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EVAPORATORS FOR CO₂ REFRIGERATION

BACKGROUND

Carbon Dioxide was first proposed as a refrigerant by the British inventor, Alexander Twining, in his 1850 patent. Its use in refrigeration systems gradually increased and peaked in the mid-1920's. After the introduction of the synthetic chlorofluorocarbon "CFC" refrigerants (i.e. R12) in the 1930's, the use of CO₂ as a refrigerant declined and finally disappeared almost completely by about 1960.

In 1974 the researchers Molina and Rowland published a laboratory study demonstrating the ability of CFC's to catalytically breakdown Ozone in the presence of high frequency UV light. Further studies estimated that the ozone layer would be depleted by CFC's by about 7% within 60 yrs and based on such studies the US banned CFC's in aerosol sprays in 1978. In 1984 British Antarctic Survey scientists Farman, Gardiner, and Shanklin, discovered a recurring springtime Antarctic ozone hole. In the 1980's the first measurements of this loss were actually documented and in 1984, when the British first reported their findings, October ozone levels were about 35 percent lower than the average for the 1960s. The U.S. satellite Nimbus-7 confirmed these results, and the term Antarctic ozone hole entered our popular language.

Ultimately this research led to the Montreal Protocol on Substances that Deplete the Ozone Layer, an international agreement to phase out the production of numerous substances that are believed to be responsible for ozone depletion. The Montreal Protocol was agreed to in September 1987, and entered into force on January 1, 1989.

All of these events triggered intense research activity throughout the 1980's to find alternate refrigerants with lower Ozone Depletion Potential (ODP) and resulted in the development of various HCFC and HFC type refrigerants. At the same time a group of engineers and academics, mostly Europeans, began to promote the use of "natural refrigerants" as being the only truly environmentally friendly fluids to use in refrigeration and air-conditioning machines. It was in 1993 that Professor Gustav Lorentzen at the University of Trondheim in Norway demonstrated that CO₂ could be a viable refrigerant, and a number of patents were issued to him relating to the use of CO₂ in vapor compression machines. Interest in applying CO₂ in various air-conditioning and refrigeration applications has grown steadily since that time.

In the industrial refrigeration world we are accustomed to using ammonia as the only refrigerant in the system. All of us know and appreciate the benefits of ammonia - it is a great refrigerant! However, because of its toxicity there is interest in using ammonia and CO₂ in combination to remove the ammonia from occupied spaces in our refrigerated facilities. Ammonia and CO₂ have very different operating characteristics which must be accounted for when designing and operating industrial refrigeration systems. These differences will be highlighted below with an emphasis on the design and operation of air-cooling evaporators.



CO₂ BENEFITS AND CHALLENGES

Health and Safety

All of us in the industrial refrigeration industry know and love ammonia for its low cost, high efficiency, and ease of application. On the other hand, ammonia's toxicity and flammability characteristics must be carefully addressed with good safety practices including proper maintenance, training, and coordination with local first responders. Depending on the amount of ammonia on the site, various Federal and local agencies (OSHA, EPA, DHS) will begin to take an interest in Process Safety Management (PSM and RMP) systems in place and other regulatory requirements. Even a small release of ammonia from the refrigeration system can create big problems including plant evacuations, required reporting to various governmental agencies, interaction with surrounding populations and local first responders, unwanted media attention, etc. The comment I hear repeatedly from refrigerated facility end users is "I love ammonia and understand its benefits, just don't put it next to my products or people."

Carbon dioxide on the other hand is referred to as being both non-toxic and non-flammable and has a safety classification of A1 according to ASHRAE Std 34 (ASHRAE 2013). Actually, CO₂ is toxic to humans in high enough concentrations. Our bodies continually expel CO₂ as a normal physiological process for a reason, too much CO₂ is bad for us. The AGA gas handbook presents the following data for adults in good health:

- 0.04%: Normal concentration of CO₂ in air.
- 2%: 50% increase in breathing rate.
- 3%: 10 minute short term exposure limit; 100% increase in breathing rate.
- 5%: 300% increase in breathing rate, headache and sweating may begin after about an hour (tolerable — but physically draining).
- 8-10%: Headache after 10-15 minutes. Dizziness, buzzing in the ears, increase in blood pressure, increased pulse rate, anxiety and nausea.
- 10-18%: After a few minutes, cramps similar to epileptic fits, loss of consciousness and shock (sharp drop in blood pressure). A quick recovery in fresh air is possible.
- 18-20%: Symptoms similar to those of a stroke followed by death.

When looking at the concentration amounts shown above for CO₂, bear in mind that even 5% concentration in air is very high compared to the 50 ppm ammonia concentration that has us all looking for the door to get out. 5% concentration is equal to 50,000 ppm, so it takes a relatively large leak of CO₂ to produce the serious reactions listed above. One of the problems with CO₂ leaking in confined spaces is its absence of odor - you can't smell it. Everybody knows when there is even a small ammonia leak, but unless you are trained to recognize the physical symptoms related to the presence of high CO₂ concentrations you won't know there has been a leak and you may be in trouble.



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Carbon dioxide is also different from ammonia in that it is heavier than air and so will tend to collect in higher concentrations at low points in confined spaces like engine rooms.

Carbon dioxide is absolutely non-flammable which is beneficial. Ammonia, while it is flammable only in a narrow range of concentration in air (approx. 16%) is nonetheless classified as B2L according to ASHRAE Std 34. CO₂ is good at displacing oxygen and is actually used as an effective fire extinguishing agent. However, what is good for fire-fighting is bad for us humans who need oxygen to breathe and avoid asphyxiation.

Because of the above risks, having properly placed and maintained CO₂ detectors is absolutely critical to health and safety in industrial refrigeration systems running with CO₂.

Cost

Carbon dioxide is abundant in nature and is widely used in many industries, including the beverage business. This tends to keep the cost of CO₂ extremely low compared to the synthetic HFC and HFO refrigerants. Similarly, the cost of ammonia is also very low because it is produced in huge quantities for use as fertilizer and in other industrial processes. Natural refrigerants are cheap!

Thermodynamic Properties

Two thermodynamic properties of carbon dioxide make it somewhat more challenging to use as a refrigerant compared to ammonia, its pressure and its critical temperature.

The higher operating pressures of CO₂ affect all of the pressure bearing components in the system, compressors, piping, vessels, control valves, and heat exchangers. Fortunately, all of the components needed to design for these higher pressures are now widely available in the market.

We are all used to designing refrigeration systems to operate well below the critical point of the refrigerant. Ammonia has a particularly high critical temperature of 270 deg F (132 deg C) and so can easily operate 'subcritically'. The high critical temperature of ammonia also produces low throttling losses and inherently high COP under most conditions. On the other hand, the low critical temperature of 88 deg F (31 deg C) of CO₂ limits efficient subcritical operation to very low condensing temperatures, around 32 deg F (0 deg C). Operation with high side temperatures above 88 deg F is possible but becomes 'transcritical'. Operation of a transcritical machine is more complicated than a traditional subcritical machine and is always less energy efficient (IIAR 2014).

In the subcritical operating range typical for carbon dioxide, gas densities are very high and the resulting refrigerating capacity (in TR/ft³ of displacement) is relatively high. This translates to compressor displacements which are much smaller than those required for an ammonia compressor having similar refrigeration capacity. The high suction and discharge pressures for CO₂ also result in compression ratios that are significantly lower compared to ammonia.

Ammonia and CO₂ Together - A Perfect Marriage



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Finding ways to cost effectively use ammonia and carbon dioxide together in cascade refrigeration systems would seem to be the perfect solution for a facility operator:

- Both fluids are 'natural refrigerants' and ultimately make industrial refrigeration safe for the environment.
- Ammonia/CO₂ cascade refrigeration systems are at least as energy efficient as a traditional ammonia refrigeration system (Christensen 2006).
- Ammonia has high operating efficiencies even at high temperatures and so can efficiently reject heat even in high ambient environments. Using ammonia on the high side of a cascade system reduces the amount of ammonia in the system and allows it to be managed in ways that minimize or eliminate the negative effects of an ammonia leak on products or people. NH₃/CO₂ cascade systems are safer for workers in refrigerated facilities.
- Carbon dioxide is non-toxic and is highly efficient at low temperatures, which is the part of the refrigeration system that is closest to products and people. Using CO₂ on the low side of a cascade system is safer for people and eliminates the risk of damage to products in the event of a leak.

