Parametric Study of Temperatures in a Cascade Refrigeration System Using NH_3 and CO_2

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1 Introduction

This project focuses on the analysis and optimization of a cascade refrigeration system, which is a refrigeration arrangement where two separate cycles work together to achieve very low temperatures efficiently. In our case, the system consists of two distinct vapor compression cycles: one using ammonia (NH_3) as the refrigerant and the other using carbon dioxide (CO_2).

The ammonia cycle is the low-temperature stage, designed to achieve ultra-low temperatures around $-50\,^{\circ}C$, making it suitable for deep-freezing applications such as the storage of seafood, pharmaceuticals, or specialty chemicals. However, achieving such low temperatures with a single cycle would result in extremely low pressures on the evaporator side and very high pressures on the condenser side, leading to inefficiency and mechanical challenges. Additionally, ammonia is a toxic refrigerant and using it throughout an entire system, including in the distribution lines near occupied spaces, poses safety risks. By coupling the ammonia cycle with a high-temperature CO_2 cycle, the system can isolate the ammonia to a compact, contained low-temperature loop, while the CO_2 handles the heat rejection and distribution side at safer, non-toxic conditions [1]. This design not only improves system performance by distributing the temperature lift across two fluids but also enhances operational safety by limiting ammonia's presence to areas where it can be carefully monitored and controlled.

In this project, we will conduct a detailed thermodynamic analysis by systematically modifying key parameters of the cascade system, including the minimum (evaporator), intermediate (cascade heat exchanger), and maximum (condenser) temperatures. By varying these temperatures, we will evaluate their impact on crucial system performance indicators such as the coefficient of performance (COP) and the total work input. This sensitivity study will allow us to identify optimal operating conditions, understand the trade-offs between energy efficiency and cooling capacity, and explore how temperature adjustments affect the balance between the ammonia and CO_2 cycles. The insights gained will provide valuable guidance for the design and operation of efficient and safe cascade refrigeration systems in industrial applications.

2 Selection of Working Fluids

The choice of NH_3 and CO_2 as working fluids is deeply tied to their thermodynamic properties. Ammonia has long been favored in industrial refrigeration due to its high latent heat of vaporization, excellent thermal conductivity, and superior coefficient of performance (COP) at low temperatures, making it ideal for the ultra-cold stage. However, because of its toxicity, ammonia is best confined to controlled equipment zones.

On the other hand, CO_2 is a natural refrigerant that works efficiently at moderate and high temperatures, has a high volumetric cooling capacity, and operates under high pressure, which helps reduce equipment size and allows it to serve as the safer working fluid in distribution lines. Additionally, both fluids are environmentally friendly, with zero ozone depletion potential (ODP) and very low global warming potential (GWP), aligning with global sustainability goals.

Another commonly used refrigerant in refrigeration systems is R134a (tetrafluoroethane), a synthetic refrigerant often applied in medium-temperature commercial systems. While R134a has good thermodynamic performance and chemical stability, it has a significantly higher GWP compared to CO_2 and NH_3, making NH_3–CO_2 combinations preferable in modern, sustainability-focused cascade systems.

3 Applications

Cascade refrigeration systems have a wide range of applications in industries that require very low temperatures beyond the reach of conventional single-stage systems. They are commonly used in the chemical and pharmaceutical industries for the preservation and storage of temperature-sensitive products such as vaccines, biologics, and specialty chemicals [2].

They are also essential in the food industry, particularly in deep-freezing operations for seafood, meat, and ready-to-eat products, where maintaining temperatures well below freezing is critical for quality and safety. Additionally, cascade systems are employed in cryogenic applications, such as the production and storage of liquefied gases (e.g., liquefied natural gas, or LNG), and in low-temperature scientific research facilities, including superconducting magnet cooling or space simulation chambers.

Their ability to deliver stable and efficient ultra-low temperatures makes them indispensable in sectors where precise thermal control is crucial.

For this work, we analyze different refrigeration cycles with the objective of extracting a cooling load of 76,210 W.

4 Simple Refrigeration Cycle with Ammonia

To begin analyzing the system, we consider a simple vapor-compression refrigeration cycle using ammonia (NH_3), illustrated in Figure 1. In this cycle, ammonia is compressed, condensed, expanded, and evaporated. Heat is absorbed from the refrigerated space during evaporation and rejected to the environment during condensation. The condensation and evaporation temperatures are set at 35°C and −40°C, respectively.

