Safety in Numbers

Images courtesy of Danfoss.

BY TERRY L. CHAPP, PE

Editor's Note: This series is excerpted from “Low Ammonia Charge Refrigeration Systems for Cold Storage White Paper” by Terry L. Chapp, PE, published by the International Association of Refrigerated Warehouses (IARW) and the International Association for Cold Storage Construction (IACSC) Refrigeration & Energy Committee.

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Part 3:

Ammonia Systems for Cold Storage

The third part of this multi-part series looks at CO₂/NH₃ cascade systems and those pumped with volatile brine.

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Part 2 of this feature series introduced a baseline cold-storage system and began to review different low-charge systems and what their anticipated energy consumption is. This month, comparing the same baseline system from Part 2, CO₂/NH₃ cascade systems and those pumped with volatile brine are on the docket.

CO₂/NH₃ cascade systems

While the direct expansion system is one of the oldest alternatives to pumped recirculated liquid systems in industrial refrigeration, CO₂ systems have roots in industrial refrigeration as far back as the 19th century. Until recently, the thermophysical characteristics of CO₂ have made it difficult to utilize CO₂ as a working refrigerant in industrial refrigeration systems in a practical and cost-effective manner. However, significant advances in materials, manufacturing, equipment designs, and controls have made it a highly viable alternative to many of the more common refrigerants and heat transfer fluids. It should be noted that at least one major cold-storage company is using CO₂/NH₃ systems extensively throughout its organization. There are several variations in using CO₂ refrigeration. In this feature, the focus will be on compressed CO₂ for the room side with ammonia serving as the initial heat sink (condenser) for the CO₂ process.

The type of system described as CO₂/NH₃ cascade isolates the ammonia refrigeration part of the system to the machine room. As with all refrigerants, the higher the temperature of the refrigerant, the higher the operating pressure. In order to limit the extremely high pressures necessary for an all-CO₂ system, the ammonia part of the system provides the initial condensing for the CO₂ system and thereby limits the pressure that would exist if CO₂ were to be condensed as in a typical refrigeration cycle (95°F condensing temperature, typically). The evaporator for the ammonia system is also the condenser for the CO₂ system.

This type of heat exchanger is referred to as a cascade heat exchanger and can be constructed in a number of different ways: shell and tube; welded plate and shell and plate. The choice of style will have a direct impact on the refrigerant charge of the system, with shell and tube requiring the most refrigerant and welded plate requiring the least. There are several different approaches taken in achieving the desired temperature levels of the coolers and the freezers. The methodology for this particular configuration has been as follows:

1. Cool the CO₂ down to the temperatures required by the coolers through the cascade heat exchanger and send to the CO₂ recirculator vessel.

2. Move the cool liquid to the coolers and freezers.
   a. Pump some of the condensed CO₂ to the cooler coils from the cascade heat exchanger. Unlike ammonia, recirculation rates are maintained much closer to 1:1, allowing almost all of the CO₂ to be evaporated. This has the added benefit of reducing the size of the evaporator and the horsepower of the pump.
   b. Supply additional liquid from the CO₂ recirculator to the freezer coils. Each coil is equipped with an electronic expansion valve to lower the temperature and pressure of the CO₂ to the desired temperature.

3. Return the gas/liquid mixture from the cooler coils and the freezer coils to the CO₂ recirculator.

4. The vapor collected in the recirculator is then sent back to the CO₂ compressor, compressed and sent to the cascade heat exchanger to be condensed while the liquid is circulated back to the coolers and freezers.
The system is relatively simple and does not require much more equipment than a two-stage pumped recirculated system would. As will be noted later, due to some of the thermophysical properties of CO2, there can be some significant savings on the equipment for this type of system. Additional advantages offered by CO2 as a refrigerant arise from its high heat-transfer coefficient at very low temperatures and favorable vapor-to-liquid volumetric ratios. This leads to:

- A more efficient cascade heat exchanger (reduced energy consumption due to a smaller temperature difference requirement);
- Smaller evaporators;
- Lower recirculation rates;
- Smaller pumps; and
- Reduced pumping power demands.