AMMONIA / CO₂ CASCADE SYSTEMS

Ammonia and carbon dioxide can be used effectively in combination in two main configurations:

- a. Ammonia / CO₂ Cascade with low-side compression
- b. Ammonia / CO₂ Cascade with low-side pumped as a volatile brine

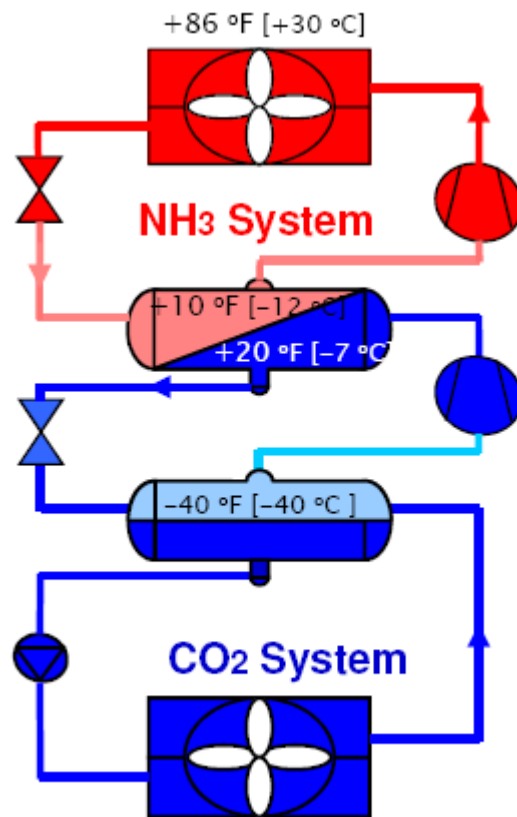
Ammonia / CO₂ Cascade with low-side compression

In this type of system carbon dioxide is used with compression on the low side to absorb heat from the load being refrigerated. Heat is transferred at an intermediate temperature level to the ammonia through a heat exchanger ('cascade heat exchanger'). In the heat exchanger the CO₂ is condensed while the ammonia is evaporated. The ammonia evaporated in the cascade heat exchanger is compressed to a pressure and temperature high enough to reject the heat to ambient.

Figure 1 below shows a typical Ammonia / CO₂ Cascade system with compression and pumped liquid on the low-side (Danfoss 2014).

FIGURE 1

Ammonia / CO₂ Cascade with Compression and Pumped Liquid on Low-Side



The compressor oil typically used in this type of system is fully miscible POE (Polyol Ester) which will become concentrated in the recirculator vessel. This requires an oil rectifier to distill the recirculator liquid and return the oil-rich liquid back to the crankcase of the compressor(s), and is similar in this regard to a pumped halocarbon system.

Various examples of this type of system are shown in the Appendix:

- Figure A1: NH₃/CO₂ with low side compression. CO₂ pumped to (3) x temperature levels. Electric or water defrosting. CO₂ reciprocating compressors with oil rectification.
- Figure A2: NH₃/CO₂ with low side compression. CO₂ pumped to (2) x temperature levels. Hot gas defrosting. CO₂ reciprocating compressors with oil rectification.



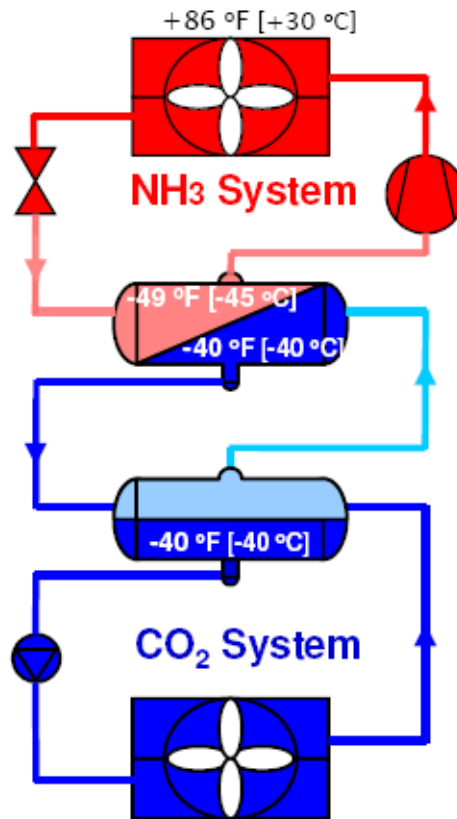
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- Figure A3: NH₃/CO₂ with low side compression. CO₂ pumped to medium temperature level, CO₂ DX to (2) x low temperature levels. Electric or water defrosting. CO₂ reciprocating compressors with oil rectification.
- Figure A4: NH₃/CO₂ with low side compression. CO₂ pumped to medium temperature level, CO₂ DX to (1) x low temperature level. Electric or water defrosting. CO₂ screw compressors without oil rectification.

Figure 2 below shows a typical Ammonia / CO₂ Cascade system with pumped liquid on the low-side but no CO₂ compression (Danfoss 2014). The CO₂ in type of system is often referred to as pumped "volatile brine".

FIGURE 2

Ammonia / CO₂ Cascade with Pumped Liquid and No Compression on Low-Side



In the system shown in Figure 2 above, since there is no compression taking place on the CO₂ side of the system, the CO₂ "brine" is kept oil-less and oil rectification in the CO₂ recirculator vessel is not needed.

An example of this type of system is shown in the Appendix:

- Figure A5: NH₃/CO₂ without low side compression. CO₂ pumped to (2) x temperature levels. Electric or water defrosting.

Note that in these types of systems, hot gas defrosting is possible but requires that liquid CO₂ be pressurized and then fed to a heat driven boiler to generate hot gas for defrosting. This type of hot gas defrost system will not be covered in this paper. Also, because of the relatively low difference between CO₂ supply liquid pressure and suction pressure, direct expansion of the evaporators would be difficult (not enough pressure difference for stable expansion valve operation) and pumped CO₂ is recommended. Evaporators for these types of systems are selected in the same way as you would for any pumped CO₂ system. i.e. Evaporators shown in Figure A5 are selected the same way as evaporators shown in Figure A1. These evaporators can be arranged for either top feed or bottom feed.

Effects of Oil on Evaporator Performance

Even though the oil used in CO₂ compressors is miscible (typically POE) and is in solution with the CO₂, there is a slight penalty to the boiling heat transfer coefficient (Kim and Hrnjak 2012). This penalty is very small (on the order of 1-3%), however, for normal operating conditions found with refrigeration evaporators and is typically neglected in ratings and selection.

Theoretically there is a very slight increase in evaporator performance in the oil-less volatile brine type system compared to the system with CO₂ compression, however, the bigger benefit is the elimination of oil rectification and problems associated with oil return.

Effects of Water on Evaporator Performance

As is pointed out elsewhere (Vestergaard 2004), the solubility of water in CO₂ particularly at low temperatures is very low in the vapor phase. This means that even small concentrations of water in CO₂ will form free water and freeze to ice when liquid CO₂ flashes at expansion valves. This will obviously cause plugging of the expansion valves and starving the coils, and must be avoided by: a) very thorough evacuation on startup, b) charging the system with dry CO₂, and c) installation and servicing of generously sized desiccant type filter-driers in the liquid line(s).

CO₂ WORKING PRESSURE

Table 1 below compares the saturation pressures for CO₂ and ammonia and illustrates the significantly higher pressures (and consequently higher strength requirements) for CO₂. Both copper and stainless



steel are completely compatible with CO₂ (Nelson 2014) and so are chosen as the tubing materials shown below.

TABLE 1					
Saturation Pressure vs Temperature					
CO ₂ vs Ammonia					
		Ammonia		CO ₂	
Temperature		Pressure		Pressure	
deg F	deg C	psia	bar	psia	bar
-60	-51.1	6	0.4	95	6.5
-40	-40.0	10	0.7	146	10.0
-20	-28.9	18	1.3	215	14.8
0	-17.8	30	2.1	306	21.1
20	-6.7	48	3.3	422	29.1
40	4.4	73	5.1	568	39.1
60	15.6	108	7.4	748	51.6
80	26.7	153	10.6	970	66.8

ASHRAE Standard 15 “Safety Standard for Refrigeration Systems” (ASHRAE 2013), sets the minimum design pressure for system components (including evaporators) in Section 9.2. This section of the standard also refers to the ASME Boiler and Pressure Vessel Code, Section VIII, as the appropriate method of determining the design (or ‘working’) pressure given evaporator dimensions and materials of construction.

Table 2 below shows the calculated minimum required design pressure for CO₂ piping and equipment, including evaporators, according to Section 9.2 of the standard. The temperature shown in Table 2 is defined as the “warmest location in the circuit” and can be considered the same as the room temperature.



TABLE 2				
Minimum Design Pressure vs Temperature				
CO2 Evaporators				
Temperature		Minimum Design		
		Pressure		
deg F	deg C	psia	psig	bar
-60	-51.1	113	99	7.8
-40	-40.0	175	160	12.1
-20	-28.9	258	243	17.8
0	-17.8	367	352	25.3
20	-6.7	506	492	34.9
40	4.4	681	666	47.0
60	15.6	897	883	61.9
80	26.7	1070*	1055*	73.8*

* Exceeds the critical pressure of CO2 so design pressure is set equal to the critical pressure.