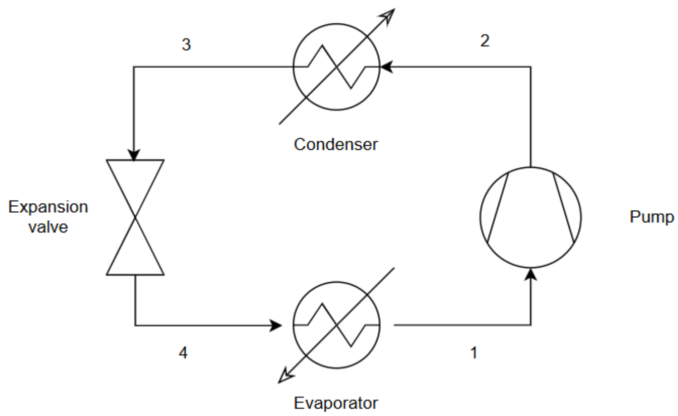

Figure 1: Simple Refrigeration Cycle Using Ammonia

4.1 Compression Stage

State 1 is defined by the evaporation temperature T_{evap} and a vapor quality $x = 1$, representing saturated vapor. Other thermodynamic properties are obtained using CoolProp.

In the first stage, ammonia is compressed isentropically with an isentropic efficiency of $\eta_{\text{comp}} = 0.85$, up to the pressure corresponding to saturation at $T_{\text{cond}} = 35°\text{C}$. Assuming an ideal isentropic process, the entropy remains constant:

$$s_{2s} = s_1, \quad P_2 = P(T = T_{\text{cond}}, x = 1) \tag{1}$$

The isentropic enthalpy at this state is obtained using:

$$h_{2s} = h(s_{2s}, P_2) \tag{2}$$

The actual enthalpy after accounting for compressor efficiency is:

$$h_2 = h_1 + \frac{h_{2s} - h_1}{\eta_{\text{comp}}} \tag{3}$$

State 2 is then fully defined by P_2 and h_2.

4.2 Condensation Stage

At this stage, the superheated vapor from State 2 is condensed at constant pressure until it becomes saturated liquid:

$$P_3 = P_2, \quad x_3 = 0 \tag{4}$$

State 3 is defined by P_3 and $x_3 = 0$.

4.3 Expansion Stage

Ammonia expands isenthalpically through a throttling device to the evaporator pressure:

$$h_4 = h_3, \quad P_4 = P_1 \tag{5}$$

This expansion is irreversible, with entropy increasing during the process.

4.4 Evaporation Stage

In the final stage, ammonia evaporates at constant pressure P_1, absorbing heat and returning to State 1 to complete the cycle.

4.5 Results

The temperature-entropy (T-s) diagram is shown in Figure 2.

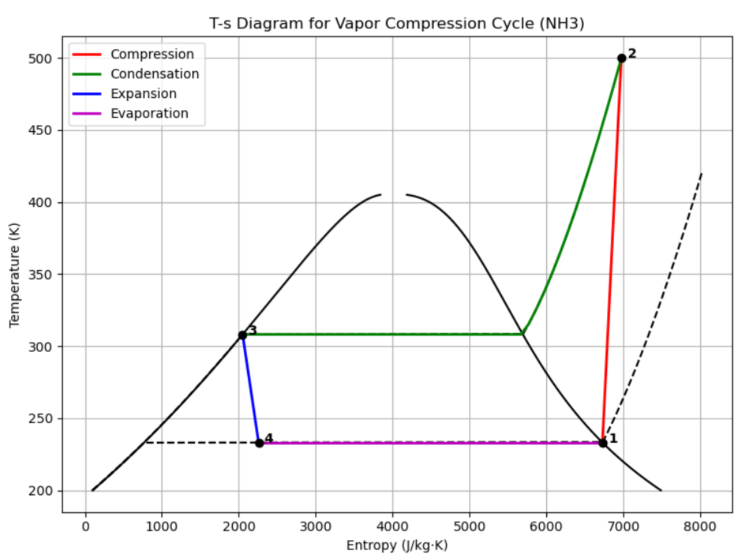

Figure 2: T-s Diagram of the Simple Refrigeration Cycle Using Ammonia

Thermodynamic states:

