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Introducing a New Ammonia/CO_2 Cascade Concept for Large Fishing Vessels

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Abstract

Until recently, fishing vessels could use R-22 to preserve the catch, but this refrigerant is no longer an option in many areas. Ammonia has always been an efficient alternative, but had been perceived as difficult and expensive to handle on the production deck of fishing vessels. Combining the strengths of two environmentally friendly and efficient refrigerants, ammonia and CO_2, and using standard industrial refrigeration components, a cascade refrigeration solution was developed for this application. The paper discusses the technical challenges, solutions, and benefits now and in the future for these types of systems.

Introduction

CO_2 refrigeration is not a new discipline, neither on land nor on ships. In fact, many consider it to be one of the first refrigeration applications. The first compressor built by our company back in the 19th century used CO_2, and ships have been equipped with CO_2 refrigeration systems for at least 100 years.

Many things have changed since then. With the introduction of halocarbon refrigerants, CO_2 was largely abandoned. However, with increasing focus on the negative environmental properties of halocarbon refrigerants, CO_2 is being "discovered" once more. However, re-introducing CO_2 in refrigeration designs involves much more than just finding the old drawings! Standards have changed, and end users expect all the functionality normally found in state-of-the-art refrigeration systems to be included with a new CO_2 refrigeration plant. Using such a high-pressure refrigerant posed quite a challenge to us, for example, to find components strong enough to handle the pressure. However, it also spurred innovation, because we had to fundamentally rethink our designs, for example, to devise an effective defrost method.

This paper describes the development and proving of the system built for the Norwegian-owned MS Kvannøy.

Presentation of the M/S Kvannøy

The Kvannøy is a 75-meter (250 ft) vessel able to fish with a purse seine or by trawling. Its equipment can fillet the catch, freeze either fillets or whole fish, and also make ice for the refrigerated-seawater (RSW) tanks, in which the catch is pre-cooled.

The refrigeration systems on the ship consists of:

- Two ammonia systems with the capacity to cool 300 m^3 (10,600 ft^3) of seawater from +18°C (64°F) to +2°C (36°F) in five hours.
- An ammonia/CO_2 cascade freezing system with 11 vertical plate freezers with the capacity to process 210 metric tons (463,000 pounds) of fish to -18°C (0°F) core temperature in 24 hours at an evaporating temperature of -49°C (-56°F). Because air-cooling plates are installed in three of the nine RSW tanks, these can also be used as -30°C (-22°F) holds. The cascade system also provides cooling for a vertical ice machine capable of producing 30 tons of flake ice every 24 hours. All evaporators, except the interstage cooler, use CO_2.

The ammonia/CO_2 system consists of:

- Two ammonia screw compressors with computer control, one of which is equipped with a variable speed drive.
- Six reciprocating CO_2 compressors, three of which are specially modified for dual use, either to provide cooling, or for dedicated use in the special hot-gas defrost system (patent pending)
- Three pumps (one as a spare) connected at the suction end to a low-pressure receiver that can hold five tons of CO_2. The low-pressure receiver contains a cooling coil for a stand-alone cooling unit, which is used when the ammonia/CO_2 system is shut down.
- One shell-and-tube interstage heat exchanger with vented double-wall tubing.

The eleven plate freezers are filled manually but empty automatically.

Production Efficiency – Theory and Practice

Like any other heat exchanger, the capacity of a plate freezer can be expressed as the product of the heat transfer area, the overall heat transfer coefficient and the temperature difference. For a plate freezer, neither the surface area available for heat transfer nor the heat transfer coefficient can be changed significantly. However, by increasing the temperature difference, we can also increase the heat transfer, which decreased the freezing time, and thus increased the production capacity of the plate freezer.

We decided that for a CO_2 system, an evaporating temperature of -50°C (-58°F) was realistic. For this system, freezing time would be about 80% of that in a typical system operating at -40°C (-40°F) with the same size plate freezer. Thus, unit production capacity would be about 125% that of the -40°C (-40°F) system—a 25% increase. We initiated a project to verify this capacity gain, among other things, and validated this 25% unit capacity increase.

Subsequently, we optimized the plate design specifically for CO_2 use. These plates had smaller channels and, as a consequence, were thinner. Thinner plates enabled us to fit more plates in the same freezer space. Thus, a 39-compartment (40-plate) CO_2 freezer could fit in the same space as a typical 37-compartment freezer, representing an additional 5% increase in unit production capacity.

In tests aboard the Kvannøy, we found that the freezing time was even quicker than the predicted 80% of a typical system. Although not verified, the authors' opinion is that using CO_2 resulted in a lower equivalent temperature drop in the suction line and plate freezer outlet hose than with other refrigerants. The high vapor density of CO_2 yields a much lower volume flow in the return system. The lower volume flow of CO_2 produces a much lower velocity in the return system when compared to other refrigerants, with substantially lower pressure loss as a result. The lower pressure loss results in an evaporating temperature in the plate much closer to the low-pressure receivers saturation temperature than seen before.

Production Efficiency versus Available Deck Space

Although an important item on a fishing vessel, the refrigeration system is rarely a factor influencing the general design of the vessel. Typically, shipyard or design agency inquiries focus on how many tons of fish can be frozen to a given core temperature in one day, given a certain amount of deck space, rather than on compressor horsepower or specific equipment.

Given this focus on unit production for each unit of deck space area, it is clear why CO_2 is an extremely promising option for the fishing industry. With the capacity gained from lowering the evaporating temperature, the CO_2 solution is extremely attractive.

Conversely, the alternatives have become less attractive. Leakage of halocarbons such as CFCs, HCFCs, or HFCs is frowned upon because of the negative effect on the environment. The impact on the environment and higher costs from added taxes make halocarbons unattractive. While ammonia is still considered an environmentally-friendly alternative, European countries have begun to set limits on the allowable size of ammonia refrigerant charges in populated areas, and the same trend is beginning in the marine sector.

Finally, with CO_2, the loss of product due to contamination from refrigerant leaks is not a problem, as is the case with ammonia. Because the product in the processing areas and in the cargo hold of large fishing vessels represents such a high value, reducing risk of product damage results in reducing insurance costs.

