). Table 3 shows the exergy efficiencies (ηex) of the components of the cascade refrigeration system.

In Figure 13, the rates of exergy destruction of all components increase with the cooling capacity of the refrigeration system. The condenser, the evaporator, the low-temperature cell expansion valve (CO_2 expansion valve), the low-temperature cell compressor (CO_2 compressor), thecascade heat exchanger, the high-temperature cold room compressor (NH_3 compressor) and its expansion valve (NH_3 expansion valve) have the highest exergy destruction rates, respectively.

Table 3: Exergy efficiencies of cascade refrigeration system components

Exergy contribution of components (ηex)						
Condense	Compressor of the CO_2 cycle	NH3 cycle compressor	Dealer of the CO_2 cycle	Dealer of the NH_3 cycle	Cascade exchanger	Evaporator
0.1	0.82	0.64	0.99	0.98	0.99	0.80

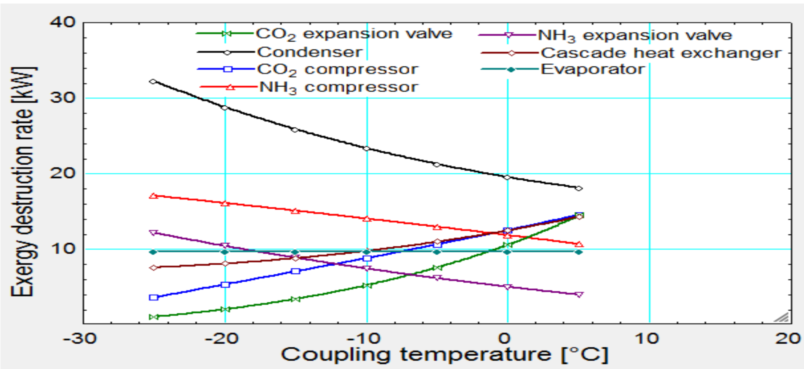

Figure 10: Variation in the rate of exergy destruction as a function of coupling temperature