- **State 1:**
  $P = 0.72\,\text{bar}, \quad T = -40.00°\text{C}, \quad h = 1553.81\,\text{kJ/kg}, \quad s = 6.7276\,\text{kJ/kg·K}$
- **State 2:**
  $P = 13.50\,\text{bar}, \quad T = 226.64°\text{C}, \quad h = 2137.15\,\text{kJ/kg}, \quad s = 6.9725\,\text{kJ/kg·K}$
- **State 3:**
  $P = 13.50\,\text{bar}, \quad T = 35.00°\text{C}, \quad h = 511.56\,\text{kJ/kg}, \quad s = 2.0494\,\text{kJ/kg·K}$
- **State 4:**
  $P = 0.72\,\text{bar}, \quad T = -40.00°\text{C}, \quad h = 511.56\,\text{kJ/kg}, \quad s = 2.2573\,\text{kJ/kg·K}$

Performance metrics:

$$q_{\text{in}} = h_1 - h_4 = 1042.26 \text{ kJ/kg}$$
$$w_{\text{in}} = h_2 - h_1 = 583.34 \text{ kJ/kg}$$
$$\text{COP} = \frac{q_{\text{in}}}{w_{\text{in}}} = 1.79$$

To deliver $Q = 76.21$ kW, the required mass flow rate is:

$$\dot{m} = \frac{Q}{q_{\text{in}}} = 0.07 \text{ kg/s}$$

The compressor work is:

$$\dot{W} = \dot{m} \cdot w_{in} = 42.65 \text{ kW}$$

5 Cascade Refrigeration Cycle

In this section, we analyze a cascade refrigeration cycle using carbon dioxide (CO_2) as the low-temperature cycle (LTC) working fluid and ammonia (NH_3) as the high-temperature cycle (HTC) working fluid. To allow for a fair comparison with the simple cycle, the evaporation and condensation temperatures are maintained at $-40°$C and $35°$C, respectively. The intermediate temperature, where heat is exchanged between the two cycles, is selected as $-10°$C for CO_2 condensation and $-15°$C for NH_3 evaporation, introducing a temperature difference of $5°$C for effective heat transfer.

The cycle layout is shown in Figure 3. Each fluid undergoes its own vapor-compression cycle comprising compression, condensation, expansion, and evaporation. The two cycles are thermally coupled at the cascade heat exchanger: the evaporator of the NH_3 cycle absorbs heat from the condenser of the CO_2 cycle.

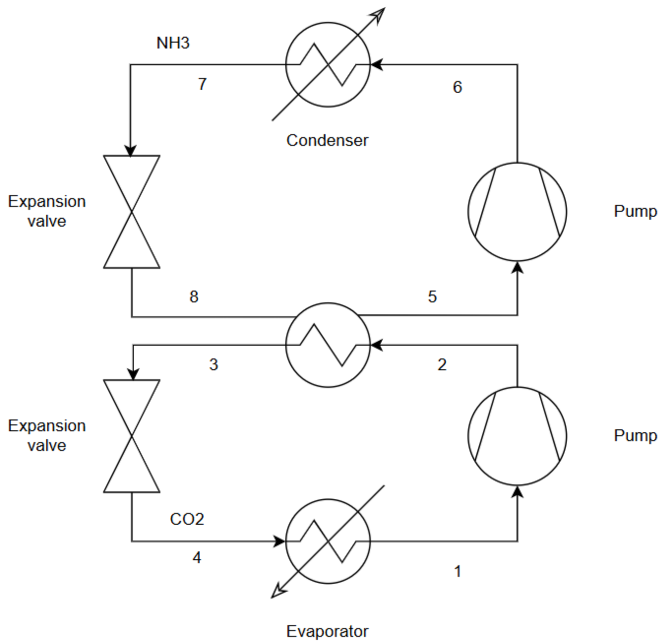

Figure 3: Cascade Refrigeration Cycle using Ammonia and CO_2

5.1 Low-Temperature Cycle (CO_2)

- **State 1:**
  $P = 10.04\,\text{bar}, \quad T = -40.00°\text{C}, \quad h = 435.32\,\text{kJ/kg}, \quad s = 2.0485\,\text{kJ/kg·K}$
- **State 2:**
  $P = 26.49\,\text{bar}, \quad T = 32.75°\text{C}, \quad h = 487.36\,\text{kJ/kg}, \quad s = 2.0831\,\text{kJ/kg·K}$
- **State 3:**
  $P = 26.49\,\text{bar}, \quad T = -10.00°\text{C}, \quad h = 176.52\,\text{kJ/kg}, \quad s = 0.9157\,\text{kJ/kg·K}$
- **State 4:**
  $P = 10.04\,\text{bar}, \quad T = -40.00°\text{C}, \quad h = 176.52\,\text{kJ/kg}, \quad s = 0.9385\,\text{kJ/kg·K}$