Cooling Efficiency

Table 1 shows the coefficient of performance (COP) in BTUh/kW units for a number of different system configurations and operating conditions. The table values represent the best efficiencies found using our own computer program. In modeling the different system options, we assumed that single-stage systems would use screw compressors without economizing, and that two-stage systems would use reciprocating compressors with a closed intercooling system. We modeled the cascade system after the system used on the Kvannøy, which used reciprocating single-stage reciprocating CO_2 compressors and ammonia screw compressors without economizing.

Evaporating temperatures for brine systems are lower than for direct systems, and thus have brine systems have lower COPs. Therefore, we did not model them. It is clear that the ammonia/CO_2 cascade system delivers a superior COP in both running conditions. In fact, the cascade COP at -50°C (-58°F) is substantially equal to the single stage COP for R-22 and ammonia at -40°C (-40°F) and not far off the two stage values.

The unique defrost system used on the Kvannøy has an effect on the COP as well. The defrost compressors' suction flow does not have to be condensed by the ammonia circuit, and consequently, the required capacity and power demand of the ammonia circuit is reduced. Since the defrost compressor is running with a COP roughly twice as good as the ammonia compressors, every kW (or BTU/hr) used in the defrost compressor motor results in double that amount saved in the ammonia compressor motors. The overall effect of running the defrost compressor is naturally dependent on the system configuration, but on a system like the Kvannøy, the COP at an evaporating temperature of -50°C (-58°F) can be increased to 2.09 during defrosting.

The careful reader might object that the act of defrosting adds heat to both the equipment and product, and that this heat would need to be removed, resulting in a reduction in COP over time rather than an increase. This is true in most applications, but not in vertical plate freezers. When the vertical plate freezer finishes a freezing cycle, the processed product is virtually one big ice block. Under normal operation, it is necessary to apply heat to the plates to release these ice blocks. As such, the calculation of total capacity already accounts for the heat of defrosting. If we were able to use some other heat source to defrost the plates, the COP would be 1.78, as mentioned earlier, but when that heat is supplied by the defrosting compressor, the change in power consumption (defrost compressor added and ammonia compressor decreased) results in a change in COP to 2.09. This high COP is naturally only valid when defrosting, which at full production is approximately 40% of the time.

Table 2 presents the total compressor suction volume flow of refrigeration capacity at the running conditions given above and with the same compressor constellation. All values are stated relative to the ammonia/CO_2 cascade system operating at -50°C/+25°C (-58°F/77°F) conditions. To be able to make an “apples to apples” comparison of the installed swept volume, it was necessary to “scale” the systems so that they delivered the same cooling capacity. This was not possible to achieve exactly with the compressor sizes available. Thus, scaling the systems to achieve the same cooling capacity required using a “fractional” compressor (e.g., using 2.7 compressors). The reader will note from Table 2 that the total compressor suction volume required for the CO_2/ammonia cascade is much less than for any other system.

Product Quality

Product quality is all-important for getting a good price. Of the factors contributing to better quality, shorter freezing time and lower end temperature are the most important ones that relate to the refrigeration system. Shorter freezing time results in smaller ice crystals and less cell damage to the product, and a lower end temperature reduces the amount of free water in the product, resulting in a longer shelf life. Both a shorter freezing time and a lower end temperature can be achieved with CO_2, because the lower evaporating temperature yields a lower temperature at the plates, and the average temperature is therefore lower, too.

Very often, blocks frozen in typical vertical plate freezers are soft on top, usually because the upper part of the plate can become very hot during defrost. Any product softness increases the chance that the product will suffer damage from handling, and such a loss in quality is deducted from the sales price. The combination of a low evaporating temperature and a short, efficient defrost at a low temperature results in blocks that are, and remain, hard in the top.

Initial Cost of the System

A fisherman will not tell you how much cooling capacity he needs. He will tell you: “I have so many square meters of production area; how many tons of frozen fish per 24 hours can your system provide?”

Generally the components on ammonia/CO_2 systems tend to be smaller than those on conventional systems (e.g., ammonia, R-22, R-404A) with the same capacity. The very high volumetric efficiency on the CO_2 compressors makes these compressors much smaller and the suction lines and valves are approximately half of the conventional system size. Production equipment, like plate freezers, can be more effective because lower evaporating temperatures, -50°C to -55°C (-58°F to -67°F), can be used. These factors help to bring the price of the system down.

However, the ammonia/CO_2 systems require more components, with a higher pressure rating, as well as more expensive materials if temperatures below –50°C (-58°F) are desired. These factors tend to raise the price. Smaller components for small systems will have small benefits, but larger systems will reap much greater gains. Thus, small industrial ammonia/CO_2 systems will be more expensive than conventional systems, but large systems might end up at more or less the same price, for the same capacity.

If the following factors are taken into consideration, the investment in CO_2 can be very attractive:

- Saved production area or more production capacity using the same area
- Improved freezing time (both theoretically and in practice)
- Improved quality of frozen products
- Better COP and thus, lower energy cost

- Savings on insurance premiums
- No risks of damage to product and minimal risk to working personnel due to refrigerant leaks
- Easy service and maintenance work
- Very low refrigerant price
- A safe future. No restrictions due to laws and regulations.

Environment

To illustrate the impact of using CO2 in lieu of a more typical system, consider the result of hypothetically replacing the existing refrigeration system in two of the largest super trawlers that have vertical plate freezers with a CO_2/ammonia cascade freezing system of similar capacity:

- For an R-22 system, such as in the “Wilhelm Van Der Zwan”: the 55-tonne (121,000-pound) R-22 refrigerant charge would be replaced with a 12-tonne (26,000-pound) charge of CO_2 and a 1-tonne (2,200-pound) charge of ammonia
- For an ammonia-brine system, such as in the “Atlantic Dawn”: the 5-tonne (11,000-pound) ammonia charge and 100 tonnes (220,000 pounds) of calcium chloride would be replaced by a 12.5-tonne (27,600-pound) charge of CO_2 and a 1-tonne (2,200-pound) charge of ammonia

The first obvious advantage of conducting these hypothetical projects would be environmental. Each ship would use only natural refrigerants that have no effect on the stratospheric ozone layer, and use no ammonia in the production areas, where a leak may create risks for both production personnel and products.

