Technical Paper #1

System Design for Industrial Ammonia/CO_2 Cascade Installations

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Abstract

Restrictions in using CFC, HCFC, and even HFC refrigerants are tightening in most countries. Potentially, this leaves natural refrigerants, such as ammonia, CO_2, propane, butane, etc. as the only practical choices. However, hydrocarbon refrigerants such as propane and butane are highly flammable and explosive and, as such, will probably never be widely used refrigerants. For most industrial refrigeration installations, the choice will be between systems using ammonia alone and systems using cascaded ammonia/CO_2. In addition, many countries/states are increasingly restricting ammonia charge sizes, and many end users do not want ammonia in work, process, or storage areas. As a result, ammonia/CO_2 cascade systems are becoming a popular solution all over the world. This paper describes various system design solutions for cascade systems. Selection of components, sizing of valves, relief valves pipes and vessels are discussed. Finally, the paper discusses safety precautions for operating and servicing ammonia/CO_2 cascade systems.

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Background

In 1866, an American, T.S.C. Lowe became the first known user of CO_2 in a mechanical refrigeration system. Up until the 1940s, a large number of installed refrigeration systems used CO_2 as the refrigerant. However, users had trouble with these systems because:

- The operating pressure for CO_2 is very high
- The triple point for CO_2 (~ 70 psig [4.8 barg] and –70°F [–57°C]) is above atmospheric pressure
- The critical point is at a relatively low temperature (~ 1080 psig [74.5 barg] and 88°F [31°C])
- Automatic control systems did not exist at the time

Due to much less challenging physical properties, ammonia and halocarbon *Freon* refrigerants dominated, and by the 1960s, almost no CO_2 refrigeration systems remained. The halocarbon refrigerants, largely chlorofluorocarbon (CFC) and hydrochlorofluorocarbon (HCFC) chemicals, began to attain near-monopoly status in residential and commercial end uses during this time.

A couple of decades later, public concern began to grow about the environmental damage wrought by releases of Freon. Phase out dates were established for both CFC and HCFC refrigerants, and the public renewed its interest in natural refrigerants, such as ammonia and hydrocarbons (HCs), as well as new halocarbon compounds (hydrofluorocarbons, or HFCs) that did not deplete the Earth's stratospheric ozone layer. However, even though the new HFC refrigerants were found to have global warming properties, the level of interest in natural refrigerants and HFCs varied greatly depending on geographical location.

In Europe, most countries made more aggressive phase out schedules than originally agreed to in the Montreal Protocol, greatly accelerating the demise of CFC and HCFC use. By 2000, CFCs and HCFCs were almost completely out of use in all Western

European countries; even the use of HFCs came into question in many countries because of their global warming properties. Very high environmental taxes and a requirement for frequent leak tests conducted by authorized companies have made it very costly and inconvenient to use HFCs in many European countries. Almost all new refrigeration installations in Europe today use natural refrigerants; the most common choice is ammonia. However, many countries began to issue increasingly strict rules for large ammonia charges, driving end users to seriously consider using CO_2 as a refrigerant. In France, the authorities have even recommended that end users with large ammonia charges install new CO_2 systems instead of expanding their existing ammonia systems.

Other drivers, such as high energy costs and potentially lower energy consumption (especially in low temperature systems) favored CO_2 as a refrigerant. Since the beginning of this millennium, use of CO_2 in industrial systems has really accelerated. This trend is quite evident: at one of the major European refrigeration companies, the count of new CO_2 installations was 4 in 2001, 18 in 2002, 23 in 2003, and 27 in 2004.

In North America, the driving factors so far have not been as strong as in Europe. HCFCs will not be totally phased-out before 2020 (65% by 2010, 90% by 2015 and 99.5% by 2020) and no major restrictions apply to the use of HFCs, although these refrigerants are very expensive. Even with recent hikes, energy prices remain relatively low compared to global prices. However, energy prices continue to increase and restrictions on larger ammonia charges already apply. In fact, many US states are becoming stricter. Thus, interest in CO_2 is rising in the US as well.

Introduction

A few years ago, when CO_2 was reintroduced to the industrial refrigeration world, the system configuration was different from what had been used in the past. To avoid very high pressures, cascade designs were specified, in which CO_2 is only used in the low stage. Another refrigerant, typically ammonia, is used in the high stage. To move heat between the two stages, a cascade heat exchanger (CHE) is used. The CHE serves as condenser for the CO_2 side and as an evaporator for the ammonia side. The CHE can be installed in parallel with the intercooler in a conventional two-stage plant; the two are different in that for the CHE to exchange heat, the temperature of the two refrigerants must be different. Resistance to heat flow in the CHE causes an efficiency penalty that a conventional two-stage system does not experience. However, in most low temperature applications, this efficiency loss is more than redeemed by the greater efficiency of the CO_2 compressor.

Ammonia/CO_2 Cascade System Advantages

The first-cost and operating cost advantages of an ammonia/CO_2 cascade system over a conventional two-stage or economized single stage ammonia system depend on:

- Actual operating conditions
- Selection of intermediate temperature
- Selection of CHE, compressor, piping, valves, and other components

Actual costs and benefits must be calculated after the application engineering is complete and optimized. In general, the size of the ammonia charge in the cascade system is typically only 10–20% of the charge in a conventional system. The exact size is difficult to quantify beforehand and depends on local rules and regulations for the use of ammonia. In addition, the benefits associated with not having ammonia in working/processing/storage areas are difficult to quantify and must be evaluated on a case-by-case basis.

Operating Costs

Ammonia/CO_2 cascade systems cost less to operate than conventional ammonia-only systems. Figure 1 illustrates the overall COP at various evaporating temperatures for both a cascade and conventional system (two-stage and economized single stage). At –25°F [–32°C] evaporating temperature, the three systems are about even. At temperatures lower than –25°F [–32°C], the ammonia/CO_2 cascade system improves relative to the other systems; the lower the temperature, the greater the improvement. The gap between the cascade and the conventional system is especially pronounced in the case of the economized single stage system, where the difference is 18% at –40°F [–40°C], 33% at –50°F [–46°C], and 43% at –60°F [–51°C]. This last point is important because economized single stage conventional systems working at –40°F [–40°C] or –50°F [–46°C] are not unusual in the industry. In the case of a two-stage conventional system, the gap is smaller at 5% at –40°F [–40°C], 8% at –50°F [–46°C], and 12% at –60°F [–51°C].

The performance differences only account for the compressors; in an actual cascade system, the condensers would manage less load and, thus, consume less power. The performance differences also only apply at full load conditions. In most industrial plants, loads vary during the day, week, and year. The CO_2 sides of most installed ammonia/CO_2 cascade systems use reciprocating compressors, which have much better part-load characteristics than screw compressors. Figure 2 illustrates the COP for the three systems at 50% part-load. The reader may note that the COP for the cascade system is 2 to 3 times greater than that of the economized single-stage ammonia system between –45°F [–43°C] and –60°F [–51°C]. At –40°F [–40°C], the COP is 80% greater, and at –30°F [–34°C], 50% greater. Also the COP gap relative to the two-stage ammonia system is more significant now: 42% at –60°F [–51°C], 29% at –50°F [–46°C], 20% at –40°F [–40°C], and 14% at –30°F [–34°C]. Because loads vary so much, owners may save greatly in operation costs by selecting an ammonia/CO_2 cascade system for low temperature systems.

First Cost

Some properties of CO_2 differ significantly from ammonia, especially gas phase density at low temperatures. Because CO_2 is much denser than ammonia, smaller components are required to handle the same mass:

- Compressors are typically 8–12 times smaller. Figure 3 illustrates this point by plotting compressor capacity at different evaporating temperatures for CO_2 (at the required greater pressure) vs. ammonia.
- Significantly, smaller suction pipes (typically 2 to 3 sizes smaller) may be used to achieve an equivalent pressure/temperature drop.
- Liquid-gas separators are smaller as well. For example, a surge drum for CO_2 can often be half the diameter of an ammonia surge drum at the same conditions.

The situation reverses for liquid density of the two refrigerants: for equal pressure losses, CO_2 liquid lines would be typically one size larger than those for ammonia. Pipe and vessel sizing will be discussed more detailed below.

Operating pressure for CO_2 is significantly higher than for ammonia for equal application temperatures, requiring more substantial compressors, vessels, valves, etc. In addition, a cascade system requires an expensive CHE instead of a relatively inexpensive intercooler or economizer vessel.