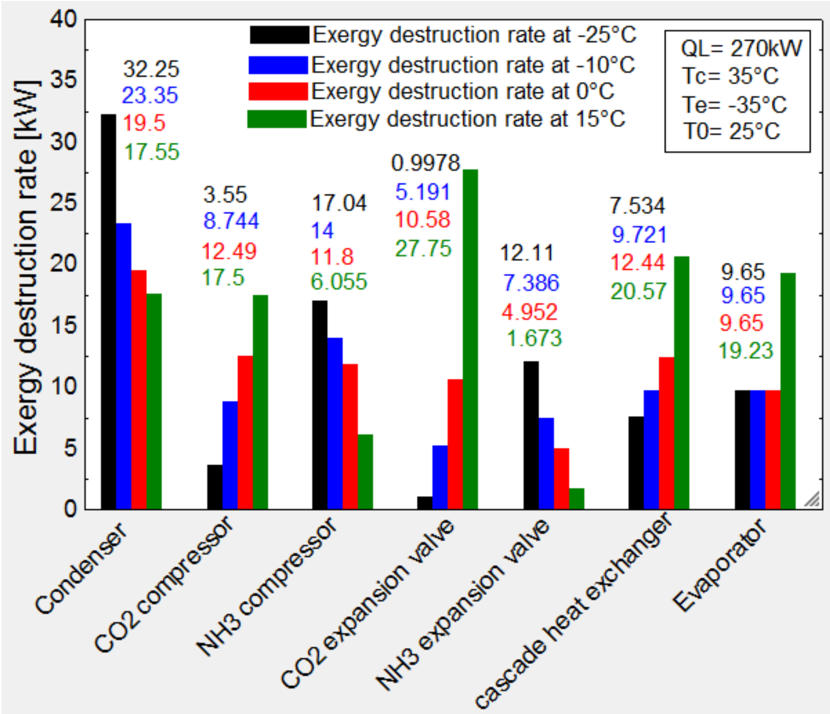

Figure 11: Exercise destruction rate of cascade refrigeration system components

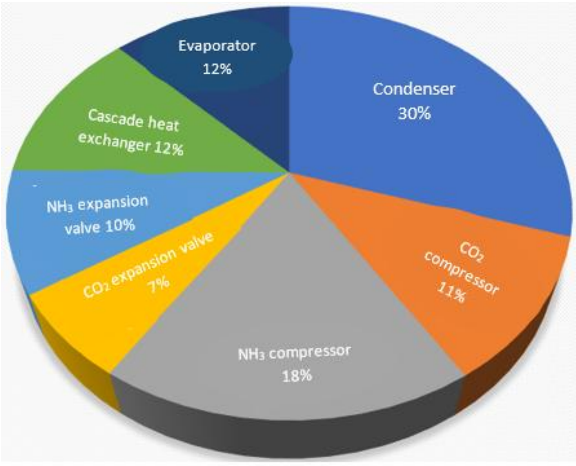

Figure 12: Distribution of energy loss in the cascade refrigeration system

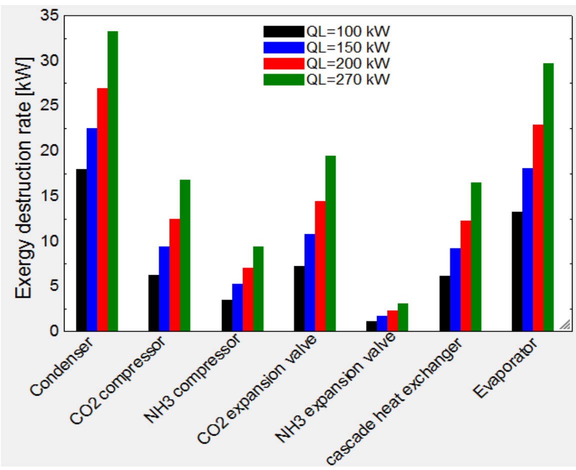

Figure 13: Variation of the exerted destruction rate as a function of cooling capacity

8. Environmental analysis of the NH_3/CO_2 cascade refrigeration system

Figures 14 and 15 show the environmental analysis of the NH_3/CO_2 cascade refrigeration system and the variation in TEWI (Total Equivalent Warning Impact) of the cascade refrigeration system operating with different refrigerants, respectively. The simulation is made by setting the condensing temperature (Tc=35°C) and the cooling capacity (Q_L= 270kW) constant. The annual average Coefficient of Performance and TEWI vary as a function of the evaporating temperature (Figure 14). The TEWI value decreases as the evaporating temperature increases because the coefficient of performance increases. Figure 15 shows the TEWI values of the cascade refrigeration system operating with different natural refrigerants in the high-temperature cell.

Of the cascade refrigeration systems (R717/R744, R600a/R744, R290/R744, R600/R744, R1234yf/R744, R1234ze/R744, R134a/R744, R404A/R744), only the R717/R744 system has the lowest TEWI value. The closest TEWI value is that of the R600/R744 cascade refrigeration system. At

-50°C the TEWI of the R717/R744 cascade system reaches a value of 18850 tons of CO_2 when the TEWI of the R600/R744 cascade system is 18890 tons of CO_2 at the same temperature. At 10°C the TEWI of the R717/R744 cascade system reaches a value of 3969 tons of CO_2 when the TEWI of the R600/R744 cascade system is 3992 tons of $_{CO2}$ at the same temperature. The R404A/R744 cascade system has the highest TEWI value (20810 tons of $_{CO2}$ at -50°C and 5574 tons of $_{CO2}$ at 10°C). Since natural refrigerants and hydrofluoroolefins are low GWPrefrigerants among other refrigerants, it can be concluded that the R717/R744 and R600/R744 cascade refrigeration systems are environmentally friendly systems.