Since the Atlantic Dawn would no longer have to bear the weight of a brine system, there might be an additional advantage in reduced fuel usage. When we compare the weight of the ammonia/CO_2 system versus the ammonia/brine system, the machinery, vessels and refrigerant more or less would weigh the same for the two systems, but the big difference is the 100 tons of brine. With information provided by ship design consultants Vik & Sandvik in Norway, we can estimate the difference in fuel usage using the following assumptions:

- A large fishing vessel will average 16 knots every 24 hours for 100 days a year.
- The energy required each hour to transport 100 tonnes (220,000 pounds) at this speed is approx 50 kWh (170,000 BTU), equal to the energy from burning 13 liters (3.43 gallons) of fuel.

Thus, the annual savings due to the reduced weight would be
13 gallons x 24 hours x 100 days = 31,200 liters (8,240 gallons) of fuel.

For land-based installations, many laws around the world limit the quantity of ammonia allowed in refrigeration installations, especially in densely populated areas. This trend is showing in the marine sector as well. This makes it very difficult to make large efficient freezing production installations with ammonia. The ammonia/CO_2 cascade system seems to be a very attractive alternative, especially when it can be designed to perform with the same or better efficiency than a traditional ammonia installation, and made with standard refrigeration components, and with the same features as typical halocarbon installations (e.g., oil rectifiers, miscible oil, hot gas defrost)

The Cascade System

In the ammonia/CO_2 cascade system, the CO_2 and the ammonia are in two separate circuits, and these two circuits come into thermal contact in the interstage heat exchanger, where they exchange heat with each other without mixing the two refrigerants. The interstage heat exchanger serves as condenser for the CO_2 system and as evaporator for the ammonia system.

In the system described herein, ammonia is used as the high temperature refrigerant, but halocarbons can also be used, if there is a specific demand to avoid the use of ammonia, and where local laws do not prohibit their use.

The CO_2 refrigeration circuit controls restrict its operation to a set pressure range to permit the use of standard industrial refrigeration components. Thus, the system can be operated, maintained, serviced, and understood by the existing staff of operating personnel and service engineers. Further, spare parts are readily available with short delivery times anywhere in the world, and can be installed by local service staff. The system piping design confines the ammonia to the machinery room, where safety precautions are relatively easy to implement. CO_2 is fed to production areas, cold stores, freezing tunnels, because it presents less risk to both people and products.

For managing potential ammonia releases, CO_2 can be mixed with ammonia, which quickly react with each other to form ammonium carbamate. In the Danish Fire-fighting School in Esbjerg, part of the training for handling an ammonia release includes using CO_2. It would be very convenient to do this in the machinery room, where CO_2 is available in good quantities. If this is done, personnel must take proper precautions to avoid exposure to high concentrations of ammonia, and prolonged exposure to the CO_2. Because it is a heavy gas, CO_2 displaces air in poorly ventilated spaces, which lowers the oxygen concentration. The American Conference of Governmental Industrial Hygienists has noted that 30-minute exposures to concentrations of CO_2 of 5% or greater have resulted in intoxication. (ACGIH, 1971) The US Navy permits emergency exposures to this concentration of up to one hour. (US Navy, 1962) A concentration of 10% or higher can cause unconsciousness and eventual death by asphyxiation over the course of hours of exposure. (OSHA, 1989)

Evaporators

Due to the high saturation pressure of CO_2, the vapor density is also high. At -40°C (-40°F) the saturated vapor density of CO_2 is 26.2 kg/m^3 (1.636 lb/ft^3), while for ammonia and R-22 it is 0.6447 kg/m^3 (0.04 lb/ft^3) and 4.9 kg/m^3 (0.306 lb/ft^3) respectively. While this has a positive effect on compressor performance, it can, if not taken into consideration, have an adverse effect on evaporator performance. As an example, consider an evaporator operating at a temperature of -40°C (-40°F) that produces a given amount of refrigeration. If we calculate the exit velocity of the suction vapor using different refrigerants, we would note that the value for CO_2 is nearly an order of magnitude lower than both ammonia and R-22, for circulation rates of 1 to 8. At -50°C (-58°F) the differences in exit velocity become even more pronounced. While this property of CO_2 naturally leads to lower pressure loss in the suction line and evaporator, the flow is likely to be a “lazy river” in the bottom of the evaporator channel. Insufficient agitation of the liquid leads to inadequate wetting of the channel wall and, as a consequence, a poor heat transfer coefficient.

As discussed earlier, the solution to this problem for CO_2 is to use smaller channels/pipes and longer circuits. As CO_2 has a much better saturation pressure–temperature relation than the above-mentioned refrigerants, it is entirely possible to find a solution that yields a good heat transfer while keeping the pressure loss and equivalent temperature penalty within acceptable limits.

From the outset, our CO_2 refrigeration efforts have focused on fisheries, and fisheries commonly use vertical plate freezers (VPFs). However, the usual suppliers of plate freezers were not in a position to examine the flow and thermodynamic behavior of CO_2 within their freezers. Thus, we needed to take the initiative to develop a program together with a plate freezer supplier if we wanted to obtain an optimized product.

Since the product side dominates the heat transfer in a VPF, we reasoned that the heat transfer coefficient on the refrigerant side had little influence unless it was extremely low. Thus, our design work focused on creating a plate with a channel that gave an acceptable flow pattern and pressure loss throughout most of the freezing period. The concept was tested in our lab, where the performance was verified.

Defrosting

A special design for defrosting CO_2 vertical plate freezers systems with hot gas is essential, where no other means of defrost is possible. Figure 1 shows a schematic of the defrost system we developed. The system consists mainly of a special version of a standard high-pressure compressor suited for 45-bar (653 psi) use and a hot gas bypass valve. The special compressor is connected to the discharge side of the CO_2 cooling compressors through a desuperheater, enabling it to produce hot gas at 45 bar (653 psi) corresponding to a condensing temperature of +10°C (50°F).