In summary, an ammonia/CO_2 cascade system requires less expensive:

- Compressors (LT side)
- Suction piping, valves, insulation, etc.
- Vessels for liquid separation (plus required insulation)

…but *more expensive*:

- Interstage vessels (CHE vs. intercooler)
- Liquid piping, valves, insulation, etc.
- Components that require pressure ratings suitable for CO_2

The appropriate weight to give these factors depends a lot on the actual case, and several other factors come into play, e.g., size of the plant, temperature levels required, piping length between components, selected temperature difference in CHE. However, in many cases the first cost for an ammonia/CO_2 cascade system will be 5–10% lower than for a conventional two-stage ammonia system, and about the same as for an economized single-stage ammonia system.

Probability and Consequences of a Refrigerant Release

Under normal circumstances, the benefits of ammonia contained in a refrigeration system far outweigh the risks. The unlikely event of an unintended release may pose a potential risk to plant workers and food products, depending on magnitude and location. However, based on public perception of this risk, many countries/states restrict ammonia charge sizes. Thus, owners have a real incentive to minimize charges.

In an ammonia/CO_2 cascade system, the ammonia charge is typically only a small fraction of the charge in a conventional ammonia system. Further, the ammonia is limited to the engine room and the condenser. Using a plate-and-frame water-cooled condenser in combination with a cooling tower can reduce ammonia charge further. See Case Study 2 below for an example of this type of system.

In production and storage areas, the cascade system only contains CO_2 which, with a few precautions taken, can be rendered practically harmless. In the event of a major leak, no products would be destroyed. The short-term (10 minute exposure) concentration limit for people is 3%, a relatively large amount. For exposure to CO_2

to be health- or life-threatening, concentrations must increase to 10–20%, which would take a considerable amount of time. In this amount of time, it would usually be possible for people to get out of the room, or to be rescued. Most likely, in the event of a major leak, most of the escaped CO_2 would form CO_2 ice/snow, which would eventually sublimate to CO_2 gas. CO_2 ice/snow on products would not be damaging at all, and could easily be wiped clear of the room.

Note: CO_2 still poses some amount of risk, however small, so gas detectors are required in rooms where CO_2 systems are installed. Normally, the detector is set to alarm at a 1% concentration, which is the long-term exposure limit in most countries. Finally, CO_2 is not flammable or explosive in any concentration; in fact, it is commonly used in fire extinguishers.

Service and Maintenance

Daily maintenance and service of an ammonia/CO_2 cascade system is similar in many respects to a conventional ammonia system, but is typically quicker and easier. Like ammonia, the high operating pressure and fast-rising hydrostatic pressure of liquid trapped in isolated components must be managed carefully. In addition, serviced components must be evacuated of air and moisture before start-up.

Evacuation is particularly critical in CO_2 systems because, unlike ammonia, CO_2 cannot absorb and does not tolerate much water. In addition, technicians must act to prevent dry ice from forming when opening a component for service.

Neither blowing refrigerant out into water basins (as with ammonia) or pumping refrigerant out with recovery units (as with Freon) is required. After isolating a component, the CO_2 contained within can simply be released into the atmosphere. In addition, when the component opened for service, no extra time is required waiting for the refrigerant smell to dissipate.

Oil draining is only necessary on the ammonia side of the system. On the CO_2 side, an oil recovery system is typically installed to automatically return oil back to the compressors.

At initial start-up and during service, air and moisture may potentially contaminate a cascade system. However, during normal operation, the CO_2 side of the system always operates at a positive pressure in all areas of the plant, thereby preventing air and moisture from entering. Therefore, no air purger is necessary but filter-driers are required to remove water from the system.

General System Design

Various System Applications

From an operational cost point of view, ammonia/CO_2 cascade systems begin to pay off when evaporating temperatures are lower than –25°F [–32°C]. So far, end users have installed these systems mostly in food freezing applications (e.g., ice cream, seafood, poultry, meat) at low temperatures (–40°F to –60°F [–40°C to –51°C]). Ammonia/CO_2 cascade systems also help to minimize ammonia charges and eliminate ammonia from working, processing, and storage areas. Figure 4 shows an example of a typical ammonia/CO_2 cascade system for a freezing application.

Most applications use pumped liquid recirculation systems similar to conventional ammonia systems. Typically, they run intermittently, which means that auxiliary equipment must be installed to maintain the CO_2 pressure below design when the main system is shut down. Most end users do this by automatically emptying the CO_2 liquid from the evaporators into the insulated CO_2 pump recirculator vessel and then starting up a small independent condensing unit (2-5 HP [1.5-3.7 kW]) with an evaporator coil in the top of the vessel to maintain a pressure setpoint. The vessels must be equipped with safety relief valves in case of power failure or malfunction of the condensing unit.

In Europe, cold storage warehouses often employ ammonia/CO_2 cascade designs. The reduced ammonia charge and elimination of ammonia from the storage area appeals to these end users. Also, the lower first cost has been a driving factor, and because many facilities also conduct blast freezing at low temperatures, lower operational costs enter the picture.

Normally, cold storage systems operate continuously. The main compressors either run or are on standby all the time, so these systems typically are not equipped with auxiliary equipment to hold pressure. Like the low temperature freezing applications mentioned above, most cold storage applications use pumped recirculation designs. The same concept has been used in numerous supermarket installations in Europe.

Many cold storage systems operate at two low temperature levels: one for freezing (typically –40°F to –50°F [–40°C to –46°C]) and one for storage (typically –20°F to –30°F [–29°C to –34°C]). In a few applications, CO_2 is also used for high temperature rooms (e.g., coolers or dock areas) at temperatures around 40°F [4°C]. Figure 5 shows an example on a typical ammonia/CO_2 cascade system for a cold storage application with freezer and cooler load.

For a high temperature room, liquid around 20°F [–7°C] is recirculated from the CO_2 high-pressure receiver. The small amount of gas that forms will migrate to the cascade cooler where it is re-condensed. In this way, CO_2 is used as an evaporating brine on the high temperature side—no CO_2 compressors are involved. Compared to glycol or other brines, CO_2 is very efficient, requiring much smaller pipes, smaller pumps, and smaller air units. CO_2 also is used this way in ice rinks.

Temperatures and Pressures

As with any refrigerant, the design evaporating temperature for CO_2 is determined by the requirements of the application at the plant. At the triple point of any refrigerant, a solid phase forms, so this is the absolute minimum practical evaporating temperature. For CO_2, the triple point is –69.9°F [–56.6°C]. To provide a safety margin that gives the control system time to react in the event of a suddenly decreasing load, the minimum evaporating temperature is generally considered to be around –65°F [–54°C].

The condensing temperature on the ammonia side is determined by meteorological conditions at the plant location and by the condenser type. Typically, for an evaporative condenser in an industrial refrigeration plant, 95°F [35°C] is selected.

What remains is to determine the intermediate temperatures, i.e., the condensing temperature on the CO_2 side and the evaporating temperature on the ammonia side, and the temperature difference (ΔT) between the two refrigerants in the cascade heat exchanger. A small ΔT will yield a high overall COP for the system and will require slightly smaller compressors, but it will also require a larger CHE. In addition, in some applications, the intermediate temperature may serve a refrigeration load, which also affects which intermediate temperature to choose. Case Study 1 shows an example of a system that includes an intermediate load.

The author investigated the effects on first cost and operating cost of different component configurations at different ΔTs for the CHE. Operating conditions on the CO_2 side were fixed at –40°F [–40°C] evaporating temperature and 23°F [–5°C] condensing temperature, with the ammonia side at 95°F [35°C] condensing temperature and various evaporating temperatures ranging from 4–20°F [–16 to –7°C]. Thus, many ΔTs, from 3–19°F [2–11K], were evaluated.

Figure 6 illustrates the effect of the different combinations on first cost of the compressor and CHE. At low ΔTs, the CHE was larger and more expensive while the compressor was smaller and less expensive. At high ΔTs, the opposite occurs. Adding the two curves together yields the total first cost of the capital investment (i.e., the part of the system affected by the choice of ΔT). The minimum total capital investment occurs around 11°F [6K].