Since copper and stainless steel are both recommended for use in carbon dioxide evaporators (Nelson 2014), Table 3 below has been constructed to show the minimum tube wall thickness needed to meet the requirements of ASHRAE Standard 15 in a CO2 evaporator operating at various room temperatures.

TABLE 3							
Minimum Tube Wall Thickness vs Room Temperature (ASHRAE Std 15)							
CO2 Evaporators							
Room Temperature		Minimum Tube Wall Thickness, in					
		SB-75 Cu Tube Diameter			SA-249 304 SS Tube Diameter		
deg F	deg C	3/8"	1/2"	5/8"	5/8"	7/8"	1"
-60	-51.1	0.010	0.010	0.010	0.010	0.010	0.010
-40	-40.0	0.010	0.011	0.013	0.010	0.010	0.010
-20	-28.9	0.012	0.015	0.018	0.010	0.010	0.012
0	-17.8	0.016	0.020	0.025	0.011	0.015	0.017
20	-6.7	0.022	0.028	0.034	0.015	0.021	0.024
40	4.4	0.027	0.035	0.043	0.020	0.027	0.032
60	15.6	0.036	0.046	NR	0.026	0.036	0.041
80	26.7	NR	NR	NR	0.031*	0.042*	0.048*

* Critical pressure used to determine MAWP.

Note that the minimum tube wall thicknesses shown in Table 3 are theoretical calculated values. In normal manufacturing practice, copper tubing with wall thickness less than about 0.016" is difficult to produce and to handle. With stainless steel tubing the practical minimum wall thickness is around 0.020". While lower temperatures may allow the use of light wall tubing and relatively low design pressures



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during normal operation, remember that the appropriate design pressure may be higher than the design room temperature in some cases. When specifying evaporator construction (tube material and wall thickness) all potential temperature/pressure conditions need to be considered, including (but not limited to):

- Startup conditions
- Peak load operation
- Abnormal loads (process temperature excursions)
- Standby conditions that occur frequently
 - o Power outages limited in time duration but which may happen with some frequency
 - o Shutdown during cleanup

Control Valve Groups for Convertible Rooms

In many cases, cold storage facilities are designed with convertible rooms. That is, with rooms which can be operated at different temperatures depending on the product being stored. Because of the high pressure of CO₂, transferring a convertible room from high temperature to low temperature operation can involve a very large change in pressure. To manage this large pressure change in both liquid and suction lines, motorized valves should be used with additional equalizing bleed valves. Piping diagrams for this type of convertible room unit(s) are shown in Figures A8 and A9.

OPTIMUM OVERFEED RATE FOR PUMPED CO₂

Reducing the overfeed rate in pumped refrigerant systems is desirable because pumping power will be reduced by the cube of the ratio of the reduction in flowrate. As the liquid overfeed rate is reduced, however, the risk of operating evaporators with the refrigerant in separated flow patterns (stratified/wavy) increases. Cooling capacity of the evaporator falls off dramatically when this occurs. With CO₂ in an evaporator having 5/8" tubes, a minimum mass flux of 41 lbm/ft²-s (approx. 1.5 TR/feed for 2:1 overfeed) is required to avoid stratified/wavy flow.



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The thermodynamic properties of CO₂ differ significantly from ammonia:

- Latent heat of vaporization is much lower resulting in higher mass flow rates for a given cooling capacity.
- The ratio of liquid to vapor density is much lower which results in lower void fractions (less tube volume occupied by vapor).
- Higher mass flux for reasons explained above (see Heat Transfer section).

These characteristics allow pumped CO₂ evaporators to be designed for lower overfeed rates compared to ammonia. Recommended overfeed rates for pumped CO₂ evaporators are 1.5:1 for coolers and 2:1 for freezers.

In comparison, to avoid separated flow in pumped ammonia evaporators, recommended overfeed rates are 3:1 for coolers and 4:1 for freezers.

DEFROST

CO₂ evaporators can be defrosted using the following methods:

- Electric
- Water
- Hot Gas
- Interlaced Warm Glycol

Electric Defrost

This method is simple and results in simplified control valves and piping. Control valve groups for this type of defrost are shown in Figures A6 through A9. There are added first costs associated with the required power cabling in addition to the operating costs from power consumed by the heating elements during defrost. The addition of return air defrost hoods shown below in Figure 3 can significantly reduce the amount of power consumed during defrosting (less heat lost to the room), reduce defrost duration, and improve the effectiveness of defrosting (coils get cleaner).

FIGURE 3
Return Air Defrost Hood



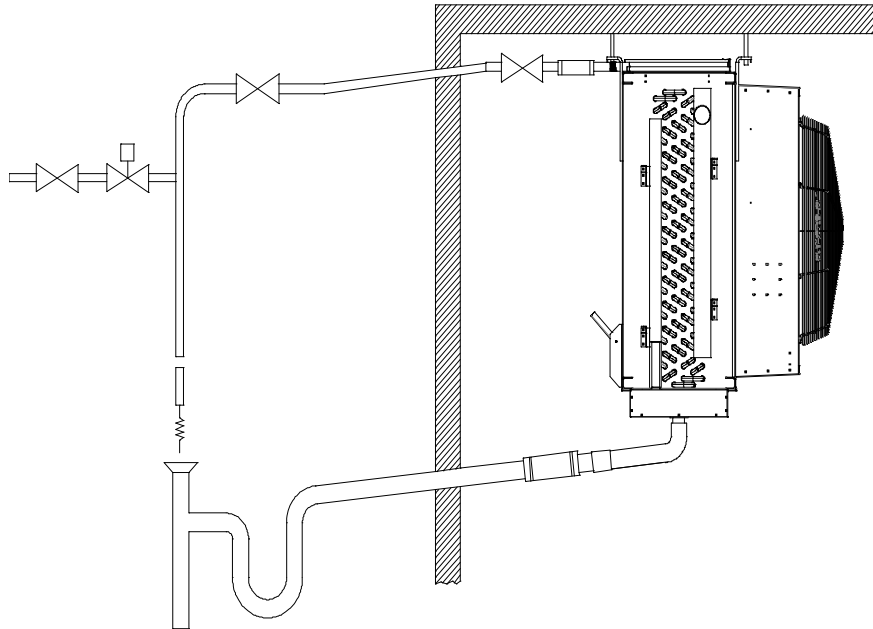
The evaporator manufacturer must insure proper placement and number of heating elements to insure a complete defrost. Also, provisions need to be made to prevent heating elements from "creeping" (moving out of the coil block over time) which can result in damage to the elements themselves and to electrical wiring.

Water Defrost

This method produces a very fast and complete defrost. Waste heat from the high side of the system can be used to heat the defrost water for reduced power consumption. Care must be taken to provide for complete drainage of water lines at the end of defrost as well as specifying and using high quality water supply valves. Use of high quality motorized ball valves is recommended for controlling water flow.



FIGURE 4
Evaporator Arranged for Water Defrost



The trick to making this type of defrost work in low temperatures is to design the evaporator cabinet and drainpan properly to manage and minimize overspray of water during defrost. Design guidelines and sequence of operation for water defrosting can be found elsewhere (Colmac Coil 2012). As with electric defrosting, the evaporator control valve group is very simple - see Figures A6 through A9.

Hot Gas Defrost

This method is very familiar to all of us in the industrial refrigeration business. The control valve groups are arranged the same as for ammonia and are shown in Figures A10 and A11. The main difference between CO₂ and ammonia of course is the pressure required to reach defrosting temperature with CO₂. Typically CO₂ hot gas provided for defrost must be at around 710 psig (50 bar) which translates to 60 deg F. The challenge with CO₂ hot gas defrost is having to add a separate compressor to provide the defrost gas. Figure A2 shows a hot gas defrost system with a swing compressor able to provide 710 psig gas during defrost and then go back to running with discharge pressure of 407 psig during normal cooling.

As with any hot gas defrost system, there must be a sufficient supply of gas coming from evaporators operating in the cooling mode to fund the amount of gas needed for defrosting (think "2 to 1 rule"). Oil return to the hot gas compressor is also something to carefully consider and manage.