The system offers several advantages:

- Service personnel and operators can easily understand it.
- It uses only standard industrial refrigeration components.
- The defrost compressor can also run in a cooling mode, so by using two defrost compressors in the system, one in cooling mode and one in defrost mode, there is always a back-up compressor available for emergency situations. This is an important safety factor as no production is possible without the capability of performing a proper defrost.
- When no defrost is performed, the defrost circuit equalizes to the interstage heat exchanger pressure, so no components are kept for prolonged periods at 45 bar (653 psi).
- Liquid drains from the evaporator under a differential pressure exceeding 35 bar (508 psi), so it will leave the plate freezer very quickly.

- The system saves energy when defrosting, as the COP of the defrost compressor is roughly double that of the ammonia compressors. This means, in fact, that if it is possible to utilize the savings on the ammonia compressors by reducing their capacity or stopping the compressors, we will save 200 kW (682 MBH) on the ammonia compressors each time we use 100 kW (341 MBH) on the defrost compressor. This creates energy saving possibilities when cold water or other means of condensing at +10°C (+50°F) are present.
- The design offers a large defrost capacity (approximately 800 to 1000 kW heating capacity (2,729 to 3,412 MBH) from this small reciprocating compressor.

We learned a lot from our experience with the system on the Kvannøy:

- The design defrosts the whole plate, hose connections and manifolds very well.
- The plates do not get hot at the top while they are still frozen at the bottom, as is often seen on typical systems, which increases the quality of the frozen blocks.
- It can be difficult to release the frozen blocks from evaporator parts that do not take an active part in the defrost, e.g., end plates, bottom plates, so heat must be actively transferred to these areas by metallic contact with the defrosting plates.
- The system offers a tremendously fast defrost, as long as there the CO_2 cooling compressors are sufficiently loaded. This is similar to a conventional system, where we need to have a minimum cooling loads to generate sufficient hot gas for a decent defrost.
- The system demands a more complicated and very fast-reacting regulating and control system to work well. Both the defrost compressor and ammonia system must be regulated and the ammonia compressors must be able to decrease capacity very quickly, as the defrost compressor “steals” approximately 800 kW (2,729 MBH) of capacity from them in a matter of seconds.
- After adjusting the regulator, the system works very well and performs an even better defrost than conventional systems.

Capacity Regulation

Each step in capacity for CO_2 compressors produces very large changes in cooling capacity for the system. At -50°C (-58°F) for a given capacity, the swept volume ratio between CO_2 and ammonia is about 1 to 35. This means that at -50°C (-58°F) suction temperature, a unit increase in volumetric capacity on the CO_2 compressors yields approximately 35 times more additional capacity than a similar increase in an ammonia system. Thus, capacity and system regulation for CO_2 systems must be capable of reacting much faster than those used with ammonia systems.

Furthermore, in systems with very big differences in capacity, as in the Kvannøy with its many plate freezers and a small capacity in the holds, it is essential to have some small capacity compressors to handle the low load situations. If only large capacity compressors such as screws were used, we would run into situations where load variations would make it necessary for a screw to stop to avoid creating “dry ice” (frozen CO_2) in the low-pressure receiver. If this happened it would be necessary to start the compressor again almost immediately, which would not be feasible for electric motor protection reasons. This would cause unintended regulation problems in the plant and it would be difficult to keep a steady pressure in the separator. The smooth capacity regulation of a screw seems to be an attractive feature, but the problem of the large minimum capacity must be dealt with in systems like the one described here.

The need to handle large and rapid changes in capacity is even more important on the ammonia side of the system when the CO_2 hot gas defrost system is used. The changes are so rapid when the defrost compressor starts up, that a hot gas bypass system on the ammonia side is necessary to keep pressures and temperatures in the system balanced while the ammonia compressors are reducing their capacity. Even with the variable speed drives on the ammonia screws, the hot gas bypass feature is necessary. Reciprocating compressors on the ammonia side will have a clear advantage, with their capability of cutting in and out all the capacity in a matter of seconds and even running fully unloaded with a minimum of power consumption for some time. This can turn out to be a useful feature on ammonia reciprocating compressors in these types of systems. The clear advantage of the screw compressors on the ammonia side is the large sizes of the available compressors to match the capacity needed.

To illustrate how fast the changes in the system are during normal operation, consider the Kvannøy system. Two plate freezers are filled with whole round herring at approximately +1°C (34°F) and started nearly at the same time. The initial load on the freezer is huge, so three reciprocating compressors start up to handle the load. After 10 minutes, though, only one compressor remains running at 25% capacity at -50°C (-58°F) suction temperature. The time required to pull the core temperature of the fish blocks down to -25°C (-13°F) was 1 hour and 45 minutes.

Our experience with the system on the Kvannøy shows that compressor controls, system control strategies, and everything else, have to be adapted to this new type of system. We cannot do just as we used to do on conventional systems, and must rethink our control systems and make them fit to the behavior of these systems.

Controlling the System

As the defrost compressor has a maximum differential pressure of 25 bar (363 psi), a maximum suction pressure of 25 bar (363 psi), and must be able to deliver at hot gas defrost pressure of 45 bar (653 psi), we need to control the system such that the compressor's operation is restricted to its allowed working conditions. This is done by combining a quick-reacting hot gas bypass valve with a suction pressure regulating valve for the defrost compressor. The challenge is to maintain the correct working conditions for the defrost compressor when the compressor begins a defrost cycle, and within seconds after removing approximately 800 kW (2,729 MBH) of capacity from the cascade interstage heat exchanger. This demands an immediate removal of capacity from the ammonia compressors to keep the CO_2 pressure in the cascade interstage heat exchanger above 20 bar (290 psi). This is necessary for the defrost compressors to be capable of producing the 45 bar (653 psi) defrost pressure and continue to work within the 25 bar (363 psi) differential pressure allowed.

If the CO_2 cooling compressors are providing insufficient capacity to feed the defrost compressor, the CO_2 pressure in the interstage heat exchanger could drop below the required 20 bar (290 psi). To avoid this, we also use the defrost compressors' capacity regulation and the hot gas bypass valve to keep the compressors' differential pressure under control. This feature secures the balance of the system. When the load on the CO_2 cooling compressors is insufficient, the defrost takes a longer time, just as with conventional systems.