Figure 7 incorporates the investment cost calculated above into a financial analysis that also includes the total electrical cost (net present value) for running the compressors. These two curves are added together to yield the total financial investment. At a ΔT around 7°F [4K], the total investment is minimum, representing the optimal choice. Many plants are designed with a 9°F [5K] ΔT, which more highly weights the minimum first cost, and this value is used in the continuing analysis below.

Having decided on ΔT, the intermediate temperature level still must be determined. For a conventional two-stage ammonia system, the theoretical optimal intermediate pressure P_i is determined by the equation:

$$P_i = (P_e \text{ x } P_c)^{1/2} \tag{1}$$

where:

P_c = Condensing pressure [psia/bara]

P_e = Evaporating pressure [psia/bara]

As an example, for a two-stage ammonia plant with a –40°F [–40°C] evaporating temperature (~10.5 psia [0.724 bara]) and a 95°F [35°C] condensing temperature (~196 psia [13.5 bara]), the optimal intermediate pressure P_i would be

$(10.5 \times 196)^{1/2} = 45.4$ psia [3.13 bara], which corresponds to a 17.5°F [–8°C] saturated intermediate temperature.

For an ammonia/CO_2 cascade system, special attention must be paid to the CO_2 condensing pressure. Design pressures for many components, including compressors, heat exchangers, vessels, and valves, are generally about 350–400 psi [24–28 bar]. A practical safety margin of 30 psi [2 bar] limits CO_2 condensing pressure correspondingly to 320–370 psi [22–26 bar], which is equivalent to a saturated temperature of 6°F to 14°F [–14°C to –10°C].

With increasing use of CO_2 in Europe over the past five years, manufacturers have responded with product lines featuring design pressures around 600 psi [~40 bar], permitting CO_2 condensing temperatures to approach 40°F [4°C]. However, these high-pressure components are often more expensive, so two questions must be asked:

1. Is there a case in which these more expensive components can be justified?
2. Are these components necessary or desirable?

To examine the benefits of higher condensing temperatures/pressures, the author compared the total system COP for two systems, each operating at –40°F [–40°C] evaporating temperature and 95°F [35°C] condensing temperature, and having a capacity of 150 TR [530 kW]:

- An ammonia/CO_2 cascade system with an intermediate ΔT of 9°F [5K]
- A conventional two-stage ammonia system with open intercooler

Figure 8 shows that for increasing intermediate temperatures, the low stage compressor COP for both systems decreases while the high stage compressor COP increases.

Figure 9 shows total COP (low plus high stage) at various high stage evaporating temperatures. The cascade system exhibits optimal performance between

12°F [–11°C] and 16°F [–9°C] evaporating temperature, corresponding to an optimal CO_2 condensing temperature (at an intermediate ΔT of 9°F [5K]) of between 21°F [–6°C] and 25°F [–4°C]. At these temperatures, saturated pressures for CO_2 are between 400–450 psig [28–31 bar]. Therefore, to maximize the benefits of low operating costs for an ammonia/CO_2 cascade system, one must condense the CO_2 at higher pressures than what normal pressure ratings would allow.

For the two-stage ammonia system, the optimal intermediate temperature is around 20°F [–7°C], in close correspondence with the theoretical optimal value.

In Figure 9, the curve for the cascade system also shows that the power consumption is relatively sensitive to changes in intermediate temperature. On the curve, system COP decreases quickly on either side of the optimal intermediate temperature. The curve for the two-stage ammonia system is much flatter around the optimal temperature; as a result, power consumption is much less sensitive to the choice of intermediate temperature.

Finally, the curves confirm that the ammonia/CO_2 cascade system has a higher total theoretical system COP.

Compressors

As mentioned above, because of the much greater density of CO_2 gas, compressors of equal displacement typically have an 8–12 times larger refrigerating capacity operating with CO_2 than with ammonia.

Reciprocating compressors (*recips*) are best used in applications with high differential pressures, high suction pressures, and low compression ratios. Screw compressors (*screws*) perform better at low pressure differentials. As a result, with CO_2, recips typically exhibit 10–15% better COP than screws. In addition, the high relative capacity with CO_2 makes it realistic to run even large industrial plants with recips on the CO_2 side.

Recips for CO_2 use are currently available with capacities up to 170 TR [600 kW] at –40°F [–40°C] evaporating temperature and maximum condensing temperatures around 40°F [4°C] and have a design pressure of 580 psi [40 bar] which means that optimal intermediate temperatures can be reached easily. The smallest screws available today for CO_2 applications have capacities around 200 TR [700 kW] at –40°F [–40°C] with maximum condensing temperatures around 18°F [–8°C]. Small rotor diameter and the high pressure differential for CO_2 screws present problems: the COPs are not very good, and they are not able to run at optimal intermediate temperatures.

Cascade Heat Exchangers (CHEs)

The CHE is the most critical and the most expensive component in an ammonia/CO_2 cascade system. Design pressure on the CO_2 side is fairly high (500–600 psi [34–41 bar]). Leaks in the CHE may cause mixing of CO_2 and ammonia, which would result in serious problems and may be fatal to the refrigeration system.

Three types of heat exchangers may be used in CHE applications:

- **Plate-and-frame** construction is very efficient, uses a very small charge of ammonia, and will not cause refrigerants to mix in the event of a weld leak. However, at these high pressures, gaskets do not work, so fully welded construction must be used. This type is the most expensive of the three options.
- **Shell-and-tube** units are available for this application, but this option has a large footprint and requires a relatively large ammonia charge. To lower the risk of mixing refrigerants at leaks in tube/tube-sheet joints, some shell-and-tube units are made with double tube-sheets with *neutral* space between the two tube-sheets. This type of construction gives a *warning* if a leak occurs. For relatively small CHEs, a shell-and-tube heat exchanger is at the lower range of prices.

- **Shell-and-plate** construction is probably the most-specified type. Advantages include relatively low cost, high efficiency, small footprint, and moderately low ammonia charge. The downside is that the plate stack (in which the CO_2 condenses) is submerged in ammonia; any leak will cause immediate mixing of the refrigerants. However, precautions can be taken to avoid total shutdown of the system, including: (1) redundant heat exchangers that can be isolated, and (2) alarm systems to detect leaks before they becomes fatal to the system. The industry now has a lot of experience using this type of heat exchanger as a CHE and leaks are very rare.

Evaporators

Most of the evaporators used in ammonia/CO_2 cascade systems are air coolers or plate freezers. If design pressures for the actual application are suitable, standard air coolers/plate freezers for either ammonia or Freon can be used for CO_2, and would yield approximately the same capacity at the same temperatures. Tube and channel velocities will be much smaller for CO_2, but the better heat transfer properties for CO_2 compensate for that. In the last few years, evaporators and plate freezers designed especially for CO_2, with smaller tubes/channels and higher design pressure, have been introduced on the market. Their customized designs have optimized both heat exchange performance and cost.

Defrost

In general, CO_2 evaporators can be defrosted according to the same principles as with conventional ammonia systems. However, hot gas defrost, which is the most common defrost method for conventional systems, becomes more complicated with CO_2. To obtain the minimum temperature required for hot gas defrost, roughly 55°F [13°C], the CO_2 pressure approaches 700 psig [48 bar]. This great pressure requires specially constructed components, including evaporators, compressors, valves, and regulators, and availability can be spotty. Even when these specialty components are available in the market, the price can be relatively high. Therefore, alternative defrost methods, e.g., electrical resistance, water, air, have been used in most of the existing ammonia/CO_2 cascade installations.

In some installations, warm glycol has been used. In this case, the heat source is a small glycol-cooled condenser in the ammonia circuit, which provides warm glycol to a separate glycol circuit in the air units. This type of system has nearly zero marginal operating costs, and its first cost is similar to electrical resistance defrost.

Vessels

Ammonia/CO_2 cascade systems require the same types of vessels as ammonia systems. In addition, except for design pressures in general, vessels for cascade systems only differ in the rules for sizing liquid separators. Where separation ability is the sole criterion considered for sizing, liquid separators and pump recirculators for CO_2 could even be smaller in diameter than those for ammonia.