As seen in Figure 1, all of the ammonia is confined to equipment residing in the machine room. Readers should note that a great deal of detail has been purposely left out of the sketch in order to focus on the basic process.

There are a number of special considerations when using CO2 as a refrigerant:

- It is important to consider what the pressure could, potentially, rise to during “stand still.” Stand still is the point at which power has been disconnected from the system (intentionally or unintentionally) and pressures can begin to build up to dangerous levels as a result of the heat transferred from the ambient environment. The simplest solution to the problem is to install a small condensing unit driven off of a generator to condense any vaporized CO2 within the system, thereby preventing an excessive pressure rise. Of course, the system will have safety relief valves installed as a preventive measure; however, it is always desirable to keep the CO2 contained within the system.

- This CO2 system is referred to as a subcritical system. By that it is implied that operation of the system is limited to within the range between the critical point (88°F) and the triple point (-70°F). In general, this allows the system to operate between 83 psi/-67°F and 507 psi/32°F. Using CO2 hot gas to defrost the evaporators will increase the upper limit of the pressure requirement to about 652 psi. A properly designed system will have no problem maintaining the correct operating conditions. However, the user must be aware of conditions that can arise during periods of standstill noted earlier.

- Water can have an extremely negative impact on CO2 evaporators and, under certain circumstances, on other system components. Therefore, the system must be kept dry. The problems associated with water in CO2 systems is directly related to its low solubility limits. In essence, this means that components can be negatively impacted by water once the solubility limit has been reached and, especially, when temperatures are below freezing. Carbonic acid formation can
Ammonia refrigerant charge—Figure 1 shows that all of the ammonia charge for this system is isolated to the machine room. As previously pointed out, the ultimate ammonia charge will depend on the type of cascade heat exchanger; the type of condenser; and the size and amount of piping connecting the CO\textsubscript{2} and NH\textsubscript{3} systems.

While it has been estimated that the ammonia charge in a system like this can be, potentially, reduced to as low as 10% of the ammonia charge in a typical pumped recirculated liquid (PRL) system, the actual numbers are likely much higher than this theoretical value. Fortunately, data exists on existing CO\textsubscript{2}/NH\textsubscript{3} cascade systems that provide actual experiences.

US Cold Storage (USCS) currently has nine CO\textsubscript{2}/NH\textsubscript{3} cascade facilities in operation. Information provided in this article does not represent any individual USCS facility, but rather data on an aggregate basis. With respect to ammonia charge, the typical ammonia charge in the USCS facilities is approximately 8 lb/ton of refrigeration (TR). Readers should note that in the summary of this article series, the theoretical value for this type of system is about 25% lower than those reported by USCS. There are a number of possible explanations for the difference between the USCS facilities and the theoretical values. A broad generalization would suggest a charge of between 4 and 6 lb/TR. It is clear that while 4 lb/TR is achievable, it may not be the most practical objective. More discussion of the factors that weigh into the final number will be found in Part 4 of this series. For the purposes of this month’s feature, the charge will be conservatively estimated at 6 lb/TR.

Energy consumption—As noted in Part 2, energy consumption is a fairly easy number to track at the plant level. However, calculating ratios such as KW/TR (KW = power of all of the motors needed to drive compressors, fans and pumps; and TR = tons of refrigeration) is much more challenging for a central system. The main difficulty in developing these numbers is in determining the actual loads that correspond to the measured kilowatt-hours. All reported figures in this feature series have to be viewed with the understanding that actual loads are almost always going to be less than design loads. USCS has estimated their average power consumption to lie somewhere between 1.8 and 2.0 KW/TR. This value, however, is based on design loads and measured kilowatt-hours, so it is clear that the actual value for KW/TR is going to be a different value.

In general, USCS refrigeration loads are fairly heavy on blast freezing. CO\textsubscript{2} is a particularly energy-efficient refrigerant at temperatures below -30°F and, if a large percentage of the total plant load falls into this category, it should not be surprising to see very attractive energy-usage figures. In addition, since these are aggregate numbers for seven different facilities, it is also likely that the actual condensing temperatures are well below 95°F. This would also have the effect of lowering the KW/TR ratio. The general conclusion here is that the true ratios are probably different than those reported by USCS, but the values do project numbers that fall within the broad range that would be anticipated for this type of system.