When too little cooling load is available, it means that not all plate freezers are running, and thus the time needed to defrost, empty and fill the plate freezer is of less importance. With sufficient cooling load on the plate freezers, the defrost compressor will give us a defrost capacity of approximately 800 to 1000 kW (2,729 to 3,412 MBH), which is sufficient to do a complete defrost on the plates, hoses and manifolds, within the maximum 4-5 minutes desired by the fishermen.

We faced a problem with variations in the defrost time, but this had nothing to do with the ammonia/CO_2 system. The problem was in communicating to the automatic emptying robot what the defrost time will be, so the automatic emptying process can begin at the proper time. The defrost time needs to be the shortest possible time when all plate freezers are in use to ensure the maximum production capacity.

However when insufficient suction gas is available to the defrost compressor, the compressor regulation reduces capacity to keep the compressor within its permitted operating range. With the capacity reduction the maximum defrost capacity cannot be achieved thus increasing the defrosting time. The reason for insufficient suction gas for the defrost compressor is that there are not enough fish to fill all freezers and thus the overall cooling capacity is low. In this situation the longer defrost time has no effect on production capacity since other freezers are available. The necessary defrost time for a plate freezer was thus signaled to the robot by the load on the cooling compressors, and it could then begin emptying the freezer at the appropriate time.

As discussed under capacity regulation, the changes in pressures and capacity on the system are much faster than in conventional systems because of the high efficiency on the evaporators and the high volumetric efficiency on the compressors. This can make it difficult to control the pressure very accurately with the compressors, as it will be necessary if we want to run a system close to -55°C (-67°F) or even just -50°C to -52°C (-58°F to -62°F) without unintentionally risking the creation of dry ice, which happens at -56.6°C (-69.9°F). We prevent this occurrence with the very quick-reacting electronically controlled hot gas bypass that will not allow the pressure to drop under a -55°C (-67°F) limit put into the system. This feature ensures we will not run into problems with dry ice in the system, no matter how the operating personnel should run the system.

System Lubricant

The lubricant used in the CO_2 system is a polyol ester (POE), and is fully miscible with the CO_2 over the whole temperature range in this application. POE has been extensively tested and approved by the compressor manufacturer. Thus, we were able to design the piping and oil return system in nearly the same way as with a typical R-22 system, i.e., with an oil rectifier. Without this feature, it would be much more difficult and expensive to install a CO_2 system in a marine application. The limited available space for large oil separators, and the fact that a fishing vessel is not always stationary, but is often subject to wave motion, makes return of immiscible oil very problematic. Also, because POE is not as dense as liquid CO_2, it would float on top of the CO_2 liquid, making oil draining or recovery very difficult.

We were unable to measure any negative effect on the evaporators' performance due to the presence of small amounts of oil in the refrigerant. As mentioned before, the freezing time was even better than anticipated.

The presence of oil in the CO_2 may actually turn out to be an advantage, lubricating components such as valves and regulators, but this has yet to be proven. We do know that the strong cleaning and poor lubricating characteristics of CO_2 can cause problems in oil free systems, such as valves and threads seizing.

POE's hygroscopic nature causes it to absorb whatever water it comes in contact with in the CO_2 system, so care must be taken to keep it dry, as with hydrofluorocarbon systems. It is also very important the POE lubricant is never charged into the ammonia system by accident, since some POEs react with ammonia and the small amounts of moisture always present. However, if the polyalpha olefin (PAO) lubricant used in the ammonia compressors should find its way into the CO_2 system, it would not harm the system. PAO will not mix with CO_2, but will float upon the CO_2 liquid in the separator.

This solves another problem for us, because we can use PAO, which is not hygroscopic, for lubricating seals, gaskets, threads, etc. during construction and assembly of the CO_2 system.

Proper cleaning of pipes, vessels, and components seems to be even more important for CO_2 systems than for typical ammonia systems. The cleaning properties of the CO_2 will remove dirt and particles very quickly and transport these contaminants to the filters and compressor. We observed that the lubricant initially charged into the compressors seems to very quickly become discolored by dirt particles, grease, or oil from the valves, compressors, and other components. However, it does not pose a problem for the compressors as long as the lubricant is properly filtered, as it is on compressors today.

Stand-alone Units

Since the saturation pressure of CO_2 at even moderate ambient temperatures is too high for convenient storage, a stand-alone unit is employed to keep the pressure down in the low-pressure receiver. With intelligent management of the system, a complete drainage of evaporators, etc., can be achieved, reducing the stand-alone pressure problem to cooling of the low-pressure receiver.

In the Kvannøy project, we considered an additional task for the stand-alone unit. If properly insulated, the transmission heat load in the cargo holds should be relatively small. Since the minimum capacity of a CO_2 refrigeration compressor is fairly large, it is very likely that the compressor will experience many start/stop situations, with excessive wear as a result. Usually the cargo holds do not need a –50°C (-58°F) temperature, so it was suggested that we transfer the hold-keeping capacity to one or two stand-alone units during periods where the cargo holds were the only duty for the system. This equipment uses small, relatively inexpensive, standard air-cooled condensing units. Naturally, the greatest benefit is for small cargo holds, such as in our case.

CO_2 Cooling Circuit

The system is diagrammed in Figure 2. The maximum operating pressure (MOP) of the CO_2 low side is 25 bar (363 psi). The parts used in the CO_2 cooling circuit are standard industrial refrigeration components. The only difference is the higher pressure rating (MOP of 50 bar (725 psi)) of the components in the defrost circuit. Besides having the higher pressure rating, the plate freezers are also specially designed to use CO_2.

The compressors are the high-pressure version of a standard reciprocating compressor (MOP of 40 bar (580 psi) discharge side).

We chose reciprocating compressors for this application because they can operate at the low capacities often required, and offer excellent COPs at both full and part load. If we had chosen a screw compressor, its minimum capacity of 15-20 % would be too much for low load situations, and would require a hot gas bypass to run most of the time, even if fitted with a variable speed drive.

Three of the six reciprocating compressors are specially modified with a maximum discharge pressure of 45 bar (653 psi), which enables each of them to serve as the defrost compressor. Since only one of these will act as the defrost compressor at a time, it means there are always two compressors in the system that can be used as a backup unit during service, or if a problem should occur. This redundancy ensures a high degree of reliability in our system, and provides a service and spare part benefit to a ship that depends completely on the functionality of the refrigeration system.