In calculating the maximum allowable velocity, W_{max}, in a vertical liquid separator, the following formula can be used:

$$W_{max} = \frac{C(\rho_{gas} - \rho_{liquid})^{0.25}}{\rho_{gas}} \qquad (2)$$

where:

C	=	a constant specific to the refrigerant
ρ	=	the density of the refrigerant

In the –10°F to –40°F [–23°C to –40°C] range of evaporating temperatures, this formula yields the following ratio for CO_2 and ammonia:

$$W_{\mathrm{maxCO_2}} = 0.45 W_{\mathrm{maxNH_3}} \qquad (3)$$

For equivalent refrigerating capacity, the volume flow for CO_2 relative to that for ammonia is only about 10%:

$$V_{CO_2} = 0.1 V_{NH_3} \tag{4}$$

The ratio of the cross-sectional area required in the separator for CO_2 relative to ammonia is:

$$\frac{A_{CO_2}}{A_{NH_3}} = \frac{V_{CO_2}/W_{maxCO_2}}{V_{NH_3}/W_{maxNH_3}} = \frac{0.1}{0.45} = 0.22 \tag{5}$$

The ratio of the required diameter in the separator for CO_2 relative to ammonia becomes:

$$\frac{D_{CO_2}}{D_{NH_3}} = \left(\frac{A_{CO_2}}{A_{NH_3}}\right)^{0.5} = 0.47 \tag{6}$$

One may conclude that the diameter for a CO_2 liquid separator would be around half of the diameter of a similar ammonia separator. However, depending on the overall system design, the separator might also be called on to function as the main liquid reservoir and would need additional capacity to store all the CO_2 liquid in the system at shutdown, as well as the total surge volume.

Piping

As mentioned earlier, for equivalent refrigerating capacity, pipe sizes for CO_2 are different from those for ammonia.

As an example, pipe sizes were calculated for a 300 TR [1100 kW] system operating at –40°F [–40°C] evaporating temperature with a recirculating rate of 1:3. Wet suction pipe sizes were calculated, and the results are shown in Table 1.

For a pressure drop equivalent to the same temperature loss, the size required for ammonia is 12" [300 mm], much larger than the 6" [150 mm] needed for CO_2. For liquid pipes having approximately the same velocity and pressure loss, the size required for ammonia would be 2½" [60 mm], much smaller than the 4" [100 mm] required for CO_2. For a pressure drop equivalent to the same temperature loss in discharge pipes, the size required for ammonia is 6" [150 mm]; for CO_2, 4" [100 mm].

Most existing CO_2 systems have been installed using the same practices regarding pipe wall thickness as for conventional ammonia installations, i.e., steel piping, Sch 40 for pipes larger than 2" [50 mm], and Sch 80 for all other sizes. Copper pipes can be used as well for CO_2.

Valves and Other Components

CO_2 systems use the same basic types of equipment as ammonia systems, e.g., stop valves, solenoid valves, check valves, pressure regulators, strainers. However, the maximum operating pressure of each component must be checked against system design pressures. Many available components are only rated for 400 psi [28 bar], which in most cases will not be enough for the high side of the system, and at high evaporating temperatures, not enough for the low side either. Some valve manufacturers now offer special CO_2 product lines rated to 600 psi [41 bar], and one valve manufacturer has upgraded all of its product lines to this pressure.

For relief valves, a good quality, re-seating type is recommended, as relief valves may pop off easily relative to those for ammonia. In ammonia and Freon systems, pop-offs are rare and relief valves in these systems are often not very reliable regarding re-seating.

For sizing of relief valves the well known formula below is used:

$$C = fDL \tag{7}$$

where:

C	=	Capacity required (lbs/min of air)
D	=	Diameter of vessel (ft)
L	=	Length of vessel (ft)
f	=	A refrigerant specific constant (0.5 for ammonia, 1.0 for CO_2)

The following precautions must be taken when installing relief valves in CO_2 systems:

- Relief valves on the liquid side cannot blow to atmosphere because dry ice will form and block the outlet. The outlet must be connected to the low side of the system.
- Where relief valves are connected to low side vessels containing saturated gas, these valves will produce dry ice in the form of CO_2 *snow* when blowing. Owners should avoid long relief headers by extending the inlet connection from the vessel to the relief valve outside the building and then install the relief valve outdoors without piping or with minimum piping on the outlet.
- Only relief valves from superheated gas, for example oil separators, can be installed with common relief headers.

Oil Management

Oil is lighter than CO_2 liquid; insoluble oil will float on top of the CO_2. In a majority of the installed ammonia/CO_2 cascade systems, a fully soluble polyolester oil (POE) has been used on the CO_2 side. An oil rectifier can recover this oil from the low temperature side, as shown in Figure 10.

The oil rectifier is principally a shell and tube heat exchanger, which, on the shell side, is passed by the high-pressure liquid before the liquid, is throttled to the low-pressure side. The tube side is connected to the bottom of the surge drum, meaning that low-pressure liquid is boiled off, and the remaining oil is directed to the suction line.

The capacity for the oil rectifier can be calculated as:

Plant capacity x 10^{-4} x Oil carry over (ppm)/Oil concentration (%)

However, at a minimum, the oil rectifier liquid supply should be at least 1% of the plant capacity. The oil rectifier does not affect the plant efficiency because the liquid used subcools the remaining plant liquid.

Typically, the oil rectifier is sized to maintain a concentration of 1% oil in the CO_2 charge. Oil carryover from a reciprocating compressor with a standard oil separator is typically 10 – 20 ppm for CO_2 operation.

Case Study #1

Cold Storage Application

The facility under consideration, a cold store built in Pennsylvania in 2004, is 175,000 ft^2 in area and 6,000,000 ft^3 in volume [16,000 m^2, 170,000 m^3]. It contains four high-rise freezer rooms designed for room temperatures down to –20°F [–29°C] and two dock areas designed for room temperatures down to 35°F [2°C]. Two of the freezer rooms are convertible, and can be switched from freezer operation to cooler operation. The present building is Phase 1 of 4; one of the future phases will include a blast freezer operating at –45°F to –50°F [–43°C to –46°C].

The ammonia/CO_2 cascade system was compared with a two-stage ammonia system; the ammonia/CO_2 cascade system was chosen for the following reasons:

- Lower first cost for refrigeration installation (5–10% less)
- Low ammonia charge, 2500 lbs [1100 kg] in Phase 1; approximately 8000 lbs [3600 kg] total for all 4 phases
- No ammonia required in storage areas
- Potential savings on operation cost with blast freezing. (In present configuration, no operational cost savings were predicted.)

Main System Components and Characteristics

A diagram of the present system is shown in Figure 5 (without blast freezing). The CO_2 side of the system consists of the following:

- One 8 cylinder CO_2 compressor: 206 TR at –30°F/20°F, 162/407 psig [724 kW at –34°C/–7°C, 11.2/28.1 barg]
- One 4 cylinder CO_2 compressor: 103 TR at –30°F/20°F, 162/407 psig [362 kW at –34°C/–7°C, 11.2/28.1 barg]
- One vertical CO_2 low temperature pump recirculator: (60"x151", 350 psig DWP) [152 cm x 384 cm, 24 barg DWP] with two 10 HP [7.5 kW] canned pumps, 3:1 recirculation rate. Vessel sized for all 4 phases.
- One horizontal CO_2 high-pressure receiver/high temperature pump recirculator: 54"x151", 600 psig DWP [137 cm x 384 cm, 41 barg DWP] with two 10 HP [7.5 kW] canned pumps, 3:1 recirculation rate. Vessel is sized for all 4 phases.
- In freezers, total 8 penthouse evaporators, 290 TR at –30°F/–20°F, 600 psig DWP [1020 kW at –34°C/29°C, 41 barg DWP], electrical defrost.
- In dock areas and USDA room, total 11 hanging evaporators: 165 TR at 20°F/35°F, 600 psig DWP [580 kW at –7°C/2°C, 41 barg], electrical defrost.
- Two shell-and-plate CHEs: total 595 TR at 11°F [2090 kW at –12°C] ammonia evaporating temperature and 20°F [–7°C] CO_2 condensing temperature. Connected to common ammonia surge drum for flooded operation. Present cascade coolers only serve Phase 1.

The ammonia side is equipped with:

- Two screw compressors: 222 TR and 307 TR at 11°F/95°F [781 kW and 1080 kW at –12°C/35°C]
- One evaporative condenser with high side floats on the outlets
- One control pressure receiver at 55°F [13°C]. Vessel is sized for all 4 phases.

The main piping on the roof is sized for all 4 phases. All valves, strainers etc. are rated for 600 psig [41 barg].