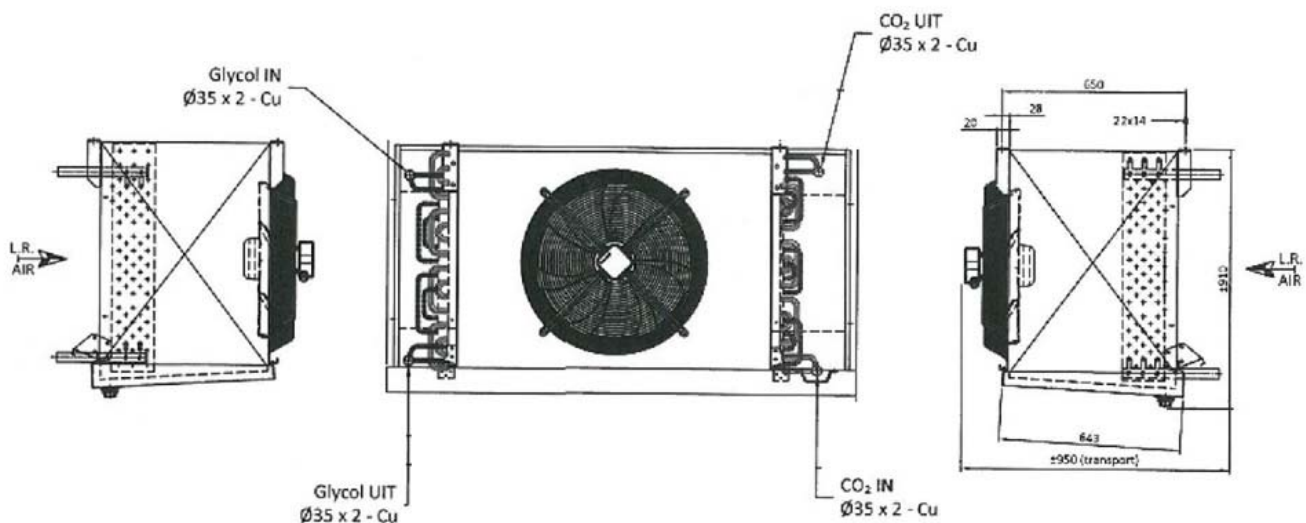
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Compared to electric and water defrosting, the more complex control valve groups as well as additional piping and the hot gas compressor may add cost to the point that this method is not financially attractive.

Interlaced Warm Glycol Defrost

In this method, heated Propylene Glycol is circulated through tubes evenly spaced throughout the coil block to warm the coil and melt frost. Figure 5 below shows an evaporator arranged for pumped CO₂ with an interlaced glycol defrost circuit.

FIGURE 5
CO₂ Evaporator with Interlaced Glycol Defrost Circuit



Glycol for defrosting is typically provided at the coil at 60 to 70 deg F (15 to 20 deg C) for 20 minutes at a heating duty that is approximately 150% of the cooling capacity (Pearson 2015). Glycol flowrate is supplied to the coil to produce this heating duty with a glycol temperature change of 10 to 12 deg F (5 to 7 deg C) at a glycol circuit pressure drop of around 10 to 15 psi (80 kPa).

While this defrosting method does offer the benefit of being able to use waste heat from the high side of the system to heat the glycol, it is quite limited in its application.

Room temperatures with this type of defrost should not be lower than 25 to 28 deg F (-2 to -3 deg C). Below these temperatures, a number of problems arise:



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1. Glycol pumping power becomes excessive,
2. The pumping pressure required to move the cold glycol out of the coil at the start of defrost will be quite high,
3. The amount of energy required to heat the mass of cold glycol coming from the coil at the start of defrost and the energy needed to cool it back down again at the end of defrost, becomes excessive,
4. Heat loss from the glycol circulating loop could become very high depending on how well insulated the piping loop is,
5. The risk of freezing glycol and bursting coil tubes due to poor mixing, improper preparation, or diluted glycol.

While the evaporator control valve groups are quite simple for this type of defrost system (see Figures A6 through A9), the added glycol heating system, piping, insulation, and control valves, add significant cost and may make interlaced glycol defrosting economically unfeasible.

CONCLUSION

The combined use of ammonia and carbon dioxide in industrial refrigeration systems offers end-users a solution which is both safe for the environment and safe for workers, products, and the public. The paper has described several possible arrangements for cascade NH₃/CO₂ refrigeration systems and has highlighted special concerns and considerations for CO₂ evaporators used in these systems.



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APPENDIX

1. Figure A1: P&ID NH₃/CO₂ with low side compression. CO₂ pumped. Electric or water defrosting. CO₂ recip compressors.
2. Figure A2: NH₃/CO₂ with low side compression. CO₂ pumped med and low temps. Hot Gas defrosting. CO₂ recip compressors.
3. Figure A3: NH₃/CO₂ with low side compression. CO₂ pumped med temp, DX low temp. Elec or Water defrosting. CO₂ recip compressors.
4. Figure A4: NH₃/CO₂ with low side compression. CO₂ pumped med temp, DX low temp. Elec or Water defrosting. CO₂ screw compressors.
5. Figure A5: NH₃/CO₂ without low side compression. CO₂ pumped. Elec or Water defrosting. CO₂ recip compressors.
6. Figure A6: Control Valve Group for Top Feed Pumped CO₂ Evaporator. Single Temperature Level. Electric Defrost.
7. Figure A7: Control Valve Group for Top Feed Pumped CO₂ Evaporator. Multi-Temperature Levels. Electric Defrost.
8. Figure A8: Control Valve Group for Bottom Feed Pumped CO₂ Evaporator. Single Temperature Level. Electric Defrost.
9. Figure A9: Control Valve Group for Bottom Feed Pumped CO₂ Evaporator. Multi-Temperature Levels. Electric Defrost.
10. Figure A10: Control Valve Group for Bottom Feed Pumped CO₂ Evaporator. Single Temperature Level. Hot Gas Defrost.

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FIGURE A1
NH3/CO2 with low side compression. CO2 pumped. Electric or water defrosting. CO2 recip compressors.

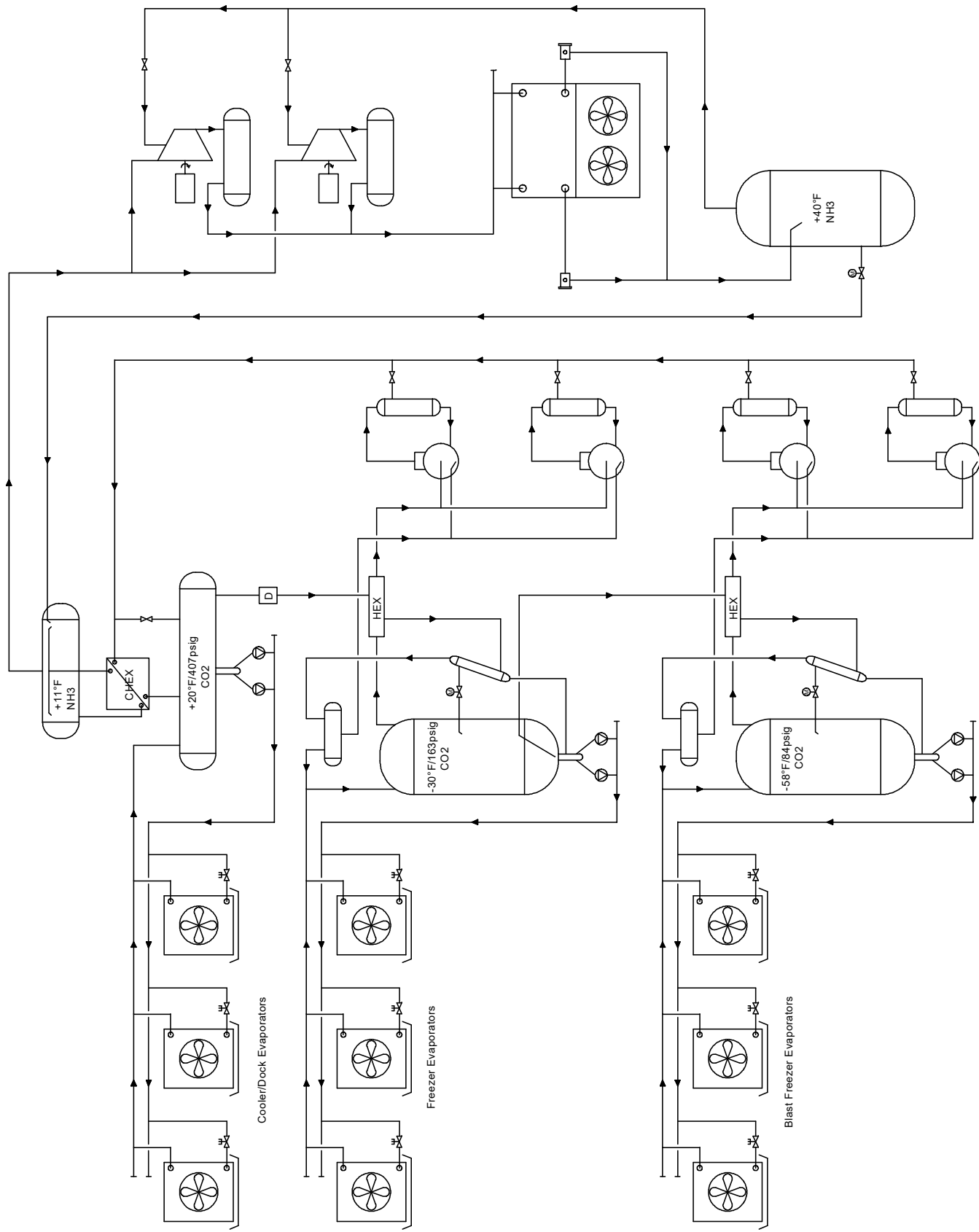


FIGURE A2

NH3/CO2 with low side compression. CO2 pumped med and low temps. Hot Gas defrosting. CO2 recip compressors.