CO_2 Condensing Circuit

The CO_2 condensing circuit is also built with standard refrigeration components, with the only demand being that they have a MOP of 40 bar (580 psi). This is not a problem today, as most common stop, regulating and safety valves meet this standard.

CO_2 Hot Gas Defrost Circuit

In this circuit, the 50 bar (725 psi) MOP makes it more difficult to use standard refrigeration products. Stop valves and safety relief valves are available with the higher pressure rating as the only difference. Solenoid valves are not available, but pneumatic or motor-activated ball valves widely used in the refrigeration industry are, and have no problem in handling both CO_2 and the 50 bar (725 psi) pressure. Normal refrigeration regulating valves are a problem, but with the special defrost system used, the pressure setting and regulation are managed by the defrost compressor and its bypass valve system, so no further pressure regulation devices are needed to keep the pressure in the right range for the defrost.

As the compressor manages the pressure, we only need to worry about what is leaving the evaporator. If it is liquid, we have delivered the latent heat of the refrigerant. If it is vapor, we must close the line to make sure that the vapor deliver its latent heat. This calls for a float valve, but standard refrigerating valves with MOP of 50 bar (725 psi) are not available. Other available float valves with the necessary capacity for defrosting production equipment in a sufficiently short time, will be extremely expensive, difficult to get with short notice and therefore not the kind of product we want to have in our system. Luckily, the steam industry has run into the same problem and has developed an elegant solution: float and thermostatic steam traps. Their only problem is their limited capacity, but using several of these in parallel gives a perfect solution to the problem with a suitable regulation performance.

Another great advantage with the defrost compressor system is that the only vessel necessary in the MOP 50 bar (725 psi) system is the defrost compressor oil separator.

R-404A Stand-alone Cooling Circuit

The MOP of the R-404A high side is 28 bar (~400 psi). The MOP in this circuit is determined by the refrigerant's properties, but R-404A was a good choice for the task. Condensing units from the commercial refrigeration area for R-404A with the needed capacity are easily available, and easy to service.

Ammonia Circuit

All components in this circuit are normal industrial refrigeration components for ammonia, and are rated at an MOP of 24 bar (~350 psi).

Approvals

The standard that governs design, manufacturing, and operation of pressure equipment for European Commission members, the Pressure Equipment Directive (PED), does not cover sea-going vessels, but the installation needs to be approved by another authority like Lloyds of London or Det Norske Veritas (DNV), an independent risk management agency. Neither Lloyds nor DNV have finalized rules at the moment that apply to ammonia/CO_2 cascade systems, but they are working on them, together with the industrial refrigeration industry. For the time being, they will approve each new installation on a case-by-case basis.

DNV has produced some preliminary guidelines that have been sent to some relevant industrial refrigeration manufacturers and contractors. In these guidelines, the rules for the CO_2 part of the system are much gentler than those that apply to the ammonia part of the system, in spite of the much higher operating pressure on the CO_2 parts of the system.

The alarm and warning levels for ammonia and/or CO_2 concentrations in the areas around the refrigerating system are, on this first ship, specially approved by DNV and to be set at the following levels.

<u>CO_2</u>

- 10,000 ppm: Warning.
- 30,000 ppm: Visual alarm and start emergency ventilation in machine room.

<u>Ammonia</u>

- 150 ppm: Warning.
- 1,500 ppm: Alarm and start emergency ventilation in machine room.
- 30,000 ppm: Complete shut down of machine room, except for explosion-proof installations. 30,000 ppm is a 3% concentration, which is 20 % of LEL, i.e., the Lower Explosion Limit, which for ammonia is 15% concentration in the air.

Service Work and Leaks

In spite of its higher pressures, it is much easier to service and repair the CO_2 system than a conventional system. Because CO_2 is cheap and harmless to the environment, we can vent it before service operations without much concern.

However, we did need to take some precautions:

- Be careful not to vent CO_2 to an area where the concentration can build to an asphyxiating level. NOTE: CO_2 is heavier than air so its concentration will be greatest in low areas, e.g., at the floor, in the basement.
- When venting refrigerant before evacuating a system or part of a system, take care to release only the vapor phase, because any liquid that comes out will form dry ice, which could block the release valve, line or hose. This could lead to an incorrect observation that the system pressure has equalized with atmospheric pressure. If the part of the system is subsequently dismantled where pressure is still present, serious injury, or worse, could result.
- Changing or cleaning filters is very easy. If a small amount of liquid is still present in the filter housing when it is equalized to atmospheric pressure, only a little dry ice will form in the filter. This dry ice can easily be removed, and the filter changed or cleaned, refitted and put back into operation.
- Properly handled, dry ice will not present a problem, and can actually be an advantage. Unlike "water ice," dry ice has a higher density (i.e., is more compact) than the liquid phase, so it will contract rather than expand. Thus, the creation of dry ice will not damage pipes, filters, vessels, or other components. This characteristic can be very useful in service work. If a liquid-containing pipe or vessel needs to be opened for service or other reasons, it can be vented to atmospheric pressure and kept vented, and the liquid turned to dry ice. The service work can be completed and the system closed again. When the pressure is subsequently raised above CO_2's triple point, the dry ice will disappear and change back to liquid. Not much refrigerant would be lost, and it makes it

possible to service CO_2 systems in situations where it would be very difficult and expensive to service a conventional system.

- In low-pressure receivers, the high density of dry ice must be considered. Dry ice will sink below the liquid, and could block the liquid inlet to the pumps, or even mechanically damage the pumps. This is a good reason for paying close attention to the minimum capacity on the cooling CO_2 compressors, to ensure that the low-pressure receiver pressure is always kept above the triple point.
- When CO_2 systems are built or opened for service it is essential that proper vacuum procedures are followed to get all moisture out from the system, to avoid the serious problems that can occur in connection with water contamination of CO_2 systems.