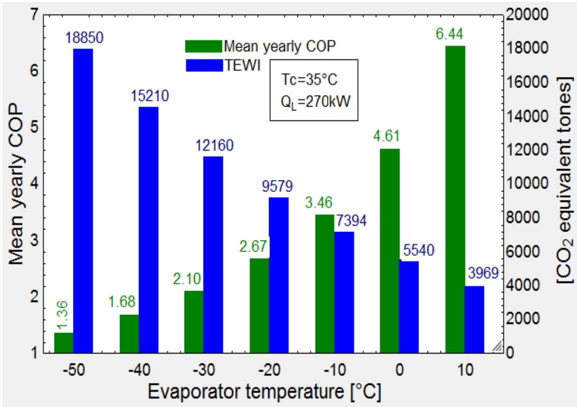

Figure 14: Variation of annual average coefficient of performance and TEWI as a function of evaporating temperature.

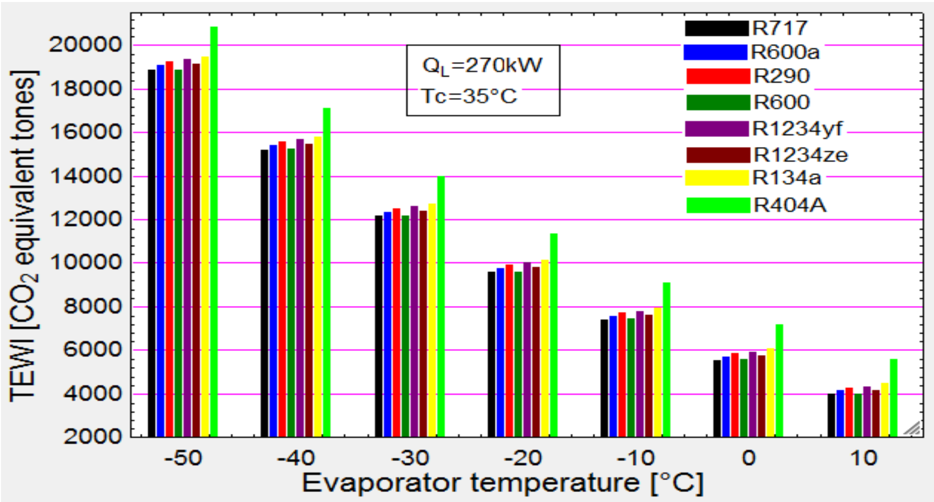

Figure15: TEWI variation of cascade refrigeration system operating with different refrigerants

CONCLUSION

Based on the concepts of thermodynamics and with the help of the performance optimisation calculation, the following relevant conclusions can be drawn from the study:

The performance of the cascade system increases as the evaporating temperature rises and decreases as the condensing temperature rises due to the diminishing of the compression ratio which results in less compressor work requirement. That induces an increase in COP. Performance is maximum for particular condensing and evaporating temperatures, and the optimum coupling temperature value can be obtained. The use of an internal heat exchanger in a cascade system reduces performance by 4.7%, whereas subcooling in both circuits increases the performance of the cascade system by 16.50%. As the isentropic efficiency of the compressor improves the performance of the cascade system increases. The destroyed exertion in the high-temperature cell components drops as the $_{CO2}$ condensing temperature rises. After the exergy analysis of the cascade system, the coupling temperature of -10°C results in a low irreversibility (i.e. a total irreversibility of 78.072kW for a system with 270kW refrigeration capacity. The condenser and compressor of the high-temperature cell have been identified as the most critical components with a total irreversibility of 48%. Exercise destruction rates of all components increase with the cooling capacity of the refrigeration system. Among 8 cascade refrigeration systems operating with different refrigerants, the R717/R744 cascade refrigeration system is the most environmentally friendly followed by the R600/R744 cascade refrigeration system. The TEWI (Total Equivalent Warning Impact) value decreases as the evaporating temperature increases and the coefficient of performance of the cascade system increases. Finally, the feasibility of cascade systems in tropical areas does not pose any particular safety problems compared to cold countries. These systems are therefore applicable without reservation.

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