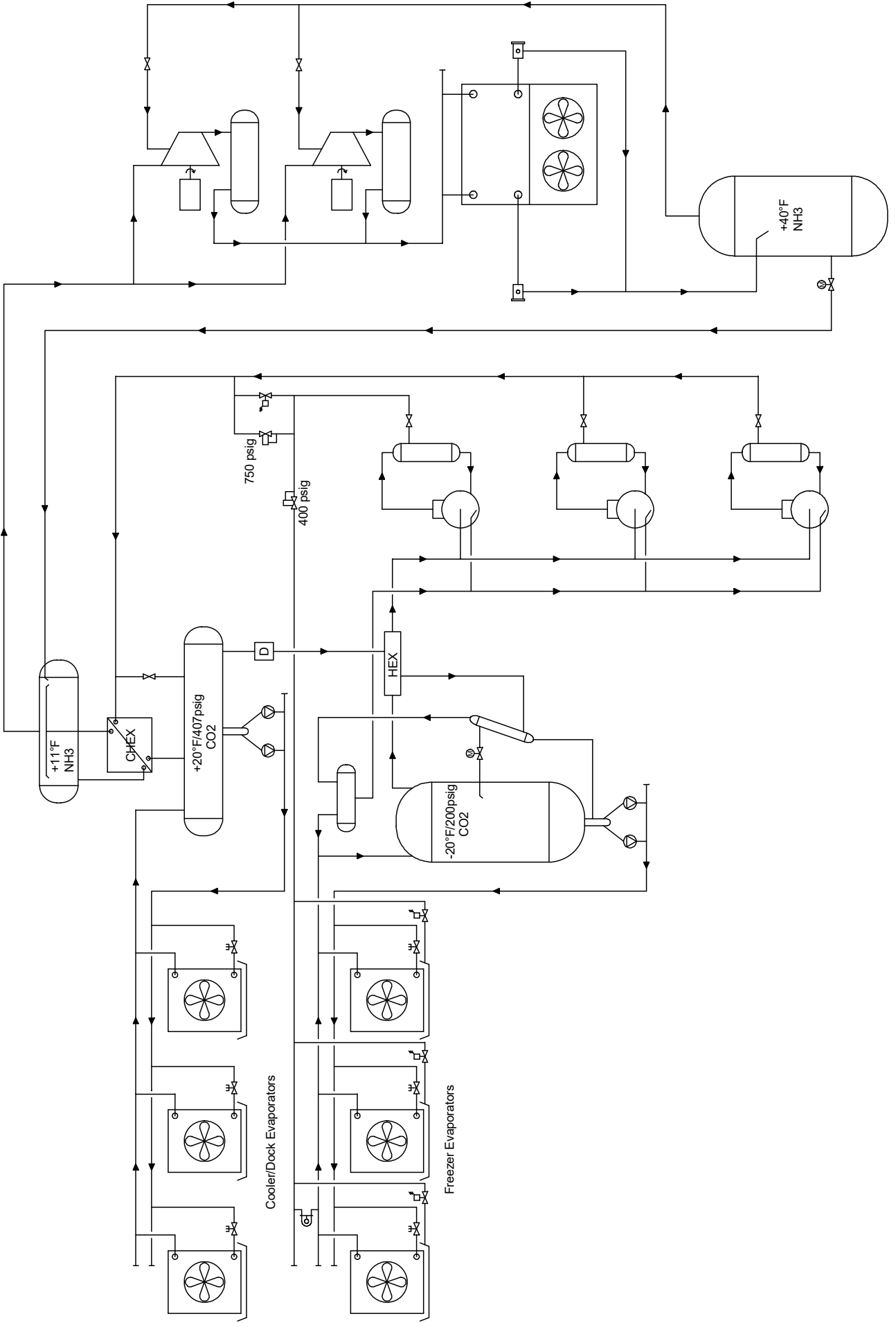


FIGURE A3

NH3/CO2 with low side compression. CO2 pumped med temp. DX low temp. Elec or Water defrosting. CO2 recip compressors.

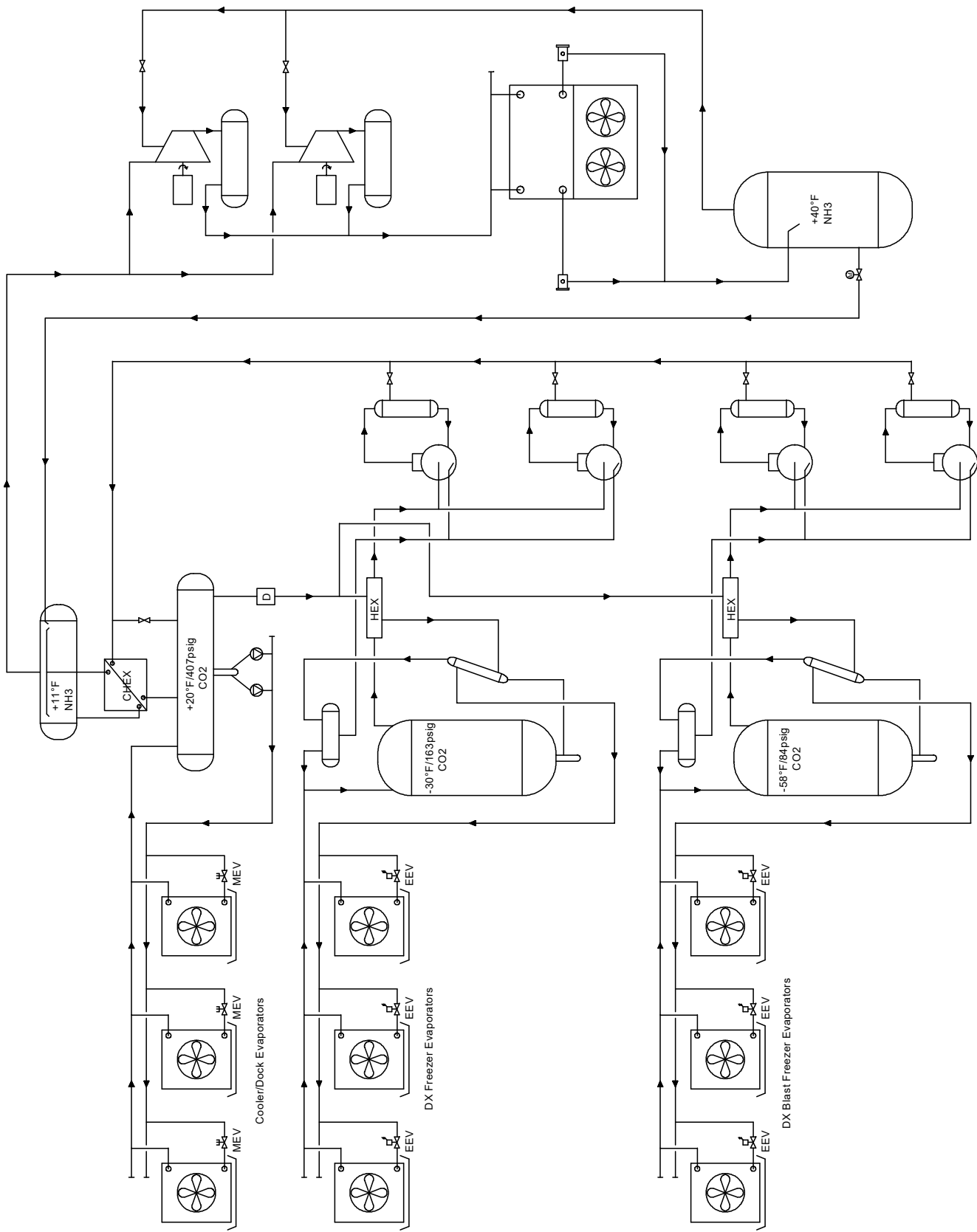


FIGURE A4

NH3/CO2 with low side compression. CO2 pumped med temp, DX low temp. Elec or Water defrosting, CO2 screw compressors.