Based on the work done by VaCom, there is strong evidence that facilities with blast freezers exhibit significant improvements in energy consumption when using CO\textsubscript{2}/NH\textsubscript{3} cascade systems. The cascade system exploits the best efficiencies of NH\textsubscript{3} and CO\textsubscript{2} compressors—NH\textsubscript{3} compressors have higher efficiencies at higher temperatures; CO\textsubscript{2} compressors have higher efficiencies at lower temperatures. The optimum temperatures for this efficiency improvement are typically lower than the -30°F target reported earlier. CO\textsubscript{2} refrigeration has also been a topic of much controversy in the industrial refrigeration arena. Theoretical simulations typically suggest energy-consumption numbers to be higher than that of a well-performing, equivalent PRL system. Some user experiences, including that of the VaCom study, suggest otherwise. The reader is referred to a very comprehensive study undertaken by the Industrial Refrigeration Consortium comparing two-stage ammonia PRL systems with CO\textsubscript{2}/NH\textsubscript{3} cascade systems.

The value for the system in the VaCom study was equal to 2.3 KW/TR. Assuming a small decrease in efficiency at a 95°F condensing temperature, the value assumed for this system will be set at 2.4 KW/TR. Taking into account the amount of theoretical work that has been undertaken in order to compare the two systems, it is probably prudent to use a value not less than that of a PRL system. For the purposes of this article, and until more actual data is compiled, the value will be set at 2.5 KW/TR.
Installed cost—One of the highly beneficial characteristics of CO₂ when compared to NH₃ is the reduced pipe diameter for CO₂ vapor compared to NH₃ vapor (liquid lines for CO₂ are typically larger than those for NH₃ but still fall into the small-diameter piping category). Because vapor lines typically represent some of the larger-diameter piping and longer runs of piping in any facility, the installed costs of vapor carrying pipe can be significant. Smaller diameters translate to savings in:

- Piping (if the same materials are used for both systems);
- Transportation;
- Welding (if the same weld procedures are used for both systems);
- Painting (if applicable); and
- Insulation.

It should not be surprising to find that a CO₂/NH₃ cascade system can be more cost effective from an installation perspective than a PRL system. However, there are many “ifs” in the above list. In this case, USCS has determined that the average installed cost of their CO₂/NH₃ cascade systems was at the low end of the scale for all installed systems. This might be attributed to economies of scale, as USCS facilities are usually quite large or perhaps favorable locations with respect to building costs or other factors.

Refrigeration contractors, on the other hand, have provided a somewhat different perspective. While the cost of this type of system is not significantly higher than the baseline system described in Part 2, three very large contractors that have both quoted and built CO₂/NH₃ facilities all stated that the CO₂/NH₃ cascade system carried a slight premium in cost over that of the baseline system. For the purposes of this article, it will be assumed that there is a 5% premium in installed cost over the baseline system, or $7,400/TR.

Maintenance cost—Other than the fact that one-half of this system uses CO₂ as the refrigerant, there is not a big difference between the CO₂/NH₃ cascade system and a conventional PRL system. Of course, as noted in the previous article (under “Special considerations”), CO₂ has a few challenges that must be carefully controlled. The two areas of special importance in the CO₂ system are the filter-driers and the oil-management system. A facility with a sound maintenance plan and a reliable maintenance staff should find no problems in meeting the special requirements of maintaining filter-driers and the oil and water stills with little increase in maintenance cost. The conclusion here is that while maintenance costs may be slightly higher than for a PRL, they will have a minimal impact. Maintenance Cost vs. Baseline = Even.

CO₂/NH₃ with pumped volatile brine (PVB)
The CO₂/NH₃ system with pumped volatile brine has many similarities to the CO₂/NH₃ cascade system. In this system, however, there is no primary CO₂ compressor. It is possible that there may be a small CO₂ compressor for the hot-gas defrost system, but there are also other options that allow for defrost without adding a compressor for that purpose. Many would describe this as a chiller system, but what distinguishes this system is the word “volatile.” Most chiller systems use water or some formulation of glycol/water and the cooling coil is no longer an evaporator but rather just a sensible heat exchanger. This has an enormous impact not only on the amount of heat-transfer fluid that is required to meet the cooling load but also on pump size, pumping power, piping size and the heat-exchanger size.