Mixing of CO_2 and Ammonia

When CO_2 and ammonia come into contact, they react and produce ammonium carbamate, a white powder that may aggressively attack metal and lead to corrosion. So, ammonium carbamate is definitely something we want to avoid creating in our system. We know from accidents in the past what can happen. Some years ago an ammonia plant was built in a developing country, and because nitrogen was not available for the pressure test, CO_2 was used. The CO_2 was not properly evacuated before the system was charged with ammonia. This led to massive problems with sludge and pollution of the oil.

We always assume that CO_2 would enter the ammonia system in the event of a leak, because the pressure on the CO_2 side of the interstage heat exchanger is much higher than on the ammonia side.

Approximately 0.03 % of the natural atmosphere is CO_2. Thus, CO_2 inevitably finds its way into ammonia plants, especially low temperature plants operating below atmospheric pressure. Over time, quite a large quantity of CO_2 can be sucked into these systems through leaks. This phenomenon plays an active role in creating some of the sludge found in leaking ammonia systems.

Ammonium carbamate evaporates fast at temperatures over 70°C (158°F). So, if a massive problem should occur with mixing CO_2 and ammonia in the system, a simple way of cleaning the system can be to just blow hot air through the system for as long time as it takes to evaporate the ammonium carbamate. The way our system is built allows us to use this way of handling such a "worst case" scenario. If an interstage heat exchanger should leak CO_2 into the ammonia system, the stand-alone unit is able to keep the CO_2 system under control while repair work is done on the ammonia system. It might be difficult to gather the necessary number of hair dryers among the crew, as commercial fishing is a very male-dominated occupation! However, other hot air sources can be used.

By using a shell and tube interstage heat exchanger, we can repair a leak on the spot. This is very important on a ship, where the removal and exchange of such a large piece of equipment would be extremely difficult, time consuming, and expensive. On our system, we used a shell and tube interstage heat exchanger with double endplates that vents to atmosphere between these plates. But on future systems we do not regard this as a precaution we need to take. We made contact with a Danish company who has delivered equipment for production of CO_2 for more than 30 years. In these systems, shell and tube ammonia evaporators from our company with single endplates have been used for condensing CO_2 without having any problems with leaks between CO_2 and ammonia.

Water Contamination

CO_2 refrigerant is sensitive to water contamination. Liquid phase CO_2 can dissolve much more water than the gas phase at the same temperature and pressure, resulting in the same problem we have with R-22: freezing up of expansion valves.

Fortunately, the acceptable limits for water content in a fresh CO_2 charge are very low, which avoids these problems.

The water and CO_2 solution is actually carbonic acid. Carbonic acid promotes galvanic corrosion, and will attack materials inside the system. We have heard reports of heavy acid attacks on steel pipes and on screw compressor ball bearings in old CO_2 systems with water contamination. Good evacuation procedures will keep the systems free of water, and must be followed before charging CO_2 into the system, and also when it is opened for service and maintenance.

Filter-driers used for R-22, R-404A, R-507 etc. can also be used for CO_2 as well, and have shown to be very effective. We suggest mounting filter-driers in a bypass line for the cold liquid line where they will be most efficient.

We use a model of sightglass that has a moisture indicator, that has been tested by the manufacture with CO_2, and that has proven to work well. The sightglass needs to be selected to match the maximum expected water content of the CO_2 at the temperature and pressure where the indicator is placed in the system.

Special CO_2 Issues

One of the special properties of CO_2 is its compatibility with elastomers, e.g., rubber, polytetrafluoroethylene (PTFE), plastics. The influence of CO_2 on these materials is pressure-dependent; when pressures increase beyond about 50 bar, compatibility can become a big problem. Rubber in gaskets can burst when pressures change rapidly, and PTFE in valve seats can change size to such an extent that it cannot be used as valve seats in many applications. These compatibility problems are a prime reason for the pressure level ranges we maintain in our system. In the 45 bar defrost circuit (MOP=50 bar), we install the minimum number of valves necessary, and these have been chosen and tested carefully to ensure compatibility. We learned not to underestimate problems of this nature during our development and testing period.

CO_2's strong cleaning properties and low liquid viscosity make it a poor lubricant. This should be considered when selecting components such as pumps and valves. We use miscible oil and do not expect lubrication problems. So far, our experience is very good, and no problems have been reported.

When designing systems with CO_2, special care must be taken to prevent liquid from being trapped in the system. If these parts of the system heat up, the resulting hydrostatic pressure can easily exceed the MOP. A lockout, tag-out procedure for service valves can be a good precaution.

Damage of valve orifices due to cavitation seems not to be a problem on CO_2 systems. In this respect, CO_2's behavior is gentler to expansion devices than ammonia. In the reverse situation of condensation-induced shock, the impact of imploding CO_2 bubbles in liquid at atmospheric pressure is approximately 3% of the value for water at the same pressure. This is one of the reasons for using CO_2 for "soft" ultrasound surface cleaning, where the cleaning is done by the implosion of bubbles created by ultrasound in CO_2 liquid.

Reconfiguring Old Systems to be CO_2 Cascade Systems

Existing ammonia systems can be reconfigured as ammonia/CO_2 cascade systems by using the existing compressors and condensers in the ammonia circuit if they are suitable for the new working conditions. Motor size on the compressors must be checked and maybe changed to match the new conditions.

On the CO_2 side, it may be cheapest to change the complete system. Production equipment like plate freezers and evaporators that need to handle hot gas defrost at 50 bar (725 psi) must be changed to specially designed evaporators for CO_2 use and/or specially approved for 50 bar (725 psi) MOP.

Valves in the existing system in many cases will not provide the necessary pressure rating. Additionally, liquid lines must be slightly larger. The suction pipes will not be correctly sized to return oil back to the compressors, as the velocity will be too low to lift two-phase flow in risers. The oversized suction lines and evaporators will require an unnecessarily large CO_2 charge. A larger charge will require a larger low-pressure receiver and larger pumps. In our system, the low-pressure receiver must also be equipped with a cooling coil for the stand-alone unit, so a new low-pressure receiver specially designed for CO_2 use is necessary.

By changing the whole system on the CO_2 side we avoid problems related to mixing old oil, sludge, and other contaminants from the old system with CO_2. Problems of this nature need to be considered, which we learned by accident in our research and development test center. Old oil that we had been unable to remove formed a thick sludge that clogged filters and orifices in our test rig.