The present CO_2 charge is 25,000 lbs [11 tonnes]. In all freezers and coolers and in the engine room, CO_2 detectors are installed. The alarm setpoint is 1% CO_2 in air. Ammonia detectors are only installed in the engine room.

Evaporators in freezers are all equipped with ducted outlets and air intake hoods. This keeps the warm air inside the evaporator during defrost. Defrost time after pump down is 10–15 minutes.

The ammonia surge drum for the CHE is fed liquid from the control pressure receiver via a modulating motorized expansion valve, which is controlled by a level probe in the surge drum.

The low temperature CO_2 pump recirculator is also equipped with a modulating motorized expansion valve controlled by a level probe in the vessel. Liquid fed from the CO_2 high-pressure receiver/high temperature pump recirculator first passes through drying filters and an oil rectifier. The tube side of the oil rectifier is fed with a CO_2 /oil mixture from the drop leg to the pumps, and the boiled off gas/oil mixture from the oil rectifier's outlet is piped to a small surge drum/oil distribution vessel. From the vessel, the gas is piped to the CO_2 compressor suction line, while the oil is collected in the vessel. The CO_2 compressors are equipped with an electrical float switch on the crankcase; upon sensing a low oil level, the switch opens an oil fill solenoid to allow re-supply from the oil vessel. The oil vessel is furnished with an electrical heater to ensure that no liquid CO_2 remains in the vessel, and oil filling is only allowed if the oil temperature is higher than the CO_2 temperature. In the beginning, this system did not return any oil, but after operating approximately six months and after 2–3 oil refills of the CO_2 compressors, the system was in balance and no more oil needed to be added.

The CHE with the ammonia surge drum is located over the CO_2 high-pressure receiver/high temperature pump recirculator. The CO_2 compressor discharge is piped to the CHE inlet, and CO_2 condensate drains by gravity to the CO_2 high-pressure receiver/high temperature pump recirculator. The wet suction return from the high temperature CO_2 evaporators is piped into the vessel and oversized vent lines are connected from the vessel to the cascade cooler inlet.

The system has been running without problems since startup in April 2005. While theoretical calculations did not show any operational cost savings, in actuality, this plant has the lowest power consumption per cubic foot of any of the owner's 30 similar cold storages that use ammonia two stage systems. This result can probably be attributed to better part load characteristics for the ammonia/CO_2 cascade system, less pressure loss in the CO_2 piping, and the customized air intake hood on the freezer evaporators which prevents warm air from escaping into the room during defrost.

Case Study #2

Shrimp Freezing Application

This case study considers a standard factory built ammonia/CO_2 cascade system installed in Thailand at a large seafood producer. This facility freezes shrimp on an IQF belt freezer, after which the shrimp are glazed and an after-freezer hardens the shrimp.

The 2,200 lbs per hour [1 tonne per hr] production is served by an 82-TR [290 kW] ammonia/CO_2 freeze package running at –63°F [–53°C] evaporating temperature. Air temperature in the freezer is –52°F [–47°C].

The seafood producer used to freeze the shrimp in a cryogenic gas freezer with CO_2 gas sprayed directly on the product. This process is very effective due to a very low freezer temperature, but also very expensive to operate. Often, these freezers are supplied and installed free of charge if the customer contracts to buy gas from the supplier for a number of years. In this way, the customer can get freezing equipment without any investment. However, operating costs for operating this system are approximately US$200 per hour, or approximately US$45,000 per month.

In the summer of 2003, the gas supply contract terminated for one of the cryogenic lines. The owner wanted something more economical to operate, did not want to make a large investment, and did not want to run and maintain a conventional plant; therefore, a lease/finance agreement was constructed. The refrigeration contractor pledged to deliver, install and run a plant with the ammonia/CO_2 package and belt freezers. The owner committed to pay monthly 75% of what he used to pay for gas monthly, minus electricity costs for the new plant (which are only about 15% of the gas cost.) After four years, the contract would terminate, and the plant would be handed over to the owner free of charge.

The principal diagram for the system is as shown in Figure 4.

Main System Components and Characteristics

- One 6 cylinder CO_2 compressor: 82 TR at –63°F /20°F [290 kW at –53°C/–7°C], 200 HP [150 kW] motor with VFD drive (1100–1500 RPM)
- One 8 cylinder ammonia compressor: 113 TR at 13°F/95°F [397 kW at –11°C/35°C], 200 HP [150 kW] motor with VFD drive (750–1500 RPM)

- One horizontal CO_2 low temperature pump recirculator: 36"x 108", 350 psig DWP [91 cm x 274 cm, 24 barg], with two 5 HP [3.7 kW] canned pumps, 3:1 recirculation rate. The vessel is equipped with an evaporating coil in the upper part. The evaporator coil is connected to a 1 TR [3.5 kW] air-cooled condensing unit using R-134a, which keeps the CO_2 pressure at approximately 300 psig (2°F) [21 barg (–17°C)] when the main compressors are not running.
- One main belt freezer: 70 TR at –63°F/–52°F, 350 psig DWP [246 kW at –53°C/–47°C, 24 barg] for 2,200 lbs [1 tonne] shrimp per hour from 50°F to –10°F [10°C to –23°C], equipped with water defrost.
- One hardening belt freezer: 10 TR at –63°F/–52°F, 350 psig DWP [35 kW at –53°C/–47°C, 24 barg] for 2,300 lbs [1.05 tonne] glazed shrimp per hour from 0°F to –10°F [–18°C to –23°C], equipped with water defrost.
- One shell-and-tube CHE: 115 TR at 13°F [404 kW at –11°C] ammonia evaporating temperature and 20°F [–7°C] CO_2 condensing temperature. The CHE operates flooded and has an integrated surge drum. It has double tube sheets to minimize the risk of mixing CO_2 and ammonia if it leaks. The condensed CO_2 drains into a pilot receiver that has a level probe. The level probe controls a motorized modulating expansion valve for liquid make up to the CO_2 low temperature pump recirculator via an oil rectifier. The oil rectifier returns oil to the CO_2 compressor. On the ammonia side, the cascade cooler is equipped with an automatic oil return system to the ammonia compressor. The cascade cooler is designed for 580 psi [40 bar] DWP on the tube (CO_2) side and for 350 psi [24 bar] DWP on the shell (ammonia) side.
- The ammonia side is equipped with a water-cooled plate and frame condenser with a high side float on the outlet, expanding liquid to the cascade cooler. The water-cooled condenser is connected to a cooling tower.

The system contains a total of 2,000 lbs [900 kg] CO_2 and 250 lbs [110 kg] of ammonia.

Both compressors are equipped with VFD drives on the motors. VFDs allow continuous capacity control that can maintain temperatures within 0.2°F [0.1K]. In addition, when operating with a setpoint only 2°F [1.1K] from the low evaporating temperature cutout, the stepless control avoids unnecessary failures.

The evaporator coils in the belt freezers are defrosted with water during the daily cleaning when production is stopped. Solenoid valves in the liquid and suction lines are closed, and a solenoid valve in a drain line from the bottom of the evaporator is opened at the same time. The CO_2 pressure in the coils increases rapidly, forcing all liquid into the drain. The drain line is connected to the top of the wet return line, in which the liquid can flow back to the CO_2 low temperature pump recirculator by gravity.

The system has been running without problems since startup in August 2003.

Conclusion

With all the benefits by using CO_2 as refrigerant, there is no doubt that CO_2 is not only a refrigerant of the past but also a refrigerant of the future.

For low temperature applications, substantial operational savings can be achieved. Where first costs are lower, the choice is obvious.

Also, the opportunity to reduce ammonia charges dramatically and eliminate ammonia from working/process/storage areas is a huge benefit.

More than a hundred ammonia/CO_2 cascade systems have been installed worldwide, of which the oldest have been in operation for more than 5 years. This proves that the system concept is reliable and safe. All known users of these systems are satisfied with the operation and performance.

So far, only a few of these systems have been installed in the USA, but increasing electricity prices, stricter regulations for using ammonia, and the impending termination date for use of R-22 should rapidly increase the number of installations in the US over the next decade.