Cycle Performance:

$$q_{\text{in,LT}} = h_1 - h_4 = 258.80 \text{ kJ/kg}$$

$$w_{\text{in,LT}} = h_2 - h_1 = 52.03 \text{ kJ/kg}$$

$$\text{COP}_{\text{LT}} = \frac{q_{\text{in,LT}}}{w_{\text{in,LT}}} = 3.39$$

5.2 High-Temperature Cycle (NH_3)

- **State 5:**
  $P = 2.36$ bar, $T = -15.00°C$, $h = 1589.68$ kJ/kg, $s = 6.3110$ kJ/kg·K
- **State 6:**
  $P = 13.50$ bar, $T = 136.60°C$, $h = 1910.52$ kJ/kg, $s = 6.4727$ kJ/kg·K
- **State 7:**
  $P = 13.50$ bar, $T = 35.00°C$, $h = 511.56$ kJ/kg, $s = 2.0494$ kJ/kg·K
- **State 8:**
  $P = 2.36$ bar, $T = -15.00°C$, $h = 511.56$ kJ/kg, $s = 2.1347$ kJ/kg·K

Cycle Performance:

$$q_{\text{in,HT}} = h_5 - h_8 = 1078.12 \text{ kJ/kg}$$

$$w_{\text{in,HT}} = h_6 - h_5 = 320.84 \text{ kJ/kg}$$

$$\text{COP}_{\text{HT}} = \frac{q_{\text{in,HT}}}{w_{\text{in,HT}}} = 3.36$$

5.3 Results

The temperature-entropy (T-s) diagram is shown in Figure 4.

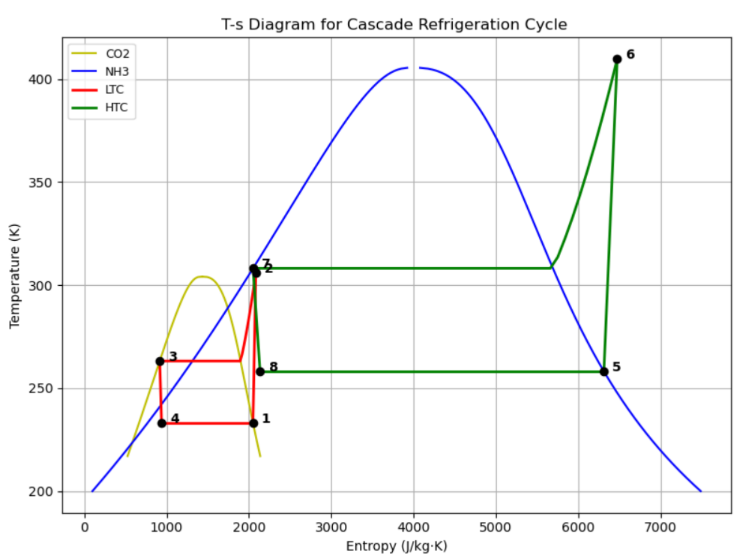

Figure 4: T-s Diagram of the Cascada Refrigeration Cycle Using Ammonia and CO_2

To satisfy a required cooling capacity of $Q = 76.21$ kW in the low-temperature cycle: **Assumed Cooling Load:**

$$Q = 76210 \text{ W}$$

Mass Flow Calculations:

$$\dot{m}_{\text{LT}} = \frac{Q}{q_{\text{in,LT}}} = \frac{76210}{258.80} = 0.2946 \text{ kg/s}$$

$$\dot{m}_{\text{ratio}} = \frac{q_{\text{in,HT}}}{h_2 - h_3} = \frac{1078.12}{487.36 - 176.52} = 3.4685$$

$$\dot{m}_{\text{HT}} = \frac{\dot{m}_{\text{LT}}}{\dot{m}_{\text{ratio}}} = \frac{0.2946}{3.4685} = 0.0850 \text{ kg/s}$$