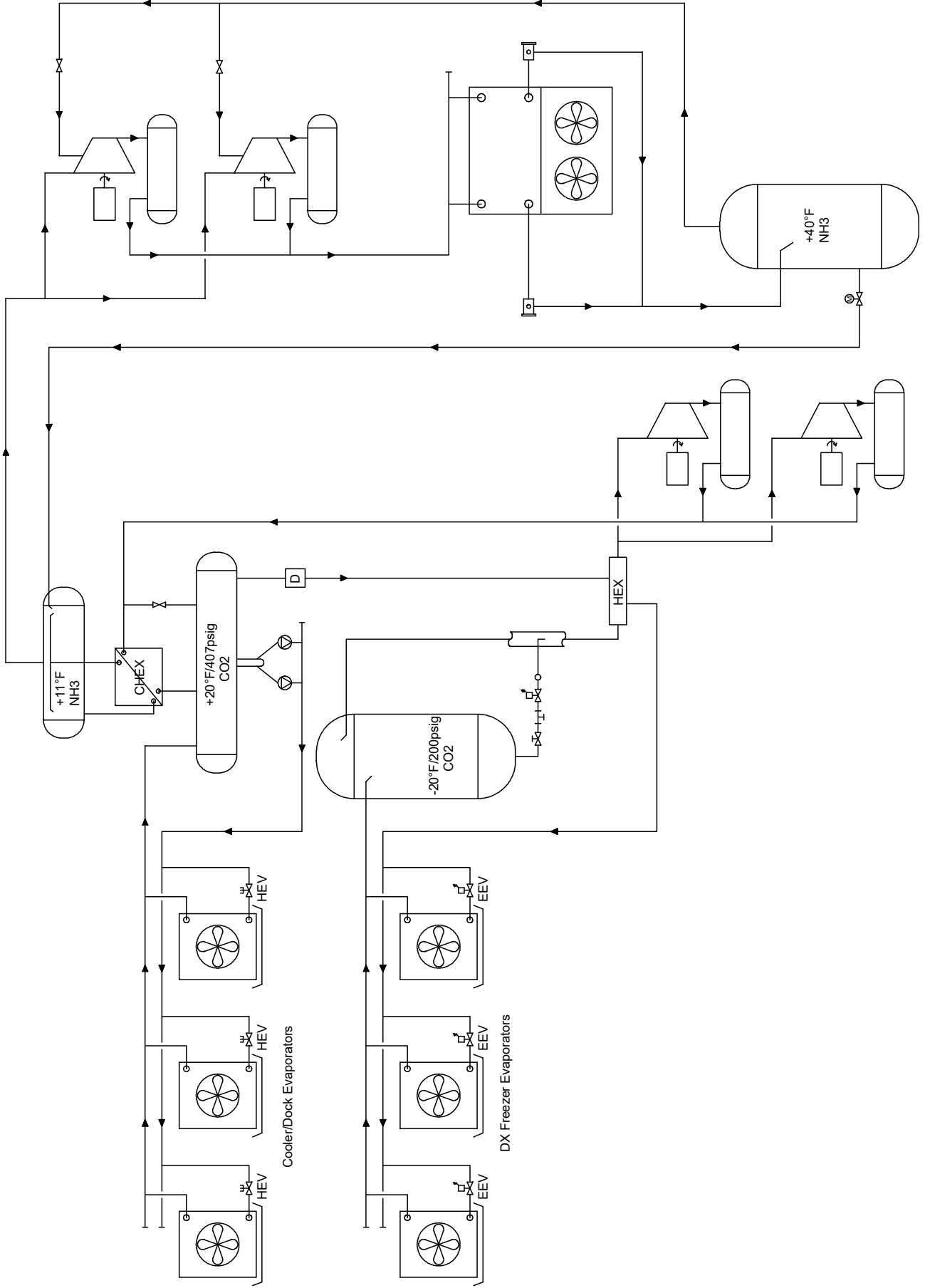


FIGURE A5
 NH3/CO2 without low side compression. CO2 pumped. Elec or Water defrosting. CO2 recip compressors.

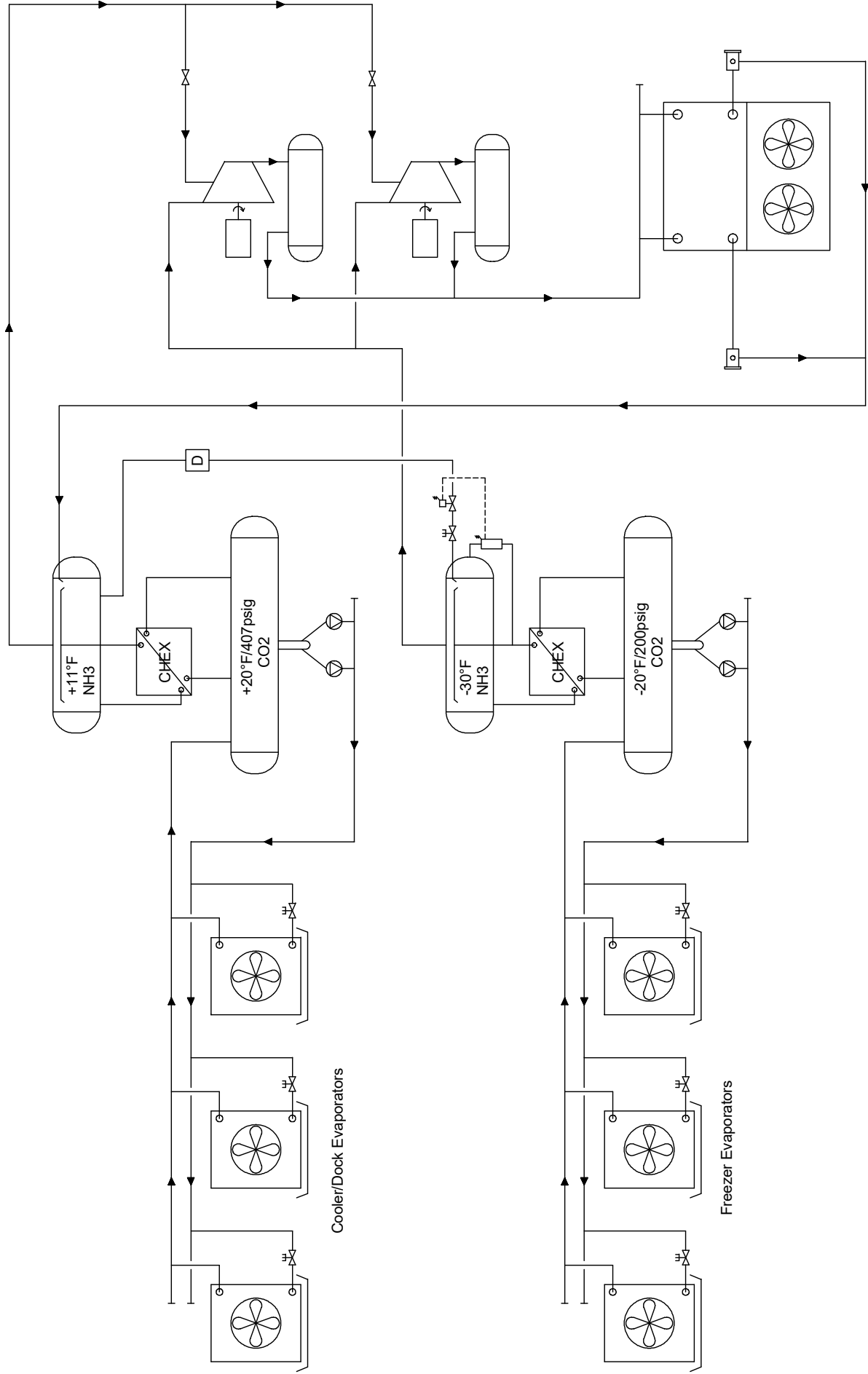


FIGURE A6

Control Valve Group for Top Feed Pumped CO2 Evaporator. Single Temperature Level. Electric Defrost.