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Figure 1: COP at Full Load for Cascade and Conventional Systems

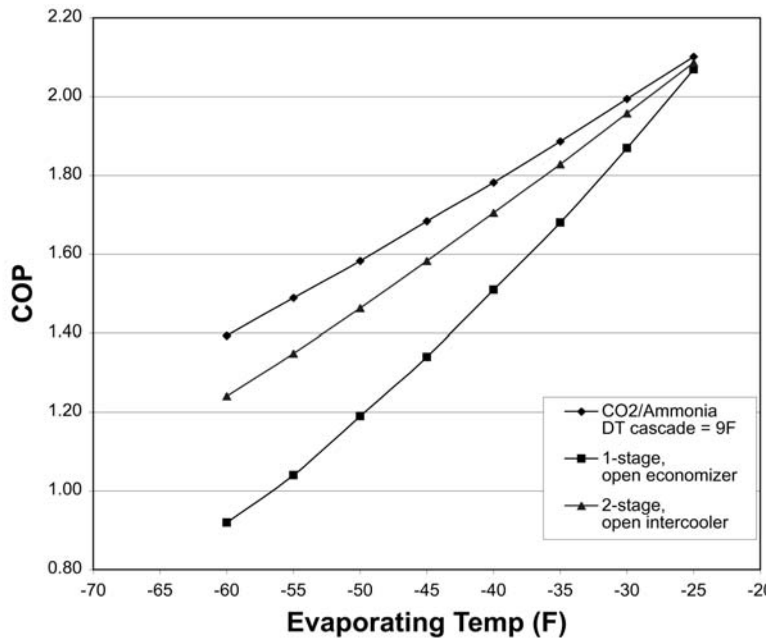

Figure 2: COP Comparison, 50% Part-load

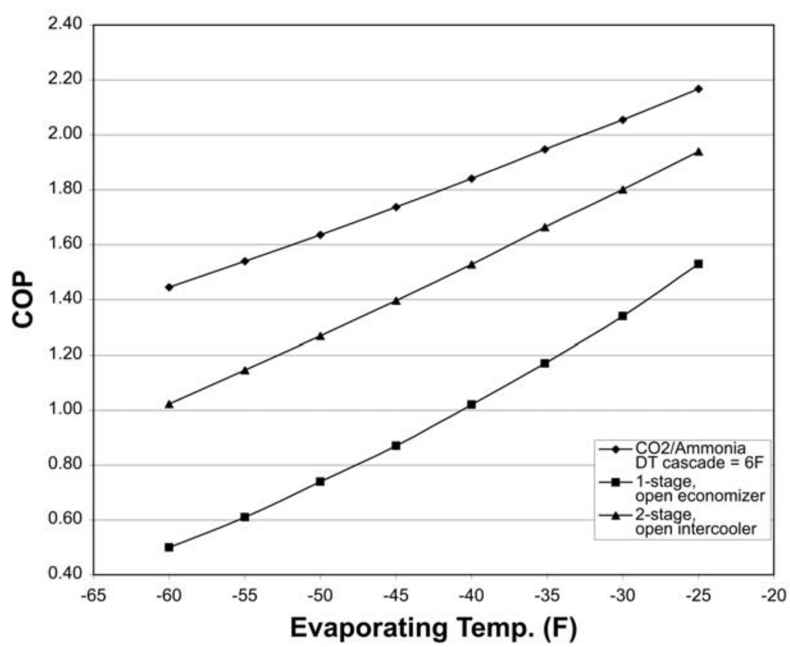

Figure 3: Compressor Capacity vs Evaporating temperature, CO_2 and Ammonia

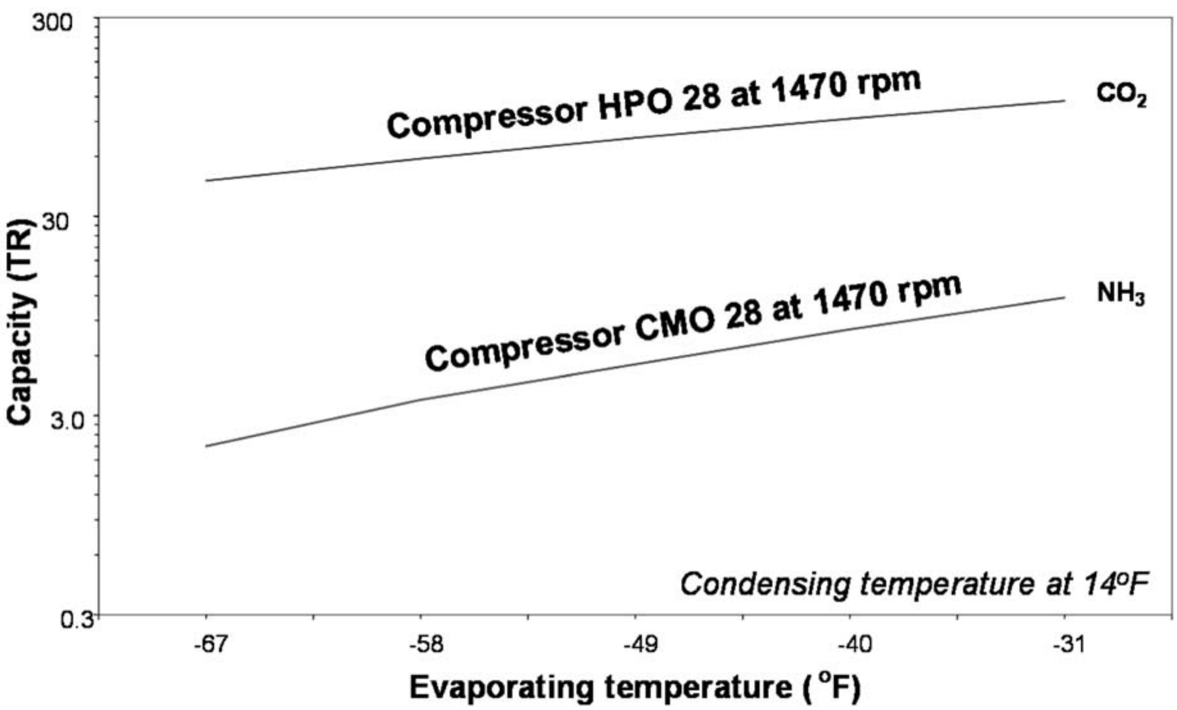

Figure 4: Typical Ammonia/CO_2 Cascade System, Freezing Application

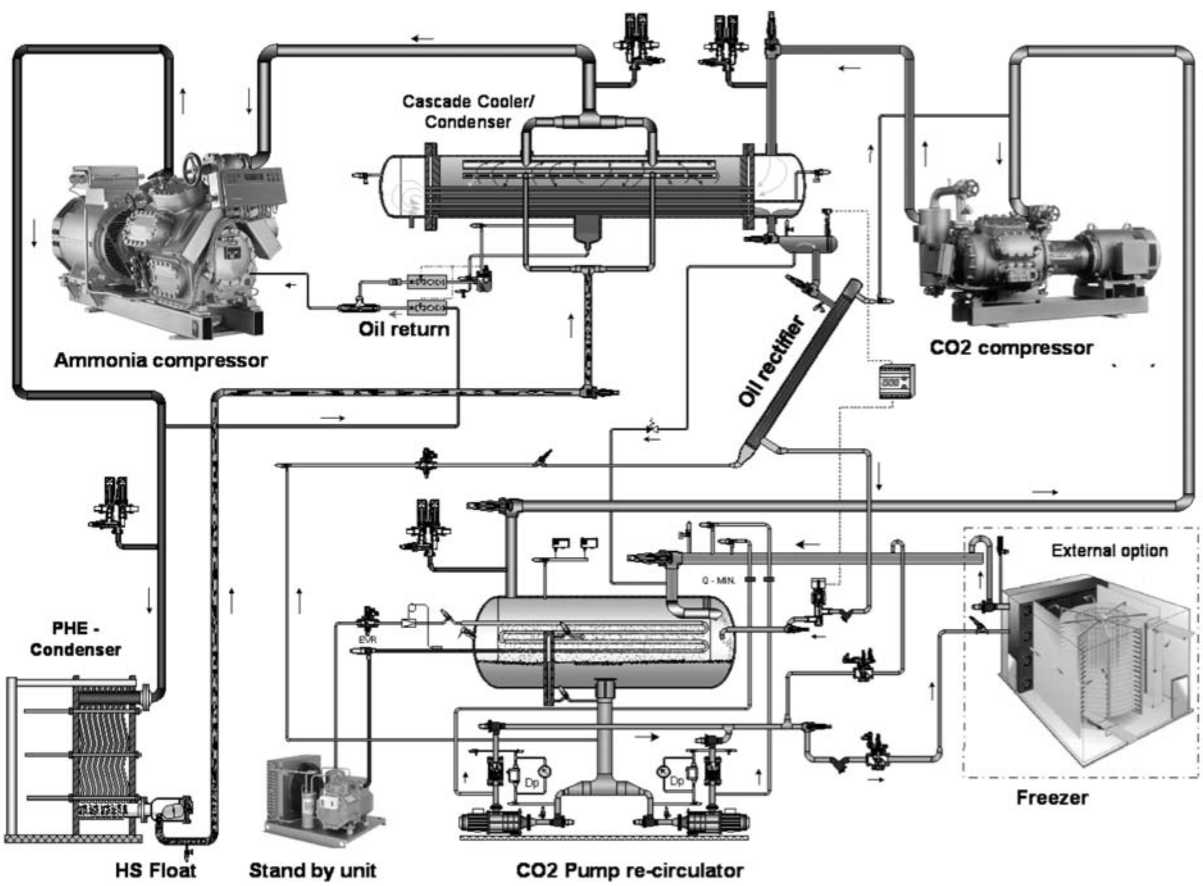

Figure 5: Typical Ammonia/CO_2 Cascade System, Freezing and Cooling Application

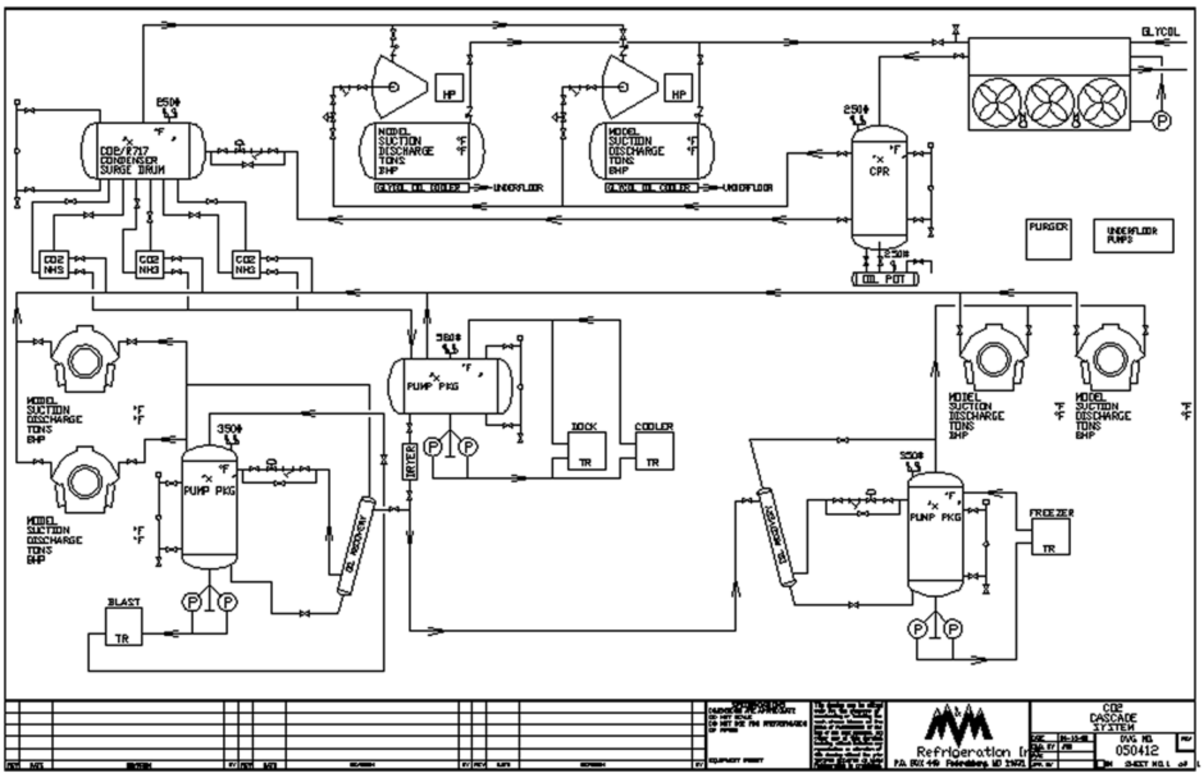