Total Heat and Work Inputs:

$$\dot{Q}_{\text{in,total}} = \dot{m}_{\text{LT}} \cdot q_{\text{in,LT}} = 0.2946 \cdot 258.80 = 76199.4 \text{ W}$$

$$\dot{W}_{\text{in,total}} = \dot{m}_{\text{LT}} \cdot w_{\text{in,LT}} + \dot{m}_{\text{HT}} \cdot w_{\text{in,HT}} = 0.2946 \cdot 52.03 + 0.0850 \cdot 320.84 = 14453.9 + 27277.4 = 41731.3 \text{ W}$$

Overall Coefficient of Performance (COP):

$$\text{COP}_{\text{total}} = \frac{\dot{Q}_{\text{in,total}}}{\dot{W}_{\text{in,total}}} = \frac{76199.4}{41731.3} = 1.826$$

This COP is consistent with the result from the simple cycle, indicating comparable performance despite the added complexity of the cascade configuration. We observe that, under these refrigerants and operating conditions, the cascade cycle yields a slightly higher COP and reduced compressor work. However, these improvements are relatively minor when weighed against the increased cost and complexity of implementing two separate refrigeration loops. Therefore, the primary motivation for adopting a cascade cycle in this context is to mitigate the risks associated with ammonia's toxicity by confining it to the high-temperature side of the system.

6 More Advanced Studies

Keeping the cascade cycle using ammonia and CO_2, we now investigate how changes in operating conditions affect the COP, compressor work, and refrigerant mass flow rates.

6.1 Modifying Intermediate Temperature

First, we analyze the impact of varying the intermediate temperature while maintaining a constant temperature difference of 5°C between the two refrigerants. As shown in Figure 5, increasing the intermediate temperature results in a lower COP and higher total compression work. This is primarily due to an increase in the work of the low-temperature cycle (LTC), which outweighs the decrease in the work of the high-temperature cycle (HTC). This behavior is expected, as higher intermediate temperatures lead to higher temperatures at State 2 in the CO_2 cycle and at State 5 in the NH_3 cycle.

Regarding mass flow rates, both refrigerants exhibit an increase. For CO_2, the increase is attributed to a higher vapor quality at State 4 under elevated intermediate temperatures, while the required cooling load remains constant. For ammonia, the increase in $q_{\mathrm{in,HT}}$ is faster than that in $q_{\mathrm{in,LT}}$, and since the HTC mass flow rate is also proportional to the CO_2 flow rate, both increase with temperature.

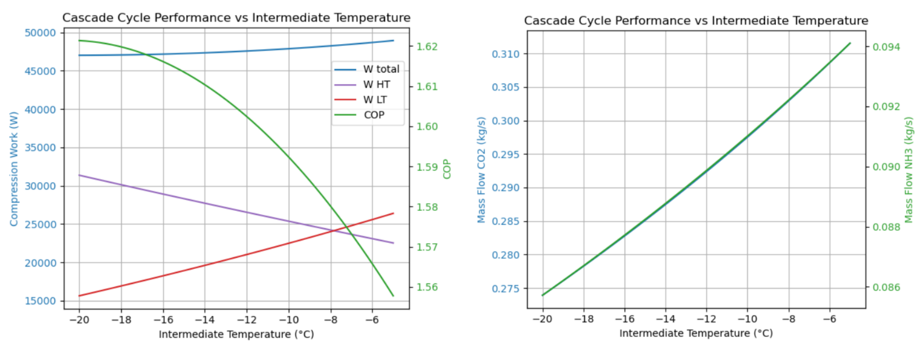

Figure 5: Effect of intermediate temperature on COP, compression work, and mass flow rates

6.2 Modifying Condensation Temperature

We now analyze the impact of varying the condensation temperature of the high-temperature cycle (HTC) using ammonia (NH_3). As shown in Figure 6, increasing the condensation temperature results in a lower COP and higher total compression work. This is primarily due to the increased compression work required by the ammonia cycle. Consequently, this refrigeration system performs worse in hotter environments.