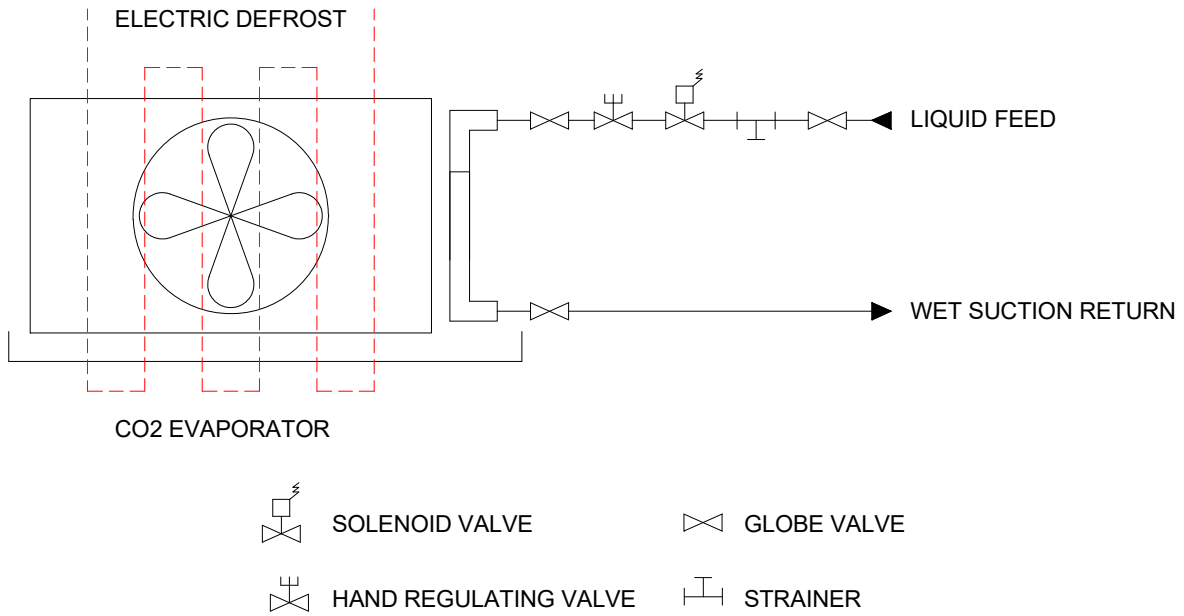


FIGURE A7

Control Valve Group for Top Feed Pumped CO2 Evaporator. Multi-Temperature Levels. Electric Defrost.

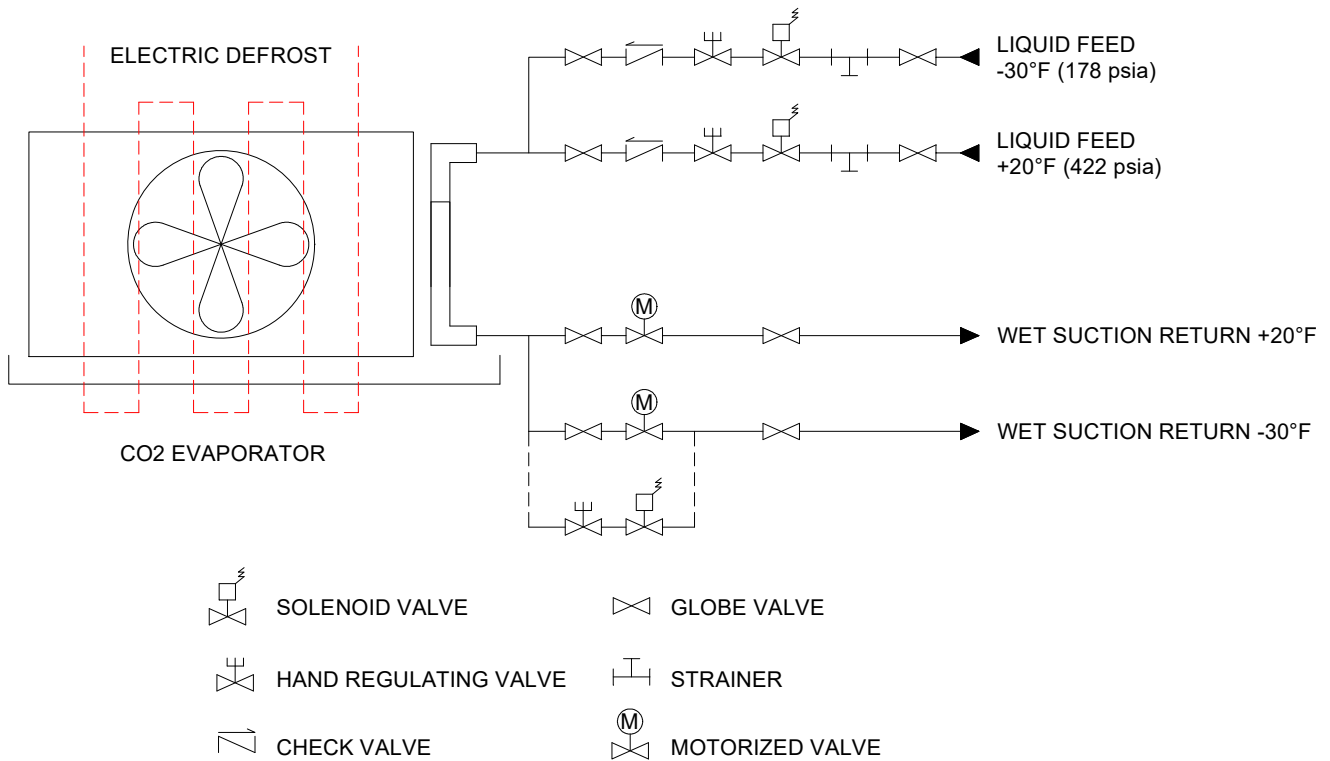


FIGURE A8

Control Valve Group for Bottom Feed Pumped CO2 Evaporator. Single Temperature Level. Electric Defrost.

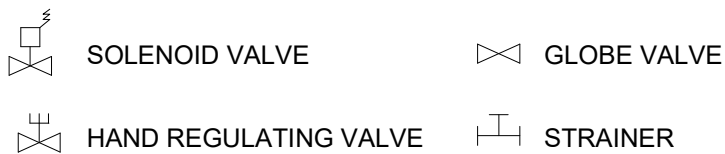
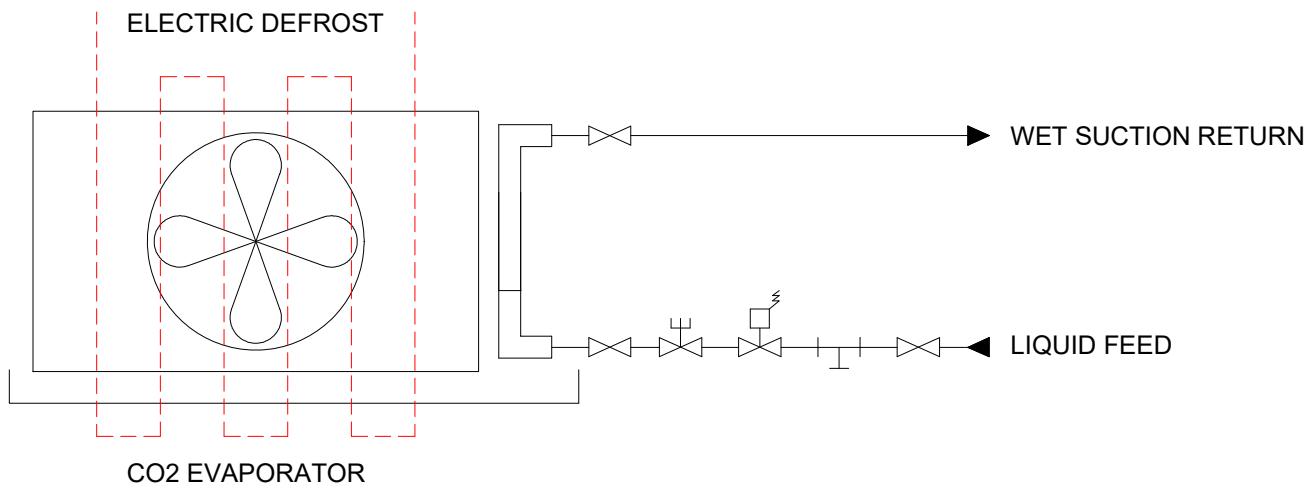


FIGURE A9

Control Valve Group for Bottom Feed Pumped CO2 Evaporator. Multi-Temperature Levels. Electric Defrost.