Figure 6: Effect of ∆T on First Cost of Compressor and CHE

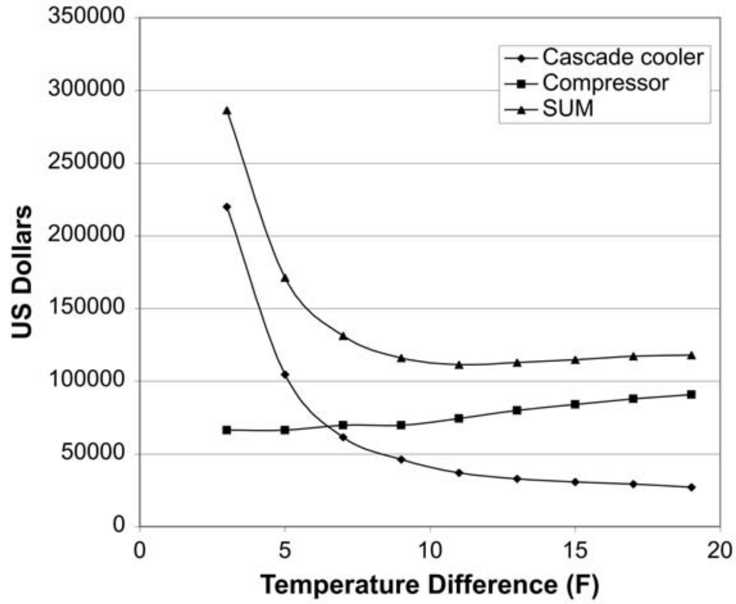

Figure 7: Optimal ∆T Considering Capital and Operating Costs

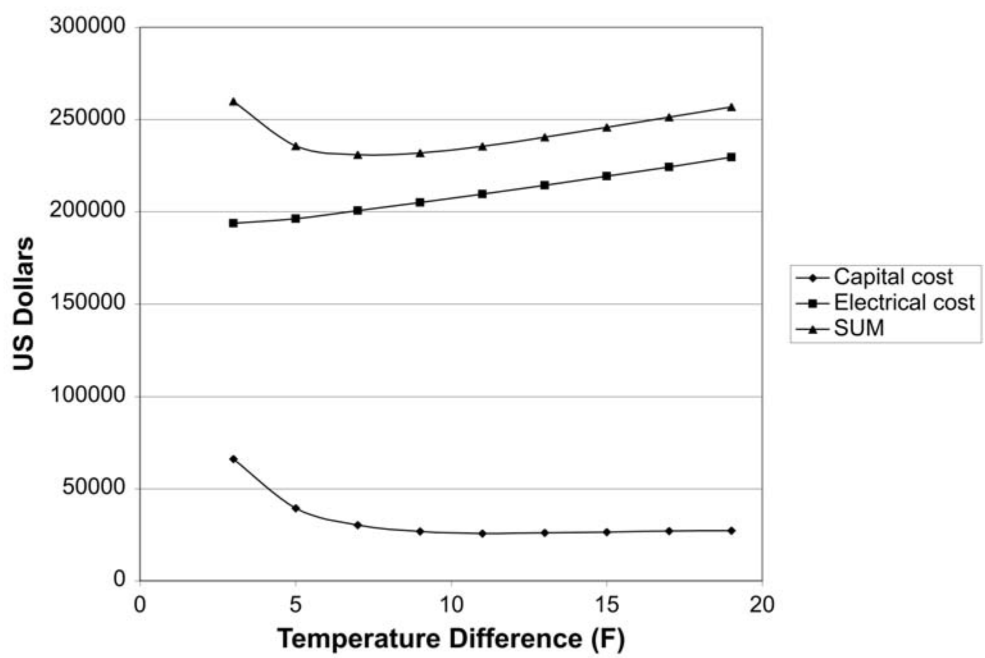

Figure 8: COP, Low and High Stage Compressors

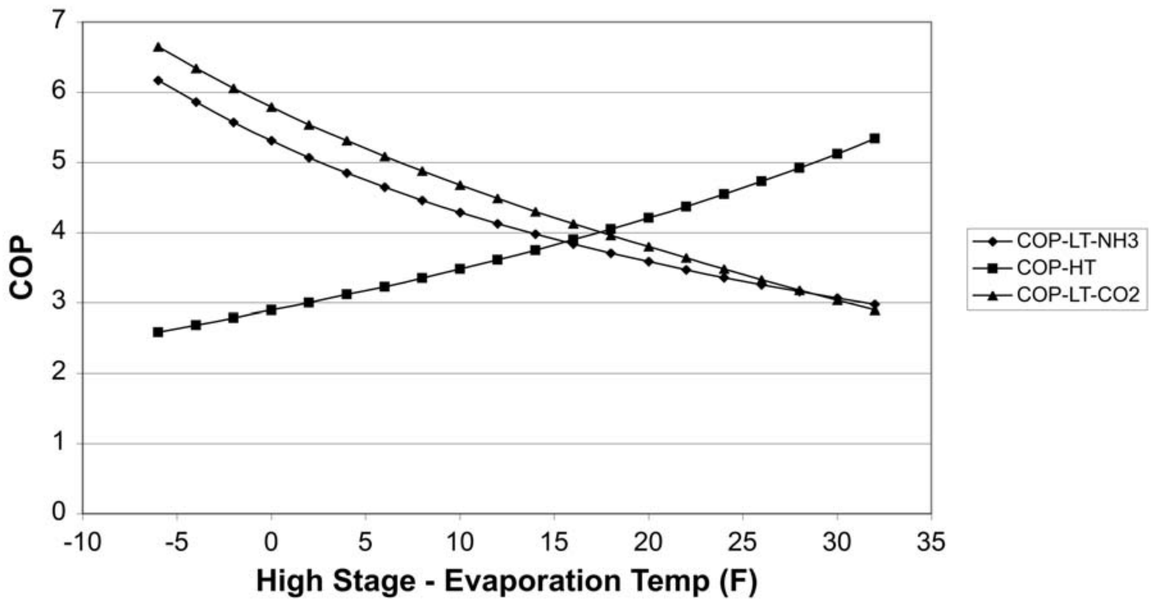

Figure 9: Total COP vs High Stage Evaporating Temperature

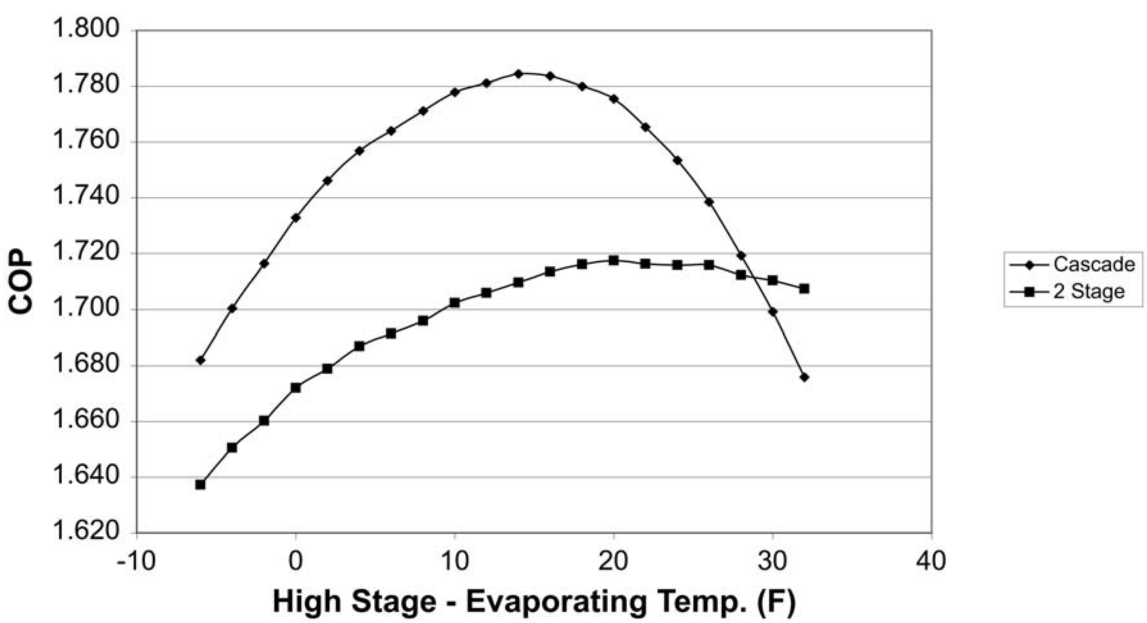

Figure 10: Oil Rectifier System

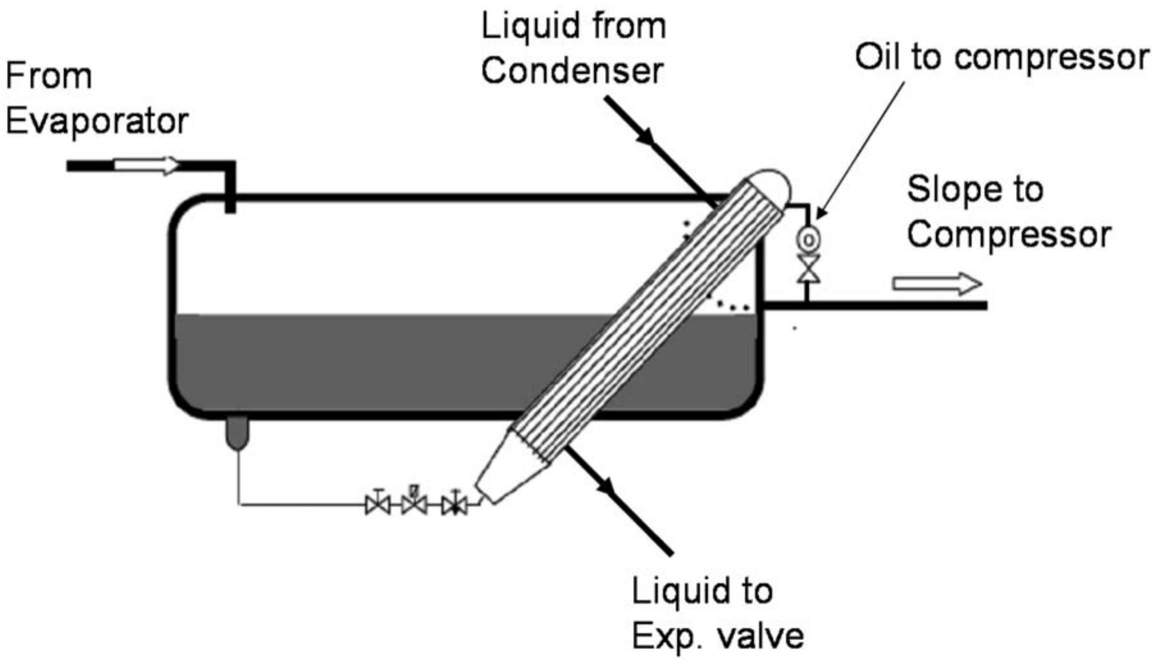

Table 1: Comparison of Piping for CO_2 and Ammonia

300 TR at -40F evaporation temperature, circulation rate 1:3

Wet suction, 300 ft pipe, two 90 bends, one stop valve

	Pipe size	Temperature drop	Pressure drop	Velocity
	in (nom.)	F	psi	ft/s
Ammonia	12"	1.01	0.31	54
CO2	6"	0.96	2.85	23

Dry suction, 100 ft pipe, two 90 bends, one stop valve

	Pipe size	Temperature drop	Pressure drop	Velocity
	in (nom.)	F	psi	ft/s
Ammonia	12"	0.29	0.089	68
CO2	6"	0.33	1	39

Liquid, 300 ft pipe

	Pipe size	Temperature drop	Pressure drop	Velocity
	in (nom.)	F	psi	ft/s
Ammonia	2½"	1.4	1.4	3.6
CO2	4"	0.3	1.05	3.5

Discharge, 100 ft pipe

	Pipe size	Temperature drop	Pressure drop	Velocity
	in (nom.)	F	psi	ft/s
Ammonia	6"	0.21	0.2	79
CO2	4"	0.14	0.9	29

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