Compared to a traditional chiller system, the CO₂/NH₃ with pumped volatile brine system has higher energy efficiency and, usually, a lower installed cost. Actual data for industrial refrigeration systems is still scarce. However, in similar systems for commercial refrigeration, it has been found that the pumping power required to circulate CO₂ as a volatile secondary refrigerant represents about 5% of the power (or a 95% reduction in pumping power) that would be required to circulate a non-volatile secondary refrigerant such as water or glycol.

The main difficulty in developing these numbers is in determining the actual loads that correspond to the measured kilowatt-hours.

CO₂ systems typically operate between 83 psi and 507 psi. What this means is that the piping is always under pressure and the ability for water or air to be drawn into the system is virtually non-existent. This is not to say that water and air, along with other contaminants, cannot find their way into the system, but the likelihood is greatly diminished.

The system description shown in Figure 2 describes a system in which the liquid CO₂ is cooled down to a temperature low enough to satisfy the cooling loads in the freezers. The liquid is cooled to the required temperature in the cascade heat exchanger and then stored in the CO₂ recirculator vessel. This same refrigerant is then used to satisfy the cooling loads of the coolers. The key component in meeting the cooler requirements is the pulse-width-modulating (PWM) valve. Normally used for DX systems, the PWM valve is used in this system simply as a metering device to supply enough refrigerant to meet the load requirements of the coolers. There is no intention of evaporating all of the refrigerant in a DX evaporator but rather simply to satisfy the room-temperature setting.

The approach taken for this particular design was chosen because of the high ratio of freezer load to that of cooler load. If the ratio was reversed, it is likely that a different design philosophy would have been adopted. Since there is no compressor and no suction accumulator, no effort is made to ensure that liquid is not carried out of the evaporators. The main driver behind this approach is the temperature at which coolers typically operate. If the cooler is set at, say 55°F, and the evaporator has been designed for a 10°F temperature difference,
the CO₂ would be operating at nearly 600 psig. While this is not an unmanageable number, it does push the limit on some of the equipment requirements (especially the pump).

The special considerations for the CO₂/NH₃ with pumped volatile brine are nearly the same as that for the CO₂/NH₃ cascade system:

→It is important to consider what the pressure can rise to during “stand still.” Stand still is the point at which power has been disconnected from the system (intentionally or unintentionally) and pressures can begin to build up to dangerous levels as a result of the heat transferred from the ambient environment. The simplest solution to the problem is to install a small condensing unit driven off of a generator to condense any vaporized CO₂ that is causing the pressure to rise. Of course, the system will have safety-relief valves installed as preventive measure, however, it is always desirable to keep the CO₂ contained within the system.

→This CO₂ system is called a subcritical system. By that it is implied that operation of the system is limited to within the range between the critical point and the triple point. In general, this allows the system to operate between 83 psi/-67°F and 507 psi/32°F. Although it is unlikely that defrost will be done using CO₂ hot gas to defrost the evaporators, doing so will increase the upper limit of pressure requirements to about 652 psi. A properly designed system will have no problem in maintaining the correct operating conditions; however, the user must be aware of conditions that can arise during periods of standstill noted earlier.

→Water can have an extremely negative impact on CO₂ evaporators and, therefore, the system must be kept dry. The problems associated with water in CO₂ systems is directly related to the low solubility limits. In essence, this means that components can be negatively impacted by water once the solubility limit has been reached and, especially, when temperatures are below freezing. The solution to the problem is to incorporate and maintain filter-driers in the system design.

→Oil is not required for the CO₂/NH₃ pumped volatile brine system.

→As noted in the second point, the use of hot-gas defrost can increase the system design pressures by as much as 145 psi and, will also add to the total ammonia refrigerant charge. Hot-gas defrost is typically provided through a compressor dedicated just for that purpose.

→Although CO₂ is a non-toxic and non-flammable refrigerant, it is heavier than air and, therefore, sinks to the ground.