Existing R-22 systems can be reconfigured as halocarbon/CO_2 cascade systems as well. The same considerations apply as for reconfigured ammonia/CO_2 systems with the following exceptions:

- The compressors can normally be reused, but working pressures, existing gaskets, o-rings, and compatibility with the ester oils might demand an overhaul, and motors are usually too small.
- It might be necessary to replace condensers, as the original pressure rating often does not allow the higher condensing pressure introduced by the new refrigerant.
- The liquid line might be a useable size, but there is still the pressure rating problem with other components. The piping system will have the appropriate design to return lubricant to the compressors.
- For both types of plants that will be rebuilt, it will result in a dramatic reduction in refrigerant charge. This might become a very attractive feature in light of all the laws and regulations being introduced for ammonia and R-22 use now and in the future.

- The price of the "new refrigerants" such as R-134a, R-404A, R-507, R-410A, etc., both now and in the future, speaks for itself, when considerable charge is needed.

Perspectives for the Future

When initially introduced to CO_2, the reaction by many is concern about the high pressures. We admit, the pressures are higher than traditionally experienced in refrigeration plants, but several applications have been running with much higher pressures for years, with hydraulics and fuel injection in car engines being obvious examples. It is our opinion that great potential lies in accepting and using higher pressures than today. We especially advocate compressing CO_2 in two stages, thus eliminating the need for other refrigerants to be used in the cascade system. This option is viable today if the cooling media can enable a condensing temperature at +10°C (50°F) (or even slightly higher) with a considerable gain in COP as a result, not to mention the reduction in installation cost. Alternatively it should be possible to shut down the ammonia or halocarbon type systems in the colder seasons, which would increase the COP, but not reduce the installation cost.

Hotter ambient conditions necessitate a transcritical or a cascade system. Transcritical systems are used in automotive air conditioning and in principle pose no problem. However, the pressures are much higher, in excess of 75 bar absolute (1088 psi), which is where the relatively small size and simplicity of the automotive systems helps greatly. While it is not a problem to design and manufacture heat exchangers, compressors, etc., it is certainly expensive and thus the overall system might not be competitive. Furthermore, if consumers are skeptical about a 50 bar (725 psi) system, one can only imagine their shock at a 100 bar (1450 psi) transcritical design proposal. Maybe refrigeration companies should employ psychiatrists in the future? In any case, classification of industrial size systems like these would present a major challenge.

The experience we gained during the design of the Kvannøy showed that it was not a problem to get components that could handle pressures above what is normally encountered in the refrigeration industry, but instead, to get them at a price where the solution could still be economically competitive. Very often, we found that we had crossed the line into the domain of products used in the chemical industry, with an enormous price increase as a result.

The solution was in many cases to ‘push’ the traditional suppliers to tell us why their components could not handle the increased pressures. Most would admit that while most components could physically handle the pressure, the original classification was performed at a pressure, which, at the time, was deemed to be sufficient for the refrigeration industry. Individual pressure tests of these components ensured their fitness for use in our CO_2 system. We benefited from being a major consumer for many suppliers, and were able to make the project happen as a result. It is our opinion, however, that as more of the industry fully grasps the importance of $CO_{2,}$ this problems will go away. It seems that for regulating equipment (e.g., stop valves), the “magic line” is at 40 bar (580 psi), and 50 bar (725 psi) for non-regulating equipment.

Careful design of the system to limit the number of components exposed to high pressures greatly simplifies the task of sourcing the correct components. The system we developed uses a minimum of regulating equipment in the high pressure defrost cycle.

One further boundary to explore is using evaporating temperatures that approach CO_2’s triple point (-56.6°C, -69.9°F). In our project, we decided to limit ourselves to -50°C (-58°F). Some classification societies require special materials when temperatures are below –50°C (-58°F), but the promise of a capacity increase of another 10% in the case of an evaporating temperature of –55°C (-67°F) should be sufficient to interest most companies.

Conclusion

CO_2 is no longer the refrigerant of the future; it is a refrigerant of the present. This project demonstrated that it is possible to make CO_2 systems with all the features we normally expect in a traditional system, and most of these features are realized in the way we are used to doing things. Furthermore, with proper design, the system can use components that are standard throughout the refrigeration industry.

In a very short time CO_2 has become a most attractive refrigerant when it comes to freezing applications. Once we had realized that the apparent problems with CO_2 refrigeration could easily be solved, we realized its substantial benefits:

- Increased production on the same equipment, or same production on smaller equipment
- Higher product quality.
- Marked improvement in COP compared to other refrigerants and system types.
- Substantial reduction in equipment size.
- Environmental friendly solution, especially when used in cascade with ammonia. Ammonia is present only in the refrigeration machinery room and the ammonia charge is much smaller, enabling production in more restricted areas.
- The loss of product due to leaks is virtually non-existent, unlike an ammonia-only solution. This should have a positive effect on product/cargo insurance.
- Recharging is inexpensive compared to other refrigerants.

With the ammonia/CO_2 cascade now being a viable option, there is no longer any reason for large halocarbon systems and there is no longer a reason to have ammonia in the production areas. The ammonia/CO_2 cascade can do the job more efficiently, quicker, and safer – both to the worker and to the planet.

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System	Evaporating/Condensing Temperature -40°C/+25°C (-40°F/+77°F)	Evaporating/Condensing Temperature -50°C/+25°C (-58°F/+77°F)
Ammonia, single stage	1.75	1.09
Ammonia, two stage	1.92	1.38
R-22, single stage	1.77	1.22
R-22, two stage	1.86	1.42
Ammonia/CO_2 cascade	2.18	1.78

Table 1. Coefficients of performance for candidate refrigeration system designs

System	Evaporating/Condensing Temperature -40°C/+25°C (-40°F/+77°F)	Evaporating/Condensing Temperature -50°C/+25°C (-58°F/+77°F)
Ammonia, single stage	1.93	3.33
Ammonia, two stage	2.33	3.85
R-22, single stage	1.75	2.85
R-22, two stage	2.01	3.05
Ammonia/CO_2 cascade	0.77	1.00

Table 2. Relative compressor displacements for candidate refrigeration system designs