Regarding the mass flow rates, the mass flow rate of carbon dioxide (CO_2) remains constant, since the cooling load is fixed. However, a higher condensation temperature necessitates a higher mass flow rate of ammonia. This behavior is expected, as state 8 (post-expansion ammonia) moves to a higher quality, and state 7 shifts further to the right in the T-s diagram with increasing condensation temperature.

6.3 Modifying Evaporation Temperature

We now analyze the impact of varying the evaporation temperature of the low-temperature cycle (LTC) using CO_2. As shown in Figure 7, increasing the evaporation temperature leads to a higher COP and lower total compression

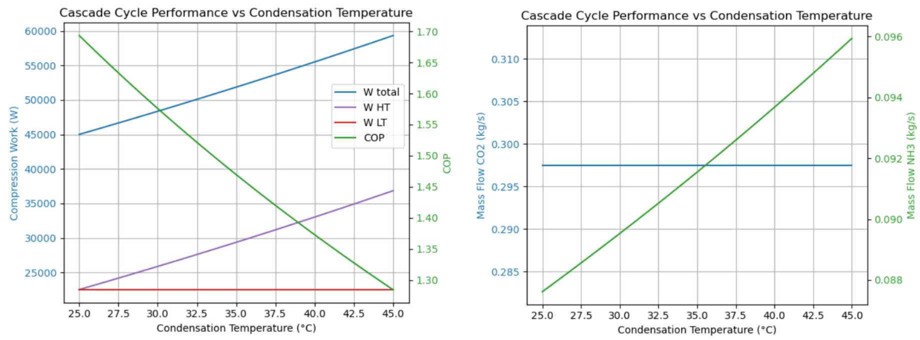

Figure 6: Effect of condensation temperature on COP, compression work, and mass flow rates

work. This is primarily due to the significant reduction in the compression work of the LTC, as state 1 shifts to a higher enthalpy. In practical terms, this means that if the target of refrigeration can be kept at a higher temperature, the system will operate more efficiently.

Regarding the mass flow rates, both decrease with higher evaporation temperatures. The mass flow rate of CO_2 decreases because states 1 and 4 move closer together, reducing the enthalpy difference. For NH_3, the decrease in mass flow rate occurs because the HTC flow is proportional to the LTC flow and also due to the reduced enthalpy difference between states 2 and 3.

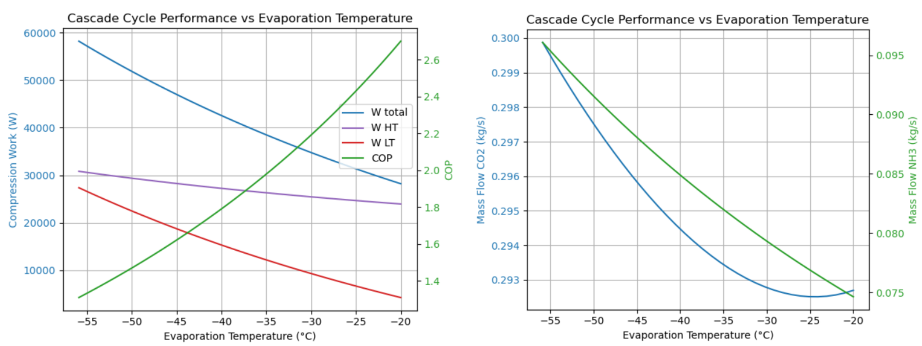

Figure 7: Effect of evaporation temperature on COP, compression work, and mass flow rates

6.4 Supercooling NH_3

We now analyze the impact of supercooling the high-temperature cycle (HTC) using NH_3. As shown in Figure 8, increasing the degree of supercooling (i.e., reducing the subcooling temperature) leads to a higher COP and lower total compression work. This improvement is primarily due to the reduced compression work required by the low-temperature cycle (LTC).

Additionally, the mass flow rate of NH_3 decreases with increased supercooling, further enhancing system efficiency. These results indicate that implementing supercooling in the HTC can be an effective strategy for improving performance.