**Figure 2 CO₂/NH₃ pumped volatile brine system.**

**Nomenclature**

EC = Evaporative Condenser
OS = Oil Separator
CO₂R = CO₂ Receiver
CHX = Cascade Heat Exchanger
HPR = High Pressure Receiver
C = Compressor

**Cooler Evaporators**

**Freezer Evaporators**
or floor level. This can create dangerous situations in confined spaces by resulting in non-breathable air. Leak detection and emergency ventilation systems are essential in the design.

Ammonia refrigerant charge—Because the system under consideration requires blast freezing, the ammonia temperature available to condense CO$_2$ liquid as low as -50°F, would have to be on the order of -60°F. For all practical purposes, this would require a two-stage refrigeration system for the ammonia. At first glance, it might appear that the charge will be quite high due to the two compressors, intercooler (optional), and a heat exchanger large enough to satisfy the entire load of the cold-storage facility. Readers should keep in mind, however, that there is minimal piping, no evaporators (other than the cascade heat exchanger), a condenser and only a modestly sized receiver.

There is not a lot of good information available at this time to determine the actual charge in a system like this. However, in an effort to stay consistent with the analysis from Welch’s paper (also referenced in Part 2 of this series), a value has been calculated using the basic numbers for the DX system without all of the piping and vessels (will be referenced in the summary of this series next month). Although it is believed that the actual ammonia refrigerant charge could be as low as 2.5 lb/TR, there is little data available to support this theoretical value at this point. Instead, the systems ammonia charge will be assumed to be the same as that of the CO$_2$/NH$_3$ cascade system, or 6 lb/TR.

Energy consumption—This system section started out with a cursory comparison to a chiller system. As noted previously, the characteristics of CO$_2$ and the fact that the heat exchangers are truly evaporators contribute to a sound energy-efficiency image. There are other aspects of the system equipment which, when combined with the characteristics of CO$_2$, continue to present an attractive energy picture. One of the main features of the refrigerant is its high heat-transfer coefficient. This leads to smaller temperature differences in both the evaporators and as the cascade heat exchanger. Along with the fact that the pumping power requirements are also low for CO$_2$, the end result is that the system efficiency is close to that of a PRL system.

Once again, there are many factors that contribute to the overall energy consumption of each system. The use of CO$_2$ volatile brine is not new but has had limited deployment in cold-storage applications. For the purposes of this feature, it will be assumed that the CO$_2$/NH$_3$ pumped volatile brine system is only slightly more energy intensive than the baseline system described in Part 2. This assumption is based on a generalized view of the “energy consumers” in this system and how they would be expected to compare with an equivalently sized baseline system. The value for the purpose of this article is 2.5 KW/TR.

Installed cost—The installed cost of the system will likely be very similar or just slightly more expensive than that of the CO$_2$/NH$_3$ cascade system. The ammonia refrigeration system will be larger and more costly, but there are no CO$_2$ compressors in this system unless hot gas is the desired route for defrost. The cascade heat exchanger will be larger for the CO$_2$/NH$_3$ PVB system, but there will be no need for an oil still. With this in mind, a conservative estimate for the cost of the system is set at $7,400/TR.

Maintenance cost—As with all of the systems under consideration in this feature series, maintenance cost as compared to the base pumped ammonia system has some beneficial aspects and a few items that will require more attention. On the plus side, since all of the ammonia is confined to the machine room, oil sumps are very manageable and oil draining is limited to a significantly reduced number of vessels. Since the volatile brine system has no requirements for oil, there is no oil-draining requirement.

On the additional cost side of the equation, filter-driers must be monitored and maintained on a routine basis. As noted earlier, a backup condensing unit is highly recommended for periods of “stand still.” This system must also be maintained on a routine basis to ensure that it is in good working order when and if the time is needed for its use.

The overall conclusion regarding maintenance costs is that any increase in cost over that of a conventional PRL system will be minimal. Maintenance Cost vs. Baseline = Even.

Part 4 of this series will visually summarize the alternative industrial refrigeration systems discussed in Parts 1–3 in an attempt to provide more insight and clarity into this rapidly evolving technology for the cold-storage industry.

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