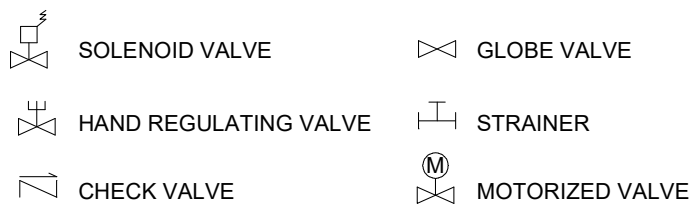
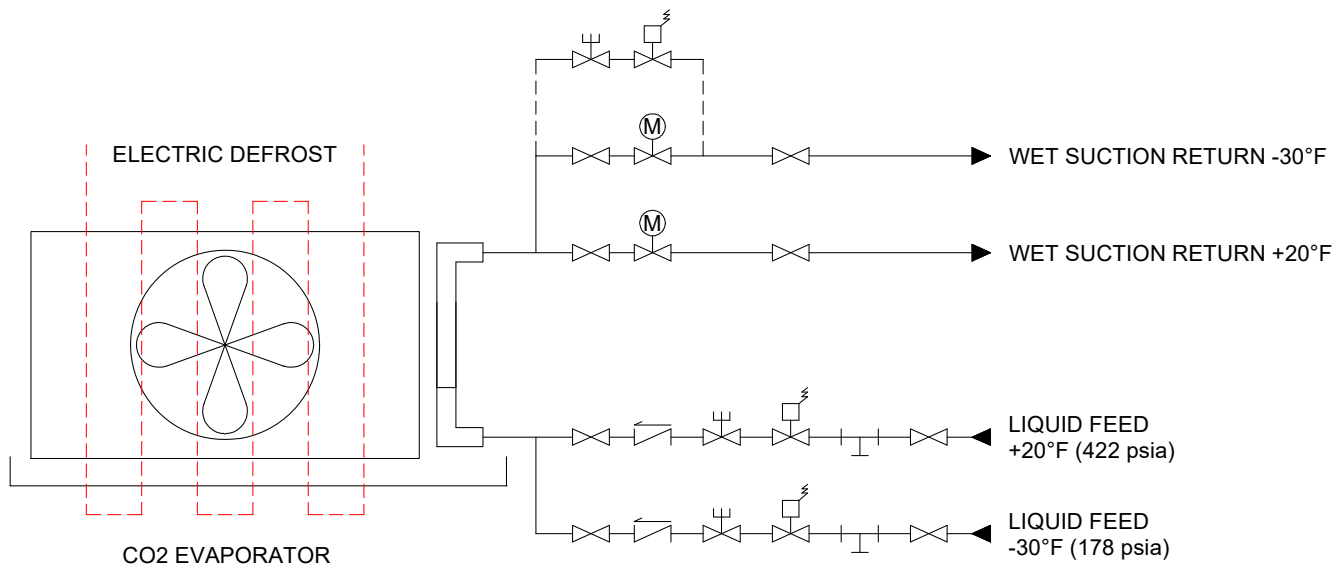
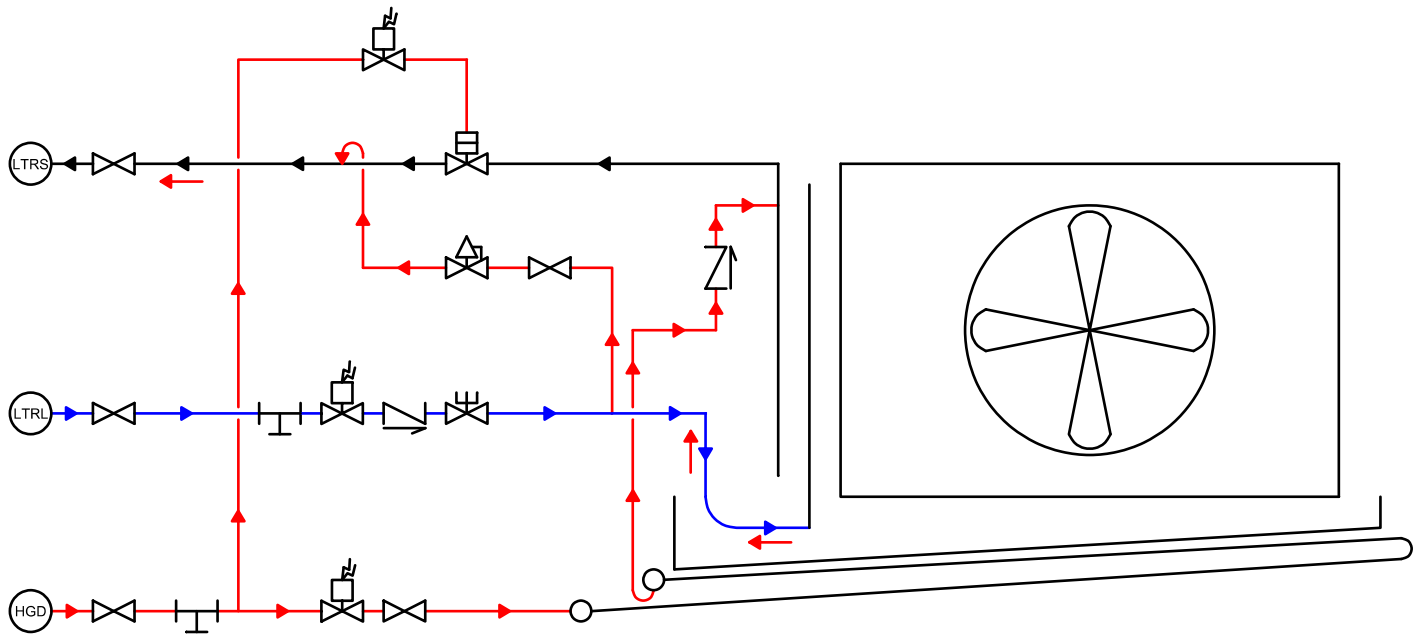


FIGURE A10

Control Valve Group for Bottom Feed Pumped CO2 Evaporator. Single Temperature Level. Hot Gas Defrost.
















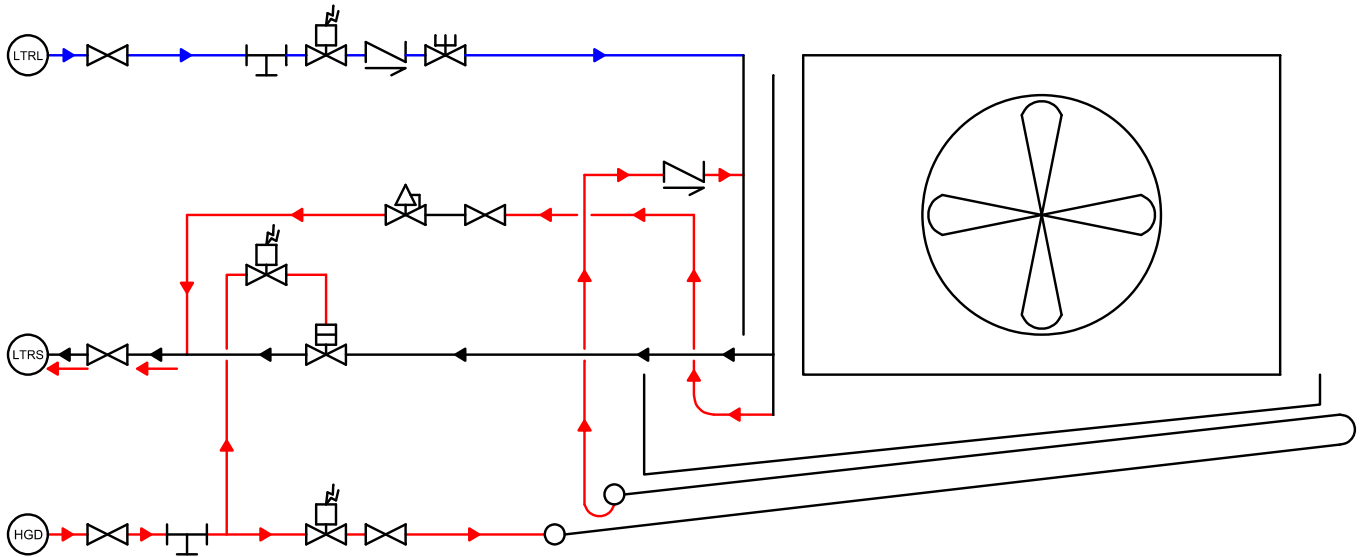













- | | | | |
|---|--------------------------------|---|-------------------------------|
|  | STRAINER |  | HOT GAS DEFROST SUPPLY |
|  | CHECK VALVE |  | LOW-TEMP RECIRCULATED SUCTION |
|  | GLOBE VALVE |  | LOW-TEMP RECIRCULATED LIQUID |
|  | SOLENOID VALVE |  | HOT GAS FLOW |
|  | DEFROST PRESSURE REGULATOR |  | SATURATED LIQUID FLOW |
|  | GAS POWERED SUCTION STOP VALVE |  | SATURATED LIQUID & VAPOR FLOW |
|  | HAND EXPANSION VALVE | | |

FIGURE A11

Control Valve Group for Top Feed Pumped CO2 Evaporator. Single Temperature Level. Hot Gas Defrost.



- | | | | |
|---|--------------------------------|---|------------------------------------|
|  | STRAINER |  | HGD HOT GAS DEFROST SUPPLY |
|  | CHECK VALVE |  | LTRS LOW-TEMP RECIRCULATED SUCTION |
|  | GLOBE VALVE |  | LTRL LOW-TEMP RECIRCULATED LIQUID |
|  | SOLENOID VALVE |  | HOT GAS FLOW |
|  | DEFROST PRESSURE REGULATOR |  | SATURATED LIQUID FLOW |
|  | GAS POWERED SUCTION STOP VALVE |  | SATURATED LIQUID & VAPOR FLOW |
|  | HAND EXPANSION VALVE | | |