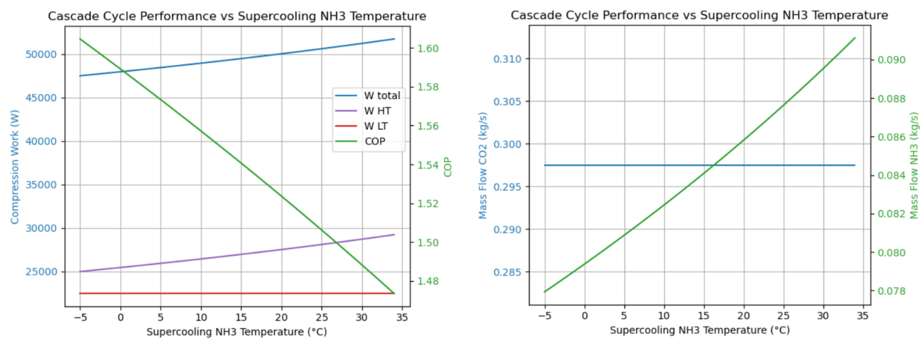

Figure 8: Effect of supercooling NH_3 on COP, compression work, and mass flow rates

6.5 Superheating CO_2

We now analyze the impact of superheating the low-temperature cycle (LTC) using CO_2. As shown in Figure 9, increasing the degree of superheating (i.e., raising the superheating temperature) results in a lower COP and higher total compression work.

Superheating increases the enthalpy of state 1, which leads to a greater enthalpy difference between states 1 and 4, requiring a lower mass flow rate of CO_2. However, the increased enthalpy at state 2 also raises the enthalpy difference between states 2 and 3 in the HTC, which, combined with the interdependence of flow rates, leads to an increased NH_3 mass flow rate.

Overall, superheating negatively affects system performance, making it an undesirable configuration.

7 Conclusions

This study provided a detailed thermodynamic analysis of both a simple ammonia refrigeration cycle and a cascade cycle using ammonia and carbon dioxide. While the baseline comparison revealed the cascade cycle's superior performance—particularly in achieving lower evaporator temperatures with a higher COP and lower specific compressor work—the extended investigations in the *More Advanced Studies* section provided critical insights into system design and optimization.

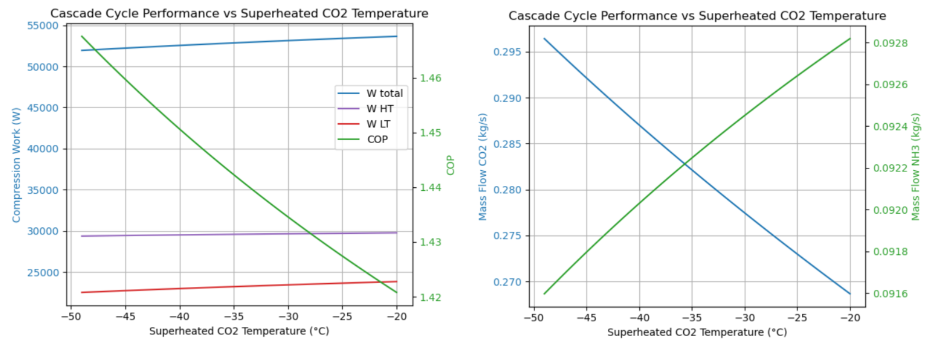

Figure 9: Effect of superheating CO_2 on COP, compression work, and mass flow rates

By conducting a parametric study on the temperature difference across the cascade heat exchanger, we demonstrated the sensitivity of the COP and the refrigerant mass flow rates to practical design considerations. We found that, for a fixed heat rejection requirement, subcooling improves system performance by increasing the COP and reducing the mass flow rate of NH_3. Additionally, lower ambient (sink) temperatures and higher refrigeration (load) temperatures both perform better. A particularly important result is that decreasing the intermediate temperature increases the COP and reduces the mass flow rate of both refrigerants, since NH_3 is a significantly more efficient refrigerant than CO_2. This implies that the intermediate temperature should be as close as possible to the evaporation temperature to maximize performance.

Future work should explore experimental validation, incorporate pressure drops, and assess system performance under dynamic load conditions to further refine the design and practical applicability of such refrigeration systems.

References

[1] Refrigerated & Frozen Foods. (2016, August 9). *Ammonia vs. ammonia/CO_2 cascade systems*. Refrigerated & Frozen Foods. https://www.refrigeratedfrozenfood.com/articles/92807-ammonia-vs-ammoniaco2-cascade-systems

[2] Kasi, P., & Marimuthu, C. (2021). Review of cascade refrigeration systems for vaccine storage. *Journal of Physics: Conference Series*, 2054(1), 012041. https://doi.org/10.1088/1742-6596/2054/1/012041